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# The Application of Advanced Spark Ignition Engine Combustion Systems for High Performance and a Better Environment

by

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# ABSTRACT

This is a thesis that brings together work conducted over a thirty year period concerning the research, development and knowledge management of high performance and low exhaust emission engines. The thesis includes nine published and refereed works that are discussed and appended.

Internal combustion engines translate the chemical energy of a fuel into mechanical work by burning the fuel with air in a combustion chamber. It is demonstrated that this process can be improved beneficially with respect to power output, fuel economy and exhaust emissions, by efficient cylinder filling and the generation of enhanced charge motion characteristics at the point of ignition. The advantages of multivalve engines, particularly with 5-valves per cylinder, and the methods of producing and measuring good air flow and beneficial amounts of tumble or barrel swirl, are described. Two patents and three novel research techniques for air flow and air motion are presented and discussed. The combustion developments carried out by the author for competition and high-performance road cars are presented as examples of the application of the theory and research.

It is appreciated that exhaust emission control is an essential feature of good engine development. The techniques for control are described together with relevant factors that determine the emissions performance of road vehicles in overall terms as well as during the legislative emissions test procedures. Different manufacturers' philosophies in this regard are discussed and a number of examples of cost-effective, high performance, low-emission engines are described.

The need for knowledge transfer in engineering and manufacturing industries is explained together with some advantages that can be derived by the learner and the tutor from e-learning. Examples of some best practice are described for collaborative projects and internationally recognised use of the Internet. The author demonstrates successful transfer of engine research and development knowledge to emerging automotive manufacturers, resulting in the launch of new engines, some of which are class-leading for power output and fuel economy.

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1. Development of the Aston Martin V8 endurance racing engine
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# NOMENCLATURE AND ABBREVIATIONS

| | | |
|---|---|---|
| A | Cross sectional area | [$m^2$] |
| aBDC | After bottom dead centre | [°CA] |
| ACT | Air charge temperature | [K] |
| AFR | Air to fuel ratio | |
| $A_i$ | Mean inlet valve area | [$m^2$] |
| $A_p$ | Piston area | [$m^2$] |
| APTEO | Advanced Powertrain Engineering Operations (Ford Motor Company) | |
| aTDC | After top dead centre | [°CA] |
| B | Bore diameter | [m] |
| bBDC | Before bottom dead centre | [°CA] |
| BDC | Bottom dead centre | [°CA] |
| BMEP | Brake mean effective pressure | [bar] |
| bTDC | Before top dead centre | [°CA] |
| $C$ | Speed of sound | [$m.s^{-1}$] |
| $C_c$ | Calibration constant | |
| $C_d$ | Discharge coefficient | |
| $C_i$ | Mean intake valve discharge co-efficient | |
| $C_{local}$ | Speed of sound under local conditions | [$m.s^{-1}$] |
| $C_s$ | Scale multiplier (constant) for a laminar flow element in a Cussons viscous air flow meter | |
| $C_v$ | Flow coefficient | |
| $C_v\theta_{mean}$ | Mean of the integral of flow coefficient across a given valve opening profile against crank angle from intake valve opening to valve closing | |
| $C_{v180}$ | Mean of the integral of flow coefficient across a given valve opening profile against crank angle from TDC to BDC | |
| CA | Crank angle | [Degree or Radian] |
| CARB | California Air Resources Board | |
| $CF$ | Correction factor | |
| $CF_{t_{air}}$ | Correction factor for air temperature | |
| CFD | Computational fluid dynamics | |
| CI | Compression ignition | |
| CO | Carbon monoxide | |
| $CO_2$ | Carbon dioxide | |
| CoV | Coefficient of variation | |
| CSSR | Cold start spark retard | |
| CVH | Compound valve hemispherical | |
| CVT | Continuously variable transmission | |
| D | Mean valve seat diameter | [m] |
| DOHC | Dual (or double) over head camshafts | |
| $DS_{max}$ | Outside diameter of the inlet valve seat insert | [m] |
| $DS_{min}$ | Inside diameter of the inlet valve seat insert | [m] |
| $DV_{max}$ | Outside diameter of the inlet valve seat | [m] |
| $DV_{min}$ | Inside diameter of the inlet valve seat | [m] |

| | | |
|---|---|---|
| EAEC | European Automobile Engineers Cooperation | |
| ECE | Economic Commission for Europe | |
| ECT | Engine coolant temperature (Sensor) | |
| EGI | Exhaust gas ignition (Ford Motor Company) | |
| EGO | Exhaust gas oxygen (Sensor) | |
| EGR | Exhaust gas recirculation | |
| EMS | Engine management system | |
| EPI | Environmental Performance Index (Rototest AB) | |
| EU | European Union (European Economic Community) | |
| f | Resonant frequency | [Hz] |
| FIA | Fédération Internationale de l'Automobile | |
| FISITA | Fédération Internationale des Sociétés d'Ingénieurs des Techniques de l'Automobile | |
| FMEA | Failure modes and effects analysis | |
| FTP | Federal test procedure | |
| GTL | Gaydon Technology Limited | |
| $H_2O$ | Water | |
| HC | Hydrocarbons | |
| HP | Horsepower | |
| IDI | Inductive digital ignition (TVS Motor Company) | |
| IMechE | Institution of Mechanical Engineers | |
| IMEP | Indicated mean effective pressure | [bar] |
| IMRC | Inlet manifold runner control (Ford Motor Company) | |
| ISM | Impulse swirl meter (Cussons Technology Ltd.) | |
| JV | Joint venture | |
| $L$ | Linear dimension (e.g. Length or Lift) | [m] |
| $L_{optimum}$ | Optimised length | [m] |
| $L_s$ | Stroke | [m] |
| L/D | Non-dimension valve lift (ratio to mean valve seat diameter) | |
| LBT | Leanest (fuelling) for best torque | |
| LDA | Laser Doppler anemometry | |
| LLP | Limited Liability Partnership | |
| $\dot{m}$ | Mass flow rate | [$kg.s^{-1}$] |
| $\dot{m}_a$ | Air mass flow rate | [$kg.s^{-1}$] |
| $\dot{m}_{a\,Std\,pbox}$ | Air mass flow rate through a pressure box under *standard conditions of pressure and temperature* | [$kg.s^{-1}$] |
| $\dot{m}_f$ | Fuel mass flow rate | [$kg.s^{-1}$] |
| M | Torque as read by impulse swirl meter | [Nm] |
| MAF | Mass air flow (Sensor) | |
| MBT | Minimum (spark advance) for best torque | [°CA] |
| MIGV | Mean inlet gas velocity | [$m.s^{-1}$] |
| MOSFET | Metal oxide Silicon field effect transistor | |
| MPG | Miles per gallon | |
| N | Engine speed | [$rev.min^{-1}$] |
| $N_2$ | Nitrogen | |
| $NO_x$ | Oxides of nitrogen | |
| NVH | Noise, vibration and harshness | |

| | | |
|---|---|---|
| $O_2$ | Oxygen | |
| OEM | Original equipment manufacturer | |
| OHC | Overhead camshaft | |
| $P$ | Pressure | [Pa] |
| $\Delta P$ | Pressure drop | [Pa] unless stated |
| $\Delta P_{pbox}$ | Pressure drop across a pressure box | [Pa] unless stated |
| $\Delta P_{ma}$ | Pressure drop across an air flow meter | [Pa] unless stated |
| $P_a$ | Air pressure | [Pa] |
| $P_b$ | Brake power | [W] |
| Pb | Lead | |
| $P_{baro}$ | Barometric pressure | [Pa] unless stated |
| $P_i$ | Inlet pressure | [Pa] unless stated |
| $P_{ref}$ | Standard reference pressure | [Pa] |
| $PbO_y$ | Collective oxides of lead | |
| PFMEA | Potential failure modes and effects analysis | |
| $\dot{Q}$ | Volume flow rate | [$l.s^{-1}$] |
| $\dot{Q}_{meter}$ | Volume flow rate through a meter | [$l.s^{-1}$] |
| $\dot{Q}_{pbox}$ | Volume flow rate through a pressure box | [$l.s^{-1}$] |
| $Q_{LHV}$ | Lower heating (calorific) value of a fuel | [$J.kg^{-1}$] |
| $\dot{Q}_{Std\ pbox}$ | Volume flow rate through a pressure box under standard conditions of pressure and temperature | [$l.s^{-1}$] |
| $r_c$ | Compression ratio | |
| $R$ | Ideal or Universal gas constant (= 8314) | [$J.kmol^{-1}.K^{-1}$] |
| $R_s$ | Swirl ratio | |
| R&D | Research and development | |
| RCF | *Roller cam follower (TVS Motor Company)* | |
| RFI | Radio frequency interference | |
| $S$ | Sulphur | |
| $\overline{S}$ | Mean speed | [$m.s^{-1}$] |
| $\overline{S}_i$ | Mean intake flow speed | [$m.s^{-1}$] |
| $\overline{S}_p$ | Mean piston speed | [$m.s^{-1}$] |
| SAE | Society of Automotive Engineers | |
| SAM | Société Anonyme Monegasque | |
| SI | Spark ignition | |
| SME | Small or medium sized enterprise | |
| $SO_z$ | Collective oxides of sulphur | |
| STP | Standard temperature and pressure | |
| $t$ | Time | [s] |
| $t_{optimum}$ | Optimum time | [s] |
| $T$ | Temperature | [K] unless stated |
| $T_a$ | Air temperature | [K] unless stated |
| $T_i$ | Intake temperature | [K] unless stated |
| $T_m$ | Air temperature at air meter | [K] unless stated |
| $T_{pbox}$ | Temperature of the air in a pressure box | [K] unless stated |
| $T_{ref}$ | Standard reference temperature | [K] |
| TDC | Top dead centre | |
| THC | Total hydrocarbons | |

| | | |
|---|---|---|
| TWC | Three way catalyst | |
| US / USA | United States of America | |
| V | Enclosed volume | [$m^3$] |
| $V_s$ | Swept volume | [$m^3$] |
| v / vpc | Valves per cylinder | |
| VM | Vehicle manufacturer | |
| VTi | Variable timing intelligent (TVS Motor Company) | |
| $Z$ | Mach Index | |
| $\eta_{f,b}$ | Brake fuel conversion efficiency | |
| $\eta_{f,i}$ | Indicated fuel conversion efficiency | |
| $\eta_{f,ig}$ | Fuel conversion efficiency for an ideal gas | |
| $\eta_{mech}$ | Mechanical efficiency | |
| $\eta_v$ | Volumetric efficiency | |
| $\lambda$ | Equivalence ratio for AFR (Lambda) | |
| $\gamma$ | Ratio of specific heats (= 1·4 for air at STP) | |
| $\rho$ | Density | [$kg.m^{-3}$] |
| $\rho_a$ | Air density | [$kg.m^{-3}$] |
| $\phi$ | Equivalence ratio from fuel to air ratio (=1/λ) | |
| $\phi_S$ | Valve seat angle | [Degree] |
| $\theta$ | Crank angle | [Radian or Degree] |
| $\omega_s$ | Angular velocity of a swirl rig paddle wheel | [$Rad.s^{-1}$] |
| $\overline{\omega}_s$ | Mean angular velocity of a swirl rig paddle wheel | [$Rad.s^{-1}$] |

# Chapter 1.

# INTRODUCTION, BACKGROUND AND OVERVIEW

## 1.1 Introduction

This is a thesis on advanced combustion designs and their benefits for spark ignition (SI) engine performance and emissions, developed from research and development (R&D) in that field conducted over a period of thirty years. It is based on the work that has produced published technical papers and patents since 1984 and other research that the author has led and to which he has contributed. The work includes the application of engine designs and developments for high power and low emissions by the author and his teams, combined with key novel elements of the transfer of such knowledge to new generations of engine developers.

This submission includes nine published and refereed works by the author on the theme of engine R&D and the engineering skills required to perform them. These show how engine performance and real world exhaust emissions may be improved by innovative design and manufacturing, how that process is implemented and how engineers in these disciplines are developed to carry out effective and well-communicated automotive R&D.

## 1.2 Background to the Publications

The purpose of the internal combustion engine is to translate the chemical energy of a fuel into mechanical work. This is achieved by burning fuel with air in a combustion chamber arranged in such a way as to translate the combustion pressure into linear force and then rotating torque, through a piston assembly and crankshaft. The modern engine is required to liberate the maximum amount of useful work with, now more than ever before, the least fuel consumption and the lowest amounts of harmful waste products. These waste products arise principally from incomplete combustion of the fuel and the accompanying, relatively minor, reactions between the constituents in the combustion space. The latter includes species such as the oxides of nitrogen ($NO_X$) that are formed under high temperature and pressure conditions, and minor products arising from combustion of impurities and additives in the fuel and lubricant.

The quest for increased power output, increased fuel efficiency and reduced exhaust emissions has been one of the greatest challenges to the automobile engine of the last forty years. The US Clean Air act of 1970 legislated exhaust emission reductions at a rate that would not have been achieved through conventional developments but this was often at the cost of fuel efficiency and engine power output. For example, compression ratios were generally reduced in order to limit maximum cylinder pressure and temperature to control $NO_x$, with a consequent drop in engine fuel efficiency. Furthermore, to clean up legislated combustion products, catalytic converters were introduced that required the engine to run close to a stoichiometric air-fuel ratio, which was far richer than that required to operate at part load, and also created an increase in exhaust back pressure, which further reduced engine fuel efficiency by increasing pumping losses and restricting cylinder scavenging. More recent legislative and market pressures on fuel economy and carbon dioxide ($CO_2$) emissions have resulted in downsizing of the swept volume of engines and a consequent desire for increases in specific power output.

Increasing the power output of an engine has been at the heart of many automobile racing formulae for a century and, during the 1980s, some fuel limitations for racing were introduced. Both these have driven improvements in combustion and cycle efficiency and the lessons learned during these developments have contributed to the development of passenger car engines that are now more powerful, more fuel efficient and lower emitters than previous designs.

The nine publications on which this thesis is based, describe a sequence of significant steps led by the author in the development of efficient engine combustion systems and the achievement of some challenging goals over the last thirty years. Adaptation of the Aston Martin V8 for endurance racing required increases in power output and mechanical robustness, whilst capitalizing on the inherently good combustion and strong cylinder block and head of the base engine [Bale (1984)]. This thesis describes the engine developer's response to a racing engine requirement where swept volume was limited by the FIA and a number of Formula One and endurance racing teams were using identical engines. Analysis of cylinder heads by the author's design team that had developed the Aston Martin racing engine, suggested that engine power output could be increased with a new design by the combination of increased valve area and optimised air motion. The hypothesis was researched with new cylinder heads on a Cosworth DFL base engine and then commercially applied to a Judd V8 for Camel Team Lotus [Bale and Downing (1990)]. It was during the latter development that the measured combustion characteristics and efficient conversion of fuel energy into work suggested that an engine with optimised inlet flow and tumble air motion could be powerful, fuel efficient and clean. Application of this premise is described in the author's paper on the high power output 2-valves per cylinder spark ignition engine developed for the Ford Falcon [Bale and Sykes (1993)]. The paper demonstrated that an engine that already has adequate valve area does not necessarily need an expensive multi-valve cylinder head when attention to detail is given to the port and camshaft design and gas exchange processes. It showed that low cost improvements to a volume production engine resulted in some 9% increase in power and 5% more torque. The power output was estimated to be 95% of the maximum benchmark for road-legal engines of the same swept volume.

A paper that embodies the heart of this thesis was published at the 1995 AE Asia technical symposium in Kuala Lumpur [Bale and Sykes (1995)]. In it, as well as reviewing the Ford Falcon engine development detailed in the previous paper [Bale and Sykes (1993)], the combustion research work appropriate to the 5-valves per cylinder concept for Formula One [Bale and Downing (1990)] was described as it had been applied to a Ford 2·5 litre V6 road car engine. The Ford V6 engine was fitted with inlet manifold runner control (IMRC) to generate inlet-induced axial swirl in its series, 4-valves per cylinder, version. The paper described the advancement of fundamental port research work, developed for Formula One, into a 5-valves per cylinder head that used tumble air motion to produce the desired combustion characteristics. This resulted in the engine being highly tolerant to charge dilution with exhaust gas recirculation (EGR) and producing lower hydrocarbon emissions at wide open throttle. Engine performance was increased and the cost and complication of IMRC was removed. Both the engine projects described in the paper were delivered on a client basis with fixed deliverables and budget, the first being for niche production and the second to demonstrate technology capability for future models. Additional research into optimised fuel injector delivery to complement the advanced port designs was also described, and development components were produced from which series parts could be derived.

The aspect of transferring engine technologies from research to production through supplier partnerships was developed in the author's 1994 SITEV paper on engineering services and joint ventures [Bale (1994)]. This paper described the challenges for an engine R&D company in its dealings with major clients and gave details of the production implementation of the technical improvements to the Ford Falcon XR6 engine [Bale and Sykes (1993)] with its enhanced performance cylinder head, via a joint venture programme. Only by such an arrangement was the technology transferred in an effective and mutually profitable way for the end-user, the manufacturer, from the source of the engine research and engine development.

One of the technical conclusions from the road-going Ford V6 5-valves per cylinder project, described above and in a further paper [Sykes (1995)], was that the engine produced lower hydrocarbon emissions at wide open throttle than the original 4-valves per cylinder version due to its improved combustion. It could be argued that since the current vehicle emissions regulations are based on a largely part-load chassis dynamometer cycle, this could be considered a null conclusion in the quest for high power output with low exhaust emissions. However, overall air quality improvement, which was the objective of the US Clean Air Act of 1970 and many pieces of legislation since, is significantly affected by all operating modes of the engine. This aspect, and particularly the importance of wide open throttle hydrocarbon emissions, was described in a paper that compared real life vehicle exhaust emissions performance with that measured on legislative drive cycles [Bale and Farnlund (1999)]. Given that a significant proportion of real world emissions derive from engine operations outside of those encountered during the legislative drive cycle, the importance of underlying good combustion, described in the author's work, becomes even clearer. The ongoing challenge to deliver the three objectives of good performance with good fuel efficiency and low emissions, has to be considered holistically and accepting that the manufacturer will bias the degree of optimisation according to needs, e.g., best emissions during the drive cycle, best fuel economy under steady state cruising and / or best performance at wide open throttle. The engine developer's delivery of optimised combustion through the techniques described in this thesis, amongst others, at engine operating conditions where the calibration of the engine and its control system minimises the negative impact on the other two parameters from the optimisation of the primary one, is fundamental to customer satisfaction and environmental tolerance.

The pressures on automobile engines from society and legislators relative to noise and exhaust pollution, fuel consumption, increased performance and reduced cost are mirrored by the pressures put on the component and system manufacturers by vehicle and engine companies. Nowhere is this more apparent than in the domain of the piston and cylinder system where the author had a senior technical role in advanced technology and research for T&N plc, later taken over by Federal-Mogul Corporation. Some of the technical solutions delivered by the author were design-related, but the complete technical armoury including materials, predictive techniques and manufacturing processes, had to be brought to the product development cycle. This was described in a published review of relevant piston and cylinder technology trends, drawing strongly on the author's knowledge of combustion, for which the piston forms one of the boundary interfaces [Bale and Neuhäuser (1998)].

All the projects reported in the forgoing publications were carried out on the basis of detailed engineering and programme management. Only by these disciplines were the ambitious technical development targets achieved on time and on budget. The other key element throughout the work was the quality of the individuals involved as co-authors, named contributors and other colleagues. The author's research and engineering career has involved much development of engines and the engineers involved in the projects. The author reported bringing these two elements together to combine the technical benefits and the learning processes of engine research to his

Institution of Mechanical Engineers (IMechE) 2001 Automobile Division Chairman's Address, published in the Proceedings of the Institution [Bale (2001)]. This paper demonstrated that by maintaining a disciplined product development process, an optimum solution could be obtained in the launch of an advanced powertrain. The same was true of the process of development and education of engineers, to which the author attested from his personal combination of research and teaching. The author continued to pass on the benefit of his engine knowledge and research ways in which engineering knowledge could be effectively transferred to young and mature engineers, such as those undertaking Continuous Professional Development (CPD). This applied to both established and emerging markets, where he had considerable first hand experience as a tutor and engine research mentor, which led to further published work by his graduate student engineers. The author has also contributed to the development of improved techniques for effective technical knowledge transfer via the Internet. This was the subject of a paper published in the Proceedings of the Institute of Cast Metal Engineers, for which he was author and presenter at the World Foundry Congress in 2006 [Ashley *et al* (2006)].

The body of work presented in this thesis documents original research for fundamental engine efficiency from competition variants of production engines, through the development of advanced concepts for the highest levels of pure racing engines, to the use of advanced combustion techniques for high performance engines with the lowest emissions as measured by real-life drive cycles. Interwoven in the engineering research is a developmental methodology whose benefit is compounded if a similar methodology to the product development is applied to the training and development of team members. The author continues to put this into practice in his consultancy business and has proven to be successful in passing on the benefits of his engine research and teaching it to, for example, TVS Motor Company in India. There he mentored the R&D department's creation of a range of in-house TVS engines and his contribution is acknowledged in publications by the company describing its technical developments [Deshmukh *et al* (2004)].

In summary, this thesis is based on nine published papers in the theme of engine research and development that show how internal combustion engine performance and real world exhaust emissions may be improved by innovative design and manufacturing, how that process is implemented and how engineers in these disciplines are developed to carry out effective and well communicated automotive R&D.

The publications presented and described in this thesis are as follows:

1. **Bale, C.J.C. (1984)** "Development of the Aston Martin V8 Endurance Racing Engine", Automotive Engineer, published by Mechanical Engineering Publications, Bury St Edmunds, UK, Volume 9, Number 4, August/September 1984, pp. 20-22.
2. **Bale, C.J.C. and Downing, I.C. (1990)** "The Design and Development of a Unique Five Valve Cylinder Head", Paper 905156, Proceedings of the Society of Automotive Engineers (SAE), XXIII FISITA Congress, Turin, Italy, 7-11 May 1990, Volume II, pp. 301-308.
3. **Bale, C.J.C. and Sykes, R.G. (1993)** "The Development of a High Output 2vpc SI Engine for the Ford Falcon", Proceedings of the Institution of Mechanical Engineers, Autotech 93 Conference, National Exhibition Centre, Birmingham, UK, 16-19 November 1993, Volume C93, pp. 43-51.
4. **Bale, C.J.C. (1994)** "The Supply of Engineering Services in the Global Car Industry", Proceedings of SITEV Technical Congress, Lingotto Centre, Turin, Italy, 15-17 November, 1994, published by Associazione Tecnica Dell'Automobile (ATA).
5. **Bale, C.J.C. and Sykes, R.G. (1995)** "High Specific Output, Low Emissions Spark Ignition Engines", Automotive Engineering Asia Technical Congress, Kuala Lumpur, Malaysia, 26-27 October 1995, Paper R95/122.

6. **Bale, C.J.C. and Neuhäuser, H-J. (1998)** "Technology trends in power cylinder systems", Automotive Technology International 1998, published by Stirling Publications Ltd., London, UK, ISSN 0950 4400, pp. 62-66.

7. **Bale, C.J.C. and Farnlund, J. (1999)** "Real-life Vehicle Exhaust Emission Performance Compared with Legislative Drive Cycles", Proceedings of the Institution of Mechanical Engineers, Paper C575/030/99, Integrated Powertrain Systems for a Better Environment, pp. 27-47.

8. **Bale, C.J.C. (2001)** "High Performance Engineering", Proceedings of the Institution of Mechanical Engineers, Automobile Division 2001 Chairman's Address and Paper.

   *This paper was published by the IMechE and first presented at the ordinary meeting of the Institution at its headquarters, 1 Birdcage Walk, London, on 4th October 2001 and subsequently as a nationwide lecture tour; nominated as the IMechE-SAE exchange lecture for presentation at the SAE World Congress in Detroit, U.S.A., March 2002.*

9. **Ashley, C., Bale, C.J.C., Millan, N., Williams, T.M. and Hendley, R.J. (2006)** "Latest Trends in Industrial Skills Development Techniques", Proceedings of the Institute of Cast Metal Engineers, World Foundry Congress, Harrogate, UK, 4-7 June 2006, Paper 197.

Other relevant published material:

10. **Sykes, R.G. (1995)** "Tickford Five Valve Per Cylinder Technology for Optimized Performance and Combustion", SAE Paper 950815. *The engine technology research reported in this paper was an extension of that published by Bale and Sykes (1995).*

11. **Deshmukh, D., Kumar, R., Garg, M., Jaffer Nayeem, M. and Lakshminarasimhan, V. (2004)** "Optimization of Gas Exchange Process on a Single-Cylinder Small 4-Stroke Engine by Intake and Exhaust

Tuning: Experimentation and Simulation", SAE Paper 2004-32-0007 and JSAE Paper 20044293, Society of Automotive Engineers Small Engine Technology Conference, Graz, Austria. 27-30 September 2004.
*The engine technology research reported in this paper was taught and mentored by the author of this thesis and is acknowledged as such in the paper.*

## 1.3 Author's Career 1975-2007

The author graduated from Loughborough University in 1975 with a Bachelor of Technology honours degree in Automotive Engineering that included a seven-month industrial placement with Aston Martin Lagonda Ltd. in its "Experimental" department, where he worked on the successful 1974 US exhaust emission compliance project. After graduation, he first held a post in production engineering at Willowbrook International Ltd. (bus and coach builders in Loughborough) before moving into his chosen field of engine design and development with Lucas Electrical Ltd. in 1976, as an assistant development engineer in the "Electronic Fuelling and Control Systems Laboratory". Here he worked on petrol injection projects for Jaguar, Rover, MG and Triumph and also represented Lucas in a tripartite working group on engine and catalyst development for low emissions, with Johnson Matthey and Jaguar-Rover-Triumph.

In 1978, the author was invited to return to Aston Martin Lagonda Ltd. as a development engineer to lead the company's re-entry to the US market, from which it had been absent for several years. During that time tailpipe emissions limits had reduced considerably, necessitating both engine and aftertreatment systems development with minimum performance loss. As well as research and development in his specialist area of engines for road and competition use, the author applied his up-to-date knowledge of exhaust aftertreatment systems. During this period he conceived and commissioned a low-cost, chassis dynamometer development facility, with rolling road, gas analysis equipment and high ambient temperature capability, for exhaust emission and cooling system development. As part of the company's engineering activities, the author also researched and applied engineering and scientific principles related to anti-corrosion treatment and windscreen wiper and reclining seat mechanisms. He also played an active role in the company's racing engine development for works and privately prepared saloon and sports cars.

Aston Martin formed a separate engineering consultancy company in 1981 and the author was one of 35 staff invited to transfer to this business, which was created to sell engineering solutions to motor industry clients. He accepted a Senior Development Engineer role specialising in engines rather than the more general product development role that was available in the parent car manufacturing company. The consultancy company, Aston Martin Tickford, grew and later became completely independent as Tickford Limited, part of CH Industrial plc, moving to a new, bespoke engineering centre in Milton Keynes in 1984 where the author was an important member of the team that specified, purchased and commissioned the engine and vehicle test facilities. The author became responsible for the engine emissions profit centre and £2·25m investments, as well as providing his emission measurement and other expertise to the 'engine test and development' profit centre. His role grew consistently and he managed more that 200 engine R&D projects for clients in engine and vehicle manufacturing, the petroleum industry and component companies, with full commercial and reporting responsibilities as well as a technical leadership role. With extensive experience of combustion measurement, air flow development and engine management, he was also closely involved in the company's research and competition engine projects that were both in-house and client sponsored work. The author used these high-profile projects and the competencies they demonstrated, to expand the company's customer base into Europe, the US and the Far East. From 1993 he led Tickford's three-year alliance with Perkins Technology Limited, which together offered engine manufacturers a comprehensive R&D service in spark and compression ignition engines and their applications. By 1996 the author had risen through the positions of 'Engine Projects Manager' and 'Engineering Manager – Engines' to become 'Business Development Director', a role that carried with it the responsibility of creating, maintaining and marketing the company's engine patent portfolio. He had been intimately involved with the novel research since the late

1980s, various findings of which were reported in papers described in this thesis herein, and successfully defended the 5-valves per cylinder patent, in person, at the European Patent Court.

This career path of fifteen years with Tickford had seen technical skills and responsibilities supplemented with commercial and managerial ones, in a successful and growing company with two UK engineering centres and facilities in the USA, Germany, Australia and Thailand. When invited to join T&N plc in 1996 as the Director of its Advanced Technology Centre for Pistons and Pins, a most difficult decision had to be made following such a long and successful career with Tickford that had resulted in a directorship and responsibility for business development for all engines operations. In order to continue his contribution to engine R&D, the author joined T&N plc's Piston Product Group in 1997 and gained a staff of 43, 18 engine test cells, metrology and metallurgy laboratories and 7 CAE seats, with a remit to develop new cylinder system technologies for the global needs of the automotive and off-highway engine industries. This provided an advanced R&D capability positioned between AE Goetze's customer-focused product engineering department and the T&N group research centre at Cawston House near Rugby. As well as providing technical and managerial leadership during this period, the author sponsored and supervised a number of PhD and MSc students and became an external examiner, as well as being the T&N Technical Directors Group spokesman on lean R&D. He provided advanced engine technical support to T&N customers and licensees all round the world, both personally and with his centre staff.

Following the takeover of T&N plc by Federal-Mogul Corporation in 1998, the new owners rationalised the group's three engine R&D centres (for pistons/pins, rings/cylinder liners and thin wall bearings respectively) into one activity in Germany. The author left the company at that time and established his own consultancy business based around his knowledge of engines and emission control technologies.

Whilst conducting consultancy contracts for clients in UK and abroad, the author has continued to publish papers in his field as well as organising international conferences and seminars and creating and delivering bespoke training in engine technology and professional engineering competencies. After a number of years as member and chairman of the technical programme committee, member of the board and the policy working group of the IMechE Automobile Division (AD), the author served as chairman of the AD in 2001-02. His chairman's paper was nominated as the IMechE–SAE exchange lecture for presentation at the SAE World Congress in Detroit in March 2002, as well as being delivered on a UK lecture tour to the IMechE's ten regional centres for automobile engineering. As AD chairman, he represented the UK on the council of FISITA and elected to work on the education committee. Completion of his term of office coincided with the handover of his 9-year role as the UK representative on the council of the European Automobile Engineers Cooperation (EAEC).

The author became a part-time member of the teaching staff at Loughborough University in 2000, lecturing and examining 'Engine Design' in Part B MEng and BEng Automotive Engineering, and as examiner of the final year vehicle engineering projects. He was already an established visiting lecturer in 'International Product Development' to the IGDS MSc course at the University of Hertfordshire, having previously been an external lecturer in engine design and development to the MSc Automotive Engineering course at Cranfield University between 1990 and 1996. In addition to lecturing duties, the author held four-year external examiner posts at the University of Central England in Birmingham and then for the University of Hertfordshire's Automotive Engineering MSc and International Automotive Engineering MSc courses. The latter role included additional duties at Hochschule für Angewandte Wißenschaften Hamburg (Hamburg University of Applied Sciences), and with the University of Hertfordshire's other academic collaborators across Europe.

The author has created and taught postgraduate automotive engineering courses in academia and industry, particularly in the automotive supply chains in UK and India, and has guided combustion and engine research in TVS Motor Company, a leading Indian motorcycle manufacturer, since late in 2000. The benefit of this input has enabled the company to develop its own product range and terminate its 19-year licensing arrangement with Suzuki Motor Corporation in November 2002. Reference is made in this thesis to the research and application work contributed by the author, which has been acknowledged in published papers by TVS Motor Company.

From late 2002 to 2005, the author was part-time Programme Executive for EuroMotor®, a network and partnership of European car manufacturers, suppliers and universities, based at the University of Birmingham. EuroMotor was formed in 1990 to improve the knowledge base of the European motor industry through high-level technical collaborative training and broke new ground by bringing together international universities and car manufacturers to address training issues on a Europe-wide basis. It developed further to include the modern developments in organisations including lean manufacture and simultaneous engineering. The author helped to expand its activity to more closely follow the needs of the automotive supply industry through its UK and European networks. The most important aspect of the project was the European dimension, bringing together staff, managers, engineers, technicians and trainers from different companies and organisations resulting in cross fertilisation of ideas and practices. A key enabler was the application of on-line and blended learning techniques for which the author created and edited suitable training material in automotive engineering and business improvement to add to the resources available through the AutoTrain® and AutoTrain-Europe Internet portals. The author's knowledge and experience proved essential in understanding the technical and business needs of engineering and manufacturing companies and engaging them in improvement and technical development programmes. He ensured that training courses from providers were fit for their purpose and appropriate in technical content and delivery methodology, to ensure that useful engineering knowledge and business techniques were acquired in the automotive supply chain.

Following the end of European Union (EU) funding in 2005, the EuroMotor and Autotrain projects were combined into a spin-out partnership and, with the author's participation, this continues to provide industrial, professional and academic institutions with e-learning courses and knowledge transfer techniques for engineering skills and business improvement. Based on his expertise in this field, the author was appointed as an honorary Research Associate by the University of Birmingham in 2006 and, since 2005, has been a visiting lecturer in automotive product development and supply chain management at the University of Warwick.

## 1.4 Thesis Overview

This first chapter has already outlined the original research work published by the author over an extended period in the automotive engine industry and described the career path of engineering and senior management in which he has developed and disseminated the body of knowledge of engine and system R&D. The remaining chapters of this thesis are summarised as follows:

Chapter 2 reviews the literature concerned with high performance spark ignition engines in the context of which the author's published work has been carried out. The review also introduces the theory of catalytic converters as a means of supplementing the combustion system to reduce engine exhaust emissions to the atmosphere.

Chapter 3 discusses 5-valves per cylinder inlet flow developments and the unique contributions made by the author in this field.

Chapter 4 describes engine combustion developments and discusses the author's and other's development of new methods of analysing air motion and optimising the combustion benefits of intake-induced air motion.

Chapter 5 discusses the limitations of different cylinder head designs and the constraints of manufacturing production engines. It describes the techniques applied by the author to minimise the effect of such constraints in delivering both high performance and competitive costs. Different multivalve cylinder heads and a unique variable geometry intake system are described.

Chapter 6 discusses the engine developers' response to their quest for improved air quality and the need to limit pollutant emissions over the whole engine operating envelope, not just that regime used in completing the legislative emissions compliance drive cycle. The author's data that compare the emissions performance of different vehicles under real world driving conditions with that of controlled driving, are discussed. The benefits of the author's combustion research and development work are put into context, as they apply to engine manufacturers and the supply chain.

Chapter 7 discusses the merits of effective knowledge transfer as a key supporting process to good engine R&D as well as the benefits of e-learning as a medium. The author uses examples from his own published work and related experience to demonstrate how the global automotive community can benefit from shared research, publication and engineer development schemes.

Chapter 8 summarises the author's input, the novelty, originality and technical significance of the work.

Chapter 9 details the conclusions of the thesis and indicates topics for further work.

The appendices contain details of the patents related to the author's work and other relevant material citations, together with the full texts of the author's publications.

Unless referenced otherwise, all the figures are the author's work.

## 1.5 Contributions to the Body of Knowledge

The author has become established over a long career as one of the knowledge-holders on engine air flow and combustion. He has led research work on engines including patented work with unique 5-valves per cylinder and stepless variable induction systems, which have been applied to engines of several major manufacturers. In addition, three specific novel research techniques have been developed for air flow and motion development under his leadership:

1. a cylinder simulation chimney for quantifying port induced tumble;
2. an empirical means of comparing flow distributions around inlet valve peripheries;
3. a variable port configuration tool to investigate the geometric factors affecting tumble generation.

In the course of his environmental R&D work, the author has contributed to the creation of the EU "Extra Urban Drive Cycle" from research on the different characteristics and candidate cycles that were considered for implementation in legislation. Relevant applications of engine, fuelling and catalyst technologies by the author have enabled employers and clients to meet exhaust emissions standards in challenging territories. The author also contributed his knowledge of engine emissions and their control to the British Technical Council for the Motor and Petroleum Industries over a number of years, and was the vice chairman of the vehicle emissions test group from 1991 to 1994.

As a direct result of his engine R&D work and knowledge, the author has formed strategic and high level relationships for his employers with some of the world's leading automotive manufacturers and he has equipped members of his teams to go on to distinguished careers in the industry.

The author has played a leading role in the learned society for his profession, the IMechE, including national chairman of its Automobile Division, technical programme leadership, formulation and delivery of international technical conferences and seminars as well as publishing his own work and that of his R&D teams. In the global automotive industry, he has contributed to the development and expansion of EAEC and its biennial congress over nine years, as well as a term of office representing the United Kingdom at FISITA.

The author has seen the benefits of his research and knowledge management, reported in this thesis, successfully applied to new engines at Tata Motors and TVS Motor Company in India. Their products are contributing to customer satisfaction, the economic use of fuel and the prevention of the pollution crises such as those that affected the US and Japan during the large scale proliferation of combustion-powered automobiles. Through his lecturing and knowledge transfer activities, the author is passing on his expertise and guiding those who, with new products and publications, are also adding to the body of knowledge in this field.

## 1.6 Closing Remarks

This chapter has described the context of this thesis, the author's publications and the author's career of engine research and its application to high performance and low exhaust emission engines. The author's contributions to the international automotive engineering profession and his contributions to the body of knowledge have been introduced. The next chapter describes some engine fundamentals and discusses relevant literature of the period of the author's work.

# Chapter 2.

# LITERATURE SURVEY

## 2.1 Introduction

This chapter describes some important engine fundamentals and reviews some of the most relevant literature of the thirty-year period of the author's work. It will explain the predominance of spark ignition combustion for high performance engines and review the factors that affect power output. In particular, there is a review of the established combustion chamber types and the main options for intake and exhaust valve arrangement. It also reviews exhaust aftertreatment and some of the main factors affecting cylinder gas flow with specific reference to valve area and the intake effects that relate directly to the author's work.

The dominant form of the internal combustion engine for passenger cars is the four-stroke, spark ignition type pioneered by Nicholaus Otto in 1876. The four-stroke "Otto Cycle" is completed in four strokes of a reciprocating piston between its extremities as part of a slider-crank mechanism, as illustrated in Figure 1.

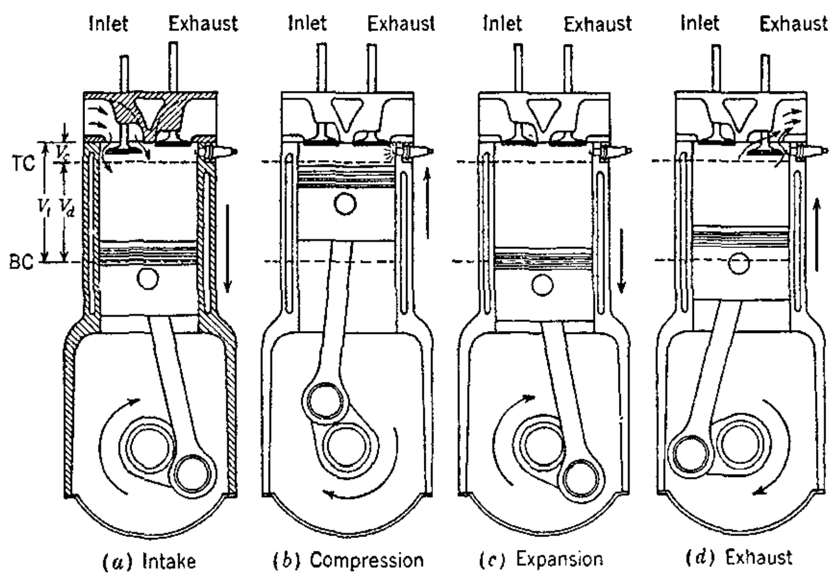



**Figure 1: Four-stroke operating cycle**
*(Reproduced from Heywood 1988a)*

The cycle begins with an induction or intake stroke, in which the outward moving piston draws charge into the cylinder through the open inlet valve, followed by a compression stroke in which the inward moving piston increases the pressure and the temperature of the fuel and air charge, towards the end of which, a spark is provided to initiate combustion. The burning fuel-air mixture rapidly increases the gas temperature and hence the pressure in the compressed cylinder space, to provide the force that propels the piston outwards in the third stroke of the cycle. The final inward stroke of the piston is used to discharge the burnt gases out through the open exhaust valve.

## 2.2 Power Density Advantage of SI Engines

Figure 2 illustrates the theoretical thermodynamic Otto cycle in terms of pressure and volume in the cylinder during the four strokes of the piston, annotated with the heat input and output and the work input and output. The instantaneous isochoric (constant volume) heat input and output at the top dead centre (TDC) and bottom dead centre (BDC) positions of the crankshaft, respectively, clearly differ from what actually happens in a working engine but facilitate comparison with other cycles, as will be shown in the following discussion.

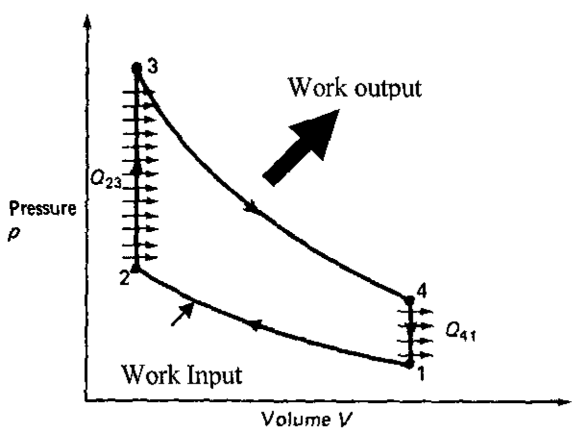



**Figure 2: Theoretical four-stroke Otto cycle**
*(Reproduced from Stone (1999a) and annotated)*

The compression ignition cycle, invented by Rudolph Diesel in 1893 and demonstrated in 1897, provides the main alternative for engine operating cycles in energy conversion, e.g. generators, motor vehicles and other heavy forms of transportation including railway and shipping. The basic four-stroke breathing and valve arrangements of the Diesel engine are similar to the Otto cycle engine but instead of inducing a fuel-air charge, the cylinder compresses air alone and the introduction of fuel into the compression-heated air is sufficient to initiate combustion, which continues as additional fuel is injected during the power stroke. Figure 3 shows the theoretical operating cycle for this type of engine working on a "constant pressure" Diesel cycle.

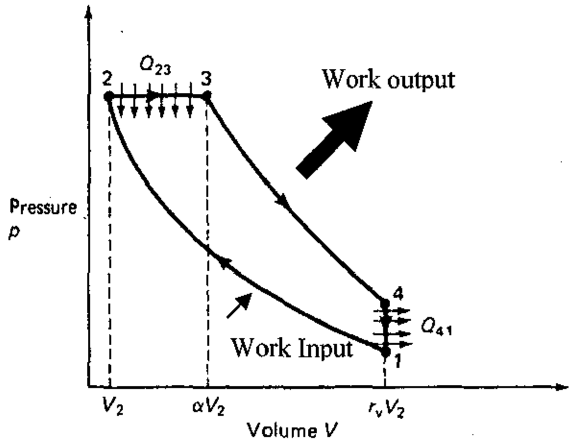



**Figure 3: Theoretical four-stroke Diesel cycle**
*(Reproduced from Stone (1999b) and annotated)*

In reality, neither the actual SI engine nor the Diesel engine adheres to these theoretical cycles. In the SI engine, the pressure rise due to combustion of the fuel is not instantaneous and hence not isochoric. The actual Diesel engine does not work to the Diesel cycle but operates more closely to a mixed or 'limited pressure' cycle as illustrated in Figure 4.

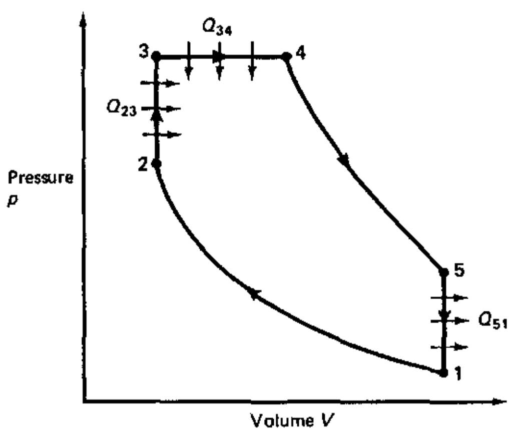



**Figure 4: Theoretical mixed, dual or limited pressure cycle**
*(Reproduced from Stone (1999c))*

In the latter theoretical cycle, there is heat addition at constant volume up to a pressure limit and, fulfilling the description as a mixed cycle, there is further heat addition as the volume increases. This is closer to what occurs in a Diesel engine. A review of the ideal gas cycles by Heywood (1988b) included a useful comparison of fuel conversion efficiency against compression ratio for constant volume, constant pressure and limited pressure cycles. Heywood's evaluation is illustrated in Figure 5.

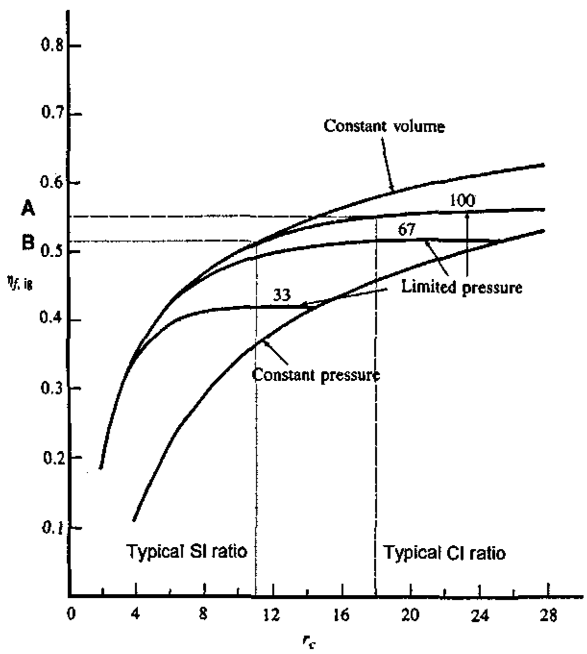



**Figure 5: Fuel conversion efficiency against compression ratio for 3 cycles**
*(Reproduced from Heywood (1988b) and annotated)*

Figure 5 exemplifies a fuel conversion efficiency benefit for the limited pressure or mixed cycle engine over the constant volume SI cycle. The pressure-limited Diesel cycle, operating at a typical compression ratio shown in the figure, $r_c$ = 18:1, has a higher fuel conversion efficiency, at value A, than an SI engine with a typical $r_c$ = 11:1, at value B. These values show how well the fuel is converted to work but the torque output will also be governed by the total mass of fuel-air charge that can be burned to increase the cylinder pressure and hence produce the work output.

The air to fuel ratio for an SI engine at maximum power, 12·5 to 13·5:1, is lower than that of a smoke-limited Diesel engine operating at approximately 18:1 AFR in a limited pressure cycle. Therefore, more fuel can be burned in an SI engine for a given air charge mass. Whereas A might be higher than B, by about 0·04 in Figure 5, the potential torque output deficit for SI of ~8% based on its fuel conversion efficiency relative to Diesel, is offset by the ~33% benefit from the lower maximum power air-fuel ratio, which provides a higher total mass of fuel in the charge.

The second significant difference between SI and compression ignition fuels is related to the combustion properties. The burn rate of the fuel limits the operating speed of the engine, if all the induced fuel is to be converted to useful work in the time available for the power stroke. Gasoline combustion systems exhibit faster burning rates than Diesel combustion systems and hence gasoline engines can operate at higher speeds. The higher operating speed (rate of doing work) contributes to the higher power potential of an SI engine over a similar engine running as a Diesel. This is discussed further and related to the factors affecting the power output of an engine, in the following Section 2.3.

The comparison of the major factors that contribute to power output of SI and Diesel engines is summarised in Table 1.

Key to symbols in this table:

↓ lower ↑ higher

| | SI | Diesel |
|---|---|---|
| Speed capability | ↑ | ↓ |
| Fuel to air ratio | ↑ | ↓ |
| Fuel conversion efficiency | ↓ | ↑ |

**Table 1: Comparison of the performance features of SI and Diesel engines**

It is because of these trends that naturally aspirated engines of the same swept volume, which are designed to produce high power, are most often of the spark ignition type. This, in turn, has led to the predominance of SI for high performance passenger car engines and those for competition in motorsports.

## 2.3 Performance Limitations of SI Engines

As stated in the previous section, despite the ideal gas cycle advantages of the SI or Otto engine, its performance is still limited by two key factors: the amount of charge that can be induced and the efficiency with which that charge can be burned to produce useful work. The brake power ($P_b$) of an engine can be expressed as

$$P_b = \dot{m}_f Q_{LHV} \eta_{f,b} \tag{1}$$

where $\dot{m}_f$ is the mass flow rate of the fuel

$Q_{LHV}$ is the lower heating value of the fuel

$\eta_{f,b}$ is the brake fuel conversion efficiency of the engine

With both mechanical and electronic fuel injection systems, it is possible to put very large amounts of fuel into the engine during operation and, since there is an optimum air fuel ratio to convert the chemical energy into heat energy, one is left with the first fundamental aspect of the SI engine, that it is an air-limited engine.

The relationships between mass flow of fuel, air charge (combining swept volume, air density and volumetric efficiency) and air fuel ratio (AFR) are as follows

$$AFR = \frac{\dot{m}_a}{\dot{m}_f} \tag{2}$$

where $\dot{m}_a$ is the mass air flow rate

and

$$\dot{m}_a = V_s \rho_a \eta_v \frac{N}{2} \tag{3}$$

where $V_s$ is the swept volume of the engine

$\rho_a$ is the density of the air

$\eta_v$ is the volumetric efficiency

$N$ is the rotational speed of the engine.

Therefore, rearranging (2) and substituting in (3)

$$\dot{m}_f = \frac{V_s \rho_a \eta_v N}{2AFR} \tag{4}$$

The relationship between charge density, temperature and pressure from the ideal gas law is

$$\rho_a = \frac{P_i}{RT_i} \tag{5}$$

where $P_i$ is the pressure

$T_i$ is the temperature of the intake air

$R$ is the universal gas constant

and
$$\eta_{f,b} = \eta_{f,i}\eta_{mech} \tag{6}$$

where $\eta_{f,i}$ is the indicated fuel conversion efficiency of the engine

$\eta_{mech}$ is the mechanical efficiency of the engine

Substituting the above into Equation (1) yields the following expression

$$P_b = \frac{V_s}{2}\frac{P_i}{RT_i}N\eta_v\frac{Q_{LHV}}{AFR}\eta_{f,b} \tag{7}$$

Therefore, to increase the power produced at a given engine speed, a larger engine ($V_s$) can be provided or some means of increasing the charge density ($\rho_a$) in the cylinder; the latter being achievable by increasing the pressure or reducing the temperature of the intake charge. Both larger swept volumes and pressure charged methods, with and without charge cooling, have been used in passenger cars and for motorsport but bring with them the penalties of additional weight, size, cost, inefficiencies and complexity. Therefore, a large body of development has concentrated on the naturally aspirated engine and ways of increasing its efficiency without an increase in engine swept volume. These are represented by improvement to volumetric efficiency ($\eta_v$) and fuel conversion efficiency ($\eta_{f,b}$) in Equation (7).

Even in cases where additional charge can be induced into a normally aspirated engine cylinder by improvements in volumetric efficiency, there is still the second fundamental limitation of the SI engine, which is burning the fuel air mixture with good combustion efficiency, hence indicated fuel conversion efficiency $\eta_{f,i}$, in order to convert the chemical energy stored within it during the power stroke. The attributes of the fuel are as follows:

- High heating Value (available energy)
- Good volatility (ability to mix with air, the other "fuel")
- Good anti-knock rating (ability to resist auto ignition)
- High rate of combustion
- No adverse effects (abrasion, corrosion, etc.)

Amongst the most important factors is the high rate of combustion, since the time available for each cycle at high engines speeds is limited. For example, at 6000 rev.min$^{-1}$ the complete downstroke of a cylinder lasts only 5 ms for 180 degrees of crank angle. Combustion lasting this entire time is inefficient from a thermodynamic standpoint since the cylinder volume is expanding and heat is being lost through the combustion chamber and cylinder walls all the time. It was noted earlier that the theoretical Otto cycle has the combustion pressure rise at constant volume. In reality, a burn time of about 1·5 ms is a more realistic target, being about 50 crank degrees at 6000 rev.min$^{-1}$.

In a controlled combustion experiment, a typical automotive air fuel mixture ignited in quiescent conditions in a sphere of 50 mm radius will take at least 20 ms to burn completely, making it impractical for use in an engine [Atzler and Lawes (1998)]. The only way that real engines can function at higher speeds is due to the effects of turbulence and charge motion and the consequentially more rapid propagation of the flame [Witze *et al* (1983) *inter alia*]. This will be described later in the chapter and is discussed in Chapter 4.

Improvements in the two important factors of cylinder filling and effective combustion have proved to be the essential mechanisms for more efficient, normally aspirated, SI engines and hence improved specific power output for the development of modern performance car engines. The brake mean effective pressure (BMEP) is a measure that enables two engines of different size to be compared since it represents the energy released by the combustion and made available to do work on the piston. The production of that pressure from the release of the chemical energy of the fuel is proportional to the amount of fuel charge that is in the cylinder and the efficiency with which the chemical energy is released and converted to work.

## 2.4 Combustion Chamber Design and Development

As early as 1862, an unpublished French patent issued to Alphonse Beau de Rochas, as reported by Heywood (1988c), outlined the conditions under which maximum efficiency in an internal combustion engine could be achieved. These were:

1. the largest possible cylinder volume with the minimum boundary surface;
2. the greatest possible working speed;
3. the greatest possible expansion ratio;
4. the greatest possible pressure at the beginning of expansion.

The first condition limits the heat losses from the chamber and is more commonly described as a 'small surface to volume ratio' for the chamber. The second condition also limits the heat loss and maximises the power output for a given cylinder pressure. The third and fourth conditions relate to the greatest extraction of the maximum amount of work from the highest pressure of combustion gases.

Other combustion chamber fundamentals, summarised well by Stone (1999d), are:

1. the distance travelled by the flame front should be minimum;
2. the exhaust valve(s) and spark plug(s) should be close together;
3. there should be sufficient turbulence;
4. the end gas should be in a cool part of the combustion chamber.

The first of these suggests that the combustion be as fast as possible, supporting high engine speeds and decreasing the possibility of knock arising due to chain reactions and combustion radicals. The second puts the hot exhaust valves as far from the end gas as possible (being close to the initiation of the mixture by the spark plug), thus avoiding end gas knock or auto ignition. The provision of "sufficient turbulence" will be dealt with more fully later regarding its ability to promote rapid combustion. However, there is a need to avoid excessive turbulence that would lead to higher heat transfer from the chamber or excessively rapid and sometimes harsh combustion in some engines. Putting the spark plug close to the exhaust valves would tend to suggest that the end gas will be in the area of the inlet valves, a comparatively cool area when adjacent to the piston crown surface (locally a high surface to volume ratio). However, this is not always possible, as in hemispherical or pent-roof chambers with a central spark plug. This is just one of the compromises that the combustion chamber designer has to make in producing the most efficient engine where it is not possible to adhere strictly to all the historical and wise design guidance that is available.

The development of high performance spark ignition engines is as old as the motor car itself and many engine developers (or tuners as they were called) were regarded as being masters of a 'black art' rather than an engineering science. As early as 1900, Wilhelm Maybach was developing the four-cylinder 17 kW Phoenix racing car engine, which made its debut at the Nice-La Turbie hill climb in March of that year, into a 26 kW, 6·0 litre unit [Voinovich (undated)]. The new version had a horizontally split crankcase in cast aluminium, cast-iron cylinders arranged in pairs and the removable cylinder heads were replaced by a cast-on design. Maybach used "Magnalium", a special aluminium/magnesium (5%) alloy, for the main bearings. The inlet valves, which were opened by intake depression in the original engine, were operated by exposed camshafts on each side of the crankcase in Maybach's version. Camshafts were used for the first time on this engine, driven by an open gear set from the flywheel. Each pair of cylinders was fitted with a spray-nozzle carburettor, another innovation at that time, and the engine had a weight to power ratio of just 4·5 $kg.kW^{-1}$, a new benchmark figure in that era noting that the engine that powered the Wright brothers historic flight in 1903, produced 9·7 kW and weighed 82 kg (8·45 $kg.kW^{-1}$) [Benedetti (2003)].

The most revered of engine developers were those who made little outside change to the engine but did their work on the ports and chambers. Early exponents were inclined to make ports open and smooth in favour of maximum air flow but, as has been discovered since, there is more to engine development than just high flow coefficients. Combustion chamber designs used squish to promote in-cylinder turbulence to speed up burning and there was development of angled and helical intake ports. These were more commonly found in Diesel engines, which rely similarly on port-induced air motion or swirl to promote mixing of the air and fuel. Axial swirl is illustrated in Figure 6.

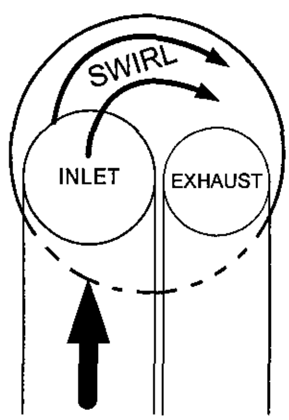



**Figure 6: Diagram of port, valve and chamber-induced axial swirl**

An example of a squish design is given in Figure 7, which illustrates an early engine by Sir Harry Ricardo, in which the motion of the charge under the compression stroke is converted to a turbulent condition by the side chamber and the restricted space, or clearance volume, on the opposite side of the chamber. The resultant charge motion is detailed in the sub-figures (4), (5) and (6) of Figure 7.

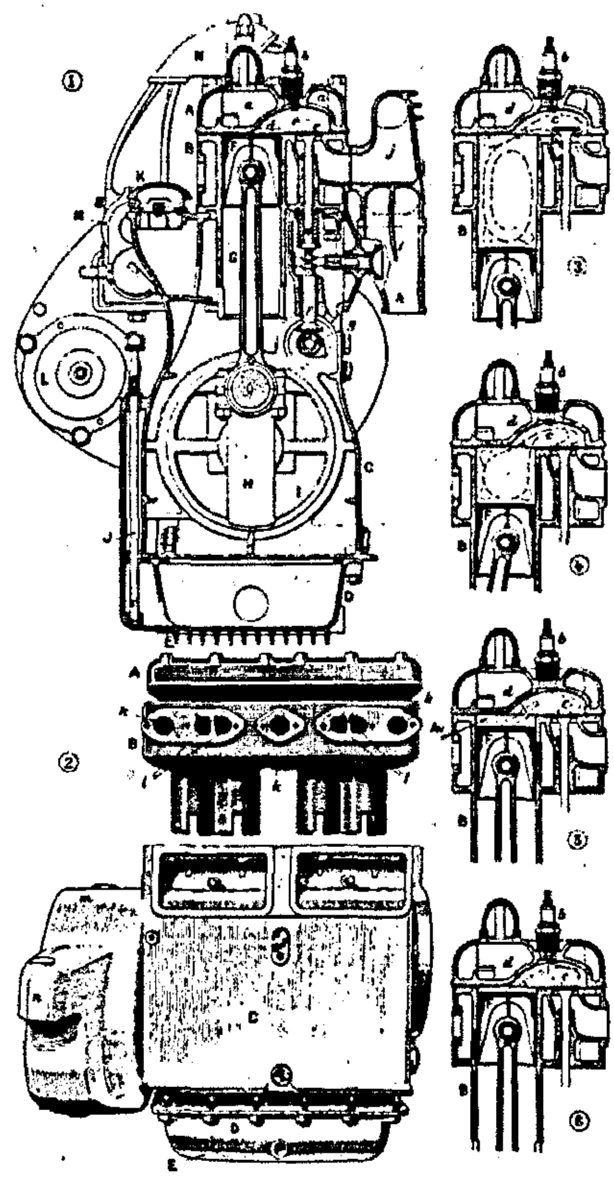


**Figure 7: Turbulent side valve engine design for the early French "Le Zebra" car**
*(Reproduced from Ricardo (1944) as exhibited by Dykes et al (1965))*

The air flow and combustion development of engines has been significantly interactive with the development of valve operation. Early side-valve engines had an elegantly simple valve actuation, as in the example by Ricardo, but suffered from severe compromise in the combustion chamber shape, being of high surface to volume ratio, with consequent high heat losses, and the extremes of poor symmetry and some long flame path lengths.

Development of overhead valve gear improved the basic architecture of the combustion chamber and ports and made multivalve applications more practical, albeit with a loss of rigidity in pushrod operated valves. The further development of overhead camshaft(s) restored a short rigid valve train but added the complexity of providing the drive mechanism from the crankshaft to the cylinder head-mounted cam drive sprocket.

## 2.5 Combustion Chamber Types

Common descriptive, but less-technical, terms for combustion chambers are "open" and "closed". These are generally terms that subjectively refer to the squish-to-bore-area relationship. With the combustion chamber normally being a cavity in the cylinder head casting, with the exception being the bowl-in-piston designs used in such as the Ford "Kent" pushrod engine, for example, and many Diesel engines, the relationship of the open area of the chamber at the head face to the area of the cylinder bore indicates whether a chamber is opened or closed. A closed chamber being one with a high area ratio to the cylinder bore and an open one having a low ratio. Alternatively one can assess the portion of the cylinder head face (fire face) that is exposed to the cylinder bore.

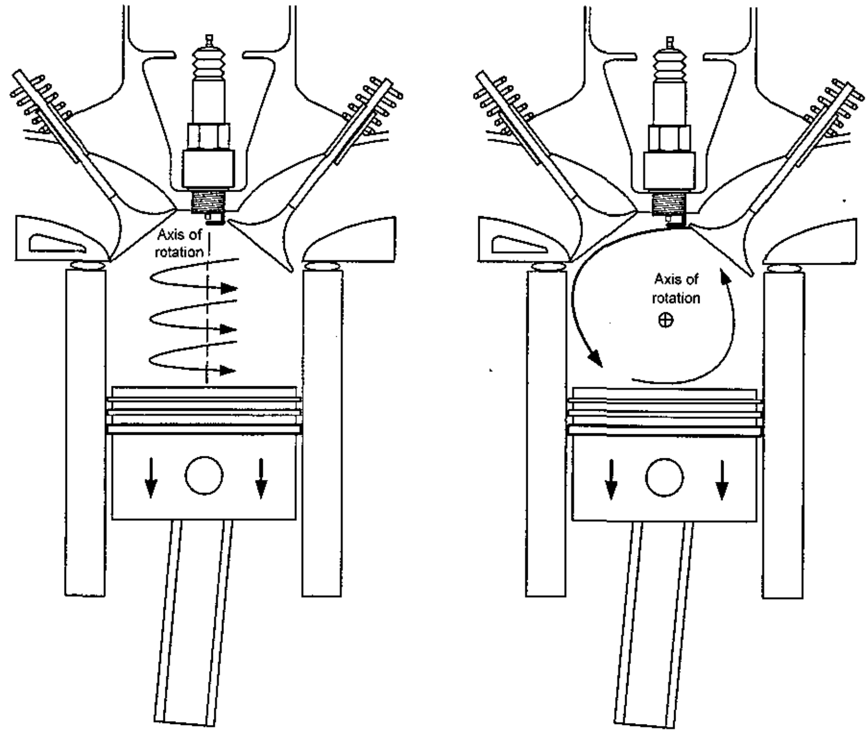
The portion of the fire face that is outside the combustion chamber but exposed to the bore is used as a squish region. Its function is to create internal charge motion that stimulates the end gas to move and increases the burn velocity because the charge moves to escape this area as the piston moves toward TDC, as illustrated earlier in Figure 7. This can be described as internal or chamber-induced charge motion because it is created in the cylinder as opposed to port-induced charge motion, which is created during the induction stroke. The other purpose of the squish area is to provide edges in the clearance volume that can disrupt the large scale air motion during the compression stroke and produce eddies and other micro turbulence that can promote charge mixing prior to ignition and flame propagation. This will be discussed further, in relation to the author's work, in Chapter 4 and Chapter 5.

It has been established that the flame front in an engine propagates across the cylinder at a speed of ~10-35 $m.s^{-1}$ [Rassweiler and Withrow (1938), Cassidy (1977), Bohacz (2000) *inter alia*] and this is substantially faster than its stoichiometric laminar flame speed of about 0·3 to 0·5 $m.s^{-1}$ [Kaminski *et al* (2000), Ju and Lee (2004), Huang *et al* (2004)]. This is the reason why gasoline can be used as an SI engine fuel. To increase burn velocity, turbulence needs to be introduced to the combustion event. In an engine, this is accomplished by the induction and compression process along with the design of the combustion chamber. During pre-mixed combustion, the effect of the turbulence, such as that created by squish or swirl of different kinds, is to break up or wrinkle the flame front, creating burnt gases in the unburned region, and *vice versa*, as well as increasing the total "length" of the flame front. This increase in the flame front area speeds up combustion [Heywood (1988f), Ju and Lee (2004)].

To typify a combustion chamber, all aspects including its shape and that of the piston crown need to be considered. In the context of this thesis, two main types of swirling charge motion in the cylinder are described to partly characterise the combustion system:

1. Axial swirl is defined as air or charge motion about an axis coincident with, or parallel to, the cylinder axis;
2. Tumble or barrel swirl is rotation at right angles to axial swirl, with the bulk motion about an axis perpendicular to the cylinder axis and parallel to the crankshaft centreline.

These two types of motion are illustrated in Figure 8 and will be discussed in Chapter 4 and Chapter 5. They are not mutually exclusive and it was not unusual, in the author's experience of cylinder head benchmarking, to find that the axis of swirl is a combination of axial and tumble motion within the cylinder space.

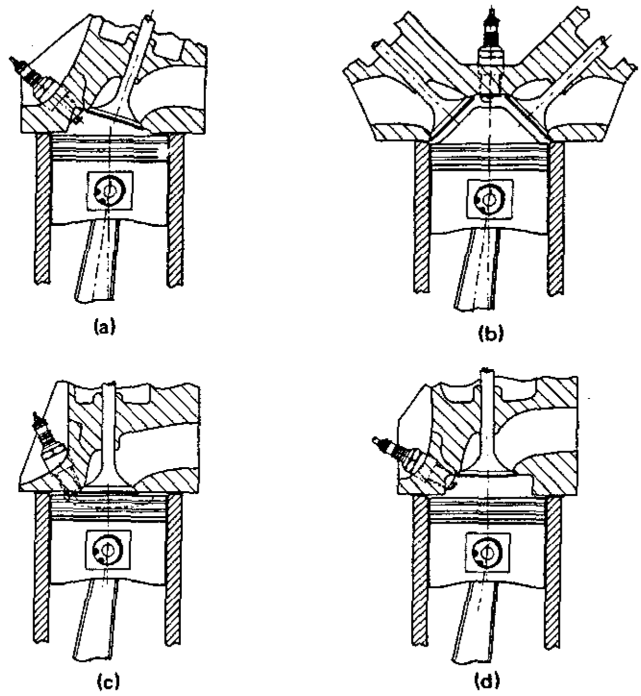



**(a) Axial Swirl** **(b) Tumble or Barrel Swirl**

**Figure 8: Definitions of axial and tumble or barrel swirl**

Regarding the main chamber features, this discussion is limited to the general forms found in most production engines as illustrated in Figure 9(a)-(d), all of which have the good features introduced earlier [Stone (1999d)]:

1. short maximum flame path length;
2. the spark plug close to the exhaust valve;
3. a squish area to help generate turbulence in the charge;
4. well-cooled end-gas.




(a) Wedge chamber

(b) Hemispherical Chamber

(c) Bowl in piston

(d) Bathtub Chamber

**Figure 9: Combustion chamber types**
*(Reproduced from Stone (1999e))*

### 2.5.1 *Wedge - Figure 9a*

The wedge design has been used over the years by almost every major manufacturer as the most logical development of the side-valve design with the valves in the top of the chamber rather than at the side. This chamber resembles an inclined "bathtub" recessed into the deck of the head with in-line valves angled to accommodate the sloping roof of this design. The spark plug is located on the thick side of the wedge and is usually positioned midway between the valves. The inherent steep walls work to mask the air/fuel flow path and deflect it to move in a downward spiral around the cylinder axis. During the compression stroke, the squish volume reduces to such an extent that the trapped mixture is moved from the thin to the thick end of the chamber.

The valve gear for this design is comparatively simple to install and, whilst not as short and rigid as that of the side valve design, the superior combustion meant that engines of this type worked relatively well. It is worth noting that engines of this type had inlet and exhaust systems on the same side of the engine which, whilst providing a useful heat path for cold start and warm up, compromised the design of the manifolds and the absolute performance of the engine due to unwanted heat transfer to the incoming charge.

### 2.5.2 *Hemispherical or pent-roof - Figure 9b*

Given that the idealised combustion chamber would be a sphere with the point of ignition at its centre, providing short and equal flame paths to all parts of the symmetrical volume, a chamber of hemispherical or pent-roof design is considered to offer the least amount of compromise for the efficiency gained. The valves are placed at the edge of the cylinder and are angled to make for as near a hemispherical chamber as possible. This has the further advantage in providing more space for larger valves than for a flat cylinder head although, to exploit this fully, it can be necessary to have twin overhead camshafts. W.O. Bentley used this design with 2-valves per cylinder for the 1948 2·6 litre Lagonda six-cylinder engine, which later went on in 2·9 litre form to the Aston Martin DB2/4 and a racing version in the DB3 [McComb (1981)]. The same basic layout was subsequently adopted by Tadek Marek for the Aston Martin 3·7 litre straight six and then retained, including the 96 mm bore and chain-driven DOHC valve gear, in his 4·8 litre V8 engine [Nixon and Newton (1991)]. Aston Martin had always planned that the DBS of 1967 would be powered by Marek's new V8 engine, which was first seen in 5·0 litre form in the Lola-Aston Martin sports racing car. Production problems, however, intervened and the DBS initially used the twin overhead camshaft straight six of the concurrently produced DB6, which, by then, had been increased in swept volume to 4·0 litre. The DBS V8 was finally announced in September 1969 and sales commenced in April 1968 in the 5·34 litre version of the engine that was later developed for a return to sports car racing [Bale (1984)]. At about the same time, other engines were being designed along similar lines with hemispherical combustion chambers and two large valves. These included the 1948 Jaguar engine by Westlake and the legendary

Chrysler "Hemi", which was the foundation of its engine range in the 1950s, 60s and 70s, and remained substantially similar until 2003 [Brain (undated)].

Other 2-valve engines used variations on the design and one of the more interesting is the Ford CVH (Compound Valve angle Hemispherical combustion chamber) designed in 1974 and in production for 1·1 to 2·0 litre, 4-cylinder engines from 1980 to 1993. The valves of this engine were inclined about an axis perpendicular to the crankshaft axis as well as the conventional disposition relative to the centre line of the head, as illustrated in Figures 10 and 11. This provided the possibility for larger valves and retained a more hemispherical chamber, especially where the valve heads were themselves dished. The arrangement also allowed for air flow gains since it moved the intake valve away from the wall and unshrouded it quickly on opening. This created a more efficient cross-flow movement of the charge during overlap and limited thermal transfer from the exhaust valve to the fresh charge. Relative to the ideal combustion chamber shape, this design offered the best surface-to-volume ratio and, as a cross-flow arrangement, enabled a short exhaust port to be used, which helped to limit heat rejection into the coolant.

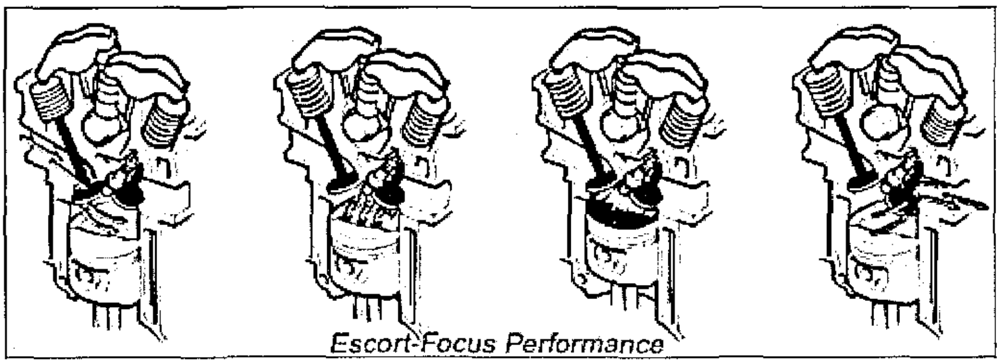



**Figure 10: Ford CVH valve arrangements through the cycle**
*(Reproduced from Escort-Performance.com)*

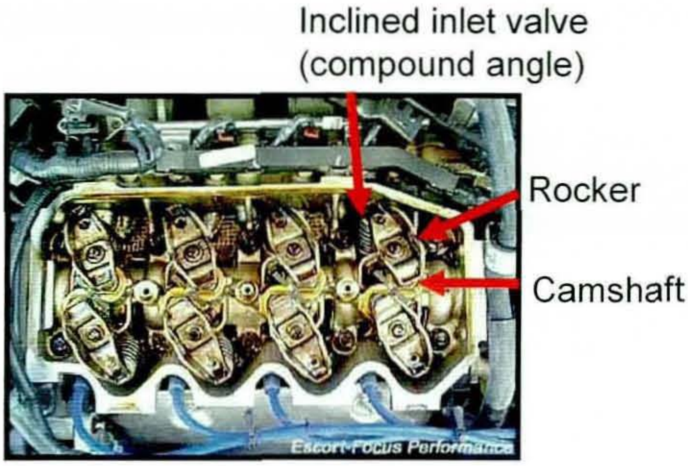



**Figure 11: Ford CVH valve arrangement viewed from above and showing compound angles**
*(Reproduced from Escort-Performance.com and annotated)*

The near-hemispherical pent-roof combustion chamber, with a similar cross section, has become typical for 4-valves per cylinder engines. This design has a near central spark plug with almost equal, short, flame paths across the chamber and hence lower propensity to end gas autoignition. At the perimeter of the cylinder across from the valves, small squish pad areas can be provided to help move the end gas over towards the spark plug and increase burn rates. With pushrod or single overhead cam designs, the valve placement requires dual rocker shafts but lends itself very well to dual overhead camshaft (DOHC) configurations. An additional benefit is the distance between the intake and exhaust valves, which further limits heat transfer. The incoming charge can also generate tumble swirl as will be described in more detail in Chapters 3, 4 and 5. Chapman *et al* (1991a) regarded the 4-valve pent roof design as the outstanding chamber design for high power characteristics and lean burn operation because of high flame speed, enhanced by tumble. This incoming charge effect was utilised even in 2-valves per cylinder hemispherical designs such as the Aston Martin V8 engine with DOHC and large valves.

### 2.5.3 *Bowl in piston - Figure 9c*

This design consists of a flat cylinder head deck with a single row of valves facing a circular cavity cast, or machined, into the piston. An annular squish region is created around the piston perimeter. Known for turbulent combustion, this design is classical for high compression Diesel engines, using a re-entrant bowl for direct injection designs and it was used by the Ford Motor Company for its UK-designed "Kent" pushrod SI engines, but was never widely adopted. The Ford engine was reported to be excessively noisy for American standards suggesting that high combustion rates were possible with this design. It is described as "open" in the sense that the piston crown recess provides an equivalent near-hemispherical combustion chamber shape, albeit inverted from the conventional arrangement and with a non-central spark plug. A further good example of this design was the Jaguar V12 engine which had only marginally inferior performance to a hemispherical head engine with twin overhead camshafts [Mundy (1972)].

### 2.5.4 *Bathtub or heart-shaped - Figure 9d*

The bathtub designation is generally reserved for any chamber that is not a wedge or hemispherical. Many engines of pushrod design have used it in varying forms. In some instances the shape of the combustion chamber was almost oval, with the later trends being the efficient heart shape to allow a higher compression ratio. The deck of the cylinder head that overlaps the piston forms two squish regions: a large area across from the spark plug and a smaller region on the opposite side. Its crescent shape is descriptive of the heart chamber. The valves are inline and are partially masked by the chamber wall, being more exposed on the spark plug side. The area across from the major squish region is generally tapered and does not have the steep wall of a wedge style. Spark plug location is optimised by biasing it towards the exhaust valve and as central as possible in the perpendicular plane. Heat transfer from the close proximity of the valves limits volumetric efficiency and detonation tolerance.

## 2.6 Valve Arrangement and Valve Area

Establishing a cylinder head design that provides high volumetric efficiency for good cylinder filling is one of the key performance parameters described above. The further measure of the fuel conversion efficiency is an indication of how the combustion chamber uses the fuel-bearing air flow and can be just as important as the flow value itself.

Further development of engine designs with overhead camshafts opened the scope for engine designers to optimise port and valve layouts without the need to allow for pushrod access to the valve gear through the cylinder head. As previously mentioned, the design complication of drive to the camshaft itself became the matter to be resolved in this case. However, the fundamental aspect of building the design around the combustion has proved, over time, to have been the dominant trend and at the core of the author's research.

It would be wrong to suggest that this trend was confined to the modern era of engine design. The Peugeot "Coupe d'Auto" engine of 1912 was a 7·6 litre, 4-valves per cylinder, dual overhead camshaft design by Ernest Henry, who had collaborated with Marc Birkigt in the development of an Hispano Suiza for King Alfonso XIII. The 4-cylinder Hispano engine featured twin overhead camshafts, hemispherical combustion chambers and 4-valves per cylinder, but Henry and Zucarelli (the chief tester for Hispano Suiza) sold the design to Peugeot. Birkigt sued and won for the theft, but Peugeot still had the design for its successful 1913 Grand Prix car. It is speculated that the 1912 Peugeot engine was one of the first to use valve overlap [Braden (1996)], recognising that gas dynamics were a limiting factor for engine performance once fuel quality and compression ratio began to increase. Other classical engine designers such as W.O. Bentley, already mentioned for his Lagonda engine, designed and sold alloy block 4-valve engines in the 1930s, albeit with two spark plugs per cylinder in the case of his classic 1930 "Speed Six".

It was the development of Formula One engines in the 1960s, exemplified by the development of the Cosworth DFV V8 Grand Prix engine, which boosted the popularity of 4-valves per cylinder designs in production. The Japanese motor industry was first to mass produce such designs based on its previous successful experience with motorcycle engines. Japanese engine manufacturers soon had a range of four cylinder dual overhead camshaft engines and European companies felt obliged to follow the trend for marketing as well as for technical reasons. Japanese "badge engineering" gave a charisma to the term "16v" that probably outweighed its technical merit, as optimum performance was often made available at higher engine speeds than those used by everyday motorists and usually at the expense of a higher maximum torque engine speed. The on-costs of the additional engine hardware and development had to be borne by manufacturers and a number of novel designs emerged to try and counter this downside of the trend in engine design. An innovative example was the single overhead camshaft of the Triumph Dolomite Sprint engine developed by Jaguar-Rover-Triumph and launched in 1973, as illustrated in Figure 12.

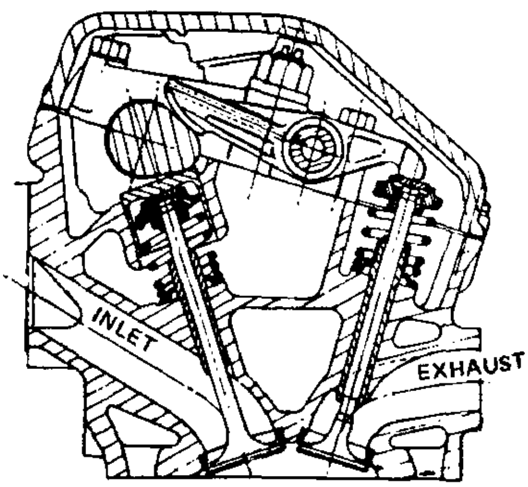
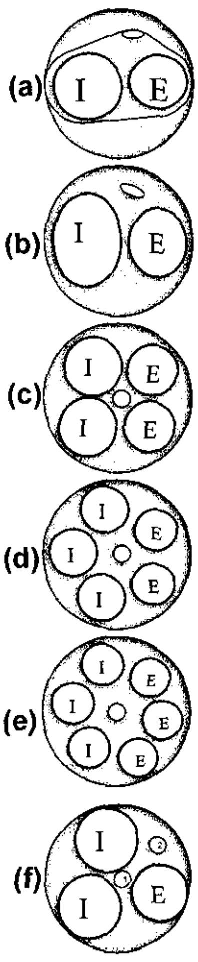



**Figure 12: 1978 Triumph Dolomite Sprint cylinder head section**
*(Reproduced from Stone (1999f) after Campbell (1978))*

With co-operation from Harry Mundy and the engineers at Coventry Climax, a 16-valve cylinder head was designed, which would fit a standard 2·0 litre version of the Triumph Dolomite in-line, 4-cylinder, engine. The 16 valves were controlled by a single camshaft with long rockers across the head to actuate the row of exhaust valves. The arrangement was clever because it negated the need for an expensive twin camshaft arrangement, and would offer all the benefits of the multi-valve layout without a completely new engine. Unfortunately, the design suffered from premature cam lobe wear but has nevertheless become a classic. The general consensus is that, with modern lubricants and design technology, the concept is practicable and at least one of the author's clients is taking a very close interest in adopting the layout.

Even the conservative, and "2-valve" dominated, US market began to adopt 4-valves per cylinder technology in the 1980s to good effect but without very much market penetration for a number of years. The output gains were small but significant and, as recently as 1990, "Popular Science" magazine reported that the mainstream engines in the US fell into three categories: about 37 kW (50 HP) per litre of displacement with pushrods, 45 kW (60 HP) per litre with overhead camshafts and 52 to 56 kW (70 to 75 HP) per litre with 4-valves per cylinder [Anon. (1990)].

Figure 13 shows some different layouts of valves and indicates inlet valve and exhaust valve areas for a common bore size. The data was calculated by the author and Sykes based on typical clearances at valve bridges and between the valves and the edge of the combustion chamber.




| | | | | |
|---|---|---|---|---|
| (a) | 2 Valve<br>Bathtub Combustion Chamber | Inlet 21% | Exhaust 15% | Ratio: Inlet/Exhaust 1·5:1 |
| (b) | 2 Valve<br>Hemispherical Combustion Chamber | Inlet 25% | Exhaust 19% | Ratio: Inlet/Exhaust 1·35:1 |
| (c) | 4 Valve<br>Pent Roof Combustion Chamber | Inlet 28% | Exhaust 21% | Ratio: Inlet/Exhaust 1·33:1 |
| (d) | 5 Valve<br>Pent Roof Combustion Chamber | Inlet 32% | Exhaust 22% | Ratio: Inlet/Exhaust 1·45:1 |
| (e) | 6 Valve<br>Pent Roof Combustion Chamber | Inlet 29% | Exhaust 21% | Ratio: Inlet/Exhaust 1·33:1 |
| (f) | 3 Valve<br>"Pent Roof" Combustion Chamber | Inlet 22% | Exhaust 15% | Ratio: Inlet/Exhaust 1·46:1 |

Key: I Inlet valve E Exhaust Valve

**Figure 13: Valve layouts and areas for a 100 mm bore design with typical valve bridge and spark plug bridge dimensions**

(a) is the vertical valve, bathtub arrangement illustrated in Figure 9(d) where the sizes of inlet and exhaust had to be traded off against each other to fit within the available bore space. Years of development of this type of engine in different places all ended up with inlet to exhaust area ratios of approximately 1·5:1. This was not unexpected given the relative difficulty of inward breathing with a small pressure difference caused by cylinder vacuum (~1 bar when normally aspirated) compared with blow down and displacement of the hot exhaust gases with a large pressure drop across the valve and a high speed of sound, which can accomplish the exhaust outflow efficiently through a smaller orifice.

(b) is representative of the Ford CVH engine, referred to earlier and illustrated in Figures 10 and 11, where the inclined valves of the more hemispherical chamber gave scope for larger intake and exhaust valves.

(c) is the classical pent-roof 4-valves per cylinder design that allowed further increases in area and a typical inlet to exhaust area ratio of 1·33:1. This ratio was mainly constrained by the increasing number and significance of valve bridges between the more numerous valves, and the adequate provision of cylinder head material at the exhaust valve bridge where heat flux densities are at their highest.

(d) is an illustration of a 5-valves per cylinder chamber. Addition of the third intake valve naturally tends to restore the evolved 1·5:1 area ratio, if the valves are all the same size. The increasing number of valves again compromises the valve bridge areas that must be provided and further work will later be demonstrated regarding the effective use of the geometrical areas described in this chapter. Nevertheless, it can be seen that this layout gives the theoretical maximum of inlet valve area at a reasonable inlet to exhaust area ratio, close to the historically derived optimum. This design will be discussed further in Chapter 3 and Chapter 5, where the 1·5:1 valve area ratio is the target value for a new design.

(e) represents a 6-valves per cylinder engine to demonstrate that the valve bridge, chamber wall and spark plug clearance areas eventually become more significant than the valve area increases from raising the number of valves. Effective inlet and exhaust open areas diminish in the example shown. Valve bridge provision is consistent with the other layouts in this analysis.

(f) illustrates the often missed option of 3-valves per cylinder in an almost hemispherical arrangement. Two alternative spark plug positions are illustrated: a central location (1), which gives the best combustion, or a side-mounted spark plug (2), which is much the easier to package. The 3-valves per cylinder arrangement has the advantages of the 4-valves per cylinder intake layout by providing adequate inlet valve area and the symmetrical arrangement that is conducive to tumble swirl. It can also have an inlet to exhaust area ratio close to the historically developed optimum of 1·5:1. Mercedes Benz introduced its successful M113 engine in 1999 with 3-valves per cylinder and uprated performance versions were developed by AMG, with twin spark plugs.

In addition to the valve area available and the valve lift, the flow performance of an inlet valve system is influenced by a number of other factors:

- Valve set width
- Valve seat angle
- Radii of valve seat corners
- Port design
- Combustion chamber shape

Some of the effects of these parameters are well described by Heywood (1988d) and he concluded that, for well-designed ports, the discharge coefficient of the port and valve assembly need be no lower that that of the valve alone unless the design is such that swirl or tumble is being generated.

At high engine speeds, the pressure drop across the valve increases and, if the valve is not of sufficient size, the flow can become choked as it reaches sonic velocity. It is useful to be able to quantify how close the flow system is to the onset of choking by relating the inlet flow velocity to the speed of sound. This parameter is referred to as Mach Index and will be more fully described in Section 2.14 and in Chapter 3, with calculations related to the author's 5-valves per cylinder engine developments.

## 2.7 The Effects of Bore and Stroke Ratio

For a given swept volume per cylinder, if the bore size is increased then there is an increase in the available valve area on an almost directly proportional basis. Thus it will allow the engine to breathe freely at higher speed. As bore increases, the stroke must decrease to keep the same cylinder size. This will decrease the mean piston speed and hence the inlet gas velocity. This is another way of demonstrating the ability of such an engine to run to higher speeds without limiting inlet gas speed. However, increasing the bore can reduce the combustion and thermodynamic efficiency, especially at low speeds, due to the greater surface area of the combustion chamber and piston in relation to the volume in the combustion chamber. The increase in the surface area exposed to combustion allows more heat to escape, thereby reducing the temperature and pressure of the internal gas available to do work.

A typical compromise for a 4-valves per cylinder road engine is to make the bore about 90-100% of the stroke. In the author's work and experience, there were many 2·0 litre displacement engines with bore and stroke of around 86 mm, which would be the dimensions of a "square" cylinder where bore and stroke are equal. For example, the Rover M-Series engine was 84·5 x 88·9 mm and the Ford Zetec was 84·8 x 88·0 mm.

For any given layout it is possible to apply typical valve areas to bore areas and also some typical flow coefficients. Some very useful "rules of thumb" were given by Barnes-Moss (1973) in his paper, "A Designer's Viewpoint" on passenger car engines, which still provides helpful guidelines for engine design students today, e.g. he suggested that a good starting point for a 2-valves per cylinder engine with a wedge or bathtub combustion chamber, is to have an inlet valve diameter of 0·43 to 0·46 times the bore (B) and the exhaust valve diameter of 0·35B to 0·37B.

## 2.8 Exhaust Valve Area and Exhaust Back Pressure

Although engine breathing tends to be dominated by the intake valve constraints in normally aspirated engines, exhaust valve area is also important despite it being easier to blow gases out that to draw them in. On the intake side of a normally aspirated engine there is only a maximum of about 1 bar of atmospheric pressure to force the air in and if there is any sort of restriction, the inlet charge gets less dense so the mass induced drops significantly. The residual cylinder pressure at exhaust valve opening applies significantly more pressure to push the gas out and any restriction results in an increase in the density that makes a small contribution to getting the mass out of the cylinder. However, it is not desirable to require the piston to push hard on the exhaust gas as useful work is absorbed. If there was an extra 0·5 bar pressure during the exhaust stroke, it would equate to losing 0·5 bar of BMEP. Therefore, the efficiency of the exhaust tract and low backpressure in the exhaust system are both desirable. Not only is there this direct effect, but if the exhaust gas should still be under pressure at the end of the exhaust stroke (as it would if the exhaust system were restrictive with a high back pressure), the residual gas in the combustion chamber would re-expand before the fresh charge started to be drawn in, thereby reducing the volumetric efficiency. Typically, mean exhaust valve gas speeds are in the range 90-120 $m.s^{-1}$ at maximum power although higher speeds are sometimes used to keep energy in the exhaust gas for good catalytic converter light-off. Designing for a higher exhaust gas speed results in less heat loss and therefore higher exhaust gas temperature at the catalyst to raise the temperature of the catalytic sites to the "light-off" operating temperature. Catalyst function and light-off are discussed in Sections 2.18 to 2.20.

Another interesting reason why exhaust valves can be smaller than inlet is that a given size of valve and port usually flows better in the exhaust direction than the inlet direction. An additional contribution to high exhaust port efficiency, which allows a smaller cross sectional area to be used without choking, relates to the speed of sound. In a high temperature medium such as exhaust gas, the speed of sound is higher than that of, say, an inlet tract. This will allow a higher gas speed for any given Mach Index and raises the mean gas speed at which choking will occur.

## 2.9 Inertia Effects

The basic volumetric efficiency discussed so far, only concerns the inlet valve and an almost insignificant length of inlet port. In real life it is impossible to separate the valve and port from the system into which they are fitted. However, as soon as any sort of pipe is attached to the valve, some extra effects occur. Although air has a low density, its momentum is significant when it is moved around at high speed.

The air flowing into the engine through the inlet tract has inertia in the form of kinetic energy. It takes some energy to get it moving and when it is running fast down the intake pipe, it has momentum. Thus it is possible that after BDC, even though the piston is stationary or on its way up the cylinder, charge can continue to flow into the combustion space provided that the inlet valve is still open. "Inertia Ram", as it is known, is particularly useful as it gives a genuine improvement in volumetric efficiency over a wide engine speed range. The typical effect is illustrated in Figure 14 where the basic efficiency of around 80% can be increased to 95% or more according to Sykes (1993).

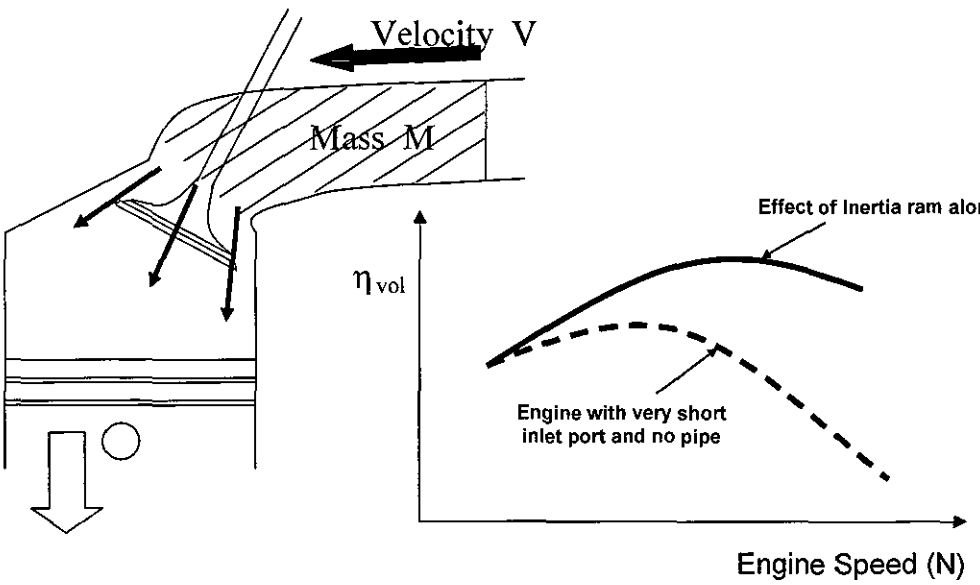



**Figure 14: Diagrams to illustrate inertia ram and its effect on volumetric efficiency**
*(Reproduced from Sykes (1993))*

By varying the diameter and length of the inlet pipe it is possible to make significant changes in the level and characteristics of volumetric efficiency. One of the most important regions is the short length of constant cross section area just upstream of the inlet guide. Here, in what is often called the "throat", the designer would typically aim for a mean gas velocity of around 100 $m.s^{-1}$ in an engine designed for road use.

The inertia of the incoming charge changes with gas speed and hence is dependent upon engine speed. It can only be effective if the inlet valve closing point allows the charge to enter the cylinder. A racing engine may close the inlet valve 80° aBDC and such engines may employ gas speeds as high as 140 $m.s^{-1}$ in the inlet port. It is normal for the throat area to be smaller than that at the valve head diameter and it is good practice to blend the narrow section out to the valve seat smoothly, with a gradual increase in cross section, so that the maximum pressure recovery takes place as the air slows. This helps to push the air into the combustion chamber. Provided that the increase in cross-sectional area is not too sudden, the valve could almost be as large as possible

The effects of inertia ram can be maximised by choosing the correct pipe diameter and length for the engine conditions. With modern simulation tools like Ricardo's "WAVE" and AVL's "BOOST", it is now straightforward to calculate with different configurations and derive the optimum dimensions for any particular engine and speed. In general, the longer the inlet pipe is, the lower will be the speed at which the maximum volumetric efficiency occurs, as illustrated in Figure 15. It can be seen that as the inlet pipe is shortened, not only does the speed rise where the maximum volumetric efficiency occurs, but also the actual amount of "ram benefit" reduces. This characteristic is well established and has been investigated since the 1960s [Lenz *et al* (1990)]. At the time of the author's work, there was no simulation software available to him and development work was all carried out through testing.

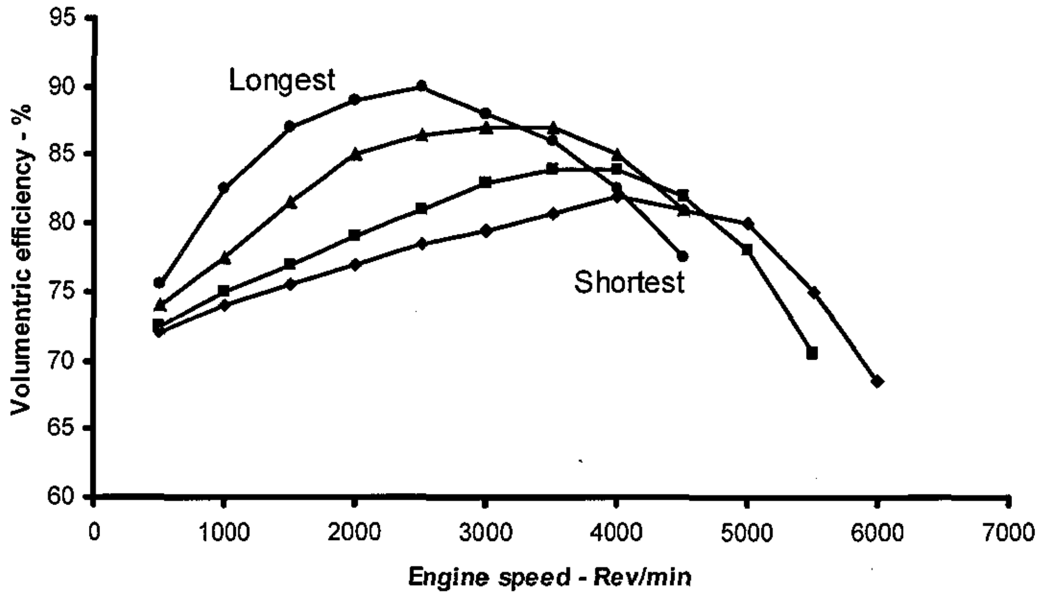



**Figure 15: Example of the effect of intake pipe length**
*(From author's confidential customer data)*

The potential amount of ram is dependent on the mass of air entrained in the pipe and the speed at which it is flowing. Counteracting this are the frictional losses that increase with both gas velocity and the length of the pipe.

A wish to increase the entrained volume by increasing the diameter of the pipe will reduce the velocity and an optimum has to be found. From the author's empirical experience, a mean gas velocity of 70-80 $m.s^{-1}$ is quite often suitable, even better if a tapered tract is designed that gradually accelerates the charge towards the throat of the port. Regardless of the size, it is always best to reduce friction and the recent trend towards plastic intake manifolds with smooth insides, or even fabricated manifolds using drawn aluminium tubes, is beneficial. In the author's experience, bends in inlet pipes, provided they are not extreme, do not have much impact. This was demonstrated by an Austin Rover engine where tests with two inlet manifolds, one with straights tracts and the other with curved tracts of 120 degrees on a centreline radius of 63 mm, gave results that lay within experimental error. This factor is fortunate since compact engine compartments often lead to complex manifold shapes if a reasonable length is to be packaged.

## 2.10 Wave Effects

There is a second effect that occurs in any pipe attached to the valve. This is where the waves of pressure oscillate backwards and forwards in the inlet pipe at the speed of sound. When the inlet valve opens, the displacement of the valve head and the charge immediately behind it, together with the descending piston, reduces the pressure at the inner end of the inlet runner pipe. This not only starts the inward flow to the cylinder but also starts a negative pressure wave that moves along the pipe in the opposite direction to the gas flow at the speed of sound, which may be of the order of ~330-340 $m.s^{-1}$ depending on the gas temperature. When this pulse reaches the open end of the intake pipe it is effectively reflected as a positive pressure pulse (if the tube were closed it would be reflected as a negative pulse). Hence, in an inlet tract that is open at both ends, an alternating positive and negative pressure pulse moves up and down the pipe. These pressure waves may aid or inhibit the gas exchange process according to dimensions and valve timing.

If it can be arranged that a positive pulse arrives at the back of the valve just before the valve closes, the pressure at the inlet valve at the end of the induction stroke is raised above the nominal inlet pressure and additional charge is forced into the cylinder. Pulses arriving at the valve part-way through the induction stroke are no help to the engine breathing. There is, therefore, a relationship between the speed of sound in the inlet, the length of the inlet pipe and the time taken from the start to end of the valve opening period that will determine whether a pulse arrives at the valve closing point. There will be times when this is a positive pulse, times when it is a negative pulse (detrimental effect) and times when it is neutral. The overall result is that as the engine runs up the speed range, there will be some speeds when volumetric efficiency is enhanced and others where the wave effects detract from it. There are a number of differing waves in the tract in addition to the rarefaction wave from the opening valve, e.g. maximum rarefaction wave from the cylinder at about mid-stroke, hence there is a combination effect to be considered.

The time for a pressure wave to propagate from the inlet valve to the end of the tract and back should correspond to approximately 85° crank angle (CA) for optimum timing [Sykes (1993), Chabry (1998), *inter alia*]. This wave reflection time depends on the speed of sound in the charge and is independent of engine speed. The time corresponding to 85° CA is inversely proportional to engine speed. Ideally, therefore, the runner length should change with speed and there are a number of systems that do this to some degree. Novel work by the author on this subject is described in Chapter 5.

Since time, distance and speed are related, the optimum tract length for different speeds can be calculated as an example, according to the following expression

$$t = \frac{2L}{C} \tag{8}$$

where t is the time for the pressure wave to return to the valve

L is the runner length

C is the speed of sound in the medium of the tract = $\sqrt{\gamma RT}$

where $\gamma$ is the ratio of specific heats (1·4 for air at standard conditions)

$R$ is the ideal gas constant

$T$ is the temperature of the medium

The time duration for the engine to rotate 85° CA needs to equate to the time for the pulse wave to pass twice (once in each direction) along the tract from the valve seat. 85 crank degrees equates to 85/360 of a revolution. Hence

$$t_{optimum} = \frac{\left(\frac{85}{360}\right)}{\left(\frac{N}{60}\right)} \tag{9}$$

where $t_{optimum}$ is the optimum time

$N$ is engine speed in rev.min$^{-1}$

Time for two passes of the wave along the length, $t_{optimum} = 14{\cdot}2/N$

Rearranging and substituting in Equation (8) yields an optimum length

$$L_{optimum} = \frac{7{\cdot}1C}{N} \tag{10}$$

where $L_{optimum}$ is the optimum length of the tract at speed $N$

Assuming that C is an average of 330 m.s$^{-1}$ for an air and fuel mixture, the optimum length at 1000 rev.min$^{-1}$ is 2·34 m and 0·39 m at 6000 rev.min$^{-1}$ in this example. This range of lengths is neither practical nor achievable. It is normal to use multiples of the optimum length as there are still beneficial effects at shorter pipe lengths utilising multiple reflections. This is because later reflections, although attenuated by losses in the tract, still arrive at the appropriate point to give some reinforcement to cylinder filling and consequent improvement in volumetric efficiency.

## 2.11 Combined Effects of Inertia and Wave Ram

The net results of wave and inertia ram are illustrated in Figure 16. This is measured data generated by Yagi *et al* (1970) of Honda Research for two values of intake length, L. It shows the general broad shape of the volumetric efficiency curve against engine speed as dotted lines while the wave effects distort this shape to the detailed curve shown. It is clear from this diagram, that unless an engine is operating over a very narrow speed range, the wave ram can quite a nuisance in creating torque dips.

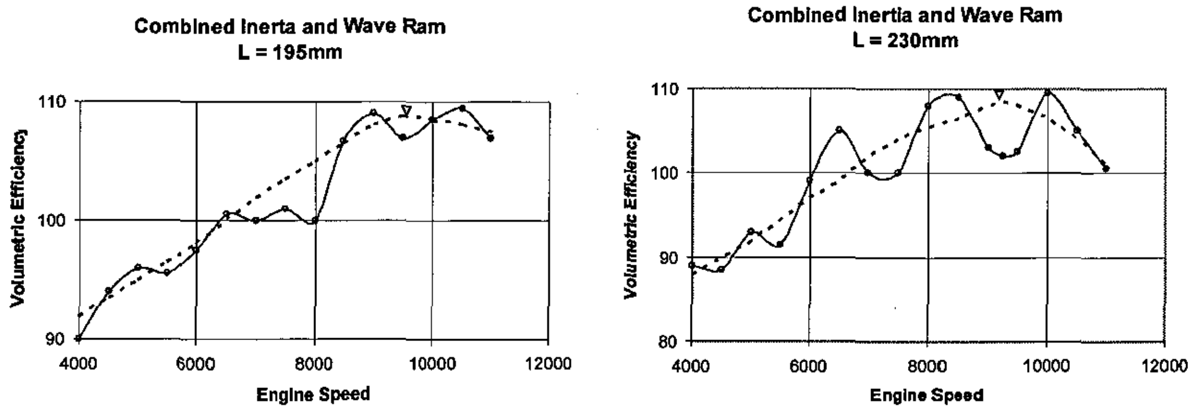



**Figure 16: Detailed volumetric efficiency plots for different intake lengths**
*(Data from Yagi et al (1970) and annotated)*

It is interesting to note that these results show the speed at which maximum inertia ram (point of maximum mean volumetric efficiency) and maximum wave ram (points of maximum resolved volumetric efficiency) occur. Both increase as the intake pipe is shortened, with the maximum volumetric efficiency occurring at 9250 and 9750 rev.min$^{-1}$ respectively, as annotated in Figure 16.

An increasingly common realisation of this effect is to have long ram pipes with a "Y" junction along its length that connects to a shorter pipe with a valve provided to open and close it. It is very hard to make an efficient flow around the junction area and some of the benefits of the change in pipe length can be lost. Further variable techniques have been designed but few are really effective and continuously variable systems have not become widely established in production at the time of writing. However, increasingly capable moulding technology and the quest for more efficiency from smaller swept volumes has improved the potential for more solutions in the future and an example is discussed in Chapter 5.

## 2.12 Acoustic Resonance of the Intake System

Only competition engines can have individual ram pipes open at the outer end because they have individual throttling of each tract, are rebuilt at regular intervals and are not subject to noise legislation. Road going engines must have an air filtration system and a means of distributing the air to the various cylinders. This "Plenum Chamber" added to the intake pipes might reduce maximum power slightly but it also introduces another effect that can be harnessed to improve low speed torque.

Consider a simple spring-mass system and its acoustic equivalent in Figure 17.

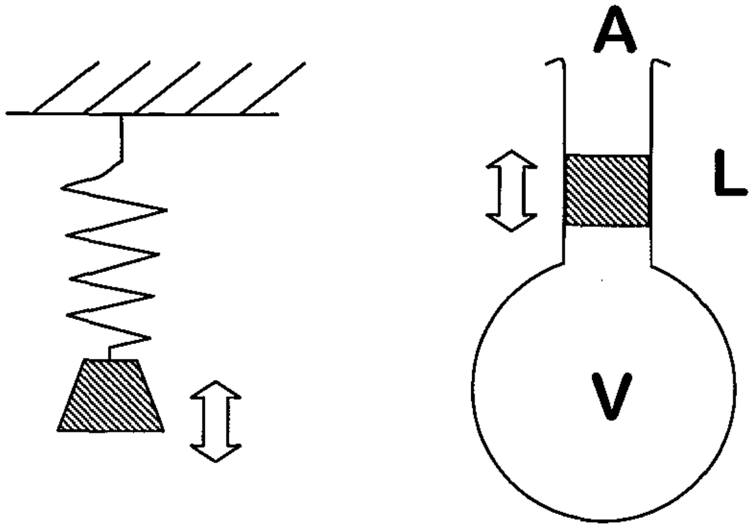



***Figure 17: (a) Metal wire spring*** ***and*** ***(b) Air spring equivalent***

If disturbed from its resting position, the metal wire mass illustrated in Figure 17(a) will oscillate at the natural frequency of the system. In Figure 17(b) the spring has been replaced by a flask of air with a neck into which a mass has been placed in the form of a frictionless piston. If displaced, this too will resonate according to the properties of the system. In fact the piston is not even necessary since the air in the column of the neck has a mass and will itself oscillate on the air spring contained in the flask. This type of device is called a Helmholtz resonator.

Kinsler shows that the natural resonance frequency, $f$ (Hz), for a system is given by the expression [Kinsler *et al* (1982)]

$$f = \frac{C}{2\pi}\sqrt{\frac{A}{LV}} \qquad (11)$$

| | | |
|---|---|---|
| where | $C$ | is the speed of sound in the medium |
| | $A$ | is the cross sectional area of the neck tube |
| | $L$ | is the length of neck tube |
| | $V$ | is the enclosed volume below the neck |

The inlet pipe leading from the air cleaner to the plenum chamber via a throttle body corresponds to the flask neck of Figure 17(b) in which the column of gas oscillates against the spring formed by the total enclosed volume of the whole inlet system. The inlet system includes the plenum chamber, the primary ram pipes and the portion of whichever cylinder happens to have its inlet valve open at the particular time. This is illustrated in Figure 18.

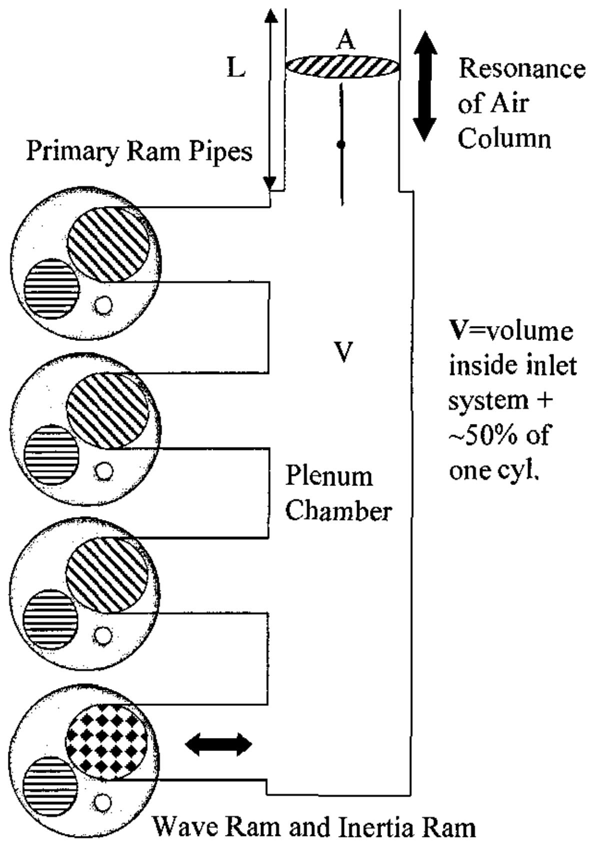



**Figure 18: Inlet manifold system as a resonator**

It is known that this resonant system will be excited by the individual cylinders each undergoing their induction strokes in turn. If there are two, three or four cylinders drawing from one plenum chamber, the amplitude of the pressure disturbance will be high. When more cylinders share the same inlet system, the disturbance frequency is higher, the flow becomes smoother and thus the amplitude is less. Vee engines can use interconnection of two plenum chambers to tune this effect and further shape the engine's volumetric efficiency and hence its torque curve.

If the frequency of the Helmholtz resonance can be arranged to coincide with the frequency of the cylinders on their induction stroke, then the pressure at the intake valve at the end of the induction stroke for each cylinder will be raised and volumetric efficiency will increase. This can be very useful for boosting low speed performance, as it helps to remove the need for extremely long lengths of pipe as per the optimised lengths calculated in Section 2.10. Unfortunately, due to the shape of production intakes and differences between branch geometries, this effect can not be fully optimised for all cylinders. Also, as with any resonant system, there are conditions where the pulsations inhibit the gas exchange and this can produce a dip in the volumetric efficiency curve at a slightly higher engine speed. The effect of adding a resonant inlet tube (or feeder tube or zip tube) and plenum is illustrated in Figure 19.

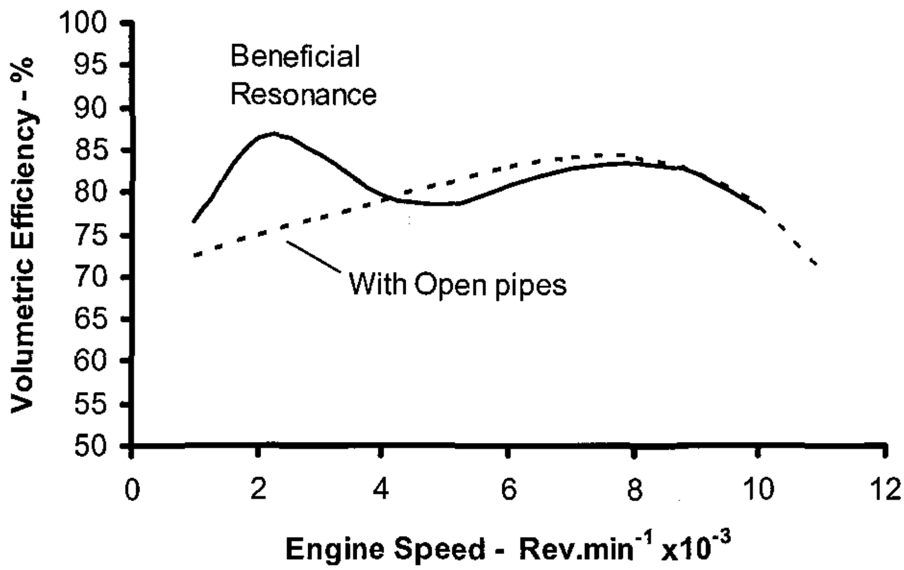



**Figure 19: Typical effect of adding a plenum chamber and feeder tube**
*(Data from Sykes (1993))*

The reason for the adverse mode of vibration is as follows. The primary ram pipe running from the plenum chamber to the valve has its own effective spring-mass system, which is another way of interpreting inertia ram. When the two spring/mass systems are in tune, the resonance is beneficial, when they are out of phase, the beneficial effect in the feeder tube is absorbed in the intake pipe and not translated to a pressure at the back of the inlet valve.

In practice, this dip is not always as bad as it might be because the high volumetric efficiency at very low speeds cannot always be translated into torque because of the need to retard the spark timing to avoid knock. Another way of reducing the dip is to reduce the volume of the plenum chamber and this can be seen in the compact plenum of current engines with diverging inlet runners to the ports, compared with designs of the recent past that used a larger volume "log" manifold that ran along the whole length of the cylinder head.

Other ways of avoiding the dip involve changing the geometry of the system. As mentioned earlier, it is not uncommon on six-cylinder engines to have two plenum chambers each feeding three cylinders. When desired, a connection can be opened between the two chambers making one volume, which immediately halves the engine speed that matches the resonant frequency. This technique can be used to avert a dip. In order to raise the speed of the Helmholtz resonance, as demonstrated in Equation (11), it is necessary to increase the cross sectional area of the feeder tube (A) or reduce the length of the feeder pipe (L) and/or the enclosed volume (V). This is interesting because to optimise the primary ram in the individual runners they also need to be shortened as the speed rises. An inverse linear relationship between primary runner length and engine speed matches the wave ram relationship and this also brings a reduction in the entrained volume. However, the Helmholtz resonant frequency varies inversely with the square root of length and volume change. It was this logic that led to one novel element of the Tickford patented system of variable induction geometry that will be described in Chapter 5, in order to create a greater change of system volume that that which occurred simply with tract length changes.

## 2.13 Exhaust Effects

The exhaust system also has gas moving in pipes and so similar effects can occur. The waves that oscillate back and forth at the speed of sound in the primary pipes will move faster than those in the inlet due to increased pressure and higher gas temperature. If a negative pulse can be arranged at the exhaust valve just before it closes, then additional gas is extracted from the cylinder. However, such effects are rarely optimised as they are only of value when the engine operates over a very narrow speed range such as a racing engine.

The avoidance of exhaust backpressure has been mentioned earlier. The optimisation of the muffler system is often a matter of debate between performance engineering and noise reduction. Increasing the energy absorption in the muffler systems quietens the engine but also reduces its ability to breath. Also, the very pulses that may be used to enhance breathing are undesirable in the noise, vibration and harshness (NVH) reduction of the engine. This general comment also applies to inlet systems where beneficial resonance may lead to increased induction noise.

In modern engines with low exhaust backpressure, the exhaust geometry and configuration are more often driven by packaging and catalytic converter light-off optimisation, than for optimised gas exchange.

## 2.14 Some Other Engine Breathing Effects

There are a number of other factors that affect volumetric efficiency and hence mass flow through the engine, including backflow, charge heating, the presence of fuel and choking of the flow:

- Backflow occurs due to closing of the inlet valve too late (often at low speed when valve overlap is optimised for mid to high speed) and incoming charge is ejected from the cylinder back into the intake tract by the rising piston before the inlet valve closes.

- Charge heating is caused by heat transfer between the fresh charge and the cylinder head and inlet manifold. This reduces the charge density and occurs primarily at low speed because of the longer residence time. The effect was reduced with the adoption of "cross-flow" cylinder head designs. However, it can be more significant in modern SI engines with direct fuel injection into the cylinder, since there is no opportunity to use the latent heat of evaporation of the gasoline to reduce charge temperature in the intake system.

- Fuel presence in the inlet tract occupies space that would otherwise have been available for air. The inefficiency caused by gasoline is about 2% and with gaseous fuels, such as petroleum gas or compressed natural gas, it can be as high as 10% [Bennett *et al* (1995)]. This factor is more significant when running an SI engine on gaseous fuel compared with gasoline; not only is the air mass reduced but there is no charge temperature reduction from vaporisation for an already-gaseous fuel.

- The velocity of the charge as it passes through the valve seat area can become limited. At high engine speeds, the inlet flow during induction can become restricted if the inlet valve area is not sufficient to prevent the flow from becoming choked, i.e. the flow reaches sonic velocity at the minimum valve flow area. Livengood (1952) and Taylor (1977) showed that the volumetric efficiency falls off significantly when the flow velocity is more than 0·5 of the speed of sound in the charge medium. This aspect will be covered in more detail relative to the author's work, in Chapter 3.

## 2.15 Exhaust Emission Control

During the last thirty years of the author's engine work, significant changes have taken place in the automotive industry and in the environment. The industry has invested heavily to meet ever increasing demands from the customers and legislators for environmental and safety improvements. The drivers for change have been:

- Improved specific power output
- Improved fuel economy
- Reduced emissions
- Reduced NVH
- Reduced cost

Of these, the emissions driver is arguably the most significant as it is a result of tightened international legislation and has implications on the application of technology to the engine and exhaust system. Improvements in the other parameters are driven by customer requirements and competitive pressures, but non-compliance with legislation will prevent a vehicle from being sold in the first place, regardless of its performance and fuel economy.

The recent development of the automotive gasoline engine has been marked with a number of clear trends. Some historically standard features have been eliminated and a number of new ones have emerged, driven by emissions legislation, market forces and the quest for fuel efficiency. Disappearing and reducing technologies and hardware have included:

- Side valve engines
- Cam-in-block
- Two stroke engines
- 2- valves per cylinder engines
- Non cross-flow engines
- Entirely cast iron engine structures
- Carburettors
- Distributors with contact breakers, springs and bob-weights
- Distributors with electronic triggering
- Multi-electrode spark plugs
- Open crankcase ventilation

Emerging and increasing technologies and hardware have included:

- Overhead camshafts
- Twin overhead camshafts
- Belt driven camshafts
- Cross-flow cylinder heads
- Aluminium cylinder heads
- Aluminium blocks
- Piston crown and bowl shapes

- Crevice volume reduction
- Three piece oil control rings
- Multivalve engines (3, 4 and 5 valves per cylinder)
- Hydraulic valve lifters
- Variable valve timing
- Port/chamber-induced air motion
- Electronic injection
- Direct injection of gasoline into the cylinder
- Lean-burn combustion systems
- Electronic ignition
- Distributorless ignition
- Controlled autoignition
- Fine centre electrode spark plugs incorporating copper and precious metals
- Closed-loop fuelling control with sensor(s) in the exhaust
- Catalytic converters
- Tubular exhaust manifolds
- Secondary air injection or "pulse air" during engine warm-up
- Exhaust gas recirculation (EGR)
- Closed crankcase ventilation systems
- Plastic intake systems

The above trends are not specific nor are they universal. For example, according to Ward's Auto World in 1996, 2-valves per cylinder was a feature of 72% of its world survey of automotive engines despite the high profile taken by 4-valves per cylinder engines in the popular car market in Japan and, later, in Europe [McCann (1996)]. The 2-valves per cylinder engine is very dominant in the cost and durability conscious US market where specific power output and fuel economy are not such high priorities.

Other factors that are being monitored and improved in modern automotive engineering are assembly energy and pollution, and end-of-life utilisation of the spent vehicle and all its component parts, which are outside the scope of this thesis.

## 2.16 Exhaust Emission Control Strategies

The vehicle impact on the environment takes many forms, of which exhaust gases can be amongst the most obvious and consequently attract most attention.

Several technologies are involved in dealing with these exhaust emissions, most of which would be harmless if perfect combustion could be achieved and if secondary reactions caused by high temperature and pressure in the cylinder, which produces $NO_x$, did not occur. The most effective method of reducing total exhaust emissions from an internal combustion engine is to burn less fuel, reducing the mass of exhaust produced. This means that whatever methods are taken to reduce the toxicity and other harmful effects of exhaust gases, fuel efficiency remains a key overall objective. This underlines the quest for increased specific power output so that smaller swept volume engines can be used to propel vehicles in a way that produces customer satisfaction and with intrinsically low fuel consumption. This strategy has the other benefit of reducing the running cost of the engine, giving additional benefit to the user or operator.

It follows, therefore, that the most effective combination of control methods revolves round the following:

1. Combustion control;
2. Post combustion control;
3. Clean fuels;
4. Fuel economy.

It would also be appropriate to point out that the benefits of all the complex and expensive technologies used in emissions control can be negated by poor maintenance. The specification of the control system to take account of this should be a further facet of the engineering process, so that maintenance procedures are properly created and any deficiencies therein, have the least impact on performance.

Current engine management systems require the minimum of maintenance and have benefited from control developments that can adapt the fuelling and ignition calibrations to an engine's current state of adjustment and wear. Further enhancements relate to the engine management system's ability to adopt a "limp-home" mode when sensors fail, and indications of malfunctions of the engine and aftertreatment systems are presented to the operator and made available for analysis by the service network.

## 2.17 Meeting Exhaust Emissions Regulations

The exhaust emissions regulations that govern the performance of engines all over the world are, for the most part, not prescriptive of the means by which the regulated levels are achieved. Often the levels can point the way towards certain technologies but the regulations do not usually specify them. Thus it is up to each manufacturer to optimise the exhaust emissions from their vehicle. The emitted products are a result of the processes of combustion (engine-out or "feed gas" emissions) plus the effects of any post-cylinder conversions, be they inherent or assisted by aftertreatment devices.

### 2.17.1 *Combustion Control*

The basic elements of combustion control, already introduced, consist of:

1. Optimising air fuel mixtures and providing sufficient oxygen to meet and mix with the fuel admitted to produce the required output;

2. Optimising the timing and rate of combustion events affecting the quality of combustion and the energy developed by it.

These fundamental elements of cylinder filling and charge combustion will be discussed further in Chapters 3, 4 and 5.

### 2.17.2 *Exhaust Aftertreatment*

Exhaust aftertreatment is essentially a cleaning up process and consists of reducing the harmful products in the exhaust after they have left the combustion chamber. Unlike combustion control that can improve thermal efficiency, aftertreatments usually detract from the efficiency of the system. This can be due to increases in exhaust system backpressure and the arrangement of piping to suit the treatment systems rather than optimising the engine breathing, in the manner described earlier in Section 2.13. Aftertreatment systems also add cost and complication. In the 1970s, shortly after the US Clean Air Act, engine-driven pump(s) or pulse energy were used to introduce air into the exhaust ports or manifold to promote additional oxidation. This was sometimes enhanced by the use of thermally treated manifolds and downpipes or by forming the downpipe as a thermal reaction chamber. However, the use of catalytic conversion of the gases has become dominant in the aftertreatment of exhaust emissions by promoting oxidation, or oxidation and reduction, of the exhaust mixtures to reduce harmful pollutants. This will be described further in the following sections.

## 2.18 Catalytic Converters

In order to explain the impacts of the utilisation of catalytic converters described in Chapter 6, it is important to understand the chemistry involved.

If complete combustion were to occur, the hydrogen and carbon constituents of the fuel would combine with the oxygen in the air to form carbon dioxide ($CO_2$) and water vapour. These would be the only products to be released into the atmosphere together with the nitrogen and other inert gases in air, according to the reaction

$$C_nH_m + \left(n + \frac{m}{4}\right)O_2 + N_2 \rightarrow \frac{m}{2}H_2O + nCO_2 + N_2 \qquad (12)$$

where $C_nH_m$ is the hydrocarbon fuel

$O_2$ is the oxygen component in air

$N_2$ is the nitrogen and other inert gases in air

$H_2O$ is water (vapour)

$CO_2$ is carbon dioxide

In reality, complete combustion does not occur under engine operating conditions and the temperatures and pressure generated during the process give rise to secondary reactions. Some of the carbon atoms are only partially oxidised, forming carbon monoxide, and amounts of both fuel (as hydrocarbon, with traces from lubricant) and oxygen remain unburned. The most significant secondary reaction is the formation of oxides of nitrogen which occurs under the high temperature and pressures in the combustion chamber. Furthermore, because the fuel and lubricant have trace elements in them such as lead (Pb) and sulphur (S), further pollutants are released from combustion that have an adverse effect on the environment. This more accurate set of products is described in the unbalanced expression

$$C_nH_m + O_2 + N_2 \rightarrow HC + CO + H_2O + CO_2 + N_2 + NO_X + PbO_Y + SO_Z + O_2 \tag{13}$$

where $HC$ is unburned hydrocarbon fuel

$CO$ is carbon monoxide

$Pb$ is lead (from fuel)

$S$ is sulphur (from fuel and lubricant)

$x$, $y$ and $z$ are collective terms for oxide states that might form

It can be seen in the products side of Equation 13 that redistributing the oxygen components and using up the expelled oxygen would improve the toxicity of the exhaust products by fully oxidising carbon monoxide to dioxide and converting (oxidising) the unburned hydrocarbon fuel. However, despite the presence of all the reactants and a hot turbulent environment, conversion does not naturally occur to a sufficient degree to meet modern legislation, even with a thermal reactor and additional air as previously described.

A catalyst is, by definition, a substance that facilitates a chemical reaction without taking part in the reaction itself. In theory, the catalyst remains intact after the promoting reaction and is not consumed. Early catalysts were placed in the exhaust stream to promote oxidation of the unburned HC and CO to produce $H_2O$ and $CO_2$. Ignoring the presence of $N_2$, $NO_X$ and impurities for the time being, the compound, unbalanced, Equation (14) can be used to describe the transition from combustion chamber to tailpipe when an oxidation catalyst is fitted:

$$C_nH_m + O_2 \xrightarrow{Combustion} HC + CO + H_2O + CO_2 + O_2 \xrightarrow{Catalysis} H_2O + CO_2 \qquad (14)$$

This process used the free oxygen in the exhaust stream and the emission control technique was based on the use of lean mixtures with excess oxygen or the use of pumped or pulsed air into the exhaust stream. When the oxides of nitrogen ($NO_X$) were restricted by legislation, the uses of lean mixtures and oxidation catalysts were not capable of meeting the standards. In fact, in the author's experience, the use of pumped or pulse air and an oxidation catalyst, exacerbated the production of $NO_X$. Extra oxygen induced into the exhaust was found to get upstream into the combustion process and further lean out the charge.

The elimination of $NO_X$ in the exhaust requires a chemical "reduction" process, the opposite of oxidation, i.e. removal of attached oxygen atoms. If all the legislated pollutants, HC, CO and $NO_X$, are to be removed from the engine exhaust stream, both oxidation and reduction reactions are required to occur simultaneously. The solution to this problem was the development of the "three way catalyst" (TWC) to convert HC, CO and $NO_X$ at the same time [Twigg (2005)]. Such a combination makes use of the oxygen component of the $NO_X$ from a chemically correct balance of fuel and oxygen in the combustion process (stoichiometric mixture) and uses the combination of combustion and catalysis to produce a favourable balance at the tailpipe. In other stream to convert HC and CO to $H_2O$ and $CO_2$, supplemented by the oxygen components from the $NO_x$. Removal of that oxygen component from the $NO_x$ leaves inert nitrogen ($N_2$) in the tailpipe and the conversion described in Equation (14) for oxidation catalysis, then becomes

$$C_nH_m + O_2 + N_2 \xrightarrow{Combustion} HC + CO + H_2O + CO_2 + O_2 + NO_X \xrightarrow{Catalysis} H_2O + CO_2 + N_2 \quad (15)$$

In order to achieve this level of control, it is necessary to maintain the air to fuel mixture close to the stoichiometric value. This is a factor that alters with the fuel composition but is typically in the range of 14·4 - 14·8:1 AFR by mass, for most gasoline engines. For this reason, a sensor is used in the exhaust stream to provide a signal to the fuelling control indicating rich or lean mixture so that the fuelling is corrected at 1 Hz intervals and maintained close to the stoichiometric value. In fact, use is made of the feedback signal to modulate the instantaneous mixture slightly rich and lean of the nominal value so as to provide the catalyst with an improved, alternating, oxidation and reduction environment.

The catalyst itself is commonly based on a ceramic monolith in the form of a honeycomb of narrow passages with at least four hundred cells per square inch of catalyst face (62 cells per $cm^2$). Some converters use a metallic substrate formed from a spiral of thin, shaped sheet and both are illustrated in Figure 20. There is a cost penalty for steel substrates but high cell densities, compact assemblies and increased temperature tolerance can be achieved.

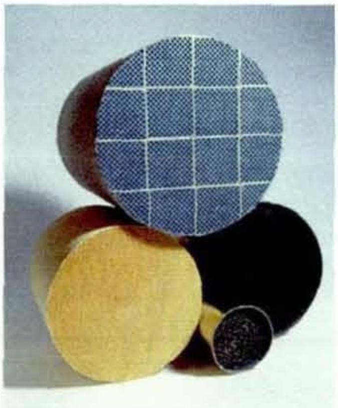


**Figure 20: Ceramic and metallic catalytic converter substrates**
*(Photograph reproduced with permission from Johnson Matthey)*

The monolith is covered, first with a "wash coat", being a sponge-like substance that increases the effective surface area, followed by the deposit of a proprietary mixture of precious metal salts based on Platinum, Palladium and, in the case of a three way catalysts, Rhodium [Twigg and Wilkins (1999)]. This is illustrated in Figure 21, in which the substrate walls show as mid grey and the highly porous washcoat with the catalytic coating, appears as light grey or white. The fillets caused by the washcoat in the corners of the ceramic monolith cells can clearly be seen in this image.

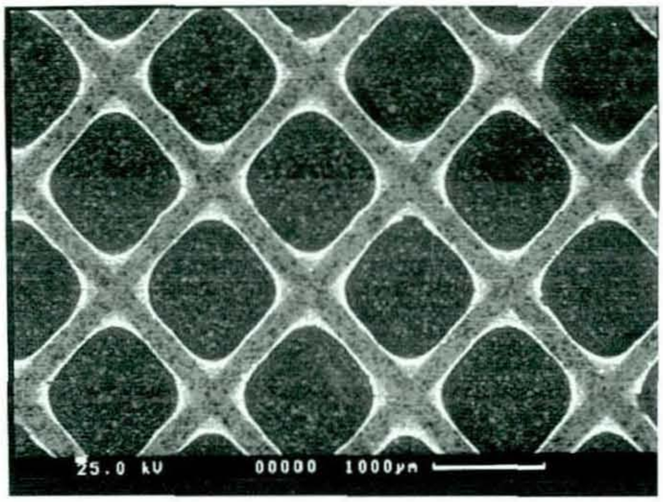



**Figure 21: Coated catalyst substrate**
*(Scanning Electron Microscope image supplied by Johnson Matthey ECT)*

The fully treated substrate is mounted and clamped into a stainless steel enclosure, not unlike a silencer, and is illustrated in Figure 22. When the substrate is of a ceramic construction, it is wrapped in a thermally insulating and shock absorbing cover. This can take the form of a knitted metallic mesh, as shown in Figure 22, or a ceramic fibrous matting to provide shock absorption and some thermal insulation, in order to reduce the outside skin temperature. Additional surface temperature management is sometimes achieved by a heat shield attached to the outer can with an air gap, or with insulation as illustrated in the figure.

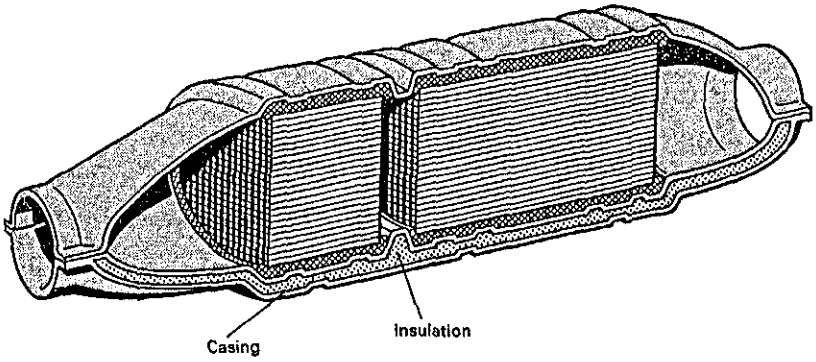



**Figure 22: Catalytic converter assembly**
*(Reproduced from Stone (1999g) after Gunther (1988))*

In theory the catalyst is unaffected by the process that it is promoting and transient compounds that are formed as by-products during the process, are reversed so that the original catalyst remains as it was at the start of the reaction. There are, however, factors that cause deterioration of the active catalytic sites. Of these, the most significant is the presence of lead in the exhaust stream. Lead causes the irreversible formation of compounds with the catalytic components and some of the reaction sites are disabled, reducing the effective working area and capacity of the converter. This is the reason why lead-free fuel had to be introduced for use in catalyst equipped vehicles. There are then other implications on the internal engine design and spark timing in order to run without the lubricity of the lead compounds and without the octane enhancement that originated their use. Other detrimental effects to the catalyst are caused by physical plugging of the substrate open area by carbon or lead compounds, erosion of the front surface by solids in the gas stream and the effects of excessive temperature which cause migration and coalescing of the catalytic sites into less effective chemical structures [Lassi (2003)].

The catalytic surface needs to be at a temperature of some 200°C in order to start converting effectively. This temperature is known as "light-off" and is achieved soon after engine start by virtue of the heat in the exhaust gas [Twigg *et al* (2002)]. The reaction at the catalyst surface is itself exothermic, which adds to the working temperature by approximately 40-70°C per 1% of CO in the gas coming into the catalyst. Conversion efficiency rises with temperature up to an optimum of 600-800°C so improved performance can be obtained by speeding up the heating of the catalyst to and above the "light-off" temperature. However, simply moving the catalyst assembly forward in the exhaust system, e.g. to the end of the exhaust manifold, would not be acceptable for prolonged high load operation with high gas temperatures that could damage the catalyst. There are, therefore, conflicting design requirements on the installation and operating temperature conditions for a catalyst to keep it hot enough to function but not too hot to reduce durability. Automotive catalysts have to maintain the legislated performance of the engine/vehicle combination for 160 000 km. The catalytic sites are adversely and irreversibly affected by temperatures in excess of 1050°C and the ceramic monolith itself can melt at temperatures above 1,350°C [Twigg *et al* (2002)]. For this reason, other strategies have to be developed for optimum vehicle emission performance including fuelling, spark timing, valve timing, catalyst location and thermal management.

In order to avoid excessive temperature at the catalytic sites, calibrations can be set to invoke fuelling enrichment in order to reduce the exhaust gas temperatures and protect the catalyst from damage. This has consequences that are discussed further in the following section and in Chapter 6. There are other side effects of the catalyst that are beyond the scope of this thesis. Such effects include the formation of compounds from sulphur impurities in the fuel and lubricant, which inhibit the function of the catalytic sites, and the release of complex hydrocarbon compounds [Rogge *et al* (1993)], some in the form of particulates that are invisible to the naked eye [Miguel *et al* (1997)].

## 2.19 Engine Fuelling for Catalytic Converters

As described above, the control of air-fuel ratio is key to optimised catalyst operation. In the case of oxidation catalysts (where no $NO_x$ regulation or emission problem were encountered), a lean calibration was required and this could be achieved with or without additional air sources.

With the need for three way conversion, more precise and sophisticated fuelling control was required. It was sometimes possible to achieve legislated levels (e.g.

where very low $NO_x$ conversion rates were needed) with carburettors or uncontrolled (open loop) fuel injection, where the fuelling was stoichiometric at enough speed and load points for sufficient three-way conversion of pollutants to take place. However, the most competent systems use fuel injection with electronic control and feedback (closed loop) information to the engine calibration. An exhaust gas oxygen (EGO) or "Lambda" ($\lambda$) sensor is needed for such applications, mounted upstream of the catalyst in order to feed a signal to the engine control unit related to the AFR conditions in the exhaust stream entering the catalyst.

The need for this level of control can be seen readily from Figure 23, which illustrates the trends of three-way catalyst conversion efficiencies against changing the air-fuel ratio of the engine exhaust in the converter.

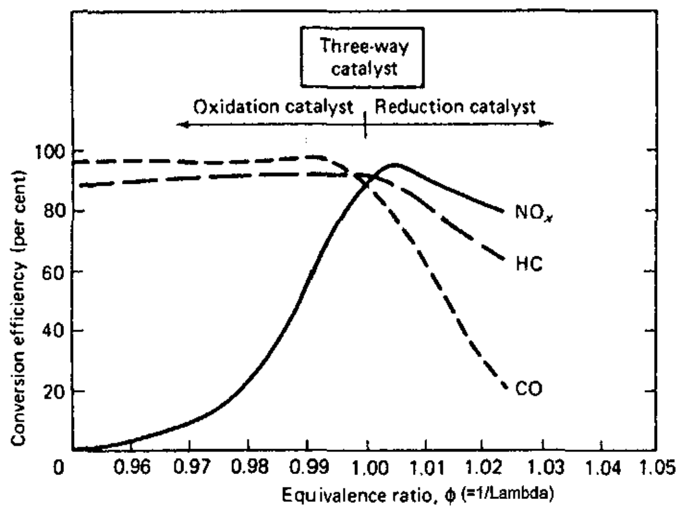



**Figure 23: Conversion efficiencies for CO, $NO_x$ and HC of a TWC as a function of air-fuel ratio**
*(Reproduced from Stone (1999h) with acknowledgement to Johnson Matthey)*

Keeping the fuelling close to the stoichiometric mixture (equivalence ratio $\phi$ or $\lambda = 1$) means that the conversion efficiencies of all the pollutants are at or close to their highest, although this is fuelling the engine considerably richer than it needs to be for best fuel economy (A) and leaner that that required for maximum BMEP (B) as illustrated in Figure 24:

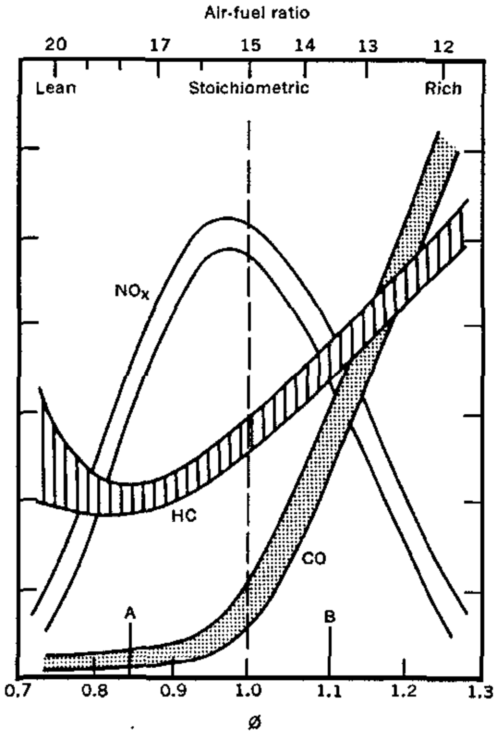



**Figure 24: Effect of air-fuel ratio on exhaust emissions**
*(Reproduced from Taylor (1985b) and annotated)*
**(Note: $NO_x$, CO and HC concentrations are not to scale)**

*This forcing of fuelling to stoichiometric ($\lambda$=1 or $\phi$=1) provides for the best tailpipe* emissions levels but in calibrating the fuelling to suit the catalyst rather than the engine, the fuel economy and $CO_2$ emissions, are penalised. Engine fuelling for maximum BMEP, at B in Figure 24, is richer than the stoichiometric value that is normally controlled by the EGO sensor. As demonstrated in Figure 23, fuelling of this level at wide open throttle in order to deliver the maximum engine power output, will cause the catalyst efficiency to be less than optimum. Under these conditions there will be increases in the levels of the three legislated pollutants and hence increases in tailpipe emissions to the atmosphere. This is particularly undesirable as both the concentrations of pollutants and the mass flow of exhaust are high at the same time. Such high load conditions are not fully included in legislative testing [Guensler (1994)] and have been shown to be important for real global air quality [Lenz and Cozzarini (1998)]. This is the subject of further discussion in Chapter 6.

Reviewing the coincidence of low levels of all three pollutants in Figure 24 around engine fuelling A, demonstrates that the "lean-burn" solution could be a sound one.

This could only be practicable if the driveability of the engine and vehicle is preserved and the misfire limit, which results in the HC curve rising up, can be moved as lean as possible to enable the low $NO_x$ regime to be used. Catalysts with the ability to convert $NO_x$ at lean ratios have been under development for some time but remain limited in operating temperature and durability [Bhattacharyya and Das (2001), Twigg (2003)]. Chemical improvements and more precise control of fuelling have meant that manufacturers are able to put less precious metal onto their converters and still achieve the same end results. Hence a realistic engineering solution for best economy, which has not become established commercially, is the combination of:

a) a lean-burn combustion system, with fuelling calibrated to be lean under as much of the driving operation as possible, coupled with a three-way catalyst to clean up the residual CO and HC under cruise conditions,

and

b) a conventional stoichiometric fuelling calibration system under acceleration and other high load operations using the capability of a TWC.

The disadvantage of such a system of emission control would be the management of the transition from lean-burn to stoichiometric operation during which driveability and tailpipe emission performance would be less than optimum using conventional SI combustion, with ever decreasing emission limits around the world [Chapman *et al* (1991a)]. The subject of wide open throttle fuelling is relevant to the discussion of real world emissions performance in Chapter 6.

## 2.20 Fast Catalyst Light-Off

The correct operation of a catalyst is dependent on its reaching a working temperature at the active sites. This is of the order of 200°C as described earlier in Section 2.18. Once the operating temperature has been reached, the exothermic reactions together with the continuing warming effect of the hot exhaust gases, ensures that the catalyst becomes suitably efficient. Hence the start-up of the catalyst is very important to achieving low tailpipe emission levels and measures such as double skinned exhaust manifolds and downpipes, close coupled catalysts and starter catalysts have all been widely used [Jeong and Kim (2001)]. There are also artificial means of heating the catalyst before and during engine start such as electrical heating and various burner systems. Each of these has its advantages and disadvantages and can be summarised as follows.

Electric heating requires energy from the battery or alternator system, at a time when the battery needs a good reserve in order to start the engine with a good cranking speed and a good ignition voltage, or while initial engine speed is likely to be low [Coppage and Bell (1997)]. The catalyst substrate has to be of a metal foil type in order to conduct electricity to be, itself, a heating element. This is not a disadvantage since many low backpressure and high performance exhaust systems use such a construction. However, pre-heating of the catalyst substrate prior to starting the engine will be counteracted by the initial convective gas flow from the exhaust that comprises the volume in the exhaust manifold and downpipe together with the contents of the first non-firing exhaust strokes of the engine. This cold gas will reduce the temperature of the pre-heated catalytic sites before the arrival of hot exhaust gases then increases it again.

Burner systems can be of many different types [Hepburn *et al* (1994), Öser *et al* (1994)]. It is sufficient to note here, however, that such systems either require fuel to be metered and ignited in the exhaust upstream of the catalyst, or the engine is needed to run in a rich enough condition to liberate sufficient amounts of hydrogen from the combustion process to be ignited in the presence of additional oxygen. The former system is subject to safety concerns in the event of a collision and has not been taken to production. A version of the latter system was developed by Ford and Cambridge University [Ma *et al* (1992)] with much of the development testing being carried out by Tickford under the management of the author. The system was known as Exhaust Gas Ignition or EGI© and used an igniter in the catalyst can in order to start a flame to heat the front face of the one of the converter substrates. The system is illustrated diagrammatically in Figure 25. It relied upon the liberation of hydrogen by the gasoline combustion process when fuelled at around 9:1 AFR, which could then be ignited in the region of the catalyst to increase the temperature of the catalytic sites in the face and foremost part of the catalyst brick.

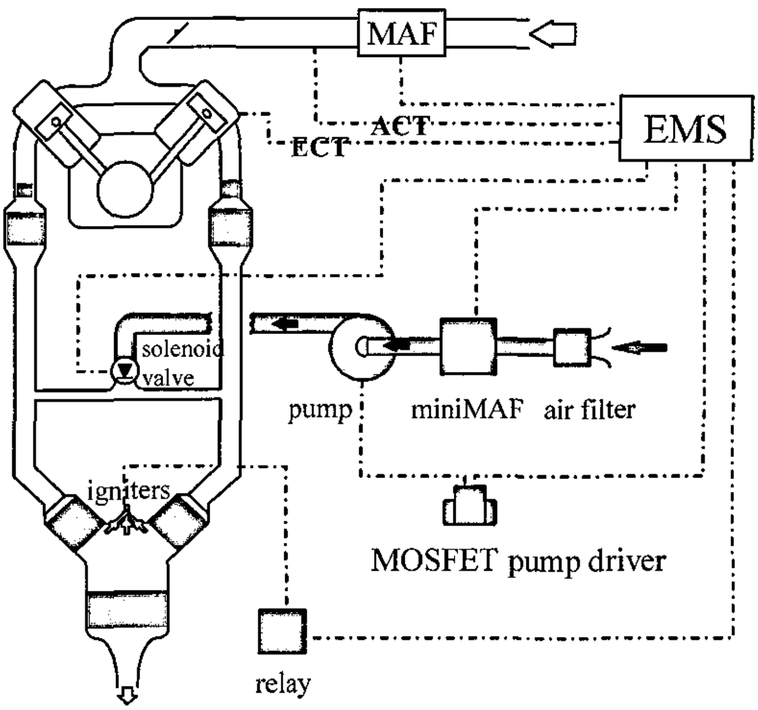



**Figure 25: Ford Exhaust Gas Ignition System (EGI ©)**
*(Schematic based on author's project knowledge and SAE 920400)*

The Ford and Cambridge system was very novel and during the development testing by the author's team, several technical improvements were made both to igniters and their position. Spark plugs caused problems with electrical noise and radio frequency interference (RFI) and were replaced by glow plugs for which BERU created some specific parts. In order to optimise the EGI flame distribution, the igniters were moved from in front of the first brick to a position in between the two bricks adding a slightly larger gap. By doing this, not only was the hydrogen-rich exhaust gas flow made more even and laminar by its passage through the front catalyst brick, but also there was a cleaner and faster light-off than would be obtained from the more exposed front brick. The front face of the rear brick was protected from contamination by debris, carbon build-up and harmful impurities which typically affect the front brick.

More conventionally, retarded spark advance and early exhaust valve opening have been effectively used to accelerate the heating of the exhaust system and catalyst. This is especially beneficial when the valve timing can be altered so that the timing strategy from cold start does not affect the whole operating speed and load range of the engine. The retarded spark leaves increased energy in the end gas and the early opening of the exhaust valve allows the high energy blow-down event to transfer the heat to the front face of the catalyst and helps to achieve the "light-off" temperature. Even without the use of early exhaust valve opening, the Ford Motor Company had a period of intense use of the CSSR or 'Cold Start Spark Retard' strategy in order for some engines to reach Euro III emission targets by starting the combustion later and leaving more energy in the exhaust gases at the blow-down point.

## 2.21 Closing Remarks

This chapter has described some important engine fundamentals and explained the predominance of spark ignition combustion for high performance engines. It has reviewed the main factors that affect power output in particular combustion chamber types and valve arrangements. It has also reviewed some of the main factors affecting cylinder gas dynamics with particular reference to valve area and intake effects that relate directly to the author's work and has put them in the context of the literature at the time the work was conducted. An explanation has been given of the factors relating to exhaust emissions and the use of catalytic converters to control the tailpipe emissions together with some of the consequent effects on fuel economy and fuelling control.

The next chapter will develop these themes in the context of multivalve engines and discuss the innovative flow work conducted by the author in 5-valves per cylinder research and development.

# Chapter 3.

# 5-VALVES PER CYLINDER DEVELOPMENTS

## 3.1 Introduction

This chapter builds on the geometrical advantages of a 5-valves per cylinder layout and describes the author's work in research and development of the design in a unique arrangement. It sets the context of the work and highlights the major results and benefits of the design in terms of the air-flow, flow coefficients and Mach Index. Reference is made to the innovative measurement techniques developed by the author and his team, as well as explaining the novel engine design work.

In this thesis, the flow coefficient ($C_v$) is defined as the ratio of the air flow through the actual port under test at a certain valve lift ($L$) to the air flow which would pass through an ideal orifice of the same diameter as the mean seat diameter ($D$), without losses and without pressure recovery.

## 3.2 Background

It is clear from the literature in Chapter 2 that multi-valve engines are not new. As well as the early motor racing engines already described, airship engine manufacturers of the early part of the 20$^{th}$ century experimented with multi-valve designs including 6, 10 and 12-valves per cylinder. A British Patent was granted to Count Lanzerotti Spina in 1928 for variants of multivalve engine including a 10-valves per cylinder engine with multiple spark plugs and a non-circular bore, as illustrated in Figure 26.

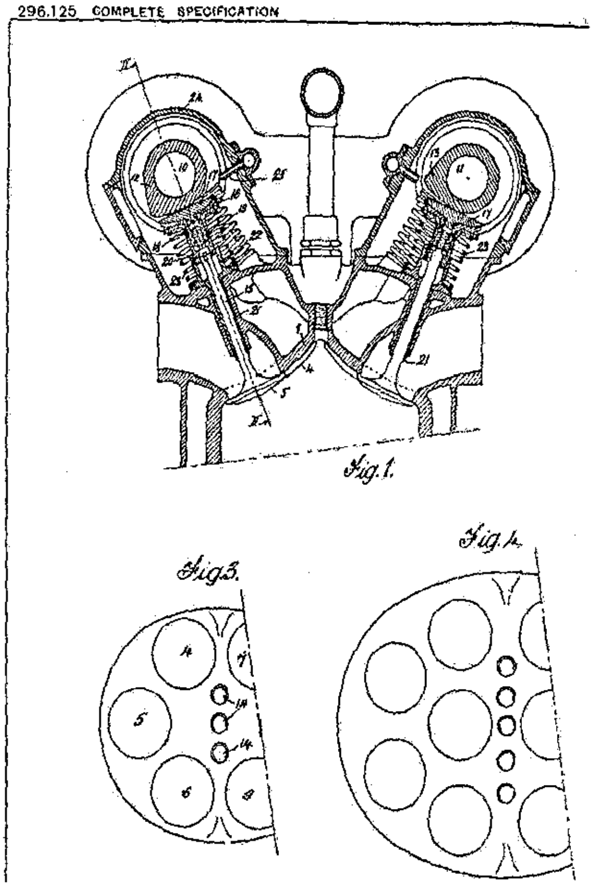



**Figure 26: Extract from Patent GB 296125**
*(Reproduced courtesy of Tickford Ltd.)*

Maybach, mentioned in Chapter 2 for his competition engine development, built airship and automotive versions of 12-valves per cylinder engines as early as 1929. The modern era arrived in 1985 when Yamaha produced the FZ750 model that had the first 5-valves per cylinder motorcycle engine. Yamaha had extensively researched the 5-valves per cylinder layout and reported its findings in 1986 [Aoi *et al* (1986)]. This and the further work conducted by the author [Bale and Downing (1990)] confirmed the advantages of such a layout over 4-valves per cylinder, which include:

1. For a given bore diameter, as already described in Chapter 2, the 5-valve layout provides the maximum achievable inlet valve area as a percentage of bore area (32 %), based on typical inlet to exhaust area ratio and material thicknesses between the valves;

2. The 5-valve layout provides an increase in inlet valve curtain area of approximately 25% relative to a 4-valve layout for the same bore diameter;
3. There is a reduction in effective individual inlet valve mass for a three inlet valve design compared with two, for the same open area, because each valve will be smaller;
4. The 5-valve layout has reduced inlet valve lift to achieve the same air flow as a 4-valve design for the same bore diameter;
5. The 5-valve layout has lower valve train loads than an equivalent 4-valve design for the same performance and bore diameter due to the lower mass of the individual components and consequent reduction in Hertzian stresses at the interface of the camshaft lobes and followers.

A 5-valve design geometrically offers a gain of approximately 25% in flow at low valve lifts compared to a 4-valves per cylinder design, due to the increased curtain area which results from the greater inlet valve perimeter of the 5-valve design. At higher valve lifts the flow becomes influenced by valve throat area and in this region the 5-valve design offers an increase of approximately 14% over a comparable 4-valve design due to its increased usage of bore area.

## 3.3 Flow Rig Testing and Analysis

Extensive air flow rig testing was performed by the author on different cylinder head designs, to realise the theoretical air flow improvement of the 5-valve design compared to an equivalent 4-valve design. The air flow rig was based on a viscous flow element to measure mass air flow and the main working components are illustrated in Figure 27.

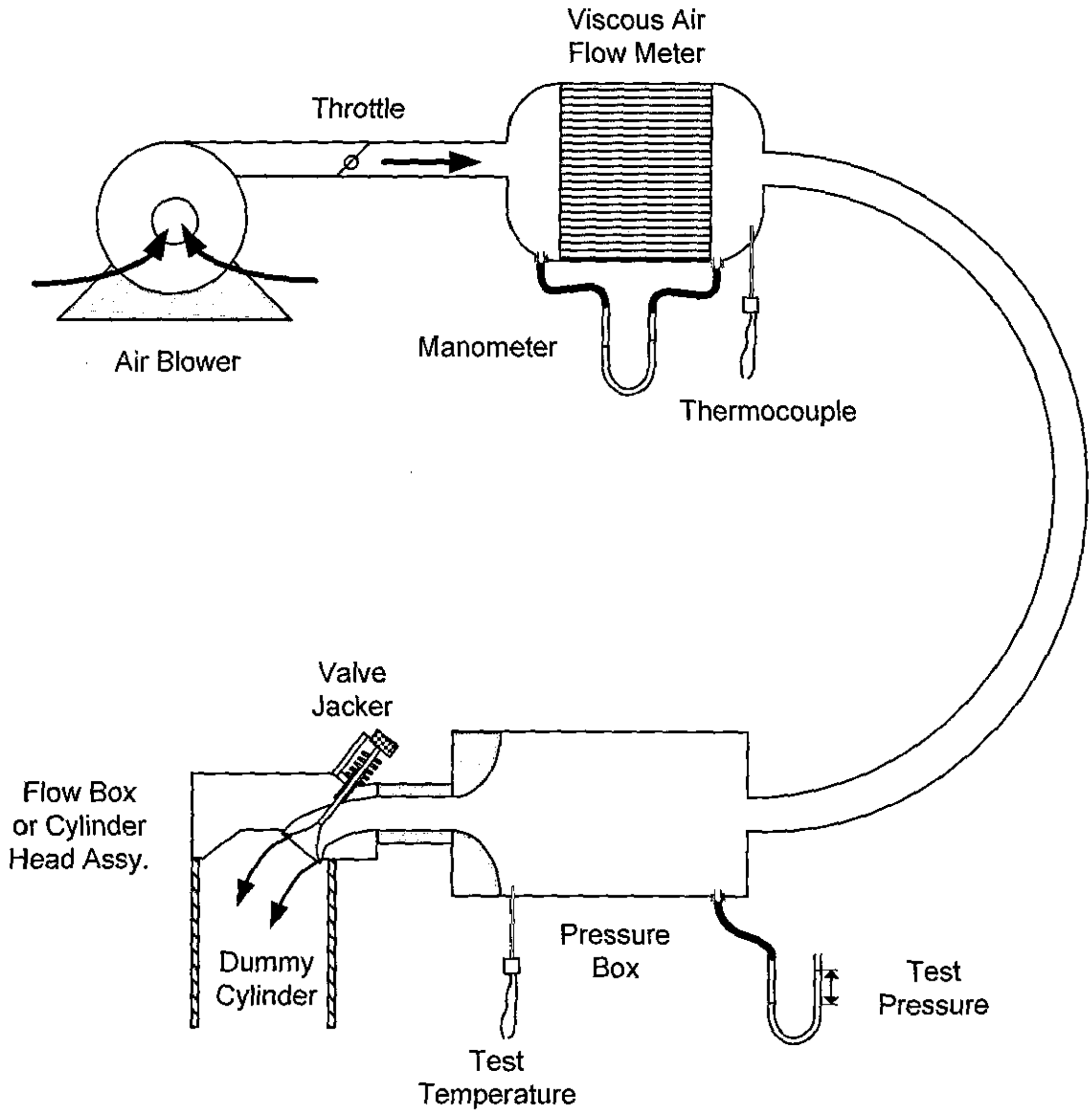



**Figure 27: Schematic of an air flow measurement rig**

A bespoke Microsoft Excel spreadsheet was used by the author to calculate the flow results, flow coefficient, Mach Index, etc., in line with the following relationships for the parameters as measured, e.g. static pressure and temperatures:

(a) The volume flow rate leaving the air flow meter is derived from the pressure drop across the air flow meter element multiplied by the basic calibration constants for the meter and a temperature dependent viscosity correction factor from Equation (16)

$$\dot{Q}_{meter} = \Delta P_{ma}\, C_s\, C_c\, CF_{t_{air}} \tag{16}$$

where $\dot{Q}_{meter}$ is the volume flow rate through the meter

$\Delta P_{ma}$ is the measured pressure drop across the mass air flow meter

$C_s$ is the scale multiplier for the laminar flow element

$C_c$ is the flow meter calibration constant

$CF_{t_{air}}$ is the temperature correction factor

(b) Mass air flow is calculated from the above volume flow rate and the air density conditions that exist at the downstream side of the meter element, and is given by

$$\dot{m}_a = \dot{Q}\left(P_{baro} + \Delta P_{ma}\right)\left(\frac{T_{ref}}{T_{ref} + T_m}\right)\rho_a \tag{17}$$

where $P_{baro}$ is the barometric pressure

$T_{ref}$ is the reference temperature

$T_m$ is the measured temperature at the meter

$\rho_a$ is the density of air at $T_{ref}$

(c) The volume flow rate at the pressure box conditions, i.e. the entry to the test piece, is slightly less than that leaving the meter because the air temperature may also be different. For these reasons the density is likely to be different from that leaving the meter and the flow rate at the pressure box entry to the test piece is give by

$$\dot{Q}_{pbox} = \dot{Q}_{meter}\left(\frac{P_{baro} + \Delta P_{ma}}{P_{baro} + \Delta P_{pbox}}\right)\left(\frac{T_{pbox}}{T_m}\right) \tag{18}$$

where $\dot{Q}_{pbox}$ is the volume flow rate through the pressure box

$\Delta P_{pbox}$ is the pressure difference between the pressure box and atmosphere

$T_{pbox}$ is the temperature of the pressure box

$T_m$ is the temperature at the air flow meter

Temperatures are measured in K

(d) Volume flow rate at standard pressure box conditions is required in order to compare tests carried out with varying ambient conditions, in the same way as for any air flow of a given density through an orifice, which is given by

$$\dot{m} = C_d\, A\sqrt{\frac{2\Delta P}{\rho}} \tag{19}$$

where $\dot{m}$ is the mass flow rate

$C_d$ is the discharge coefficient of the calibrated orifice

$A$ is the area of the orifice

$\Delta P$ is the pressure drop across the orifice

$\rho$ is the air density

Therefore, to correct the flow rate to that which would occur at the same standard pressure difference (almost invariably set to 254 mm water gauge for the purposes of automotive cylinder head air flow testing) but with a pressure box air density at a standard barometric pressure of 760 mm Hg and with an air temperature of 288 K (i.e. 15°C), a further calculation step is necessary based on the flowing expression:

$$\dot{Q}_{Std\,pbox} = \dot{Q}_{pbox}\sqrt{\left(\frac{P_{baro} + \Delta P_{pbox}}{P_{ref}}\right)\left(\frac{288}{T_{pbox}}\right)} \tag{20}$$

where $\dot{Q}_{Std\,pbox}$ is the volume flow rate at standard conditions

$\dot{Q}_{pbox}$ is the volume flow rate at measured conditions

$P_{ref}$ is the reference pressure

(e) Finally, the mass flow that would occur with standard pressure in the pressure box can be derived from the volume flow at standard conditions in the pressure box multiplied by the air density at those conditions:

$$\dot{m}_{a\,Std\,pbox} = \dot{Q}_{Std\,pbox}\,\rho_a \tag{21}$$

where $\dot{m}_{aStd\,pbox}$ is the mass air flow rate at standard conditions

($\rho_a$ was taken as $1.2254 \times 10^{-3}$ kg.m$^{-3}$ at standard conditions of 760 mm Hg and 288 K)

## 3.4 Calculation of Mean Flow Coefficient and Mach Index

It is intuitive that a port that is well shaped will flow more air than one that has, for a worst-case example, a sharp right angle bend. Furthermore, if a valve train is included that is capable of lifting the valve rapidly off its seat to maximum lift and returning it equally rapidly, the effective flow will be improved (noting that the dynamic limitations associated with the mechanics of valve trains may not allow this). A method has been developed to measure the flow efficiency of a port and valve on a flow rig, taking into account the valve lift curve to produce an overall flow efficiency number that goes beyond the simple flow analysis described above. This will be described later in this section and it is also appropriate to relate the parameter Mach Index, as introduced in Chapter 2.

Where there is any restriction in an engine flow stream, such as an inlet valve, as the flow-rate through the valve becomes higher, the gas velocity through the valve seat area and the pressure drop across it will both increase, hence restricting the volumetric efficiency of the engine. The limiting value of the velocity, at which the flow will become choked, is the sonic velocity in that medium. Thus it is useful to express the average flow velocity from the requirements of the engine at any un-throttled speed, as a function of the sonic or choking velocity. This parameter is known as inlet valve Mach Index, sometimes referred to quite descriptively as "Gulp Factor". The Mach Index provides an indication of the extent to which the engine may be choked (or "gulping") for inlet flow.

Early work in the USA by Livengood, Rogowski and Taylor [Livengood *et al* (1952)] derived a correlation between volumetric efficiency and the Mach Index and highlighted the importance of valve flow coefficients and valve area. Even at that time, consideration of valve timing, together with assumed inlet and exhaust pressures, could result in an indicated output estimation, with mechanical friction losses allowed for, to obtain the predicted power curve. Before the fluid flow predictive tools such as computational fluid dynamics (CFD) were developed, it has always been most useful and economically attractive to be able to examine the likely effect of changes made in valve size, shape or timing, port shape, etc., before actual physical alteration is made.

Since the analysis was based on a steady-state flow rig, the accuracy of any output predictions decreases with the increase of valve overlap and the presence of strongly pulsating exhaust, or ram effects derived from the inlet manifold, which were described in Chapter 2. These can significantly alter the volumetric efficiency of the engine from the steady flow conditions of a flow rig.

Livengood's work showed that volumetric efficiency initially stayed almost constant up to a Mach Index of about 0·5 at which point its value reduced significantly. This is illustrated in Figure 28.

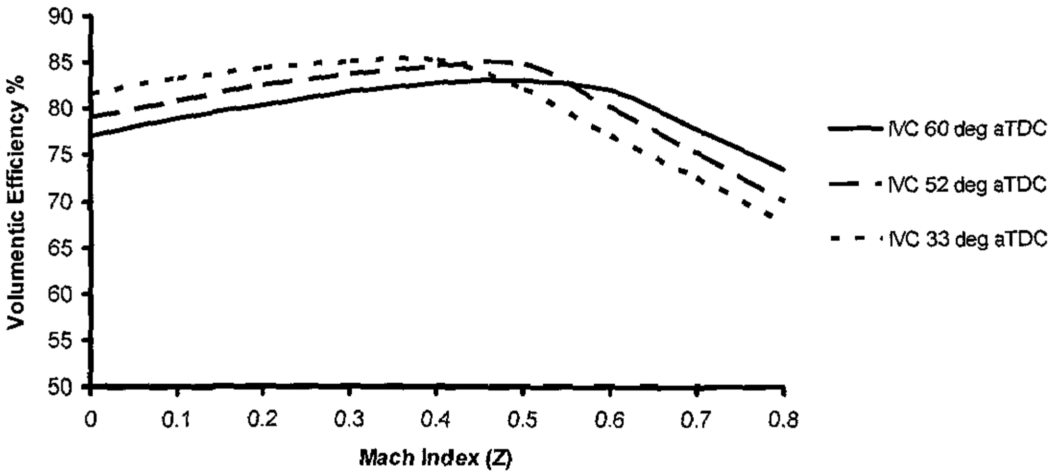



**Figure 28: The effects of Mach Index and inlet valve closing point on volumetric efficiency**
*(Data from Sykes (1993))*

Further research showed that the level of the "plateau" and the Mach Index at which the turn down point occurs, are both affected by the valve timing, particularly the valve closing point. The work initially reported by Livengood *et al* (1952), was further detailed by Taylor (1985a) who confirmed that the relationship between volumetric efficiency and Mach Index was reasonably independent of valve timing and pressure drop over a range of measurements. This is illustrated in Figure 29. Other authors, including Hayward and Stone, have also based their texts on Livengood *et al* and Taylor.

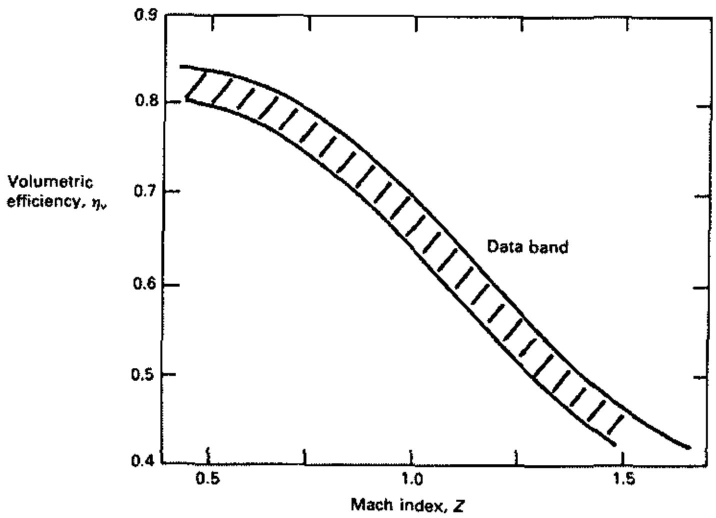



**Figure 29: Volumetric efficiency as a function of the Mach Index**
*(Reproduced from Stone (1999j), adapted from Taylor)*

The characteristics illustrated in Figures 28 and 29 correlate with some other effects frequently experienced when trying to modify and engine to give more power. If the inlet valve area starts to become restrictive at certain speeds and it is not convenient to fit a larger valve, then a common modification is to fit a longer period cam. Due to charge loss and inefficient start of compression at low speeds, this usually reduces the low speed engine torque but lets the engine breath better at higher speeds. This is due to improved cylinder filling from ram effects as the inlet valve closes, described in Chapter 2, thereby increasing volumetric efficiency at high engine speeds, to give more power.

If, conversely, plenty of valve area can be provided, the engine can breathe adequately to high engine speed even with a short inlet valve opening period. A short inlet valve period also gives best torque at the lower speeds because of reduced charge loss during the shorter inlet and exhaust valve overlap, while the inertia ram is low, as explained in Chapter 2. However, achieving a desired valve lift within a short valve opening period means that the valves have to be opened and closed more quickly. The stiffer the valve gear, the better the chance of making this work properly to give a good combination of high power and a broad speed range of high torque.

According to Livengood *et al* (1952), Mach Index ($Z$) can be expressed in the following terms

$$Z = \frac{\overline{S}_p}{C_{local}} \frac{\pi}{4} \left( \frac{B}{D} \right)^2 \frac{1}{Cv_{\theta mean}} \tag{22}$$

where $\overline{S}_p$ is the mean piston speed

$C_{local}$ is the speed of sound in the flow medium

$Cv_{\theta mean}$ is the mean valve flow coefficient

$B$ is the bore diameter

$D$ is the inlet valve seat mean diameter.

Heywood (1988d) reported the work of Taylor (1985a) and co-workers and provided a rather more succinct derivative expression for Mach Index

$$Z = \frac{A_p \overline{S}_p}{C_i A_i C_{local}} \tag{23}$$

where $A_p$ is the piston area

$A_i$ is the valve flow area

$C_i$ is a mean valve flow coefficient based on the area $A_i$

The calculations to derive flow coefficient and Mach Index therefore begin with a calculation of the mean flow coefficient. This combines the breathing abilities of the valves and ports, as described above for fixed valve openings, with the characteristics of valve opening and closing determined by the camshaft and valve gear. The process is based on an integration of the performance of the valve flow system taking into account the time that the valve is open. For a valve that is opened quickly and stays close to maximum lift for a long duration, the performance of the port and valve at and near to maximum lift is the most significant contribution to the actual engine behaviour of the system. In this way it is possible to determine a representative single value for the flow performance, which can be compared with alternative designs and configurations.

To obtain the mean flow coefficient, the figures from the valve flow coefficient, $C_v$, versus valve lift curve, are transferred to the valve lift diagram to form a $C_v$ against crank angle diagram, as illustrated in Figure 30. The value of mean $C_v\theta$ is the average of this combined curve over the whole of the valve period shown in Figure 30.

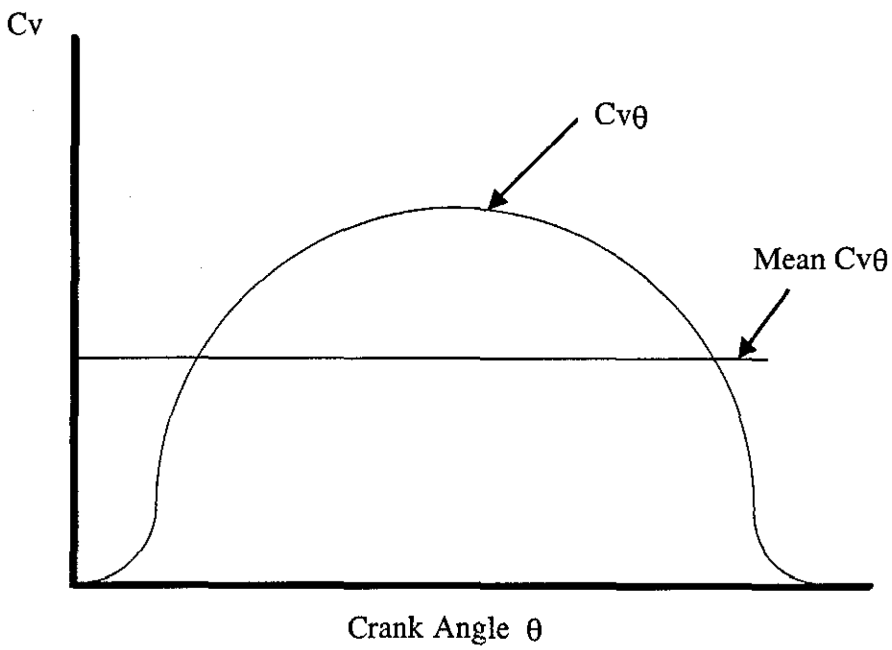



**Figure 30: Flow coefficient plotted against crank angle showing Mean Cv θ**

The calculation required is

$$C_v\theta_{mean} = \int_{ValveOpenAngle}^{ValveCloseAngle} \left( \frac{C_v\theta}{CamPeriod} \right) d\theta \tag{24}$$

The author shares the view that calculating the average of the flow coefficient over the whole of the valve period is slightly irrelevant since the air flow into the engine will not necessarily increase as the valve period is lengthened. This is because the opening of the valve before TDC is to avoid any partial vacuum at the start of the downward piston stroke and to allow any possible inward draw of charge due to an inlet pressure differential to the cylinder space, which may be available from the exhaust outflow. It is not part of the normal inlet flow. The forward flow across the cylinder from the inlet valve, which produces tumble swirl, does not necessarily occur before TDC, or after BDC, when inlet mass flow is low. Keeping the inlet valve open after BDC allows for the benefits of inertia ram, as described in Chapter 2, to improve the cylinder filling, especially at high engine speed. Since this is a dynamic effect, the use of different valve lift curves and periods in calculating mean $C_v\theta$ could indicate erroneous trends. Therefore, integrating mean flow coefficient and tumble swirl ratio between TDC and BDC, rather than over the full valve opening period, gives a more realistic parameter for comparison of different port designs and valve lift profiles. The preferred mean $C_v\theta$ is therefore calculated over the 180 crank degrees between TDC and BDC, the actual induction stroke, and is illustrated in Figure 31. This can be termed mean $C_{v180}$ or $C_{v180mean}$.

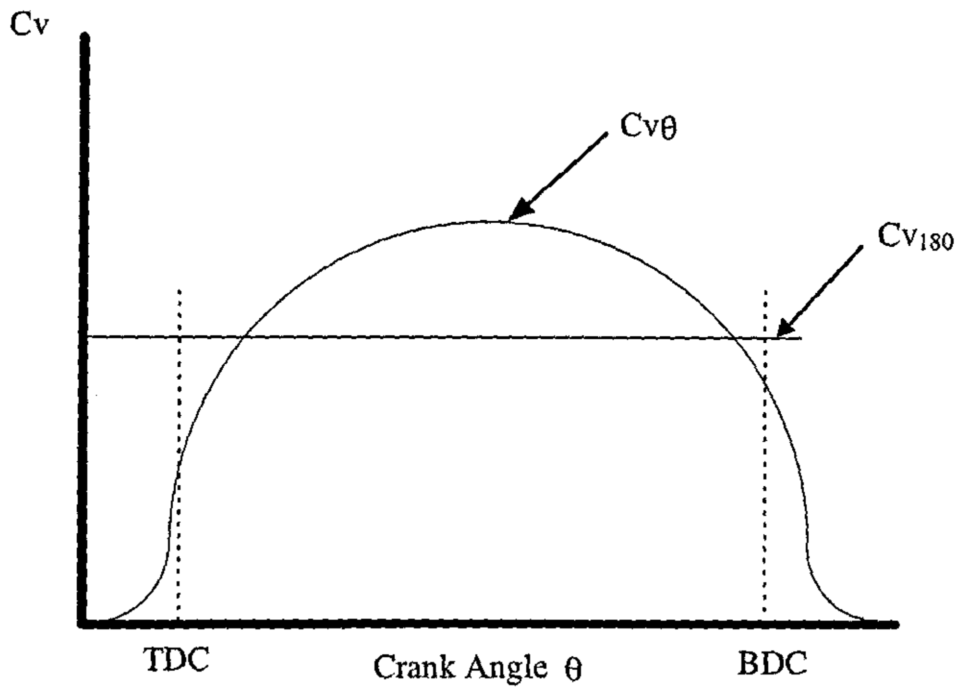



**Figure 31: Flow coefficient plotted against crank angle showing mean $C_{v180}$**

The calculation in this case is

$$C_{v180mean} = \int_0^{\pi} \left( \frac{C_v \theta}{\pi} \right) d\theta \tag{25}$$

Mach Index, previously defined in Equation (23), can alternatively be expressed in the terms of mean $Cv_\theta$ (or mean $Cv_{180}$) flow efficiency, as

$$Z = \left( \frac{\overline{S}_i}{C_{local}} \right) \times \frac{1}{Cv_{180}} \tag{26}$$

where $\overline{S}_i$ is the mean inlet gas velocity (MIGV) at the valves.

The quoted Mach Index figures for engines are usually related to the maximum rated speed but it is important to underline that this simple model ignores any dynamic effects in the inlet tract as described earlier and in Chapter 2.

This analysis reinforces the intuitive need for design optimisation on the induction side of the cylinder, which is the subject of much of the work of this thesis. This is more important than that of the exhaust valve and port, where blow-down from the high residual pressure in the cylinder at valve opening, and subsequent positive displacement, mean that higher pressure differential is available across the valve and port flow system. This high pressure differential, together with the higher limiting flow speed in the hot medium of the exhaust gas, as explained in Chapter 2, makes the exhaust flow less dependent on port design than inlet flow.

## 3.5 Inlet Flow Development

With all cylinder heads, flow out of the inlet valves is reduced due to shrouding by the combustion chamber walls and interference between the flows from adjacent valves. Work by the author [Bale and Downing (1990)] confirmed that the increase in inlet valve curtain area for a 5-valve layout over a 4-valve layout did not result in a similar increase in air flow. This was because of the relative inefficiency of multiple inlet valves. Figure 32 shows the relevant areas for a 4-valves per cylinder layout indicating where the two inlet flows interfere and where the proximity of the combustion chamber wall also reduces the flow efficiency.

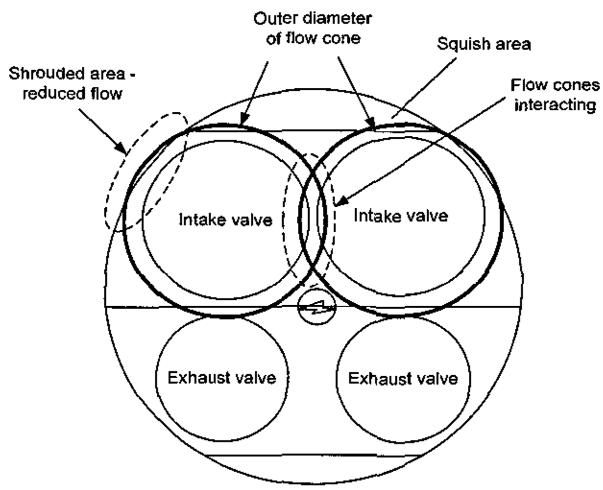



**Figure 32: Areas of valve shrouding in a typical 4-valves per cylinder layout**

These effects were particularly significant on a 5-valve design due to the closeness of the combustion chamber wall to the outer pair of inlet valves and the additional interference that occurred between the flow from the central inlet valve and the two outer inlet valves. This is a factor which partly negates the increase of available valve area that provided by a third intake valve.

Therefore, in addition to basic air mass flow measurements, the author studied the air flow profile around each valve using the apparatus illustrated in Figure 33. At the time that this work was conducted, CFD analysis code was not as developed, reliable and cost effective as it is today and the author's company did not possess any such predictive software. Therefore, an innovative empirical approach was adopted in order to quantify the flow field at the valve outlet and develop it to a more optimised condition. The method adopted was clearly novel at the time, having not been found in the literature of the period.

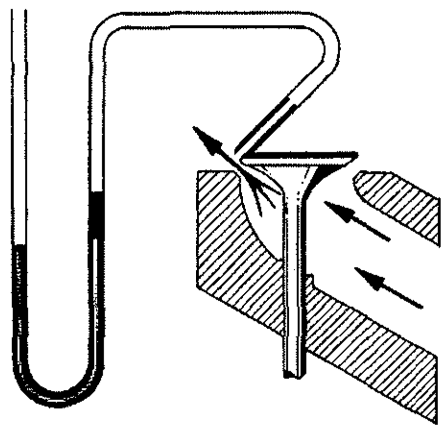


**Figure 33: Schematic of author's apparatus for measuring valve flow distribution**

The novel apparatus developed to research the flow distribution, consisted of an inlet valve with a small hole through the valve seat and at right angles to the seat and hence the intake flow. The hole was connected through a drilling in the valve stem to a multi-slope manometer capable of resolving small pressure differences by use of a low density fluid, being a proprietary mix of alcohol, dye and water. The gap between the valve head and valve insert seat acted as a venturi when air flowed and the greater the air velocity at a given point the lower the static pressure at that point due to Bernoulli's relationship. Therefore, high flow corresponded to a lower static pressure as measured by the manometer.

Tests were made by measuring the relative valve seat static pressure profile for each inlet valve at 1 mm lift steps up to 10 mm, with all the inlet valves being opened simultaneously. The pressure profile was measured by moving the valve through a complete rotation in 30 degree increments and measuring the static pressure at each point. Analysis of the relative pressure profiles showed regions where flow was reduced due to flow interference between adjacent valves or shrouding due to the combustion chamber walls, as illustrated in Figure 34. The radial axis in each part of the figure represents the relative pressure drop at the valve seat from which the flow rate at the twelve points around the periphery of the valve is inferred. Each origin represents no pressure drop (flow) from the relevant valve and a circular plot would indicate even flow distribution. A larger enclosed area of the plot indicates a higher total flow.

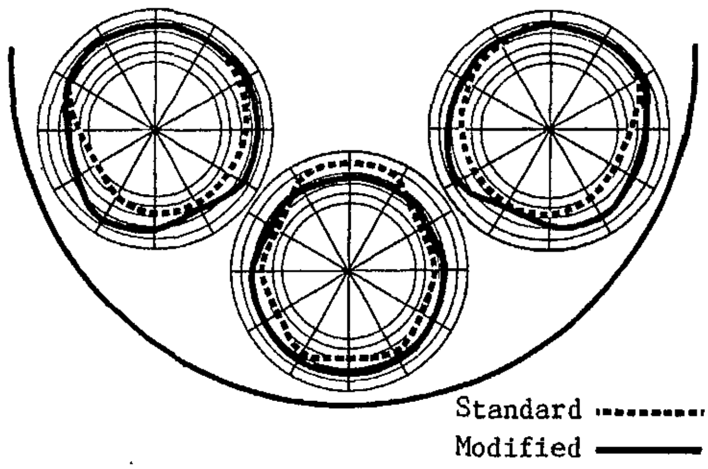



**Figure 34: Polar relative pressure plots illustrating flow distortion round the periphery of the inlet valves of a 5-valves per cylinder design before and after development**

The pressure profile at the start of the development is shown by the dotted line in Figure 34. The attenuated flow due to the inter-valve interference and the effect of the proximity of the chamber wall are clearly apparent from the local distortions in the relative pressure profile. Port development work and local relief of the chamber wall, as described in the following development stages, improved the position to that shown in the solid line in the same Figure 34. The distorted flow from the central inlet valve, where there was an uneven high flow outlet biased towards the centre of the combustion chamber, was reduced and the total flow from the valves was improved. The development work to improve the flow distribution and reduce the shrouding effects was carried out on the steady state flow bench illustrated earlier in Figure 27.

In the study of flow coefficients, it is common practice to plot results against relative valve lift rather than absolute valve lift. The use of the non-dimensional representation of valve lift as a proportion of mean inlet valve seat diameter, L/D, allows comparison of the data with other designs regardless of size or valve layout. In the course of the author's work, the value for the mean valve seat diameter, D, was established by the following procedure, using measurements illustrated in Figure 35.

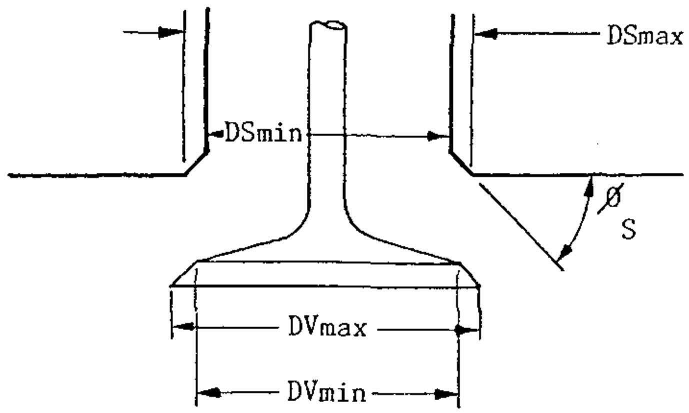



**Figure 35: Measurements used to calculate mean seat diameter D**

1. Measure the inner and outer diameters of the inlet valve seat, $DV_{max}$ and $DV_{min}$;
2. Measure the inner and outer diameters of the cylinder head valve seat insert, $DS_{max}$ and $DS_{min}$;
3. Take the minimum diameter of the seat for both valve and cylinder head set and note the larger;
4. Take the maximum diameter of the seat for both valve and cylinder head set and note the smaller;
5. The above diameters constitute the inner and outer diameters of the valve to seat contact patch and D is calculated as the mean of the two values.

Figure 36 shows flow coefficient $C_v$ against L/D ratio for the author's successful 5-valve cylinder head at several stages of development.

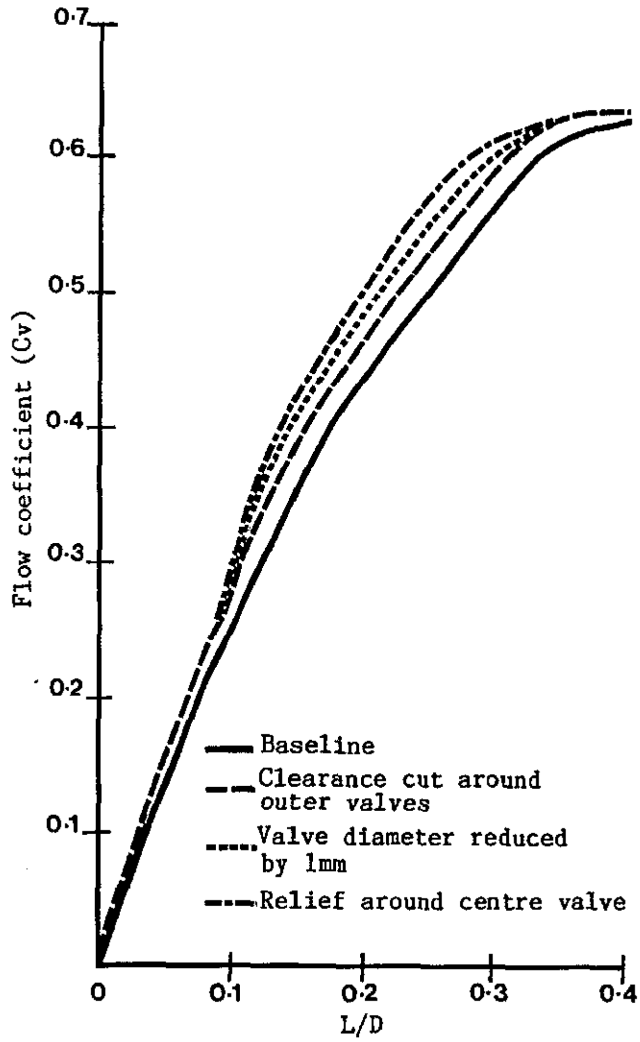



**Figure 36: Effect of air flow modifications**

The significant stages of development of the design were as follows:

Stage 1

The solid line in Figure 36 shows the baseline measurement of original design. The flow coefficient ($C_V$) of the initial design was found to be lower than a comparable 4-valve design but mass flow was still higher due to the greater valve area of the 5-valve design. The low flow coefficient was investigated by the relative pressure profile measurement described above. This indicated that the limited flow performance was primarily due to shrouding of the flow from the outer inlet valves by the combustion chamber walls.

Stage 2

The outer inlet valves were unshrouded by machining a clearance cut in the combustion chamber at a radius centred on the valve centreline and inclined at the valve guide angle relative to the cylinder head. Figure 36 shows an improvement in flow at all valve lifts. Further unshrouding of the combustion chamber around the outer valves was not possible within the confines of the bore.

Stage 3

The diameter of all the inlet valve heads were reduced by 1 mm. This improved the clearance around and between the valves and gave substantial improvements at L/D ratios of 0·1 to 0·35.

Stage 4

Relative pressure profile measurements showed that shrouding of the outer valves by the chamber had been reduced as far as was practical within the constraints of the bore diameter. However, these measurements also showed that flow interference between the centre and outer inlet valves was still significant and that the flow out of the centre inlet valve was being restricted at the sides and back of the inlet valve (nearest to the combustion chamber wall). This was causing extra air to be forced out of the front segment of the valve resulting in interference with the flows from the two outside valves. The chamber was therefore relieved around the centre valve to give a minimum clearance of 3 mm between the outer edge of the valve and the combustion chamber; this gave a further improvement in flow coefficient as illustrated in Figure 36.

Figure 34 showed the corresponding effect of the reduction in chamber shrouding and flow interference, developed during the above Stages 1 to 4, on the relative pressure profile around the inlet valves compared to the original design.

Additional testing showed that further material removal from the combustion chamber around the central inlet valve did not give any significant improvement in air flow but would lead to loss of compression ratio. This was because the clearance volume in the combustion chamber would have been increased and any prospect of recovering this by a higher piston crown would have been offset by deeper valve pockets.

Stage 5

The inlet ports to each valve were flowed individually by closing the valves of the other ports, filling the inoperative ports with modelling clay and streamlining the entrance to the port at the junction of the three inlet ports in the cylinder head. Testing of the ports without this filling had shown that results were inaccurate if this procedure was not followed. This was because the open end of the ports with closed valves caused a disturbance to the flow through the port of the open valve.

The flow performance of the centre port is illustrated in Figure 37. The port performance up to a L/D ratio of 0·21 was good, but at higher ratios did not achieve that measured for high performance 4-valve ports.

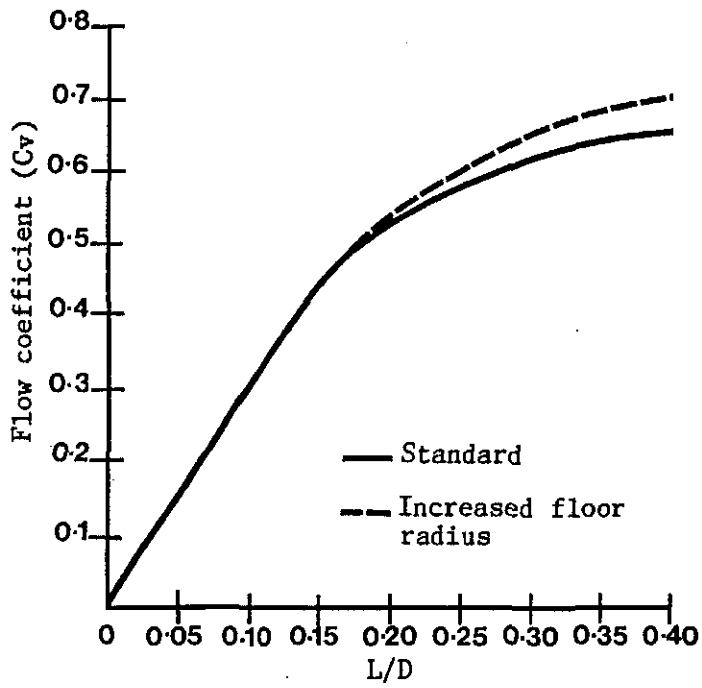



**Figure 37: Effect of port floor radius on flow coefficient**

Further investigation showed that the port geometry was similar to that of the high performance 4-valve port, except for the radius between the port floor and the valve seat as detailed in Figure 38.

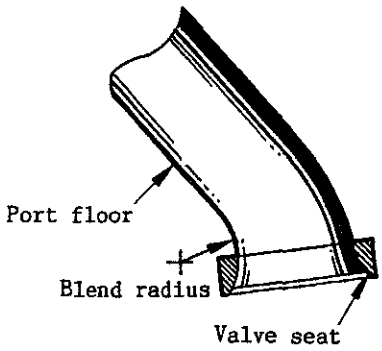



**Figure 38: Schematic diagram of port floor radius**

This radius was increased and the effect of this change was illustrated in Figure 37. The efficiency of the centre port now equalled that of the 4-valve ports investigated. The flow performance development steps for the outer inlet ports are compared with that of the improved central port in Figure 39.

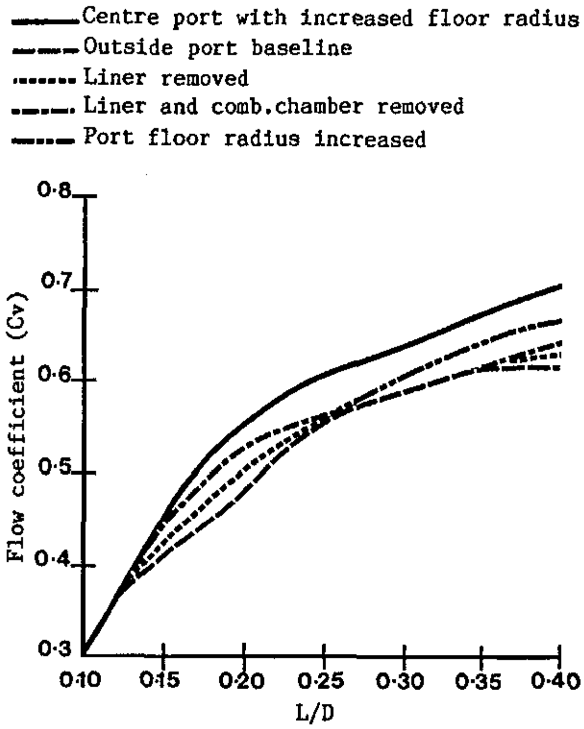



**Figure 39: Effect of liner and chamber shrouding**

Figure 39 also demonstrates that the outer inlet ports were less efficient than the centre port, particularly in the L/D range of 0·125 to 0·25. This was believed to be due to the angle of approach, in a horizontal plane, of the port to the valve or shrouding of the valve by the combustion chamber and cylinder liner. A series of tests was made to investigate the effect of each factor.

Firstly the cylinder liner was removed; this gave a small improvement in port efficiency in the L/D range of 0·125 to 0·25. The combustion chamber was then completely removed around the valve to give an unrestricted exit from the valve. This gave a further increase in port efficiency in the L/D range of 0·125 to 0·25, but no improvement elsewhere. The port floor to valve seat radius was then increased as for the centre valve; this gave a similar improvement in the L/D range above 0·25. These tests concluded that the lower efficiency of the outer inlet ports was due to the geometry of the port relative to the valve seat. It was not possible to change the angle of the outer ports on the two versions of the design developed at that time. It was intended to design and develop a further version with an improved angle of approach for the outer inlet ports.

Air mass flows for the fully developed versions of the two 5-valve cylinder heads were compared to the corresponding 4-valve designs of the same bore diameter and the results are illustrated in Figure 40. This shows that the 5-valve layout was capable of flowing significantly more air.

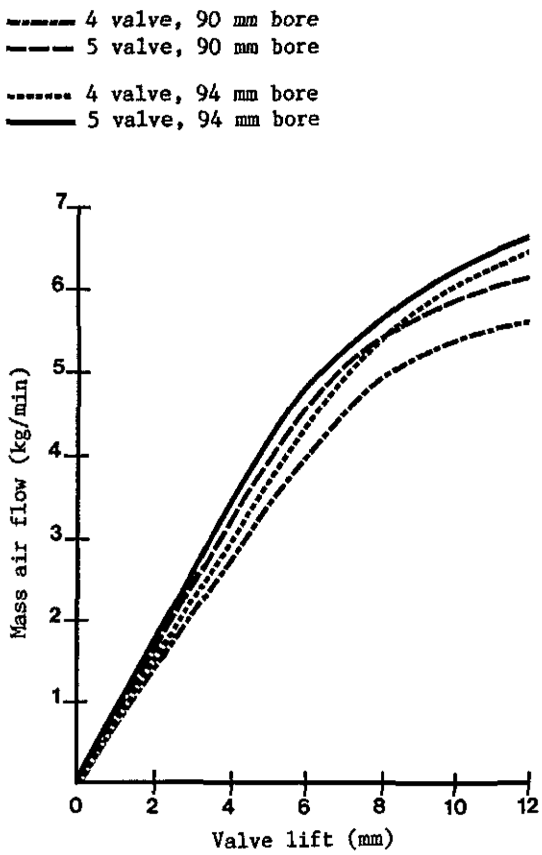



**Figure 40: Inlet mass air flow summary**

Having demonstrated routes to improving mass flow of charge into the cylinder, further attention was given to maximising the engine performance through the most efficient and complete combustion of the fuel.

## 3.6 Closing Remarks

This chapter has described methods of measuring and improving port flow using rigs, particularly addressing the particular issues arising in multivalve engines. A novel technique for measuring the flow distribution around valves was described. The unique Tickford 5-valves per cylinder design was developed using these methods and applied to two engines between 1988 and 1990.

Chapter 4 will describe how these flow improvements can be complemented by improvements in the mixing and combustion processes.

# Chapter 4.

# COMBUSTION DEVELOPMENTS

## 4.1 Introduction

This chapter discusses the different modes of intake port and combustion chamber-induced charge motion and the beneficial effects on combustion. Different methods of quantifying air motion using air flow rigs are described as well as the way in which calculations can be made in order to compare one configuration, and one test method, with another.

Having developed the inlet port and system to optimise the volumetric efficiency of an engine, it was necessary to consider how best the charge may be burnt in order to deliver the maximum useful work and, as previously stated, a significant element of this combustion efficiency can be aided by charge motion. Conventional "swirl" motion was described in Chapter 2 as being inherent in most 2-valve engines. For engines with symmetrical inlet valve configurations, be they 2 inlet valves, 3, or more, an alternative type of motion has to be considered.

## 4.2 "Tumble" or "Barrel" Swirl

The first development of a tumble port engine was probably carried out by the late Keith Duckworth of Cosworth during the development of high performance, Ford-based, 4-valves per cylinder engines like the FVA (FV = 4-valve, A=Series) and BDA (Belt Drive A-Series), which he started in the 1960s. He later went on to develop the highly successful and long-lived 3·0 litre DFV (Double Four Valve) engine in Formula One, followed by its derivatives, which included the 3·5 litre DFY, DFZ and DFR in Formula One, the 3·3 litre DFL for endurance racing and the turbocharged DFX for CART racing in the US. It is unclear whether the reasons for the superior performance of the "tumble swirl" engine were fully apparent to Duckworth at that time but he certainly understood that developing an engine solely on a flow bench was folly and that improvements made to flow bench performance may be disadvantageous to a practical, running engine [Robson (1995)]. Duckworth's design philosophy was to avoid "surplus combustion chamber volume" by the use of a pent-roof (rather than hemispherical) combustion chamber and a high compression ratio with a flat-top piston. This meant that the valve angles had to be adjusted to keep the clearance volume down and, by a completely original piece of thinking, he was responsible for the modern narrow-angle, 4-valve head. The 4-cylinder FVA engine of 1966 had a valve included angle of 40 degrees and the DFV of 1967 used a 32 degree angle. Duckworth was also an advocate of designing intake ports that were not too large in order to keep inlet velocities high. It was said that he could look at an engine, feel around in the ports with his fingers and know if it was going to flow or not [Robson (1995)]. History has proved Duckworth to be generally right, although the literature does not reveal whether he appreciated the exact impact that narrow valve included angle and good port flow had on tumble at the time.

Engine developers at Rover who were working on the K-Series and other engines, reported some understanding and application of tumble or barrel swirl [Benjamin (1988), Chapman *et al* (1989), (1991a) and (1991b) *inter alia*] with indications that the phenomenon may have been independently observed and understood during their programmes. It is almost certain that the effect of this type of charge motion was observed and implemented before the science and analysis of the flow and combustion mechanisms were fully understood. Racing engines, as previously mentioned, were historically highly developed for power output, largely without the use of combustion analysis or extensive flow-bench testing; what flow testing there was, being with the purpose of maximising flow coefficients.

Tumble swirl on its own, in the form of the induction-induced gross motion that is measured on the flow bench rig with air motion equipment, is not very supportive to combustion, other than the benefits that accrue from improving the mixing of fuel and air in the charge prior to the initiation of combustion. This will be expanded on as this chapter develops.

Since the mean inlet gas velocity is related to the mean piston speed and the ratio of bore area to inlet valve seat area, the inlet gas issues from the valve opening at, typically, about 10 times the piston speed, giving it considerable kinetic energy. The relationship is

$$\overline{S}_i = \overline{S}_p \frac{B^2}{D^2} \tag{27}$$

where $\overline{S}_i$ is MIGV past the valve

$\overline{S}_p$ is mean piston speed

B is the cylinder bore diameter

D is the inlet valve mean seat diameter

For an engine that has been designed to develop tumble, Figure 41 illustrates the initial way in which the induction flow separates from the port centreline as port velocity increases and the way the main flow is thereby directed across the top of the valve, resulting in a rotation of the charge in the cylinder about an axis parallel to the crankshaft.

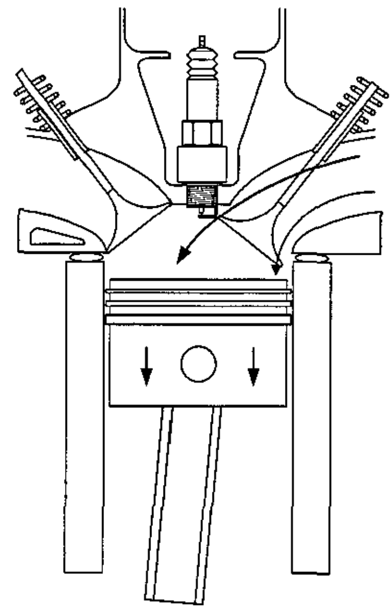


**Figure 41: Diagram of the start of induction-induced air motion**

As the piston continues its downstroke, the momentum of the rotating charge is increased and, when BDC is reached and the piston reverses its direction, the charge motion is maintained and, preserving angular momentum, accelerates in the reducing cylinder space, as illustrated in the following Figures 42 and 43.

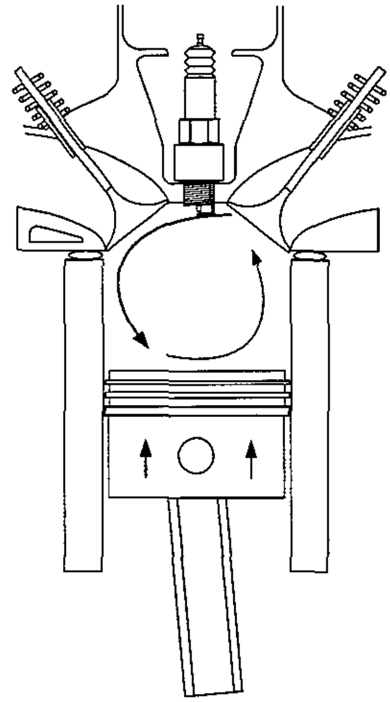


**Figure 42: Diagram to illustrate induction-induced air motion at the start of compression stroke**

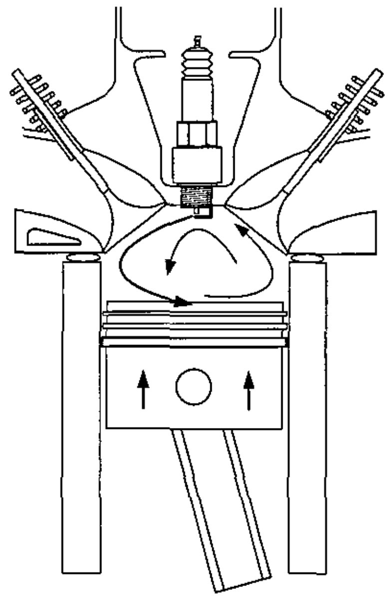


**Figure 43: Diagram to illustrate induction-induced air motion near the end of the compression stroke**

The rising piston accelerates the gross air motion and the intrusions in the combustion chamber (chamber edges, valve edges, spark plug) break up the flow and begin to form eddies that assist in mixing the fuel and air.

As the piston approaches TDC and the point of ignition, the gross air motion has been broken down into microturbulence [Witze *et al* (1983), Floch *et al* (1995) *inter alia*], which further enhances mixture homogeneity and provides physical displacement of the flame front. This is diagrammatically illustrated in Figure 44.

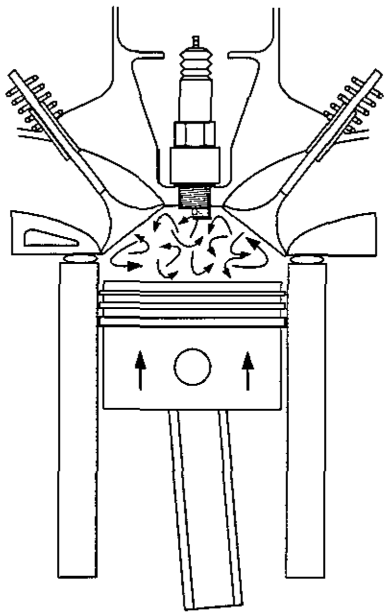


**Figure 44: Diagram to illustrate microturbulence at the point of ignition**

Once the mixture is ignited, the combustion process has two propagation mechanisms, chemical flame progression through the mixture and physical displacement across the combustion space, by which rapid combustion can be achieved, as illustrated in Figure 45. The structure of the flame front has been shown to develop as it propagates across the chamber with increasing amounts of "wrinkling" caused by the turbulent flow field, into a convoluted, thick, turbulent flame "brush" [Heywood (1988f)], which produces burn rates greater than that of a simpler laminar flame produced in a quiescent combustion chamber.

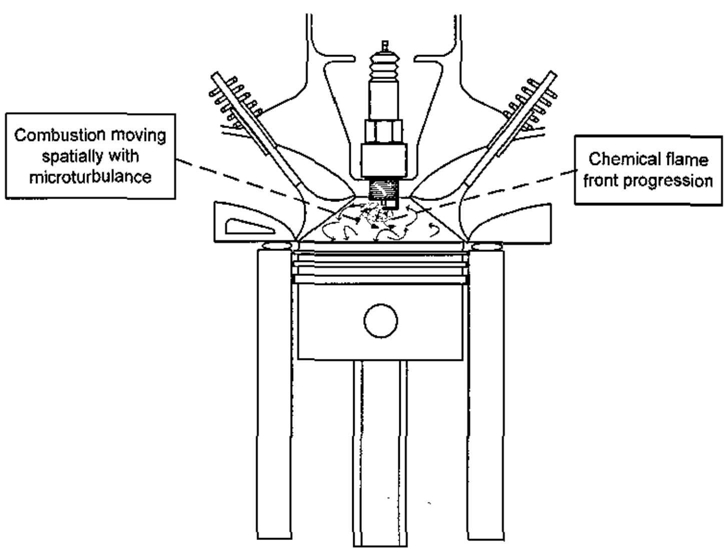



**Figure 45: Diagram to illustrate chemical and spatial flame progression**

Work that was being conducted around the same time at Ricardo using Laser Doppler Anemometry (LDA) [Kyriakides and Glover (1988)], showed strong correlation between turbulence intensity and the 10-90% burn time. It was shown that tumble air motion was a more effective way of generating turbulence at top dead centre than axial swirl, confirming the author's empirical findings based on steady-state flow rig and engine measurements.

The nature of swirling flows in an actual operating engine is difficult to measure and can only be carried out with modifications such as the LDA work referred to above. Improvements in the development and validation of simulation tools in the last thirty years, such as CFD, have made design analysis faster and more accurate but these tools were not available for the author's research work described in this thesis.

It is beneficial to carry out some development work on axial and tumble swirl in advance of a firing engine being available. To reduce the number of design variants to be tested, comparative measurements can be taken from cylinder heads or flow box models, using a flow bench and swirl measuring equipment. In its simplest form, axial swirl measurement can be assessed using a coaxial aerofoil blade pivoted on the same centreline as the cylinder and mounted at the open end of the cylinder bore. The rotational speed of the blade is read as the angular velocity of the swirl and the ratio of this speed to that of the engine speed equivalent to the test condition, gives the swirl ratio.

In order to remove the intrusion into the cylinder space and the effect of the rotating blade on the air flow, work was reported by Tippelmann (1977) describing the development of a steady-flow impulse torque swirl meter, which has become known commonly as a "Tippelmann" meter. Air is blown through the inlet port in the same way as for mass flow and flow coefficient determination, with an actual or dummy cylinder in place, carrying the impulse swirl meter. This device uses a honeycomb element to straighten the rotating air flow and measures the torque imparted by the swirl using a strain gauge. The torque is a measure of the angular momentum of the air flow through the plane of the flow-straightening element front face that has been imparted to the cylinder air flow by the port and combustion chamber. The measuring principle is illustrated in Figure 46 and the construction of the author's impulse swirl meter is illustrated in Figure 47.

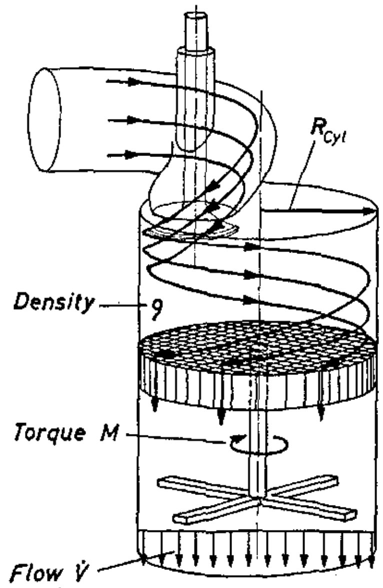



**Figure 46: Measuring principle of the Tippelmann flow rectifier**
*(Reproduced from Tippelmann (1977))*

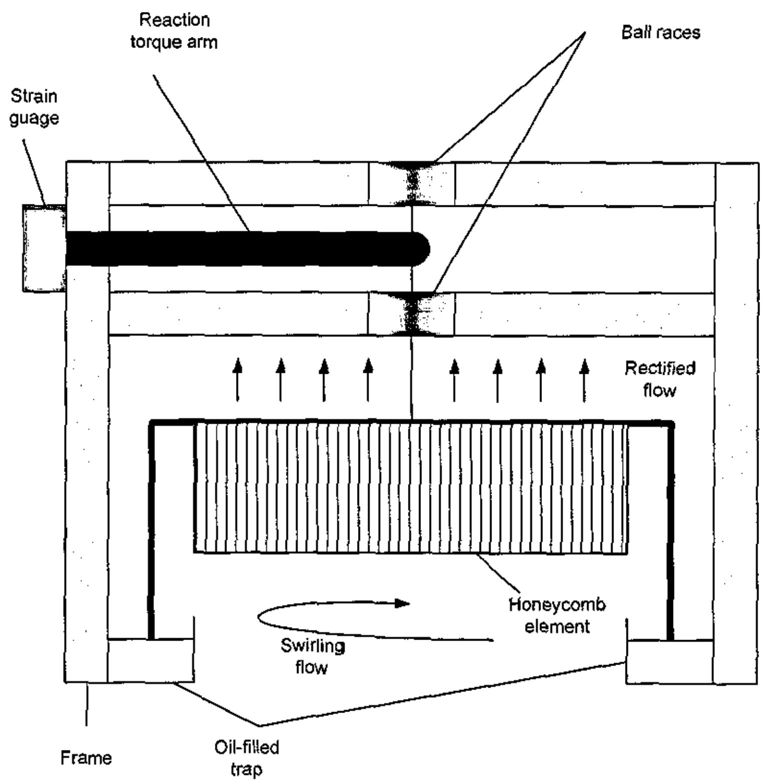



**Figure 47: Schematic diagram to show author's impulse swirl meter (Cussons ISM)**

The impulse swirl meter is designed to measure swirl in the air flow drawn into an engine cylinder generated by the engine inlet port. Earlier methods of measuring swirl such as swirl vane meters or the paddle-wheel type swirl meter were the only tools available prior to Tippelmann. The earlier equipment was dependent on the velocity profile of the air stream, whereas the impulse type of meter overcomes this limitation by responding to the total angular momentum flux in the swirling air flow. The Cusson's ISM impulse swirl meter used by the author, as illustrated in Figure 47, uses a swinging honeycomb matrix which is restrained from rotation by a strain-gauged load cell to totally arrest the angular swirl component, thereby measuring the resultant angular impulse as a torque. It was combined with the air flow bench as illustrated in Figure 48, a schematic of the apparatus used to measure air flow and swirl at Tickford Ltd.

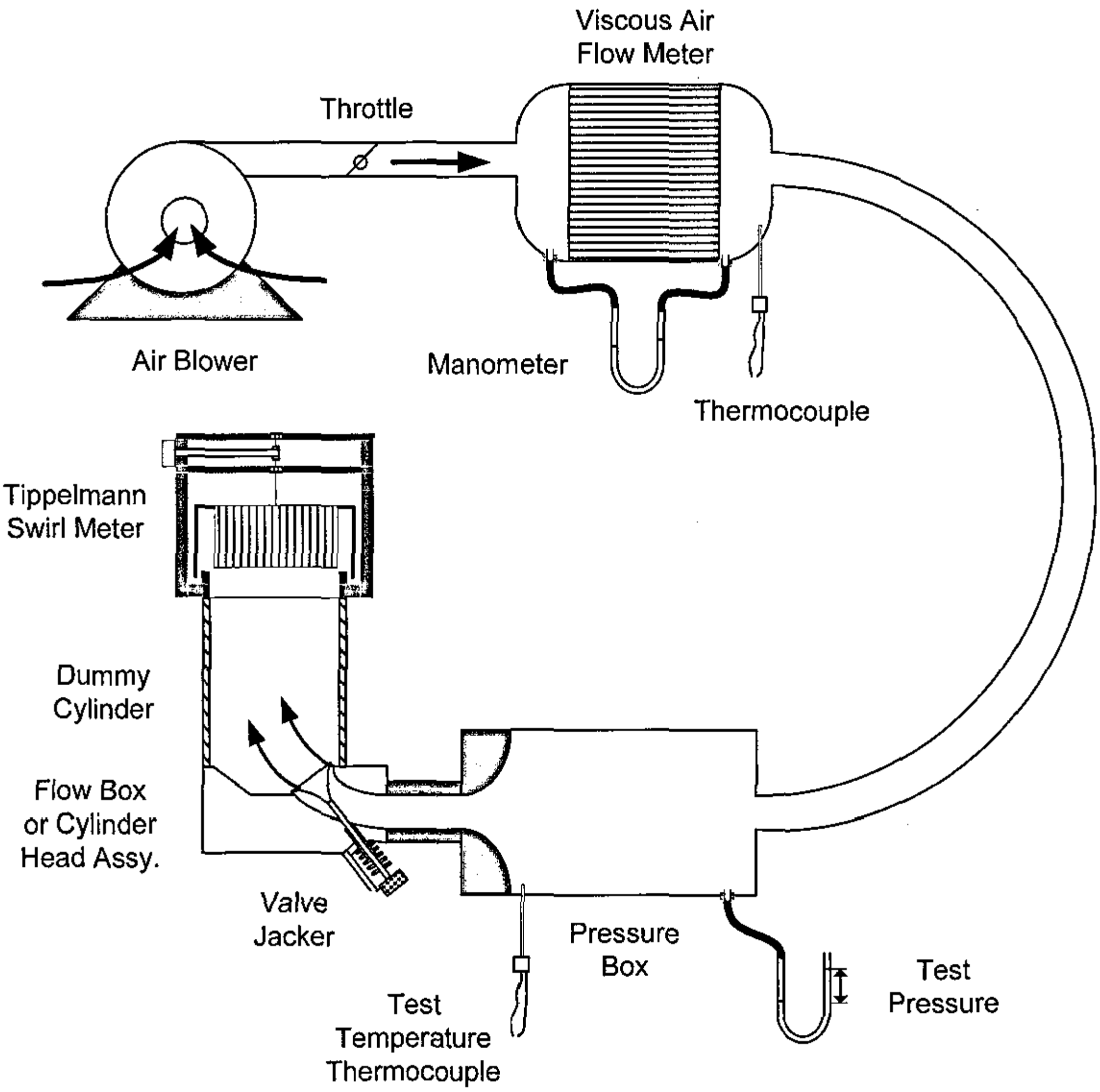



**Figure 48: Schematic of the author's air flow rig apparatus with swirl meter**

The disadvantage of the impulse swirl meter is in the analysis. The traditional "swirl ratio" of any given design at a standard test condition when measured with the paddle wheel mounted on the cylinder centreline, could be derived from the following expression [Heywood (1988e)] to give a direct ratio of the two shaft rotational speeds

$$R_s = \frac{\omega_s}{2\pi N} \tag{28}$$

where $R_s$ is the swirl ratio

$\omega_s$ is the angular velocity of the paddle

$N$ is the rotational speed of the crankshaft equivalent to the test condition generating the air flow

With an indirect, non-contact method of evaluating the swirl momentum, further calculations are required in order to give a comparative value of swirl ratio. This has become desirable due to the wealth of prior art in this field and the understanding of the relative links between combustion measurements and values of swirl ratio derived using the older, direct measurement technique.

During the induction stroke of a reciprocating engine, the inlet valve open area, the flow and the resultant angular momentum in the cylinder, change with crank angle. The same is not the case for the flow and angular momentum in flow rig tests, which are approximately constant for any given inlet valve opening and can be used to estimate engine swirl in the manner after Tippelmann (1977) using an impulse swirl meter to measure an angular torque, M. This method, with further analysis, can be carried out to evaluate the equivalent solid body angular velocity at the end of the induction process [Heywood (1988e)]. This is given in Equation (29), using the crank angles at the start and end of the intake process, the impulse swirl meter torque reading and the mass flow at the valve lift corresponding to each crank angle

$$\varpi_s = \frac{8}{B^2}\left(\int_{\theta_1}^{\theta_2} M\,d\theta\right)\Big/\left(\int_{\theta_1}^{\theta_2} \dot{m}\,d\theta\right) \tag{29}$$

where $\varpi_s$ is the angular velocity of the end of intake flow

| | |
|---|---|
| M | is the impulse swirl meter torque |
| $\dot{m}$ | is the mass flow at the corresponding inlet valve lift |
| $\theta_1$ | is the inlet valve opening angle |
| $\theta_2$ | is the inlet valve closing angle |

Hence the swirl ratio can be calculated in terms of the rig test results and engine geometry, according to Equation (30) given by Heywood (1988e)

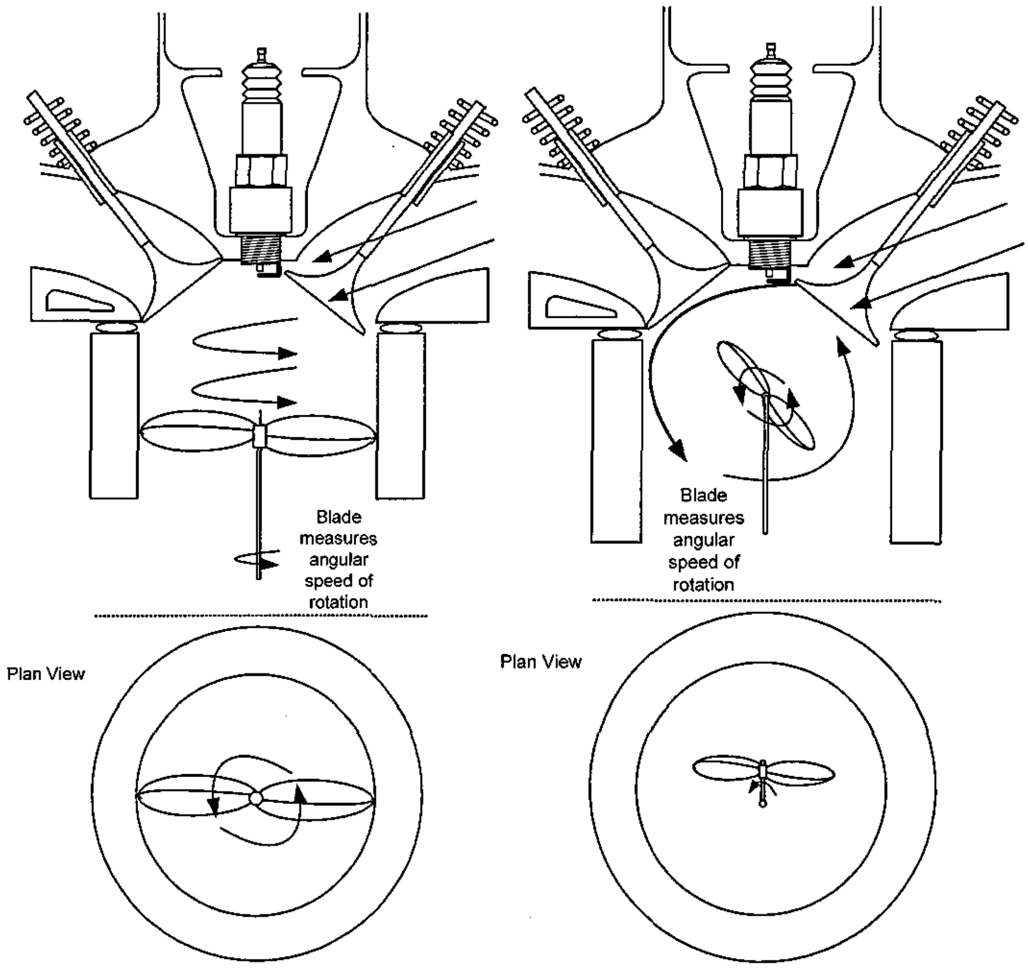
$$R_s = \frac{\omega_s}{2\pi N} = \pi\eta_v B L_s \left( \int_{\theta_1}^{\theta_2} (A_i C_d)\, C_s\, d\theta \right) \Big/ \left( \int_{\theta_1}^{\theta_2} (A_i C_d)\, d\theta \right)^2 \tag{30}$$

where $L_s$ is the stroke of the engine

It is understood that the actual engine swirl patterns are not characterised by this analysis. The motions and flows observed in the intake stroke that are evaluated above, change significantly during compression due to the reduction in volume, the presence of disturbances, e.g. combustion chamber shape and edges, squish areas, valve head protrusions, the spark plug, etc. This results in a loss of the gross charge motion and the development of mixing and microturbulence as described earlier. It is these two factors that have an effect on the combustion process. The static bench methods are useful to give an indication of the total momentum available to contribute to this effect after intake the valves are closed, for any given fixed geometry.

## 4.3 Measurement of Tumble or Barrel Swirl

Since the axis of rotation of tumble swirl is perpendicular to the cylinder axis, measurement of the motion is more complex than for axial swirl. For axial swirl, an aerofoil blade, mounted concentrically with the cylinder, can be used to measure the motion as illustrated in Figure 49a. It is possible to modify the installation of a small rotating aerodynamic blade in such a way that it can be introduced into a tumble field so as to demonstrate or compare the phenomenon using air. One possible variation of this is illustrated in Figure 49b. However, the partially open end of the cylinder for instrument access, the intrusion of the measuring device influencing the flow and its inability to indicate the total bulk air motion means that this method is not ideally suited to scientific research.




(a) Axial Swirl (b)Tumble Swirl

**Figure 49: Diagrams to illustrate "Blade" methods of measuring axial and tumble swirl**

For axial swirl, the measurement of the bulk air motion within the cylinder space below the combustion chamber was already established as the accepted method of characterising the design of port and combustion chamber features in relation to combustion performance. Both the "turbine" blade method and the impulse swirl meter were in use and remain so, but the steady state tests do not reflect the dynamic situation in an engine. Furthermore, they do not measure the turbulent conditions at the point of ignition, which is the critical condition that influences the combustion. However, as a means of comparing potential designs and reducing the amount of physical testing, the steady-state flow bench measurement technique has merit provided that the results are correlated to combustion measurements. This was demonstrated, in the context of tumble or barrel swirl, by the Rover Group, [Chapman *et al* (1991b)] as illustrated in Figure 50 and Figure 51.

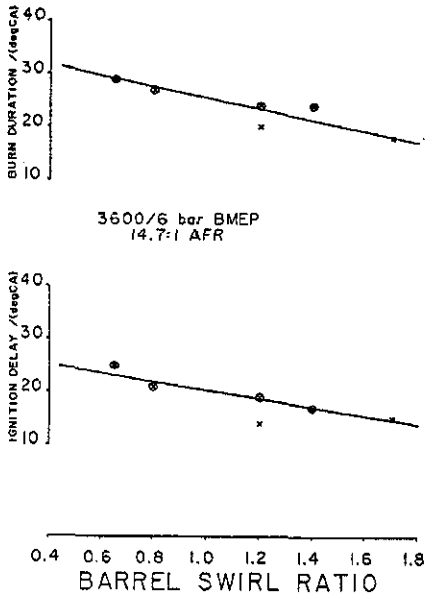



**Figure 50: Correlation between rig-measured swirl ratio (Rs) and burn duration**
*(Reproduced from Chapman et al (1991b))*

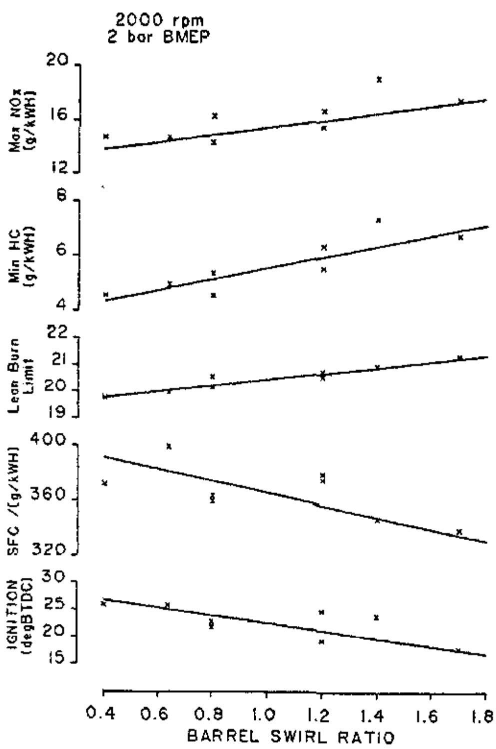



**Figure 51: Influence of rig-measured Rs on part throttle performance measures**
*(Reproduced from Chapman et al (1991b))*

As explained above, the direct measurement of tumble swirl under bench conditions could not be made with the non-contact Tippelmann meter due to its construction. What was required was a means of simulating the piston crown and converting the tumbling bulk motion into a form that could be evaluated in the horizontal plane of the meter. Work was being carried out independently by the author at Tickford and by others at Rover Group, on techniques that would facilitate such a measurement.

Since the tumbling motion is perpendicular to the axis of the cylinder bore and in the plane of the inlet ports, an adapter is required to allow the rig air escaping from the cylinder space to pass through the swirl meter in the plane of measurement. This is provided by removing part of the cylinder wall above the simulated piston crown and creating a passage up which the rotating, exiting air travels. At the upper end of this passage, the impulse swirl meter can be mounted to measure the tumble swirl momentum in a similar fashion to axial swirl, with an apparatus as illustrated in Figure 52.

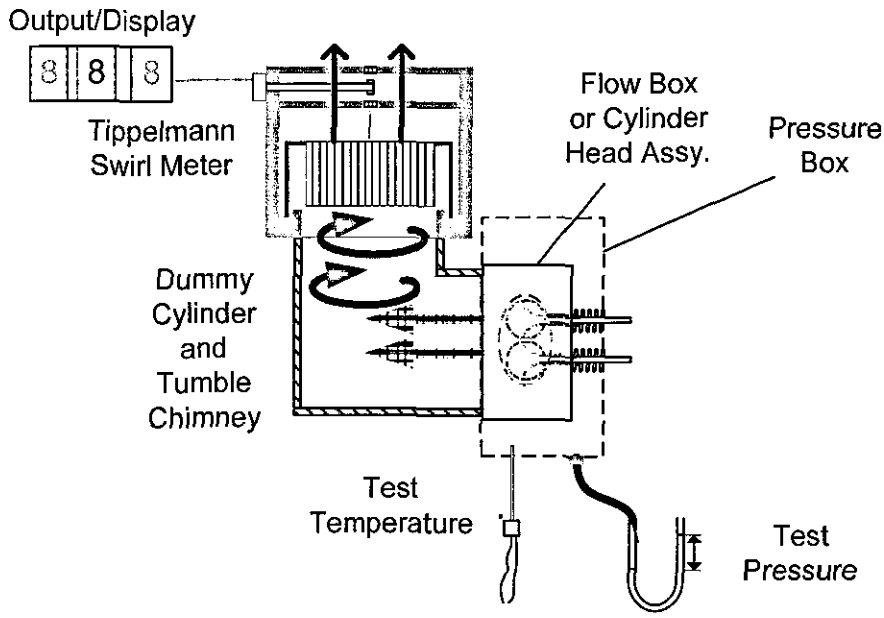



**Figure 52: Schematic of the author's air flow rig adapted for the measurement of tumble**

Gaydon Technology Limited (GTL), part of the Rover Group, and engineers from Austin Rover Group powertrain development team, reported the development of its apparatus [Chapman *et al* (1991b)], which consisted of a square passage, up which the exiting air travels, and three critical dimensions illustrated in Figure 53. They reported the following key rig design guidelines:

1. Use a head face flange thickness (not shown) of 13 mm to allow full generation of the tumble flow rather than direct exit of the air through the square passage;
2. Use a square passage of the same dimensions as the engine cylinder bore diameter;
3. The length of the square passage should be 1·25 times the bore (B) from the cylinder centreline to the mounting face of the impulse swirl meter giving an overall passage length of 1·75 B.

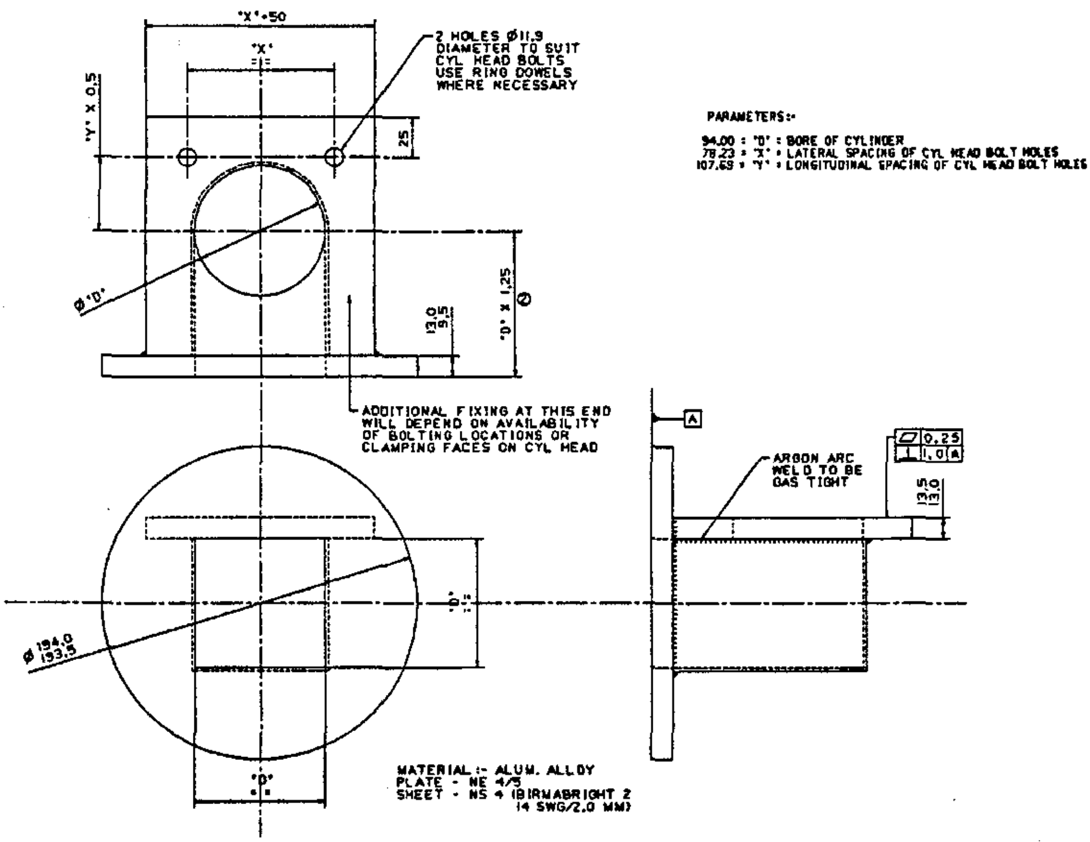



**Figure 53: GTL barrel swirl adapter**
*(Reproduced from Chapman et al (1991b))*

Research at Tickford by the author and colleagues, took a different approach in order to preserve the bulk air momentum created by the inlet port, valve and combustion chamber during tumble measurements on the air flow rig. In the similar apparatus developed independently, the tumble chimney outlet passage was circular so as to minimise edges and corners that would disrupt the flow transition from tumble in the dummy cylinder space, to axial motion in the plane of measurement by the impulse swirl meter. Since the translation of the motion was key to being able to measure the momentum in the cylinder space, preservation of the gross "air flywheel" was important for the measured results to characterise the port design. Another difference in the two adapters was that the Tickford part was nearly always machined in a single piece from a dimensionally stable PVC plastic material as opposed to Aluminium in the case of the Rover/GTL adapter. This was done to provide dimensional precision and consistency, freedom from welded joints, light weight and robustness.

Given that pure tumble is not always produced, and in fact may be deliberately changed by the provision of asymmetric ports, it is beneficial that the tumble chimney adapter mounting angle can be rotated relative to the cylinder head in order that the plane of maximum swirl can be identified. This allowed the author to derive additional information about the air motion beyond that which had previously been reported [Thring (1979), Benjamin (1988), Arcoumanis *et al* (1993)], concerning the axis of the tumbling flow field. This feature is applicable to both types of tumble chimney described above and, after the purchase of a Tickford air flow rig and a training programme, this capability was exploited by a U.S. customer for the development of a 3-valves per cylinder engine in which each of the two inlet ports was designed differently to extract a combination of part load and full load benefits from the compound angle tumble and swirl. Informally in Tickford and the client company, this was called "Swumble".

Tumble can most readily be created in engines with two or three inlet valves when the inlet valves are disposed in a conventional way; that is symmetrically about the cylinder centreline and all on one side of the cylinder head. In this respect, any axial component of swirl about the cylinder centre axis from one valve is balanced out by the contra-rotating component from the other symmetrically opposite inlet valve. Therefore, the only persisting component is the downward tumbling motion perpendicular to the cylinder axis. It is possible to generate some measure of tumble component in 2-valves per cylinder engines where the axis of the incoming charge can be directed across the cylinder centreline. Research on the Tickford air flow bench, with its rotatable plane of measurement, identified the overall resultant compound motion and the relative contribution of the axial and tumble components.

At this point it is important to restate that it is not the motion of the incoming charge that is significant but rather the motion of the charge derived from convective flow effects in addition to the microturbulence at the point of ignition, which is the end result of the induction momentum. This phenomenon of compound angle induction-induced air motion was one of the features of the development of the Aston Martin V8 engine both for competition and for enhanced road engines. Although not the main thrust of the development programme described by the author [Bale (1984)] for which mechanical aspects for durability in 24h racing were paramount, the combustion performance of the Aston Martin engine was found to be very good from both the output and emissions standpoints. The key elements in this achievement were maintaining a modest inlet port diameter, to maintain a means inlet gas velocity of the order of 100 $m.s^{-1}$ and a port floor design and port to valve angle that created some tumble motion to complement the natural swirl of the 2-valves per cylinder design. These factors, coupled with the almost hemispherical combustion chamber, led to the high performance and low emissions variants of the engine, which were developed without the benefit of air flow or air motion measurement. In respect of the low emissions variant, the combustion system was so good that the Aston Martin V8 was one of the last sports car engines to be emissions compliant with one specification in all the fifty states of the USA with four, twin-barrel carburettors and open-loop three-way catalysts. It retained this level of equipment up to the 1980 model year when it was fitted with electronic port injection in order to allow the full use of feedback control and stoichiometric catalyst operation that was described in Chapter 2.

## 4.4 Closing Remarks

Much of the emphasis of the author's work described in this chapter was focused on achieving the maximum power output from a given swept volume of engine with the best volumetric efficiency. Work carried out on the flow bench, and with apparatus to quantify the flow and tumble characteristics of each port design, allowed the selection of promising designs to be built for engine testing. Some of the author's techniques and apparatus were shown to be novel. The controlled use of the bench techniques discussed here, proved effective for compressing engine development time.

The studies of combustion, power output and fuel consumption in these running engines confirmed the trends seen in the bench results and built on the correlation between flow bench and combustion measurements. The author's work further demonstrated that the better combustion systems not only had the capability of more optimal heat release but that they were also capable of low exhaust emissions and of running with dilute charge.

The utilisation of these characteristics together with the engine design considerations for volume production, are described in Chapter 5. The different ways of designing 5-valves per cylinder engines will be discussed along with a practicable means of harnessing intake wave effects to improve volumetric efficiency across the engine speed range.

# Chapter 5.

# MULTIVALVE CYLINDER HEAD DESIGNS AND LIMITATIONS

## 5.1 Introduction

This chapter discusses the development requirements for racing and production engines and describes the introduction of more sophisticated engine designs for production applications. It also compares the two main 5-valves per cylinder design concepts with respect to their flow and combustion performance relative to hardware complication. Additional explanation is given with respect to work conducted by the author and his team to accommodate production constraints and innovative research and development techniques for optimising engine breathing.

The technical and commercial drivers for on-road and high performance competition engines are often the same. They may be capacity limited (i.e. swept volume is dictated by regulation in motorsport and by taxation or market-image for road cars) or subject to an engineering form of "peer-group pressure" where one manufacturer's introduction of a technology leads to proliferation of that technology by others. This situation was certainly true in the 1980s when 4-valves per cylinder engines were widely introduced by Japanese car manufacturers despite the poorer specific torque and fuel economy they sometimes produced. These spawned a range of engines in Europe from other manufacturers like Ford, General Motors (Opel) and Rover, which considered that there was a market need for a 4-valves per cylinder engine in their ranges. The increased component cost and complexity of the 4-valves per cylinder engine were largely ignored in the quest for market position and the different output characteristics of the engines soon became apparent. It was common to find maximum torque speeds occurring higher up the engine speed range and "double peak" torque curves were seen from engines such as the Rover 2·0 litre M series and the Ford 2·5 litre "Modular" V6. The latter is illustrated as an example in Figure 54.

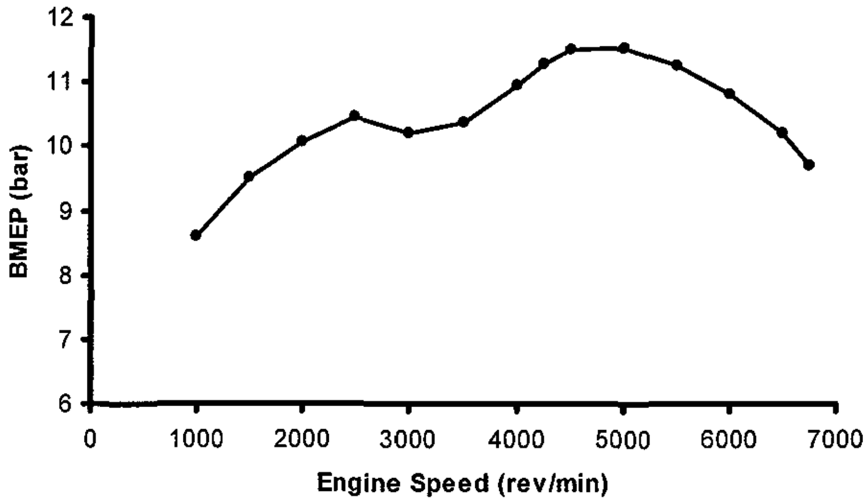



**Figure 54: Measured BMEP curve from 2·5l Ford V6 4-valves per cylinder engine**

## 5.2 Different Approaches to 5-Valves per Cylinder Head Design

For the reasons set out earlier in this thesis, ample valve area can yield both power and torque advantages when coupled with appropriate valve timings. Yamaha was the first engine manufacturer of the current era to develop engines with 5-valves per cylinder for its range of high performance motorcycles [Aoi *et al* (1986]. The design was later used in passenger cars by Toyota [Bradshaw (undated)], with an example illustrated in Figure 55, and then Ferrari with the F355. Audi introduced the first high volume car engine with 5-valves per cylinder in 1995 with a 1·8 litre engine in the A4 model, for which they paid a royalty to Yamaha for use of its design.

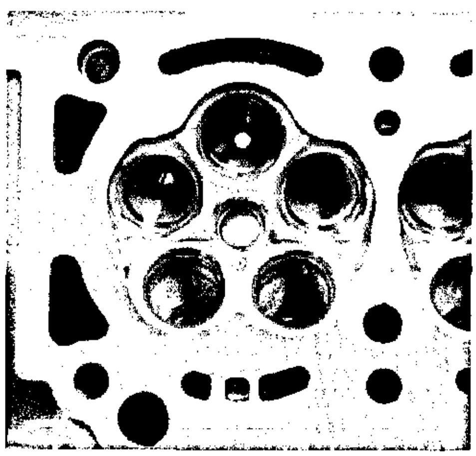


**Figure 55: Toyota 4A-GE 20v 1·6L released in Japan in June 1991**
*(Photograph courtesy of Tickford Ltd.)*

The Toyota 4A-GE 20-valve engine produced 118 kW at 7400 rev.min$^{-1}$ compared with 103 kW at 7200 rev.min$^{-1}$ for the previous 4-valves per cylinder engine, but with different intake and exhaust manifolds, different valve included angles and the advantages given by variable valve timing. The maximum torque of 162 Nm at 5200 rev.min$^{-1}$ compared with 141 Nm at 6000 rev.min$^{-1}$ for the 4-valves per cylinder version that it replaced. This represented an increase from 8·8 to 10·2 bar BMEP. The cylinder head of the 4A-GE engine is discussed in more detail in Section 5.6.

The Yamaha 5-valves per cylinder design, as also used by Toyota, Ferrari and Audi, is mechanically elegant, as illustrated in Figure 56. It has a conventional twin overhead camshaft construction, built round a 4-valves per cylinder layout, with direct acting bucket tappets on all the valves.

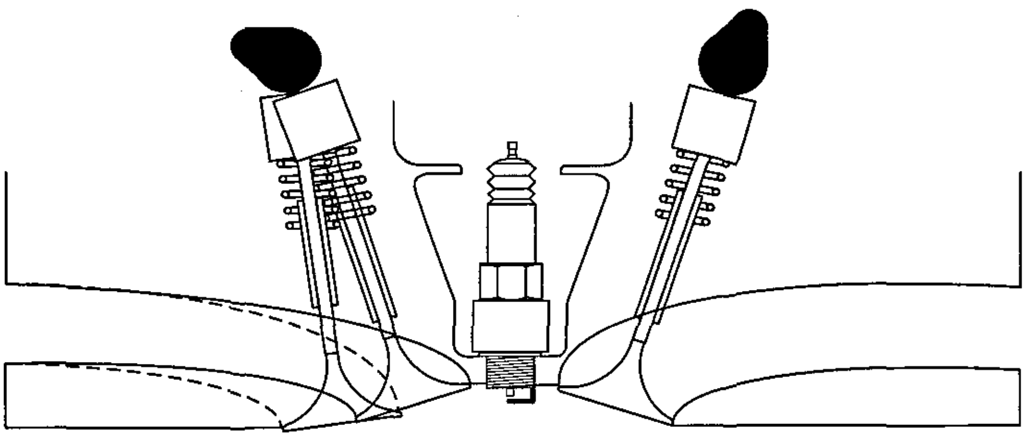


**Figure 56: Diagrammatic section through typical Yamaha 5-valves per cylinder design (only one of the outer pair of intake valves is shown)**

The intake camshaft has three lobes on a common shaft with the result that the singleton (centre) intake valve has to operate at a more vertical angle than the other two. Figure 57 shows the differences that this design means to the port arrangements for the outer pair (shown as solid) and the single central port (shown with a dashed outline).

In the context of the preceding discussion and Chapter 4, it can be seen that the outer pair of intake valves will act in a similar fashion to the intake valves of a 4-valves per cylinder engine but the central valve will have a port shape that is as not as suitable for generating tumble.

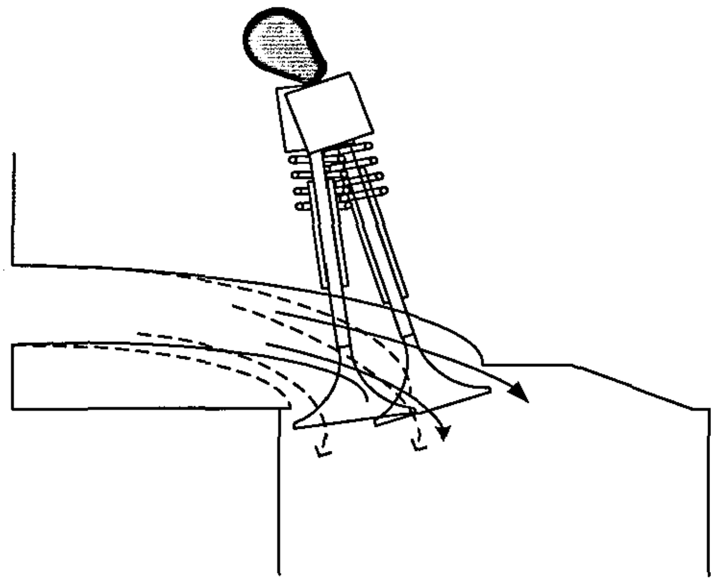


**Figure 57: Tumble generating flows in Yamaha 5-valves per cylinder design**

The flow lines in Figure 57 illustrate how the tumble generating potential decreases the further the entry point of the charge becomes from the cylinder centre. The most active flow (large arrowhead) is that over the top of the twin intakes (solid port outline), possibly with some support from the flow that can be provoked into breaking away from the port floor of these two valves (small arrowhead). Due to the downward shape of the central port, it is hard to imagine any contribution to the tumble momentum of the other two valves (solid lines with arrowheads) from the flow out of this valve (dashed lines with arrowheads). Airflow bench tests by the author's team suggested that the flow from this central valve actually reduced the tumble motion created by the other two valves, due to the "contra-rotating" flow from the upright port and valve.

Sykes (1995), a senior member of the author's team, went on to describe the comparison between the Tickford and Yamaha designs in his SAE paper, as illustrated in Figure 58. The upper part of the diagram shows the angle of the central singleton intake valve and the suggested paths of the outflow from the top and bottom of the valve in section. The middle diagrams show the flows from the pair of outer valves and in each case the axis of the other valves are shown. Solid lines indicate tumble-supporting flows and dotted lines show tumble-damping flows. The length of the lines is indicative of the strength of the effect. The lowest part of the diagram suggests a summation of the tumble-supporting and tumble-damping flows.

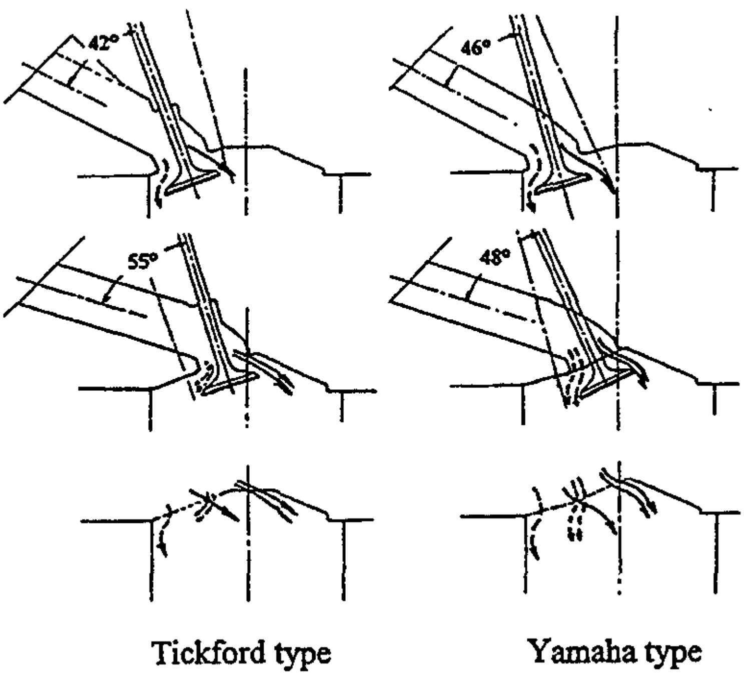



**Figure 58: Comparison of predominant flows**
*(Reproduced from Sykes (1995))*

Flow bench and engine testing confirmed that although the Yamaha design had increased valve area and adequate flow coefficients to improve the volumetric efficiency of the engine and reduce the Mach Index, the resultant engine power output was not similarly increased in the amounts that the valve area change suggested. This finding is discussed further in Section 5.5, which compares flow bench measurements of the Toyota 4A-GE cylinder head that uses the Yamaha design, with a Tickford 5-valves per cylinder flow box for a client engine and a benchmark 4-valves per cylinder head designed by Ford.

A further observation of the Yamaha design is that the open, pent roof part of the combustion chamber changes its characteristic as it reaches the central intake valve where the combustion chamber tends towards more of a squish area, with a reducing cross section and large surface areas adjacent to it. The layout would suggest a slowing flame front with an increase in heat losses in this area of the chamber as illustrated in Figure 59 and Figure 60.

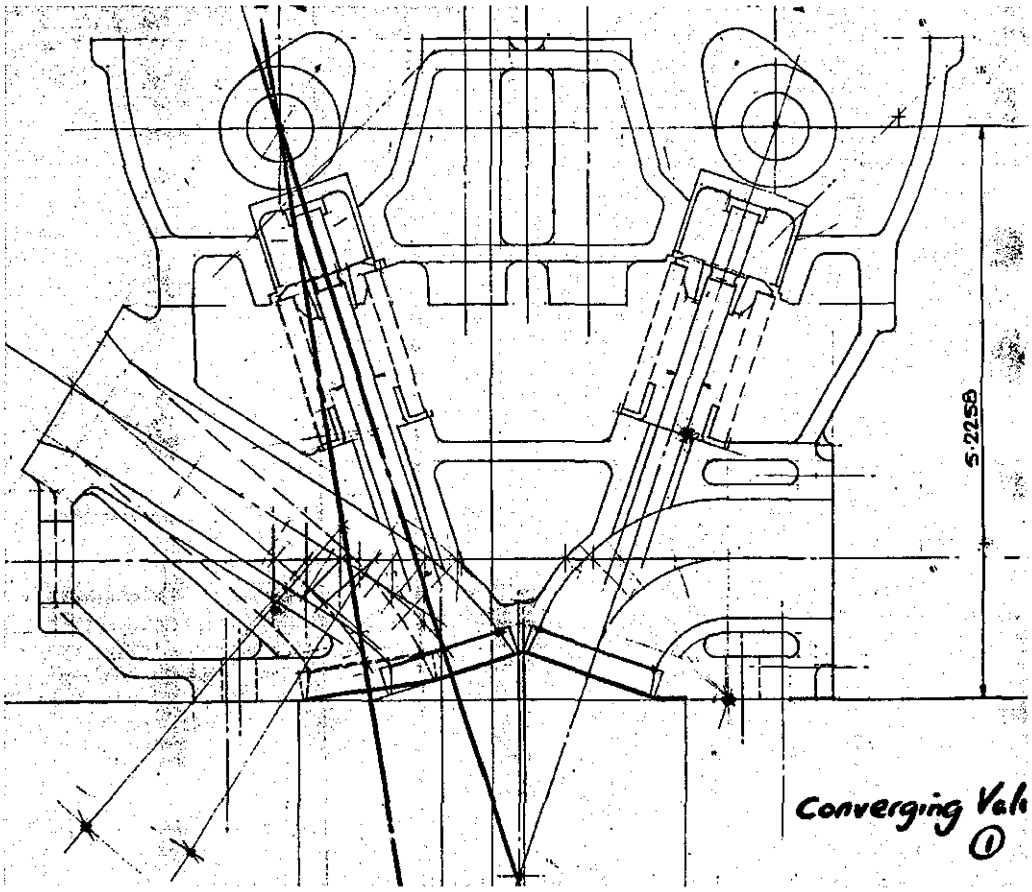



**Figure 59: Yamaha 5-valves per cylinder geometry with converging valve angles**
*(Reproduced courtesy of Tickford Ltd.)*

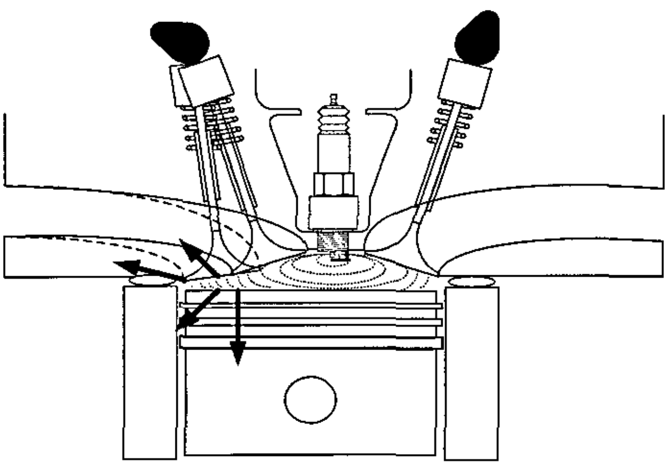


**Figure 60: Yamaha layout indicating flame front and heat loss paths**

The approach taken at Tickford was different. The author's primary objective was a port, valve and combustion chamber arrangement that would be capable of high flow, significant tumble and high burn rates. Since the initial development was conducted on a competition engine, air-fuel ratio was always optimised to best torque and the objective was simply to get high volumetric and combustion efficiencies in order to liberate the maximum power from the limited swept volume permitted in the regulations for Formula One and Group C1 endurance sports car racing (both were then a 3·5 litre swept volume maximum). The arrangement of the valves and valve train is illustrated in Figure 61.

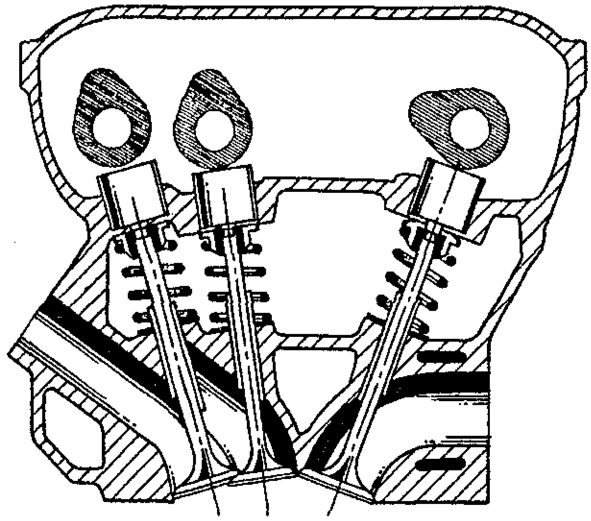


**Figure 61: Tickford Formula One 5-valves per cylinder design**

It can clearly be seen in Figure 61, that the three intake valves form an open chamber, without a constrained area under the intake valves, in order to maintain flame front speed and reduce heat losses compared with the Yamaha design. Combining this with the other essential feature of the Tickford design, which was that all three intake ports were capable of generating tumble, compared with the central port in the Yamaha design that reduced induction-induced tumble activity, demonstrated why the novel Tickford designed head was able to burn large amounts of charge quickly and efficiently.

The author's initial research and development was conducted on a 3·3 litre, 90 mm bore, Cosworth DFL engine as used in sports car racing. For maximum simplicity and reliability, and without excessive concern for unit price, the engines were designed and built with three camshafts per head as illustrated in Figure 61. This was the simplest and stiffest way to accommodate the divergent inlet valve stem angles away from the combustion chamber. It was recognised that this arrangement moved away from the mechanical simplicity of the Yamaha design but it was necessary in order to capture the combustion benefits that resulted from high tumble and low heat losses in a compact but open chamber.

The project was described in Chapter 3 and detailed in the author's paper presented at the FISITA congress in Turin in May 1990 [Bale and Downing (1990)]. The paper included references to the valve timing optimisation, where extended camshaft opening periods were avoided, leading to a good spread of torque and power, and some description of the valve dynamics issues that were addressed. By the time the paper was presented, patents had been applied for in all the major automotive territories and Camel Team Lotus had commissioned the application of the new 5-valves per cylinder design to its Judd 3·5 litre engine for the 1989-90 Formula One season for very sound reasons.

Due to the very high costs of development, there have always been a limited number of manufacturers of engines for Formula One and this meant that a number of teams used nominally identical power units. Having successfully developed and proved the benefits of its 5-valve design on the Cosworth DFL base engine, the author's company, Tickford, made the technology, and its power increase, available to racing teams. Given that lap times just 1% slower than the "pole position" (fastest practice lap time) were often enough to demote a car to the back of the grid after qualifying and certainly to be lapped by the leader several times in the race, the availability of up to 5% additional power from the 5-valves per cylinder engine was a key competitive advantage. This was taken up by Team Lotus in order to provide it with a cost-effective derivative of the Judd, 4-valves per cylinder, "customer-spec" engine, which was also supplied to other teams.

As soon as the 5-valve engine was available, the Lotus drivers, Nelson Piquet and, more particularly, Satoro Nakajima, usually the slower of the two, benefited from the higher power of the 5-valves per cylinder engine to improve their position on a number of occasions. However, because the more powerful 5-valves per cylinder engine was introduced mid-season when it was not possible to increase the fuel tank capacity in the Lotus chassis, on circuits where fuel consumption was a potential issue, the less powerful 4-valves per cylinder engines were used in some races to ensure that the cars completed the course with the available fuel. Formula One rules then allowed Lotus to run 5-valves per cylinder engines in qualifying so as to improve their drivers' grid positions. The size of the fuel tank would have been corrected in the following season but for the fact that Lotus was offered a "works" engine by Honda, which replaced the Judd engine completely.

## 5.3 Application to Road Engines and Dilute Charge

The research and development of 5-valves per cylinder competition engines that had begun to be reported [Bale and Downing (1990)], demonstrated that a number of key lessons were ready to carry into mainstream automotive engine design. These included:

1. Increased available valve area for increased performance;
2. The ability to produce high output without long overlap valve events;
3. Increased understanding of the parameters that produce tumble;
4. A rig tool for rapidly assessing different designs on the air flow rig, measuring air flow and air motion;
5. An ability to produce a high output, fast burning, engine capable of supporting high charge dilutions without loss of combustion stability.

Development work that was already being undertaken by the author's team for Ford of Australia on the creation of performance engine derivatives such as the Falcon XR6, were enhanced by the application of tumble and port floor features that were balanced between cost-effectiveness (minimum tooling changes) and improved performance and economy [Bale and Sykes (1993)].

The particular tool developed by the author's team and used for rapid assessment of proposed port shapes was described in the AE Asia Technical Congress paper [Bale and Sykes (1995)] and at the SAE World Congress [Sykes (1995)] and is illustrated in Figure 62. No evidence has been found in the literature to suggest that this was anything but a novel approach to this research.

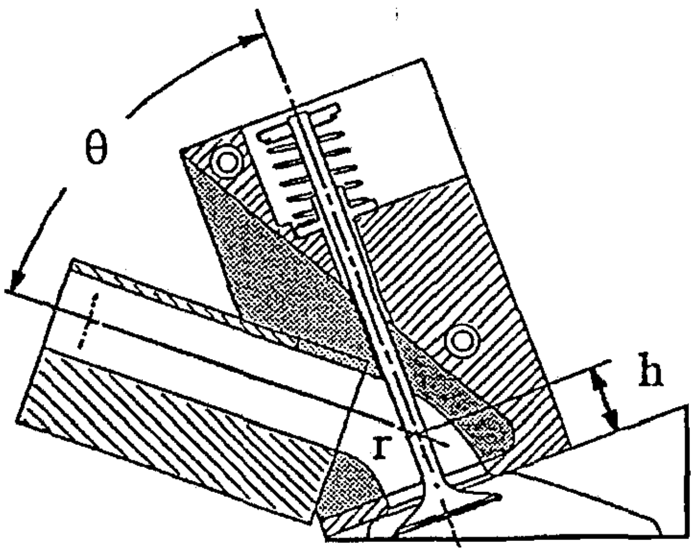



**Figure 62: Tumble development tool invented at Tickford**

The port configuration tool was made in two parts, joined together at a pivot and a fixed point shown as double circles in Figure 62. The arrangement allowed the port to valve angle (θ) to be changed as well as the height of the intercept (h) of the port and valve centrelines from the valve seat face. Independently of these, the port floor radius (r) could be changed and this was achieved by making the tool in two halves with the port floor radius routed out by a circular cutter prior to assembling the compete tool for rig testing. A programme of work was undertaken to systematically change the three main parameters of r, h and θ independently, to determine the effects on tumble and air flow. The main port diameter and the valve seat remained constant with a typical area relationship between them. Using this apparatus, some fundamental guidelines were developed for the design of inlet ports to generate high tumble ratio with the maximum possible air flow. These were:

1. The sharpness of the radius (r) between the port floor and the seat throat was found to be very significant in promoting the breakaway of the inlet flow from the port floor to exit over the top of the valve to create tumble;
2. Small intercept height (h) promoted tumble since more port flow is directed across the top of the valve head at all valve openings;
3. Increasing the port to valve angle (θ) from 40 and 50 degrees promoted a rapid increase in the measured tumble ratio.

The study of these parameters helped the author and his team to produce designs that retained flow efficiency, a reduction in which was the inevitable consequence of high activity ports, be they swirl, tumble or through the use of port deactivation.

Bench measurements of a wide variety of different cylinder heads and flow boxes were made during the research so that the effect of intake port and combustion chamber shapes could be assessed. The database of mean flow and tumble figures was maintained and added to so that the performance of any given design could be compared with others. From the selection of results illustrated in Figure 63, it can be seen that there are many cylinder head designs that are nowhere near the optimum on either scale and also that there is an outer envelope, which confirms that, as tumble increases, the maximum air flow potential decreases. The 5-valves per cylinder design for the Ford V6 was clearly in line with this optimised envelope along which increased flow could be traded off against reduced tumble.

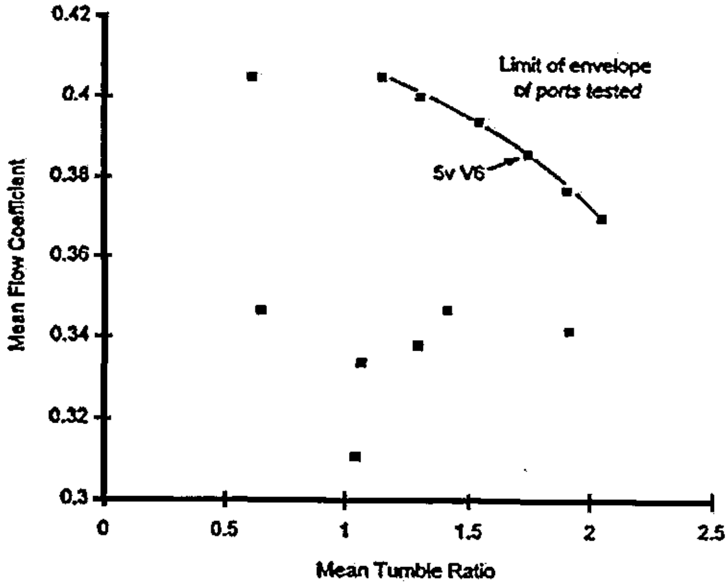



**Figure 63: Mean flow coefficient against mean tumble ratio for a selection of cylinder heads and flow boxes.**

The requirements for a compact design and minimum piece cost meant that a production approach to the valve gear was necessary for the 5-valves per cylinder Ford V6 engine. The use of three camshafts per head with gear-drive was not acceptable and yet the advantageous valve arrangement was fundamental to the function. The chosen solution was a twin overhead camshaft design that used much of the standard Ford hardware, including direct acting bucket tappets on the outer pair of inlet valves, and indirect actuation of the central intake valve via a rocker arm with hydraulic lash adjustment at the pivot post, as illustrated in Figures 64 and 65.

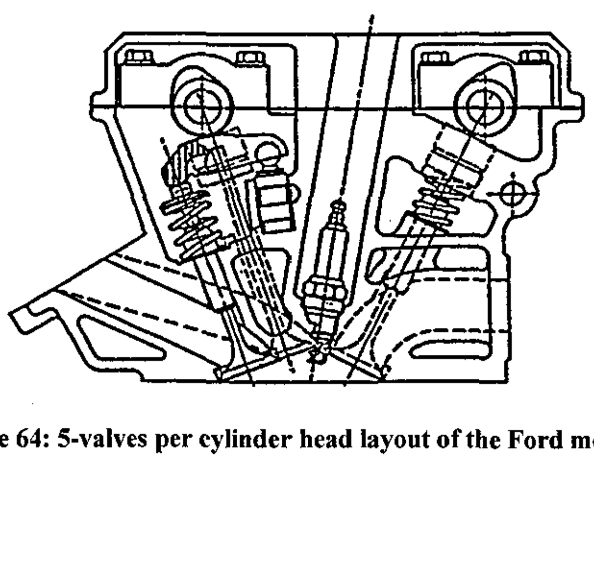


**Figure 64: 5-valves per cylinder head layout of the Ford modular V6**

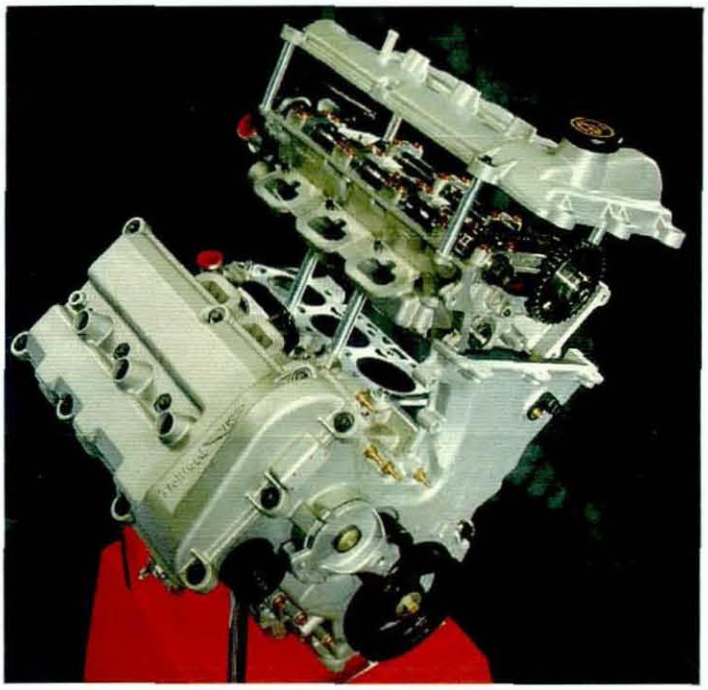


**Figure 65: Ford 30v modular V6**

The fundamental concept of working from an idealised combustion chamber out to the valves and valve gear was carried over from the Formula One designs and a similar 3·5% gain in valve area was achieved. The production-feasible combustion chamber and ports are illustrated in Figures 66 and 67.

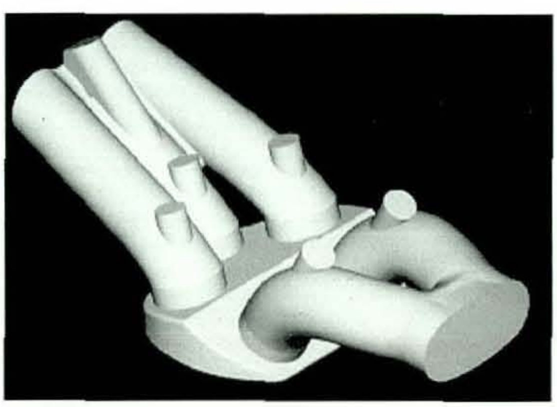


**Figure 66: 5-valves per cylinder port arrangement for Ford 2·5 litre V6 - CAD model**
*(Courtesy of Tickford Ltd.)*

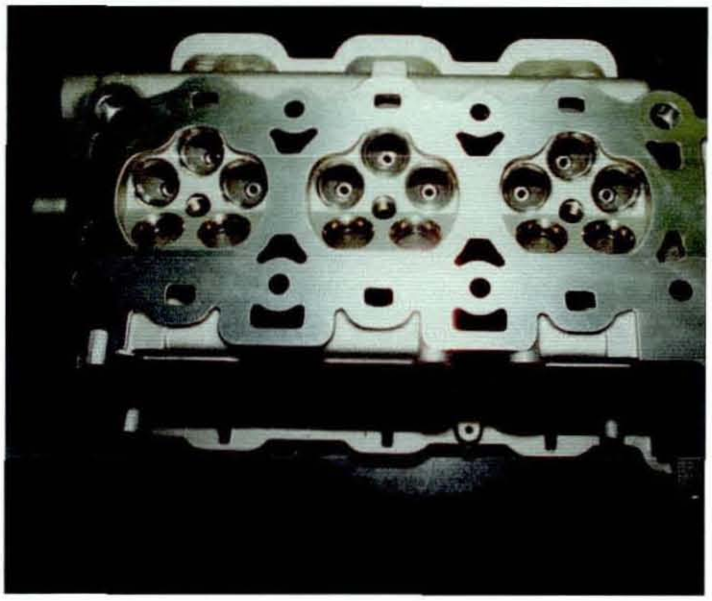


**Figure 67: 5-valves per cylinder Ford 2·5 litre V6 - Combustion chambers**

Ford specified a compact layout for this technology demonstration project, with a target to reduce the vertical height of the camshaft covers by 40 mm to facilitate a lower bonnet (or hood) line. The reduction was achieved in the final design although this meant that it was not possible to retain roller-finger followers, which were used in the original 4-valves per cylinder layout. Even though this was a retrograde step from the point of valve gear frictional losses, calculation of the valve spring requirements indicated that the lower individual inertia of the three intake valves could be controlled by lower load springs than those used in the standard production engine. This partially mitigated the sliding contact arrangement for valve actuation and reduced the contact stress between the camshaft nose and the followers. Comparative valve train testing at Ford Advanced Powertrain Engineering Office (APTEO) in Allen Park, Michigan, showed that the frictional losses of the 5-valves per cylinder heads were acceptable though actual values remained confidential. The overall package requirement was met and is illustrated in Figure 68, as a comparison with the standard 4-valves per cylinder engine (with roller followers and IMRC), and as a complete assembly in Figure 69.

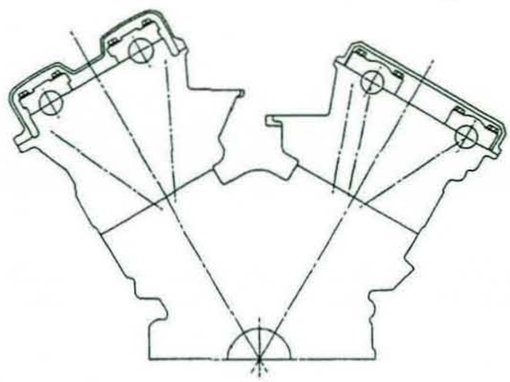



**Figure 68: Comparison of cylinder head profiles**
*(Reproduced from Sykes (1995))*

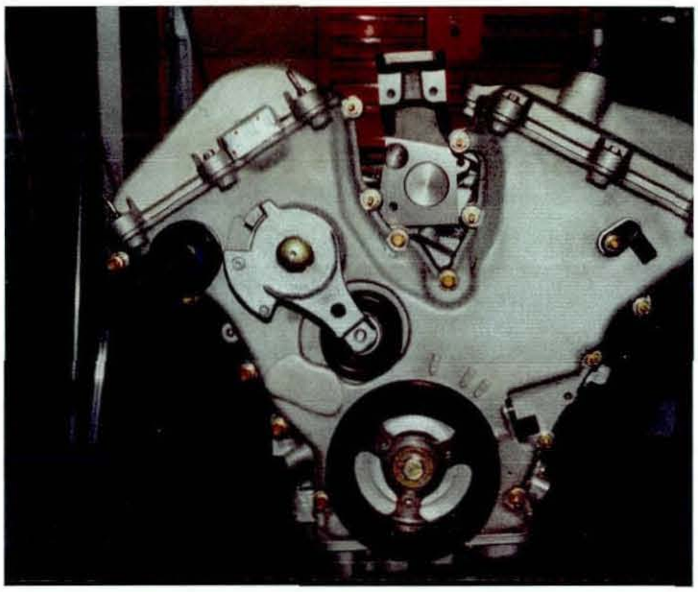


**Figure 69: Ford 30v modular V6 engine – front view**

One of the objectives of the Ford 5-valves per cylinder road engine project was to prove that part load burn rate and EGR tolerance could be matched or bettered without the use of IMRC which used additional and costly hardware to achieve port deactivation and hence create axial swirl under part load conditions. Considerable work was conducted to improve the fuel spray distribution and EGR calibration [Sykes (1995)] with the end result that the tumbling 5-valves per cylinder engine closely matched the 0-10% burn angles of the base 4-valves per cylinder engine. The 10-90% burn had a slightly different response rate to EGR concentrations, as illustrated in Figure 70. This shows that the burn rate was slightly higher up to 20% EGR. Combustion stability was measured by the coefficient of variation (CoV), being the standard deviation of the IMEP for the cycle data recorded, divided by the mean of the sample. The CoV of IMEP for the base and the 5-valves per cylinder engines were almost identical up to 20% EGR. This has to be taken in context as the 5-valves per cylinder engine produced 10-12% more torque than the standard engine over the important mid-speed range and a maximum power increase of approximately 8%. This derived from a geometrical increase in valve area of 3·5%.

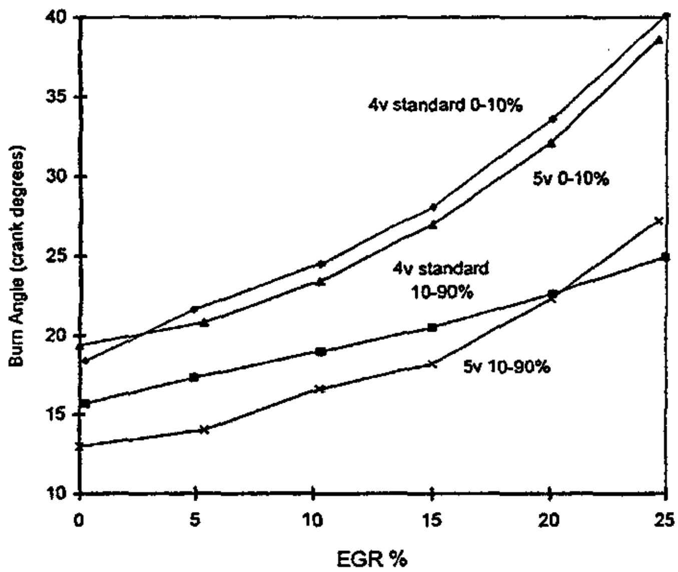



**Figure 70: Part load burn rates 5-valves per cylinder vs. 4-valves per cylinder**
*(Reproduced from Sykes (1995))*

## 5.4 Production Constraints

At the start of the 5-valve cylinder head project for the 4-valves per cylinder Ford Modular V6, a small delegation of engine design and development engineers from Tickford, led by the author, visited the Ford engine assembly plant in Romeo, Michigan, in order to gather an appreciation of the way in which the engine would be built in volume. This was, in itself, a marketing coup as no other engine design consultancy had ever asked for such a visit prior to starting work on a design for Ford. The realities of tool access and automated assembly were readily acquired and it was clear that the design manual, which upheld the ideals for clearances, access and methods, could be compromised in some areas if technical justifications were sufficient.

Another significant constraint on the design of production engines is tooling. Here the philosophy that might be taken for a small run of competition engines with maximum output, has to be changed completely. Designs must accommodate production tolerances, fast assembly without hand finishing and the fact that every Cent spent on tooling changes or replacement will be scrutinised and argued. After five years of consulting in the mainstream engine design arena, the author's engine team at Tickford had successfully made all the necessary transitions that were required for designing production engines, as opposed to pure competition engines. This was typified by the visit to Romeo mentioned above. A clear understanding of the factors affecting engine performance is necessary when moving away from the optimum design to that which can economically be manufactured and put into production.

This competence had already been demonstrated in the development of the port for the 2-valves per cylinder Ford Falcon XR6 engine [Bale and Sykes (1993), Bale and Sykes (1995)]. The desired port configuration for optimum performance, illustrated in Figure 71, would have required a new cylinder head coolant jacket core in the main head casting, in order to ensure adequate metal thickness in the area of the 50 mm radius at the port floor, in addition to the new port core itself. The additional coolant core change would have incurred a significant and very undesirable programme cost.

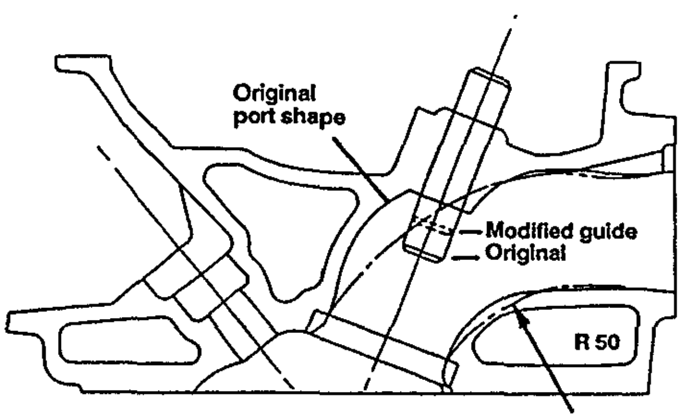



**Figure 71: Low line gradual expansion inlet port superimposed on the original shape with guide modification**

Data from the air flow bench development of the port showed inferior flow performance of the port with a reduced radius but since the production tooling cost consequences were so significant, a test engine was built with an alternative 30 mm design. In dynamic flow engine test conditions, the 30 mm port floor radius, as illustrated in Figure 72, actually showed a performance improvement over the standard design. This is just one of the author's experiences that have shown that port development carried out on a steady state air flow bench can lead to imperfect results with a fired multi-cylinder engine running under dynamic conditions.

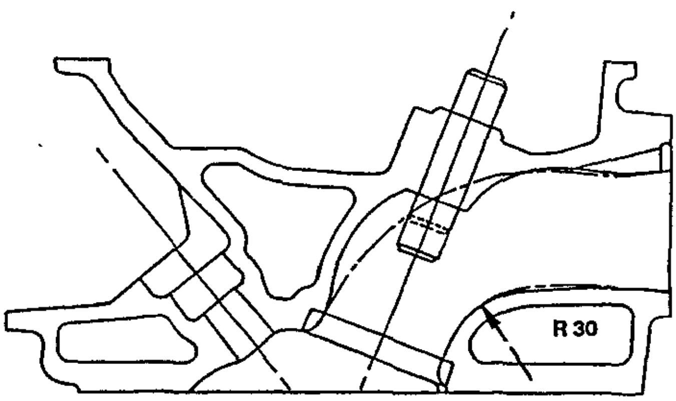



**Figure 72: Original and productionised intake port and valve guide design**

Small modifications to the exhaust port to prevent flow separation between the valve seat and the port floor, by raising the port floor and providing a larger radius, meant that the standard head casting only required relatively inexpensive changes to the port core shapes. Finally, shortening of the valve guide, also illustrated in Figures 71 and 72, optimised port flow and valve stem support without major impact on manufacturing.

Other modifications to the XR6 engine, including the camshaft profile (but not lift as this had implications on valve springs and cylinder head machining) resulted in an engine that gave 95% of the maximum that could be expected of a road engine of that swept volume. Comparing the performance of the modified 2-valves per cylinder Ford engine against that of a competitor engine of the same 4·0 litre swept volume and six cylinders, showed that maximum torque of the XR6 was just 3% less but with far lower cost due to the competitor having twin overhead camshafts and 4-valves per cylinder. The comparison is illustrated in Figure 73.

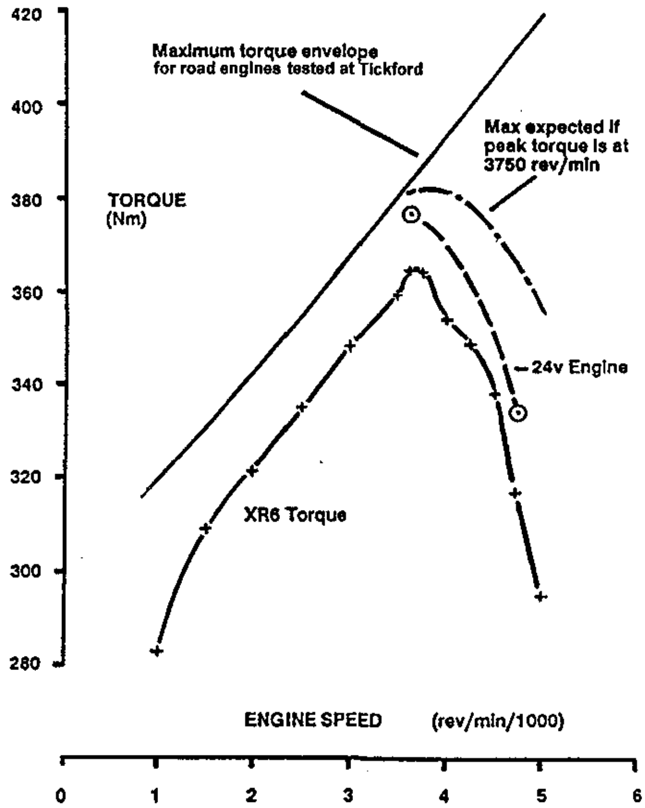



**Figure 73: Comparison of XR6 performance against the likely maximum and a 4-valves per cylinder competitor**

## 5.5 5-Valves per Cylinder V8 Design - Case Study

Based on the author's 5-valves per cylinder research and the successful development of the 5-valves per cylinder, 2·5 litre V6 technology demonstrator for Ford APTEO, the author won an order to design and supply a 5-valve flow box to suit a client's 3·4 litre V8 engine. This work was conducted to assess air flow performance and design feasibility. The specific power target was 67·1 $kW.l^{-1}$ at an engine speed of 7500 $rev.min^{-1}$. A realistic port layout was required along with a design scheme for a suitable valve operating mechanism.

To support this activity, the author reported a parallel evaluation of the 5-valves per cylinder Toyota 4A-GE cylinder head design and compared the performance of the ports. This production design, using the Yamaha concept, was mentioned in Section

5.2 and illustrated in Figure 55. More detailed description of the Toyota design is given in the following Section 5.6.

Since the performance targets specified by the client in terms of combustion stability, combustion rate and idle quality were similar to those of the European-designed Ford Zeta engine (later re-badged Zetec), further comparative flow and tumble data was taken from the Zeta engine. The 1·8 litre Zeta was used, being the variant of the design with least compromise in the design of the cylinder head and combustion chamber, compared to the 1·6 and 2·0 litre versions. Although the combustion characteristics were similar, the specific power output required of the 5-valves per cylinder V8 design was 25% higher than that of the 53·5 $kW.l^{-1}$ Zeta engine.

This new road-going 5-valves per cylinder design scheme was based on a production 3·4 litre V8 cylinder block which had a bore diameter of 82·4 mm and a stroke of 79·5 mm. The main physical target placed on the design was to achieve the minimum camshaft centre spacing in order to allow for the possibility of a future variable valve timing mechanism. Scope for moving the manifold face, and hence port angle, was limited due to the requirement to use the existing, dual length inlet manifold with minimum modification.

Since this was a confidential client project, not all the results can be included in this thesis. However, most of the relevant aspects of the design are discussed and illustrated, with comparative data for the author's design, a Toyota 5-valve design and a benchmark 4-valve per cylinder head.

### 5.5.1 *Inlet ports*

The inlet port throat diameter was designed to give a nominal MIGV of 100 $m.s^{-1}$ at 7500 $rev.min^{-1}$. This figure was within the range found by the author to be optimum for good volumetric efficiency when using typical valve opening periods for road-going, multi-valve engines. The inner seat diameter was designed to give a MIGV of 80 $m.s^{-1}$ at that point.

An inlet valve head diameter of 26·0 mm and stem diameter of 5 mm were used. These sizes gave an efficient port with a high Cv and reasonable insert spacing. Tests using 26·5 mm diameter valves showed little overall improvement in flow; an increase in flow was seen at high lifts but flow at intermediate lifts was lower. The graph of Cv against L/D is illustrated in Figure 74. The plot shows that the Tickford design, in comparison to the Toyota design, gave slightly higher values of Cv at intermediate lifts, but lower values at high lifts.

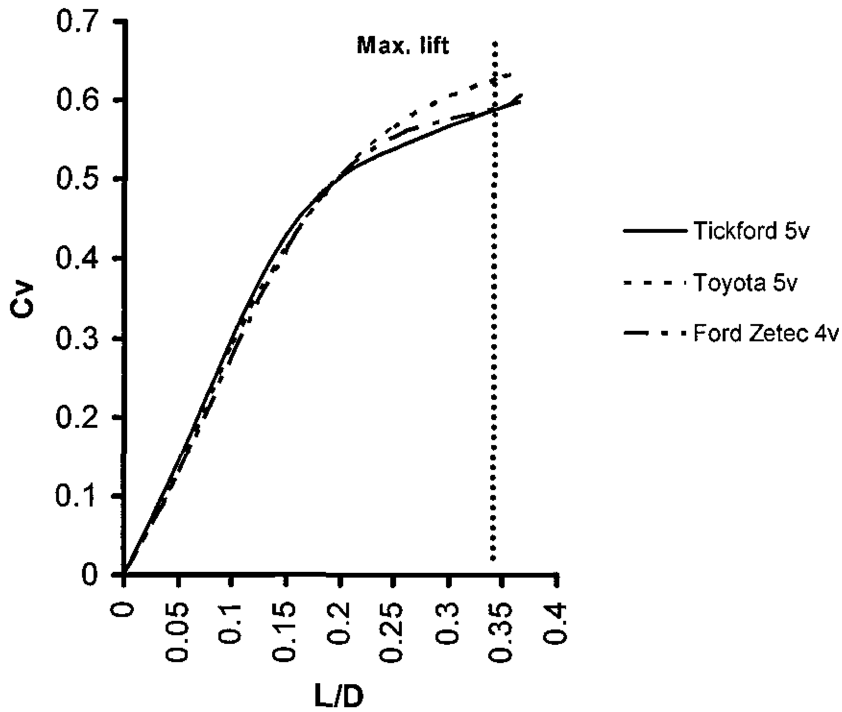



**Figure 74: Flow coefficient against non-dimensional valve lift**

Figure 75 shows inlet air mass flow against valve lift. The results demonstrate that the Tickford and Toyota cylinder heads had similar mass flows up to 4 mm lift but above this, the Toyota design gave higher mass flow.

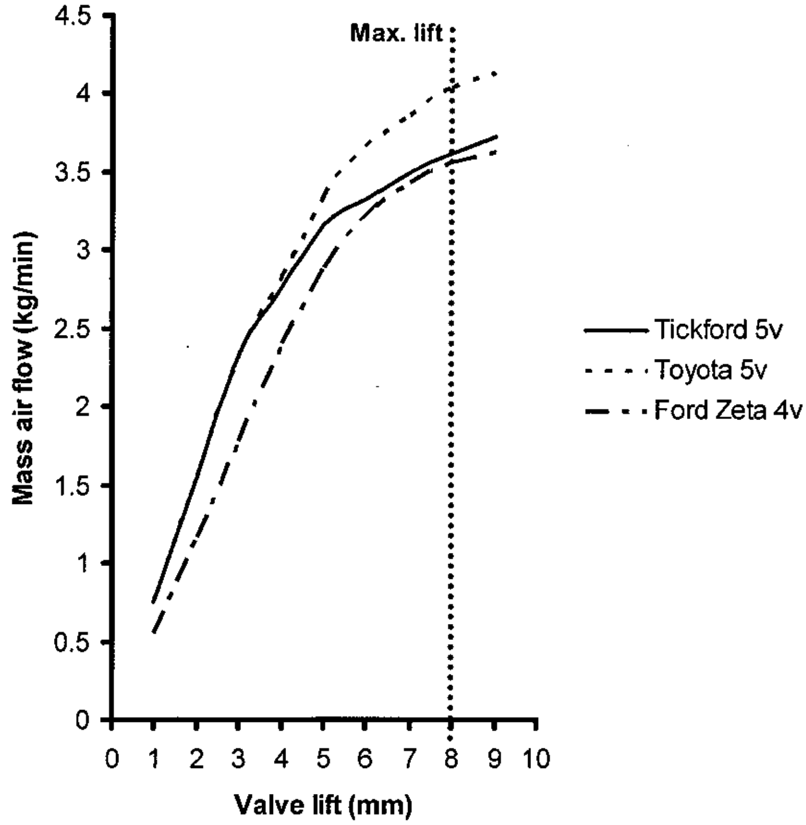



**Figure 75: Comparison of inlet port mass air flow against valve lift for 3 designs**

Both 5-valve designs exceeded the 4-valves per cylinder Zeta engine results. The better high lift flow of the Toyota was attributed to the steep port angle and high port floor as illustrated in Figure 76. This was because the high port floor allowed air, which breaks away at the inner corner above a certain lift, to re-attach before reaching the valve and therefore utilise the whole periphery of the valve. Both of these features that resulted in superior high lift flow, also resulted in low tumble swirl. This was a good illustration of the trade off between tumble swirl and high lift flow, as discussed earlier in this chapter.

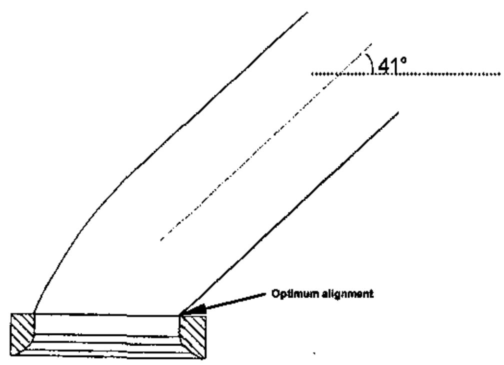



**Figure 76: Inlet port in optimum position relative to insert**

The Ford Zeta engine produced high levels of tumble swirl and was regarded by many Ford competitors as the benchmark engine of the time. The value of integrated tumble swirl ratio between TDC and BDC and the tumble swirl ratio at high lifts achieved by the Ford Zeta engine were similar to, and a little higher than figures from other manufacturers at that time. These had been measured by the author in benchmarking cylinder heads for different clients. With this in mind, together with the combustion performance targets, it was decided that the tumble swirl values achieved by the Ford Zeta were a reasonable target and, as explained in Section 3.4, the tumble swirl ratio, integrated between TDC and BDC, would be used. A mean swirl ratio of between 1·5 and 2·0 was therefore set as an objective for the V8 design, against the Zeta measurement of 1·9. The plot of mean Cv against mean tumble swirl ratio is included as Figure 77. It demonstrates that the new flow box achieved a mean tumble ratio of 1·74, which was within the target range.

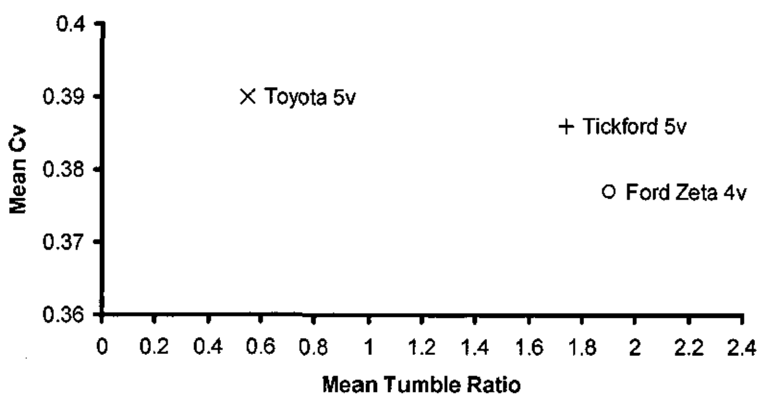



**Figure 77: Comparison of mean flow coefficient against tumble swirl ratio for 3 designs**

Figure 78 is a graph of tumble swirl ratio against L/D. It confirms that the new 5-valves per cylinder flow box produced more tumble swirl at low and intermediate lifts than the Ford Zeta, but slightly less at higher lifts. High tumble swirl at higher lifts is more beneficial than at lower lifts due to the higher mass flow rates which results in a greater contribution to total swirl momentum. The Toyota design produced much less swirl than either the Tickford 5v or Ford Zeta cylinder heads as illustrated in Figure 77 and the following Figure 78. This reason for these results was explained in Section 5.2.

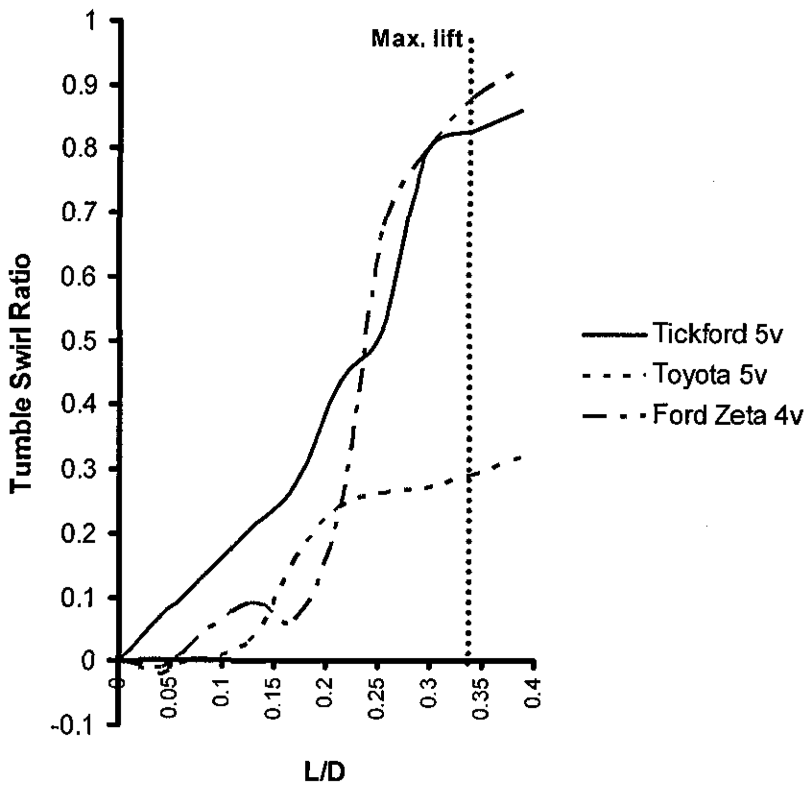



**Figure 78: Tumble swirl ratio comparison against non-dimensional valve lift**

### 5.5.2 *Exhaust ports*

An exhaust valve head diameter of 26·8 mm and stem diameter of 5 mm were used in the V8 design. This was calculated using an inlet to exhaust area ratio of 1·5. As explained in Chapter 2, this area ratio lies in the optimum region when valves are competing with each other for space.

The exhaust port was given a circular cross section at the manifold face. This was used in preference to an oval section which usually required four bolts for adequate sealing. A circular section usually only required two bolts per port and as a result, the cylinder head and exhaust manifold could be both simpler and cheaper.

### 5.5.3 *Component spacing in the combustion chamber*

Acceptable material spacings between combustion chamber components were based on the author's experience with high performance multi-valve engines, given in Table 2.

| Components | Spacing (mm) |
|---|---|
| inlet insert - inlet insert | 3·0 |
| exhaust insert - exhaust insert | 7·6 |
| exhaust insert - inlet insert | 4·1 |
| outer inlet insert - chamber wall | 1·0 |
| exhaust insert - chamber wall | 0·8 |
| inlet insert - spark plug (min) | 6·5 |
| exhaust insert - spark plug | 4·3 |

**Table 2: Combustion chamber component spacing (Tickford)**

These values make an interesting comparison with the generally closer spacings used by Toyota, which are given in Table 4 in Section 5.6.1.

### 5.5.4 *Valve gear design and layout*

The production 4-valves per cylinder customer engine had the camshaft centre spacing of 114·3 mm required to accommodate the client's proposed variable valve timing mechanism. This dimension was retained for the 5-valves per cylinder design as advantages were seen in having the same camshaft centre spacing to the current production variant to enable commonisation of machining on the front face of the cylinder head and carry-over of the cam cover. Consistent with the literature discussed in Chapter 2, the design philosophy behind the combustion chamber was to keep it as compact as possible thus avoiding the need for a domed piston to achieve the desired compression ratio. A flat top piston has a preferable combustion chamber shape, lower piston weight and, in this case, provided commonality with the production 4-valves per cylinder engine variant.

The two outer intake valves were actuated via bucket tappets whereas the central valve was operated via a finger follower, pivoting on a hydraulic lash adjuster in a similar manner to the 5-valves per cylinder, 2·5 litre Ford V6 design described in Section 5.3. The unique Tickford 5-valve design, in which the inner and outer valve axes diverge from the valve heads, precluded the use of three inlet bucket tappets unless two inlet camshafts were used as had been the case in the earlier, higher speed and much less cost-conscious, Formula One designs. The use of two inlet camshafts was not feasible for a high volume engine from economic and complexity viewpoints. Alternatively, three finger followers could have been incorporated so all three inlet valves were actuated by the same type of mechanism. This method was not used because the inlet camshaft would have needed to be displaced towards the outside of the cylinder head in order to lie between the inner and outer valve axes. To maintain the same camshaft centre spacing as the 4-valves per cylinder head, the exhaust camshaft would then have required moving towards the centre of the head. The overall result would have been a more vertical exhaust valve resulting in increased chamber volume.

Bucket tappet dimensions were made common to both inlet and exhaust valves. The tappet bucket diameter could accommodate a valve velocity consistent with achieving 8 mm lift within the estimated valve opening period. Shims under the bucket tappets rather than disk shims on top of the tappet were adopted to minimise tappet diameter. The 12 mm diameter hydraulic lash adjuster for the central inlet valve was based on a prototype made available by INA Bearing Company. A clearance of 4·5 mm was used between the finger follower and the casting except in plan view, where a small machined clearance between the finger follower and the adjacent tappet bore wall was designed in order to accommodate any lateral rocking.

The inlet camshaft was supported by bearings mid-way between each cylinder. The exhaust camshaft was supported by bearings between each cylinder's pair of exhaust valves. Due to lack of space between the bucket tappets on the inlet side, the bearings were positioned between cylinders. This did, however, allow wider bearings than would be possible if they were located between tappets, also permitting four bolts per cap. Bearings on the exhaust side had two bolts per cap. Hollow dowels were included to locate cam bearing caps at the request of the client. Common practice among many manufacturers at that time, especially the Japanese, was not to use dowels to locate bearing caps and the author's team normally only used dowels on racing engines.

The cylinder head bolt dimensions and assembly tool clearance were defined by the client. Access to the cylinder head bolts via a hole in the bottom camshaft bearing meant that the cylinder head could not be fully assembled prior to fitting to the cylinder block. This was obviously a disadvantage in comparison to most modern 4-valves per cylinder designs where single bearing caps can be used on the intake side as well as exhaust, allowing access to all cylinder head bolts when the head is fully assembled. At that time, the client company was looking to implement valve train designs into its future engines in the US that were capable of running for 100 000 miles without adjustment. This was possible with hydraulic lash adjusters. However, the possibility of reaching this objective with mechanically adjusted bucket tappets was much less certain. Therefore, there was further discussion about the use of three finger followers, each with hydraulic lash adjuster, to meet the durability requirement.

### 5.5.5 *Fuel injector*

A single spray injector, with a spray pattern cone angle of 10 degrees, was located in the inlet manifold just upstream of the manifold to cylinder head interface. The injector orientation was such that it sprayed fuel on to the back of the central inlet valve in a conventional manner. This was done in order to take advantage of whatever natural charge stratification occurred within the cylinder, where the rich mixture would be biased towards the centre of the chamber because the central inlet valve was aligned with the spark plug. Some of the injector spray did, however, go down the two outer ports.

Double and triple spray injectors were investigated and effective prototype solutions had been developed for the Ford 5-valves per cylinder, 2·5 litre V6 project [Sykes (1995)]. At that time, they were only available in bottom feed configuration which did not suit the client's 3·4 litre V8 installation. If the preferred double or triple spray injectors were to have become available in a top feed type configuration, only minor changes to port profile would have been required to accommodate them.

### 5.5.6 *Compatibility between 82·4 and 89·0 mm bore sizes*

During the project, the customer decided that two versions of the engine might be included in his product planning; one having an 82·4 mm bore diameter and the other increased to 89·0 mm bore. The design scheme continued to be based on the 82·4 mm bore but considerations of compatibility with the 89·0 mm bore are discussed here.

Ideally, the 89·0 mm bore variant would have had increased valve centre spacing in order to accommodate the larger valves required to maximise the performance benefit of larger displacement. Combustion chamber component spacing would ideally have been the same as those of the 82·4 mm bore variant. In practice, different valve centre distances between variants would have increased machining cost, when dedicated multi-spindle machining heads were used, due to the need for different machining tool sets for each cylinder head. It was decided that it was more cost effective to maintain valve spacing and allow increased valve sizes by compromising component spacing. The multi-spindle machining heads could then be common between variants with only a change to cutters.

It was decided that the design optimised for the 82·4 mm bore configuration, would be modified in order to equalise the degree of compromise on the two variants. To facilitate this, the inlet valve insert spacing on the 82·4 mm bore was increased by moving the inner valves closer to the chamber wall, accepting that mid range flow would reduce slightly, in order to use larger valves on the 89·0 mm bore without reducing insert spacing so much. The design recommended to the client by the author partially compromised both designs, rather than affect one greatly and the other not at all.

### 5.5.7 *Spark plug*

A standard, taper seat spark plug, with 14 mm thread diameter, was adopted for the design scheme. Its position within the combustion chamber was determined from acceptable material thicknesses between spark plug and valve inserts. This did not allow sufficient room for the hydraulic lash adjuster if the spark plug axis was vertical. The spark plug axis was therefore inclined at 10 degrees relative to the cylinder axis in order to provide clearance for the outer inlet valve's hydraulic adjuster without compromising the combustion chamber layout.

### 5.5.8 *Coolant path*

Water jacket sand sections were designed in liaison with Zeus Aluminium Products, the chosen prototype castings company, for optimum section thickness and casting suitability. Casting wall thickness was nominally 4·5 mm with strengthening as required in areas of high temperature and thermal stress such as the combustion chamber walls and valve bridges. No problems were envisaged when casting the cylinder head with the customer's production specification aluminium alloys.

The inlet port had a water jacket around the bottom of the port only. The water jacket around the exhaust port extended around the whole port to maximise valve guide cooling.

### 5.5.9 *Manufacturing effect of port machining tolerance*

The author's earlier work on port geometries to generate optimum tumble swirl was complemented by studies carried out for a major manufacturer on the impact of port core shift on flow and tumble swirl performance of its 4-valves per cylinder heads. For this V8 design, the port floor intersected the throat at a point coincident with the inner corner of the insert throat bore as illustrated in Figure 76. This was the optimum position for both flow and tumble swirl where there was a sharp corner between port floor and throat. The client's manufacturing process had to be evaluated in order to determine the variation in relative position of the port floor and insert as a result of core shift and insert machining tolerance. Initial tests with the port floor slightly under-cutting the back of the insert showed that flow and tumble swirl were only minimally affected. An example of this undercut is illustrated in Figure 79.

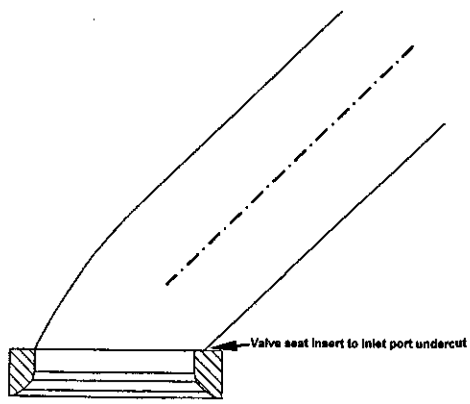



**Figure 79: Inlet port undercutting the back of the insert**

Raising the port floor and therefore leaving a step between the back of the insert and port floor reduced tumble swirl. Making sure the intersection point between the port floor and the insert bore was always below the back of the insert, as illustrated in Figure 79, was found to be worth considering as a method of minimising tumble swirl variation in production volume of engines. This would, however, have meant that in some cases, a portion of the back of the insert was unsupported, the feasibility of which would have required further investigation. The problem of variation in tumble swirl with port floor height was not different from that experienced with production 4-valves per cylinder engines.

# 5.6 Review of Toyota 5-Valves per Cylinder Head Design

## 5.6.1 *Design overview*

The Toyota 5-valves per cylinder engine type 4A-GE was released in 1991. The engine had a swept volume of 1·6 litre, a bore of 81·0 mm diameter and a stroke of 77·6 mm. The bore diameter of the Toyota engine was therefore slightly smaller than the engine on which the Tickford client design scheme, described in Section 5.5, was based (82·4 mm). The specific power output of the 4A-GE was quoted as 73 $kW.l^{-1}$. A cylinder head was obtained for air flow testing and design review. In addition to the observations made in previous sections, a general overview is followed by the author's comments on some particularly interesting design features and differences in design philosophy.

The Toyota design was based on the Yamaha layout introduced in Chapter 3, in that a single inlet camshaft operated all three inlet valves whose axes converge from valve heads to valve stem tips. The Toyota inlet valves were operated by 23·5 mm diameter, direct attack, inverted bucket followers which appeared not to have any method of adjustment. It was thought possible that bucket followers with selected thickness crowns were used but, when they were measured, they were all the same. This confirmed that the Toyota production system was extremely well controlled.

The minimum spacing between tappets bores was 2·0 mm. Oil was conventionally fed to the front camshaft bearing and then to the other bearings via oil drillings in the camshaft. The valve head sizes used on the Toyota and Tickford designs are given in Table 3.

| Engine | Inlet valve | Exhaust valve | Inlet to exhaust area ratio |
|---|---|---|---|
| Tickford 5v | 26·0 | 26·8 | 1·41 |
| Toyota 5v | 26·5 | 26·0 | 1·56 |

**Table 3: Valve sizes used by Tickford and Toyota**

All inlet and exhaust valves had stem diameters of 5 mm. The inlet valve sizes in Table 3 make an interesting comparison in the light of the smaller bore size of the Toyota engine and the inconsequential results of trials of 26·5 mm inlet valves in the Tickford design, described earlier.

Table 4 shows the combustion chamber component spacing of the Toyota 5-valve cylinder head. The outer valve heads of the Toyota were much closer to the combustion chamber wall than on the Tickford design even though they were only 0·5 mm larger in diameter. The combustion chamber wall was machined around the two outer valves with a cutter whose axis was concentric to the valve stem. This guaranteed a consistent valve head to chamber wall clearance and suggests that the larger valve diameter was compromised by chamber shrouding. Shrouding occurs when a valve head is too close to the chamber wall, which inhibits flow from the valve at intermediate lifts. This was evident on the Toyota design from the poor intermediate-lift flow coefficients illustrated in Figure 74. Valve spacing on Yamaha-type layouts was, however, usually dictated by the bucket tappet diameter required to obtain a particular lift and the minimum allowable material thickness between tappets.

Most of the component spacings were similar to and slightly smaller than those used by Tickford. The most notable difference being the exhaust insert spacing which was

7·6 mm on the Tickford design, compared with 5·2 mm on the Toyota. It would have been interesting to see the casting section in this area of maximum heat flux. The combustion chamber component spacings measured on the 4A-GE engine are given in Table 4.

| Component | Spacing (mm) |
|---|---|
| inlet insert - inlet insert | 2·3 |
| exhaust insert - exhaust insert | 5·2 |
| exhaust insert - inlet insert | 3·3 |
| outer inlet insert - chamber wall | 0·7 |
| exhaust insert - chamber wall | 1·3 |
| inlet insert - spark plug (min) | 5·3 |
| exhaust insert - spark plug | 4·5 |

**Table 4: Combustion chamber component spacings (Toyota)**

### 5.6.2 *Other notable Toyota design features*

#### *5.6.2.1. Inlet port splitter design*

The Toyota 4A-GE inlet ports split into three closer to the valve than had been observed by the author in other two and three inlet valve designs. Splitting occurred approximately 5 mm before the back of the valve seat insert. A splitter design of this type may have had advantages in terms of minimum wall wetting and resultant lower hydrocarbon emissions. However, it was felt that this possible advantage was outweighed by the resultant sudden expansion and reduced flow velocity which occurred at approximately 12-32 mm from the inner seat, which is illustrated in Figure 80. It was noted that the Toyota port was approximately 50 mm longer than the Tickford port, as marked in the figure, and therefore there was greater potential for wall wetting due to the Toyota injector being further from the back of the valve. Injectors in both engines were located at similar positions relative to the inlet manifold to cylinder head interface.

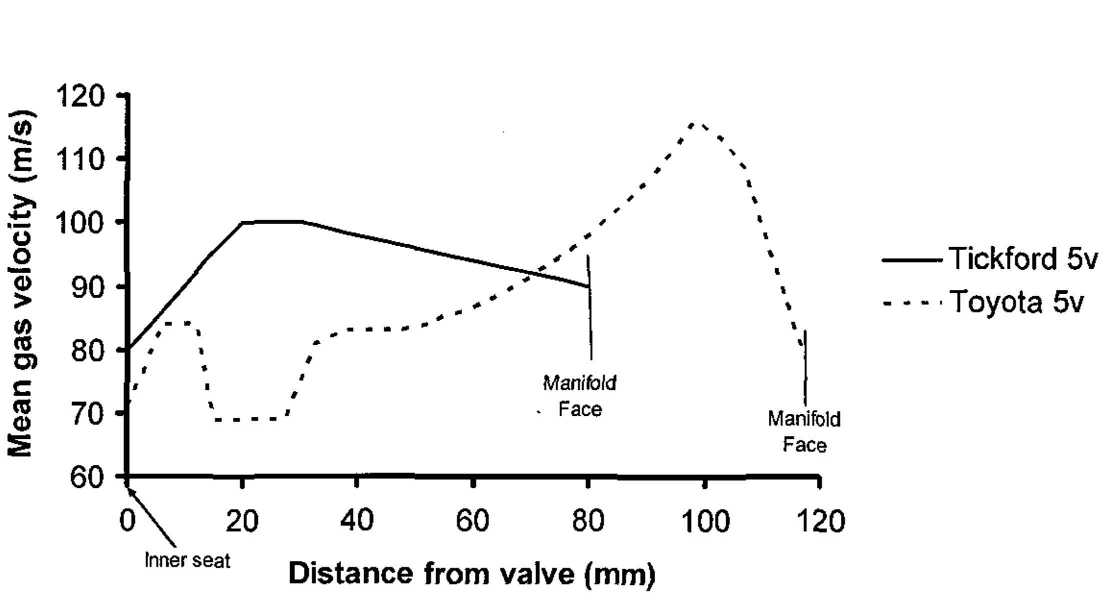



**Figure 80: Plot of calculated mean gas velocities along the inlet port**

From the graph of gas velocity against distance along the port illustrated in Figure 80, it can also be seen that gas entering the Toyota cylinder head at the manifold face, accelerated over the first 20 mm, reaching its maximum MIGV of approximately 116 $m.s^{-1}$. This velocity is at the upper limit of what the author would specify for a high output road engine. From the minimum area section to the region before the valve guides, the area increased gradually. Just prior to the machined throat section between 86 and 106 mm down the port, the cross sectional area increases abruptly giving an equally rapid reduction in theoretical velocity. Although such expansions do not always show up as reduced flow on a steady-flow air flow rig test, under pulsing conditions in a running engine, the continual expansion and contraction loses energy in the induction gas and, therefore, volumetric efficiency. After the sudden expansion and contraction at 86-106 mm the velocity was constant for a further port length of 5 mm, after which it reduced rapidly to value of 70 $m.s^{-1}$ at the inner seat, plotted as coincident with the zero of the x-axis in Figure 80.

Figure 80 also illustrates MIGV in the Tickford port. Gas entering the port accelerates gradually, reaching a maximum velocity of 100 $m.s^{-1}$ when the port splits into three, then decelerates progressively to the valve seat allowing some pressure recovery.

### *5.6.2.2. Angled machining of inlet port throats*

There was an unusual feature which suggested that Toyota's machining of the inlet port throats was carried out in two stages. This is illustrated in Figure 81. It appeared that a 50 degree included angle cutter was used first to form the angle which leads into the valve seat and ran-out some way beneath the insert into the port. Another cutter then formed a cylindrical bore at an unusual angle of approximately 17 degrees to the valve axis to a depth beneath the insert of approximately 17 mm. The result of this machining was to remove the re-entrant angle on the inner corner of the insert, formed in the first cutting operation, to leave a smooth transition between the port and valve seat insert.

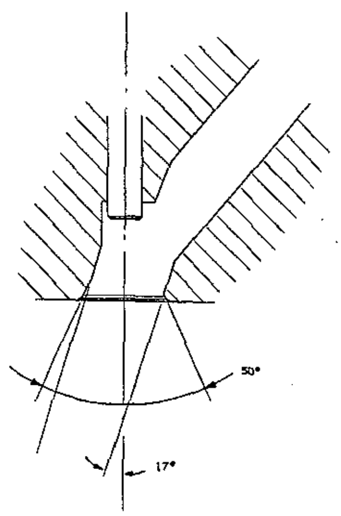



**Figure 81: Sketch of inlet port throat machining on the Toyota 4A-GE engine**

Port flow was expected to be very consistent due to the repeatable throat profile that this achieved and although the port floor height might have varied due to core shift, the high port floor meant that this particular design would probably not be sensitive to variations in port floor height. Tumble swirl, although low, should at least be repeatable.

If the throat machining had been carried out coaxial to the valve axis, rather than at an angle to the valve axis as was the case in older Yamaha designs such as the FZR 750 and FZR 1000 motor cycle engines inspected by the author and his team, the port would have imparted little tangential velocity to the mixture entering the cylinder, other than that resulting from effective port inclination due to chamber roof inclination. As a result, tumble swirl for the Toyota head would have probably have been even lower. This hypothesis is supported by the author's test on the Yamaha motorcycle engine heads, which produced lower tumble swirl ratios than the Toyota design [Bale and Downing (1990)].

#### 5.6.2.3. *Inlet camshaft bearing layout*

Four of the five inlet cam bearings were positioned directly over the cylinder head bolt centres. Only the most forward inlet-side cylinder head bolt was accessible with the cylinder head assembled. In order to gain access to the cylinder head bolts, a separate casting, housing the lower half of the inlet camshaft split bearings was used. This was located to the cylinder head via two hollow dowels. Upper bearings halves were then bolted through the lower bearing casting into the cylinder head. Dowels were not used to locate the upper bearing halves.

The lower half of the front camshaft bearing was an integral part of the cylinder head casting as a result of being offset towards the front of the cylinder head. The unsupported span of camshaft on cylinder 1 was therefore approximately 25 mm longer than on the other cylinders.

In conclusion, the author considered that the Toyota engine was very capable with regard to inlet port flow but, as discussed in Chapter 3, the interaction between the flow stream from the central valve and the flows from the outer pair resulted in low levels of tumble. With this combination and the application of variable valve timing, the performance of the engine was competitive for the era, having a maximum BMEP of 10·2 bar.

## 5.7 Tickford Variable Geometry Intake System Development

At about the same time as the 5-valves per cylinder developments were being carried out, the author's team was also working on other means of improving engine breathing and performance with a view to increased specific power output and a combination of high torque, low exhaust emissions and good fuel economy. This resulted in the creation and development of another invention, a stepless variable geometry induction system.

The need for this device arose during the development of the Rover K-Series 1·4 litre 16-valve engine being undertaken at Tickford under contract from Rover. One area of development was the intake system and, in the absence of reliable simulation tools at that time, a lot of test work was being conducted on intake manifolds of different designs, varying both sizes and configurations. As a result of the necessarily high amount of fitting work changing pipe lengths, diameters and bends, an engine test technician mentioned that he wished he had a lever in the control room that would rapidly re-configure the intake for further testing. The author's chief engine designer, Alistair Lyle together with Richard Sykes, the author's senior technical manager, successfully designed a mechanism for a stepless variable geometry system to vary the length of the primary inlet runners. Having created the initial concept, the author's team then went on to develop the design into one that not only provided variation of the primary intake runners, but also the feeder pipe and the system volume.

The incremental system volume change was necessary in order to optimise the intake system as a Helmholtz resonator by changing the entrained volume at a higher rate than that achieved by lengthening the primary intake runners and plenum feeder pipe. By adding blank-ended volumes within the rotating portion of the system, the total effective volume of the system could be changed at a higher rate. This enabled the system to more closely follow the formula for Helmholtz resonance described by Equation (11) in Chapter 2.

The inclusion of three variable geometries (primary runner length, plenum feeder pipe length and effective manifold volume) was a significant innovation that took the moving manifold concept well beyond the work conducted by Daimler Benz, which was the subject of a patent in 1954 [Gassman (1954)]. The novel work by the author's team was granted patents in all the territories where applications were made and a paper was presented at the IMechE seminar in June 1991 on "Variable Geometry Engines" [Sykes (1991)]. The design created a lot of interest by reporting how the new invention combined wave ram, inertia ram [Engelman (1953)] and plenum chamber acoustic resonance. An early model used to explain the design, is illustrated in Figure 82a and 82b.

**(a) End view showing primary inlet runner** **(b) Side view showing feeder pipe**

**Figure 82: Model of Tickford stepless variable intake**

The following Figures, 83 and 84, show the workings and construction of the device as described in the first patent granted for Great Britain in 1988, GB2205609A, which established its priority date for other territories.

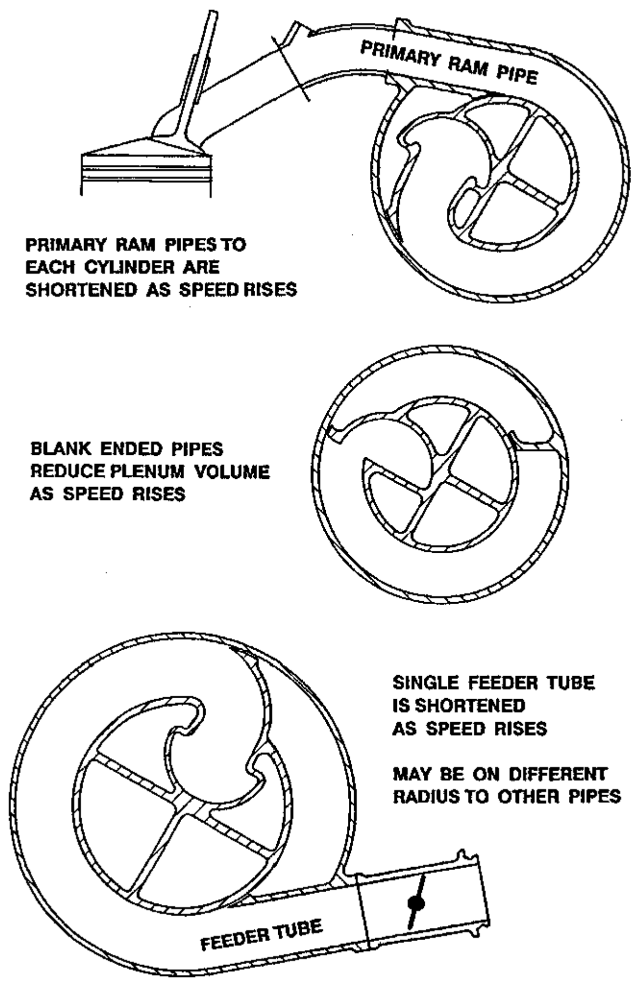



**Figure 83: Cross sections through Tickford stepless variable intake**

The three parts of the figure show, from top to bottom, the primary runner geometry change, the system volume change capability and the plenum feeder tube geometry change. The plenum chamber is the bell-mouth cross-section in the plane of the Figure 83, which is also labelled in Figure 84. All three moving ducts could be designed on different radii to vary the rate of change of length and volume with rotation. Rotation was managed with respect to engine speed.

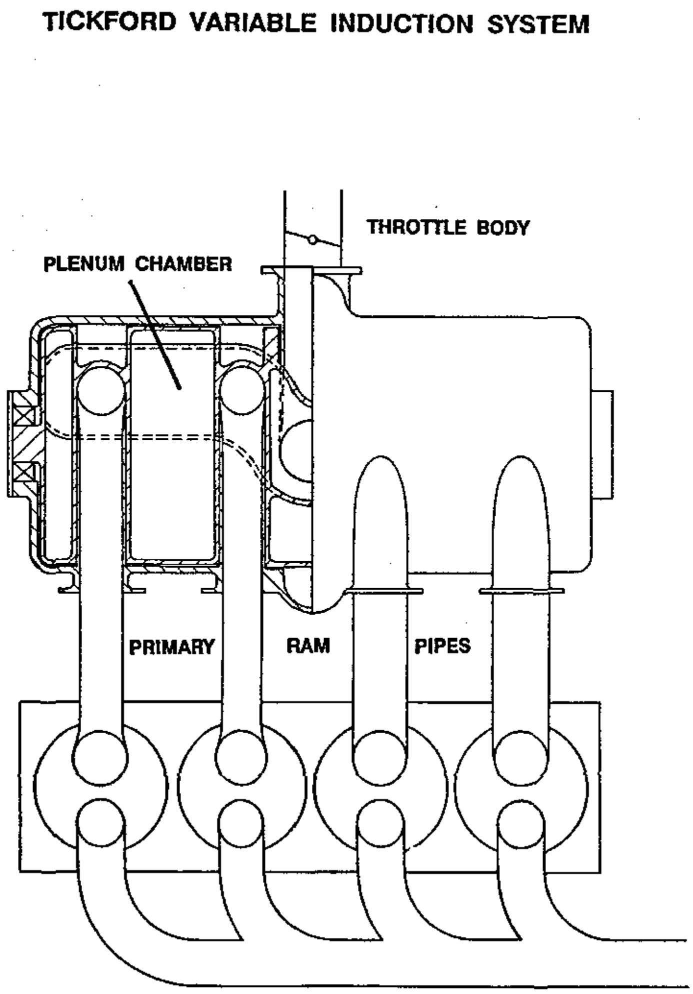



**Figure 84: Part section through Tickford stepless variable intake system**

The invention was shown at the IMechE Autotech 91 congress and the 1992 SAE World Congress exhibitions as well as in presentations to the world's leading engine manufacturers including: BMW, Ford, General Motors, Jeep (Chrysler), Mercedes-Benz, PSA, Renault and Volkswagen. Work was subsequently undertaken with the French plastic moulding company, Mecaplast SAM, in order to develop the design into a volume production design, using its lost-core moulding technology. At the time that the author left Tickford in 1996, work was underway on control systems development to regulate the position of the system according to the engine speed and load conditions.

As the volumetric efficiency of an SI engine is constrained by the throttle, there is no need for tract lengths and volumes to follow the engine speed under all conditions. The control system was defined such that the device would move to a "stand-by" position at other than wide open throttle conditions. The selected stand-by position was at maximum tract lengths and volumes on the basis that the most likely conditions for use of a fully open throttle would be acceleration from lower in the engine speed range. Only when the throttle was fully opened did the control system position the intake system according to the engine speed.

The device was originally created for the rapid, empirical development of induction systems. The increasing complexity of fixed geometry induction systems by manufacturers, coupled with constant pressure for higher specific power and torque served to make the application of this invention into a realistic proposition. The development of lost-core moulding techniques and small but powerful actuators combined to create a feasible situation. Whereas the fully developed Tickford device has not been seen in volume production, significant elements are found in the intake system of the intake system supplied by Pierburg for the BMW V8 engine in the "7-series" car [Hirschfelder *et al* (2002)] and illustrated in Figure 85.

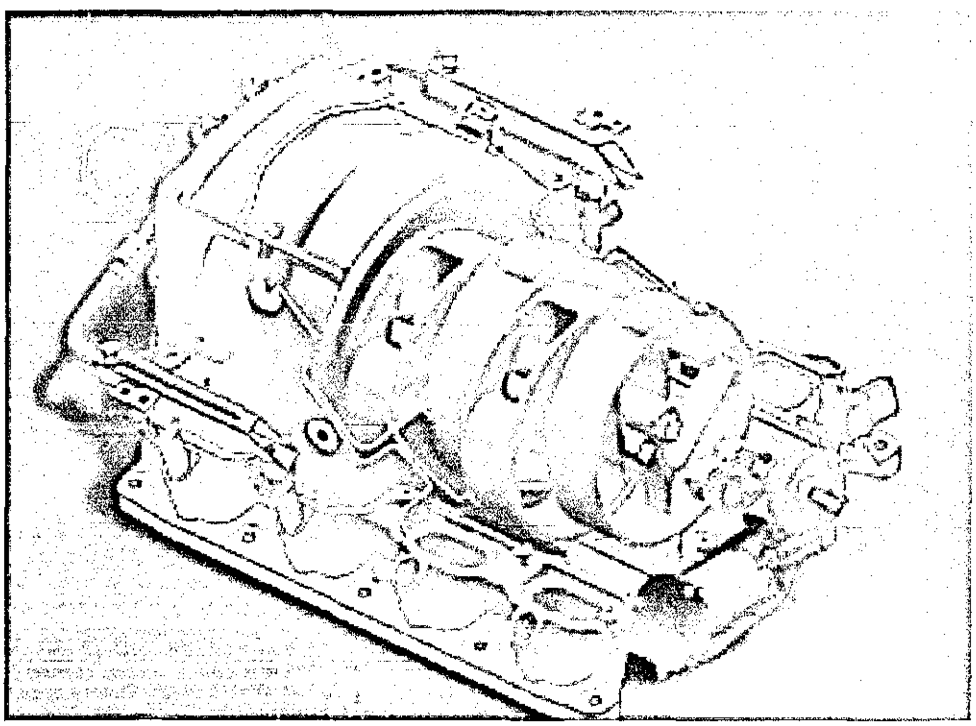


**Figure 85: BMW 7-Series V8 stepless variable intake system by Pierburg A.G.**
*(Reproduced from Hirschfelder, et al (2002))*

## 5.8 Closing Remarks

This chapter has discussed the application of designs that were initially developed for engine development and racing, into production feasible, 5-valves per cylinder engines with low emissions and with packaging and product image benefits. The author's work has been shown to go beyond the point of optimisation for power output and placed the invention of designs and development tools in the mainstream of automotive engines. The implications of production engine manufacturing on the design of cylinder heads have been discussed and examples presented.

It has been demonstrated that the author's work in 5-valves per cylinder research and its application to several client engines, has technical merits over cylinder heads that use the Yamaha 5-valves per cylinder design. This chapter has also described further novel work conducted by the author and his team inventing a new apparatus for researching the port geometry factors affecting tumble generation and a patented steplessly variable intake manifold.

Chapter 6 will discuss the larger implications of creating sound engine designs that enable engines to run efficiently (from the measures of power, output, fuel economy and exhaust emissions) in real duty cycles and not just to demonstrate compliance with legislative standards. The implications of engine manufacturers divesting some of the roles and responsibilities of engine design, particularly in developing markets with different legislation and engineering ability, will also be introduced.

# Chapter 6.

# REAL WORLD EMISSIONS PERFORMANCE

## 6.1 Introduction

This chapter discusses the operating regimes of legislative drive cycles and the relative emissions performance of some vehicles on and off these cycles. Emission measurements carried out by the author, and by others, are contrasted with legislative cycle results in a way that reinforces the importance of good combustion and wide-ranging emission control capability. It goes on to describe the ways in which effective contributions can be made to achievement of these and other challenging engineering goals by the use of supplier know-how and partnerships.

## 6.2 Emissions Test Cycles

In order to meet the market demands of customers choosing to drive performance vehicles whilst maintaining compliance with ever-tightening exhaust emission legislation, engine manufacturers must develop the use of combustion systems capable of supporting high performance and low emission outputs over a wide range of engine operating conditions. This is shown to be important for global air quality in order that engines have acceptable real world emission outputs in diverse conditions and not confined to those speeds and loads encountered during the legislated emissions measurement cycles that are enforced by various territories during the type approval process [Lenz and Cozzarini (1998)]. Different territories have different drive cycles, each of which relate to some representative vehicle operation that may be encountered in customer use, standardised to facilitate comparison of vehicle emission performance against legislated standards. The emission tests are conducted to a given procedure using a test vehicle being operated to the drive cycle whilst mounted on a chassis dynamometer. This is an arrangement of rollers fitted with actual or simulated inertia and programmable resistance that permits the vehicle to be driven as if it were propelling its own weight, but in a test laboratory with controlled ambient conditions and with sampling equipment attached. The exhaust gases are collected and analysed and, from the corrected concentrations and a measure of the total exhaust volume emitted, the mass pollutants can be calculated for comparison with the regulated standards. The European drive cycle, developed by the Economic Commission for Europe (ECE) is recognised as being rather artificial whereas the US Federal Test Procedure, known as FTP, is much more transient, having been established from real journey data. The speed of the FTP drive cycle is illustrated against time in Figure 86 and comparable data on the on the relative use of different engine loads required during the drive cycle are illustrated in Figure 87.

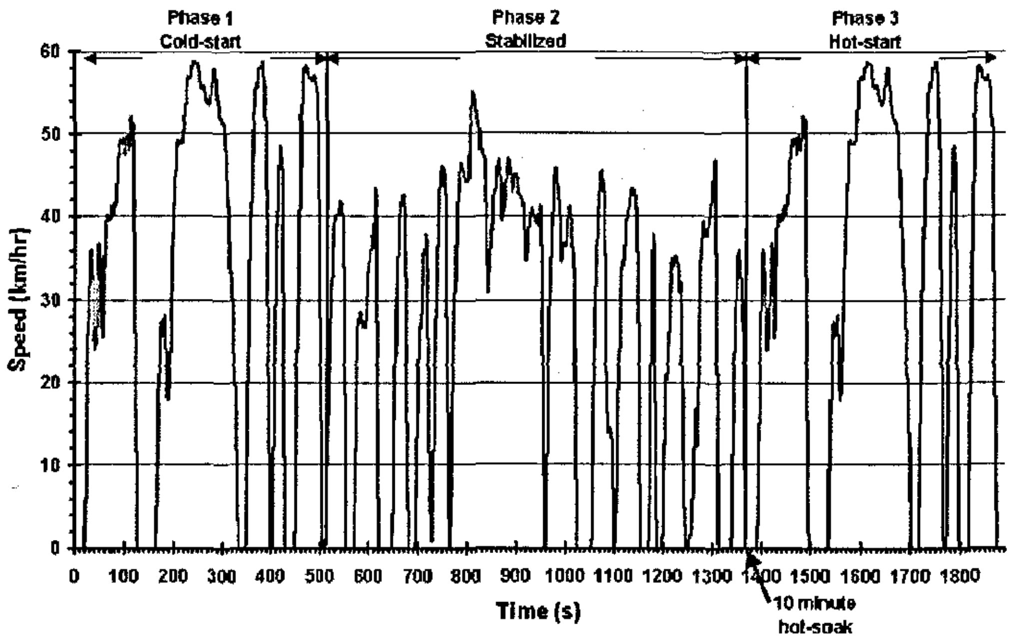



**Figure 86: US FTP drive cycle**
*(Reproduced from McDonald et al (2001))*

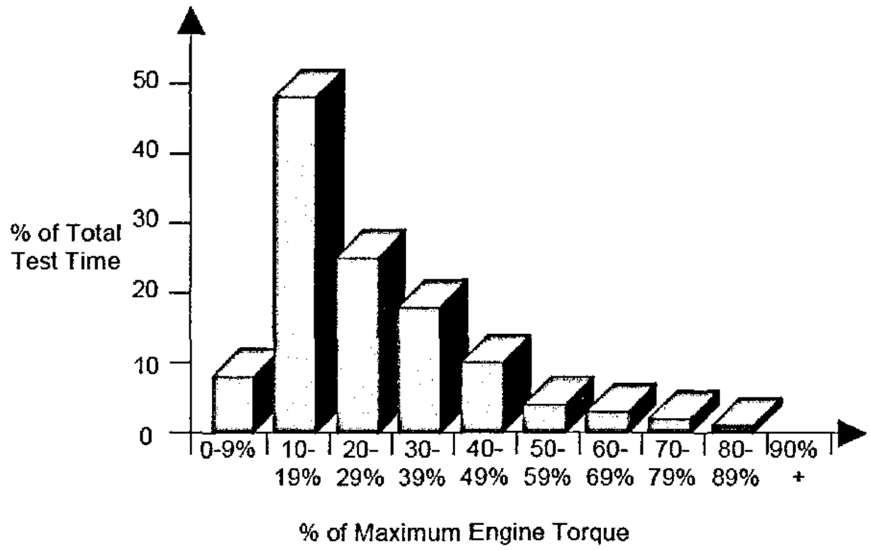



**Figure 87: Engine loads used on US FTP drive cycle by a small Rover car**
*(Reproduced from Akehurst et al (1999) from data by Simner (1995))*

Because of the widespread use of over-fuelling by engine manufacturers' calibrators to limit catalyst gas inlet temperatures, the importance of which was explained in Chapter 2, there were legislative proposals in the late 1990s [Walsh (1996), CARB (1997), ECE (1998)] that would have required the control by stoichiometric three-way catalysts to remain effective right up to full load. Operation outside of the optimum air-fuel ratio, also described in Chapter 2 and illustrated in Figure 88, means that a TWC is much less effective in converting the harmful pollutants.

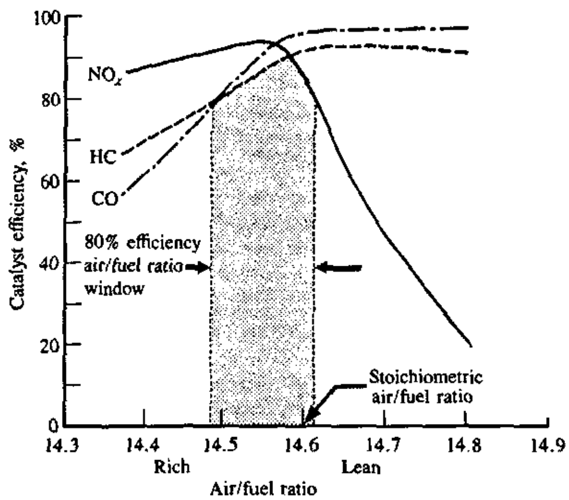



**Figure 88: Conversion efficiency for CO, $NO_x$ and HC for a three-way catalyst as a function of exhaust gas air-fuel ratio**
*(Reproduced from Heywood (1988g))*

Maintaining the calibration in the catalyst three-way window of operation, i.e. without over-fuelling, means that other adjustments, such as ignition timing and breathing limitation, are necessary to prevent the converter temperature exceeding the capability of the precious metal compounds or the substrate, as discussed in Section 2.18. These adjustments, in turn, result in a reduction in the maximum engine power output. Despite the fact that the proposal to mandate fully stoichiometric calibrations was not enacted, there was clearly recognition amongst legislators and manufacturers that emissions compliance was then, and currently, only demonstrated by use of one or more driving cycles which, of necessity, could not cover the entire operating range of the engine and vehicle combination. The subject of fully stoichiometric calibrations remains a future prospect for real-world air quality improvement.

There are different approaches to this scenario from different manufacturers. Some engine and vehicle combinations that comply with legislative cycle measurements are significant polluters outside the test cycle operating envelope. There are also engine and vehicle combinations that have combustion systems, emission control systems and fuel/ignition calibrations that deliver better emission control at speeds and loads not encountered during legislative testing [Guensler (1994), Bale and Farnlund (1999)]. Advanced combustion systems such as the tumbling 5-valves per cylinder system already described in this thesis, can be extremely useful in this regard. Further development of the author's team's design reported by Sykes (1995) made it clear that the combustion chamber activity, brought about by the high tumble design, affects the amount of enrichment beyond (richer than) stoichiometric fuel air ratio, at which maximum performance is attained. Intense activity that promotes mixing of the fuel and air and rapid combustion, as in the tumbling 5-valves per cylinder configuration, minimises the power losses resulting from running at stoichiometric rather than the leanest fuelling for best torque (LBT). Further work was indicated at the time (1995) to fully quantify the potential advantages of high tumble port and chamber designs.

Work was conducted by the author [Bale and Farnlund (1999)] in which the emission performance of a range of vehicles was evaluated over a wide range of operating conditions and compared with their performances in legislative emissions testing. The authors developed an Environmental Performance Index (EPI), first created by Rototest AB in Sweden, which produced a numerical factor combining weighted totals for the legislated pollutants when measured over an extended operating range of speeds and loads, which could, in turn, be used to rank vehicles in terms of real-world "cleanliness".

The author also investigated the different strategies and technologies adopted by the manufacturers to make their products legislatively compliant. The work gathered some measure of the costs of the emission control devices and the differing technologies that had been used for each vehicle in the research. It was concluded that the ranking for legislative testing performance was very different from that of the extended EPI, with examples given in Table 5, and that manufacturers had applied very different approaches in terms of the type and cost of the emissions hardware fitted to ensure compliance with legislation. The findings also showed wide variation in tailpipe emission performance "value-for-money".

| Vehicle | Position in ranking List | Rototest EPI | Quoted Emissions over European Drive Cycle | | |
|---|---|---|---|---|---|
| | | | THC g/km | CO gm/km | NOx g/km |
| VW Lupo 1·4 16v Manual | 1 | 64 | 0·157 | 0·053 | 0·044 |
| Audi A4 1·8T Manual | 25 | 95 | 0·798 | 0·143 | 0·121 |
| Peugeot 306 XS 1·6 Manual | 54 | 124 | 1·049 | 0·151 | 0·048 |
| Mitsubishi Space Wagon 2·4 GDI Manual | 80 | 196 | 0·070 | 0·040 | 0·050 |

**Table 5: Quoted emission results and EPI ratings for a selection of vehicles**
*(Adapted from Bale and Farnlund (1999))*

Table 5 shows a sample of vehicles and their ranking, in column 2, for real-world cleanliness as calculated using the EPI measure. The EPI values are given in column 3, which take into account the exhaust emissions levels up to and including wide open throttle. This is contrasted by their type-approval exhaust emissions results that are quoted in columns 4 to 6 [Kraftfahrt-Bundesamt (1999)], which would suggest a different ranking order for low emissions. A full table of results was included in the author's paper [Bale and Farnlund (1999)] included in the thesis appendices.

The author's work was later reinforced by Samuel *et al* (2005) who investigated emissions and fuel economy from a typical Euro IV passenger vehicle over the legislative cycle and a number of more aggressive cycles. They also found that real-world emissions levels were significantly higher than the certified legislative emission levels, noting that carbon monoxide emission was the most affected specie in a gasoline powered application. Other literature, discussed in Chapter 2, has already shown the difference in emissions with respect to driving style on emission control systems of earlier generations. An example that summarises this well, is illustrated in Figure 89.

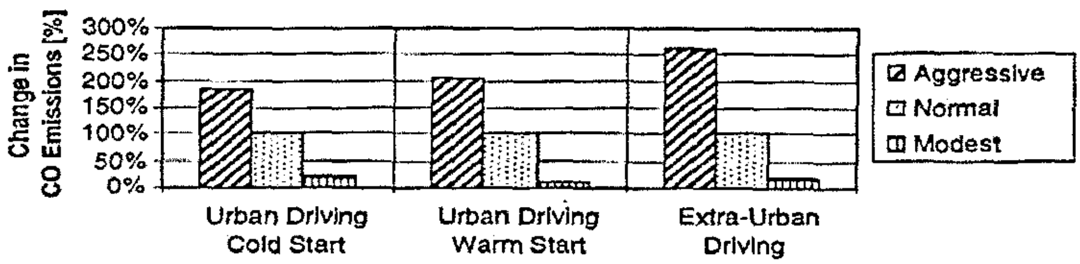



**Figure 89: Effect of driving style on CO emissions**
*(Reproduced from Lenz and Cozzarini (1998))*

## 6.3 Contract and Joint Research and Development

Much of the author's basic knowledge and experience of exhaust emissions measurement was gained whilst developing engine fuelling and emission control measures with Lucas and with Aston Martin Lagonda Ltd. The author created and ran the vehicle emissions measurement facilities for Aston Martin and then Tickford and was a member of the Vehicle Emission Test Group of the British Technical Council for the Motor and Petroleum Industry, and Vice Chairman from 1991 to 1994. During this period he provided exhaust emissions testing and associated results interpretation, on a contract basis, for vehicle manufacturers (VMs), original equipment manufacturers (OEMs) and the petroleum industry. Results remained strictly client confidential but trends and relative efficacy of different emission solutions led to a substantial expertise and value as an engineering service.

During his experience with Tickford, the author was also part of the team that secured both quality (ISO 9001:1994) and supplier (Ford Q1) accreditations that attested to the company's working procedures and philosophies. This coincided with a period of transition in the methods of supply for automotive industry world-wide as the VMs and OEMs fought to reduce their costs and increase efficiency. One strategy adopted by many vehicle manufacturers was the restructuring of the supply chain so that they only dealt with a reduced number of "Tier 1" suppliers who became responsible for the lesser suppliers. This approach was discussed in the author's paper on the subject [Bale (1994)] and is illustrated in Figures 90 and 91.

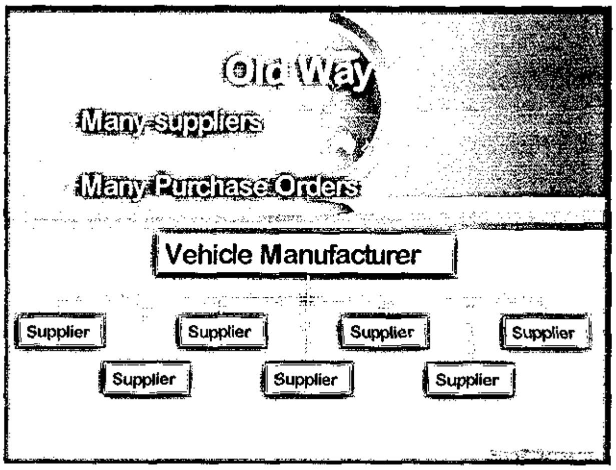



**Figure 90: Traditional supply chain**

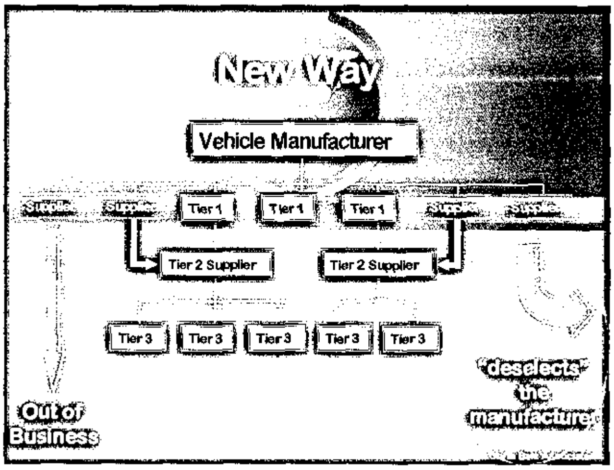



**Figure 91: Revised supply chain**

The significance of the change to engineering service and component supply companies was the need for the lower tiers to undertake design, development, testing, assembly and scheduling of parts and services. This required programme management and engineering skills to be transferred from the VMs, and increasingly from the larger OEMs, to lower and lower tiers of the supply chain to spread the financial burden of product development and legislative compliance. In terms of an engineering service supplier, such as that by which the author was employed at the time, this meant adopting processes and procedures used by the vehicle manufacturers and opened up a whole new range of customers in the supply chain.

The author found that most lower tier supply chain companies did not have the necessary engineering skills and experience to carry out all the elements of the engine and vehicle development and type-approval programmes that were being passed into the suppliers' responsibility.

Engineering service companies took a useful role in assisting component suppliers with projects while the suppliers adjusted to the new way of working. Investment in new skills, new staff and new engineering tools all took time. The VMs, of course, liked to do things as a step change and engineering companies were there to provide the capability and resources to achieve the short term solutions for VM and OEM component suppliers. Some component suppliers, recognising that there was a peak of activity during the engineering phase of a new product or application that then diminishes during the product cycle, continued to use consultancy companies as their engineering department and chose not to build their own. This gave them the additional flexibility to deal with continuous engineering and new product introduction at the same time.

During the same period, the world automotive market was expanding to new and developing territories. Significant incremental volumes of sales were available but territory requirements such as legislation, local fuel quality, climate, road surfaces and terrain, customer preferences, etc., made much of the manufacturer's mainstream product unsuitable. In most cases, exhaust emissions regulations were less stringent than in Europe and the US. This meant that less costly solutions could be used to meet the legislation whilst still contributing to reasonable local air quality. What was required were "derivative" vehicles adjusted for the new market needs, but without re-engineering the whole car for what was often localised assembly. In the cases where fuel quality or composition was very different, the re-engineering was multi-faceted; that is the fuel system needed attention as well the engine management mapping, the exhaust system, etc. The suspension, the interior trim and equipment also, typically, needed adapting to local needs so a broad range of capability was needed in the supply chain or engineering source. In the vehicle manufacturer, so many different departments may had been involved that the programme management became a challenge and service companies had to become expert at managing fast-track, multi-functional projects, understanding the customers' needs and constraints, and executing the programme with the minimum load on in-house resources.

The ultimate manifestation of a committed customer/supplier relationship was found by the author to be the formal "Joint Venture" (JV). The operational needs of a joint company are underpinned by the financial and resource commitment of both parties, resulting in commonality of purpose and roles for both shareholders that, in turn, give rise to fast response to market needs and local content without risk of damage to either "parent" or constant recourse to guidance.

## 6.4 Case Study

With a major contribution from the author as Business Development Director for Engines, Tickford was selected by the Ford Motor Company of Australia to create and operate such a JV company in Melbourne, Australia, with production of 11 500 vehicles per annum. An example of one of the projects has already been introduced [Bale and Sykes (1993)], the Falcon XR6 with its enhanced 2-valves per cylinder 6-cylinder engine described earlier and shown as a product in Figure 92. The market driver for Ford in creating the JV was to deliver its product wish for enhanced vehicles outside the product development cost and manufacturing constraints for such low-volume models.

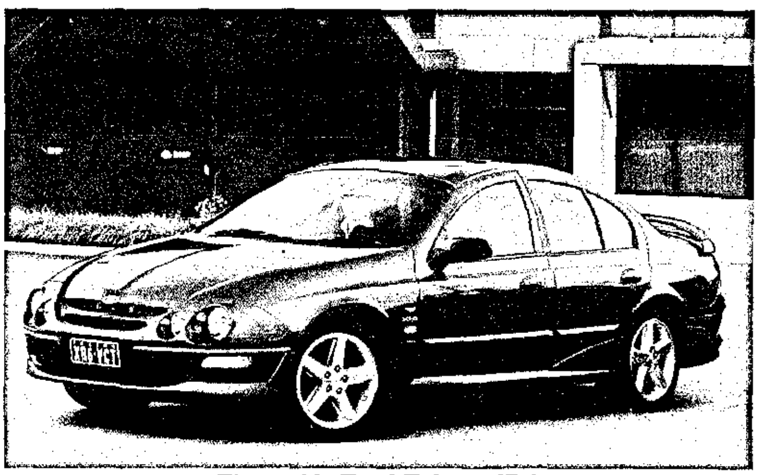


**Figure 92: Ford Falcon XR6**
*(Photograph courtesy of Tickford Ltd.)*

The power and torque achievement of the XR6 developments described on Chapter 5 is summarised in Figure 93, demonstrating the 8·8% increase in peak power and 4·9% more torque compared with the standard engine, achieved with virtually no increase in piece price and seamless local manufacturing integration.

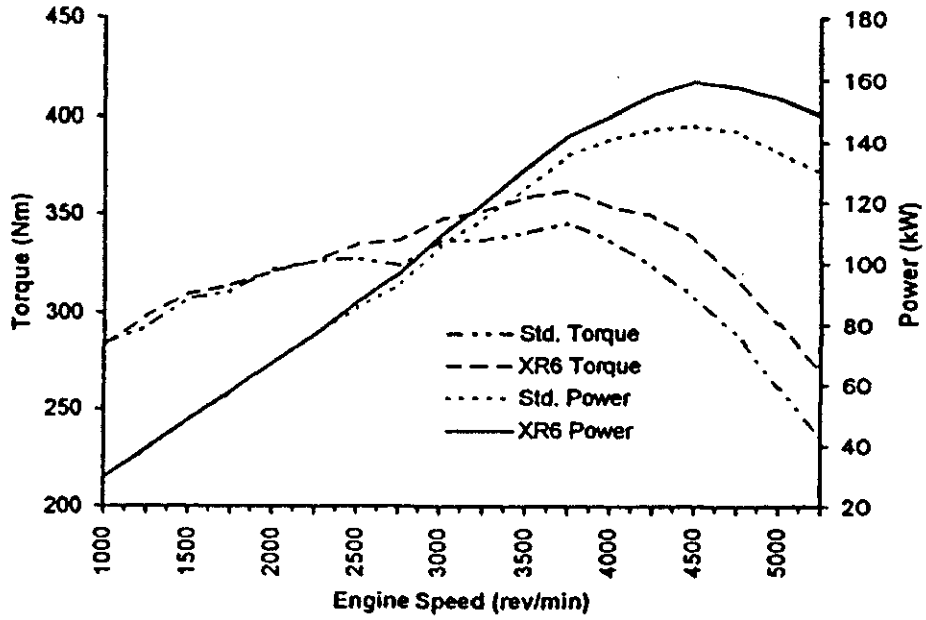



**Figure 93: Engine development of Ford XR6**

In between sharing the work and the rewards comes the inevitable need for those involved to share some risks. But there was the, not always so obvious, need to share in the excitement also. In this case study, with the type of product manufactured, there was indeed some excitement to be had. The JV had to be fully integrated into the Ford product policy and planning processes, right through to the launch of the product into the marketplace. The manufacturing data systems also had to be linked and information freely exchanged.

In the product realisation projects, the optimum route for creating the special vehicles was planned and agreed between the main factory and the JV manufacturing team well ahead of the production introduction date. Cars were built as far a possible down the main production line, but often to a "delete" specification, without the parts that would not be required in the final vehicle, or with essential items replaced by slave components. The latter would be returned to the main plant for re-use on further deliveries to the JV. This avoided wastage when new components were later substituted to create the defined product. This situation meant that much of the engineering of the product was done for line manufacture and not for low volume. Such things as the author's derivative engine had to be machined and assembled down the main assembly line even if the castings were unique, such as the cylinder heads that incorporated new ports and camshaft profiles, which were described in detail in Chapter 5.

Based on the author's own experience, a properly formed JV company has proved to be the best relationship for which to aim. But the will of both parties to want it to work, and the importance of that initial set of ground rules, were not to be underestimated.

The author's ultimate aim of a successful engineering supplier and customer relationship was like any other purchase in which both parties are satisfied with the services and funds exchanged. In a good relationship this can be described as a "win-win" situation. In a JV this became "win, win, win" as all three managements were satisfied. Like any good marriage, the outcome of a good JV is most effective when both parties contribute and recognise that each must take out good value for what they put in, not only for the day but also for the future. This proved to be the case for the Australian JV where the JV profit share that accrued to Ford of Australia represented a significant percentage of that company's total profit from all its operations, which included vehicle sales, parts sales and financing.

In achieving the delivery of emission compliant and customer-satisfying products, the author has found that the supplier who does not invest in his people, facilities and quality systems, is bound to fail in the recent tide of change in the supply chain. The vehicle manufacturer that does not recognise that these value-adding features of a supplier have a price, is deluding himself and risks becoming a non-preferred customer with the best-in-class providers of component, systems and engineering services. The inevitable knock-on effect on his own business will be the true price of such a false economy.

The author has found that the most efficient solution in engineering services has to draw from the lessons learned from world class production suppliers:

1. The customer shares enough of his plan to enable the supplier to invest in the right skills and facilities;
2. The supplier has the temerity to have core resources available to meet the timetable;
3. The customer honours the suppliers' investments by using the programme resources in a flexible framework of fair price and delivery;
4. The supplier honours the customer's commitment by delivering the project on time and on budget without seeking to exploit every minor deviation in the engineering programme.

The author concluded that customer/supplier relationships can cover a very wide range of "togetherness". Customers can be very geographically distant from their suppliers and *vice versa*. Tickford proved that this did not need to be a problem in the geographical sense, even when working with a client in Australia. This suggests that geography is often used as an excuse for poor relationships when it is not the real cause. True partner relationships allow both purchaser and supplier to become more efficient and prosper at lower cost. The main point being that no matter where the relationship stands at any point in time, it can always be made better by "Continuous Improvement" with the aim of attaining and maintaining a world-class performance.

## 6.5 Promoting Design Responsibility in the Supply Chain

The author continued to promote the contribution of design and development skills from the supply chain through work with lower tier suppliers in UK and abroad. Some of this work attracted support from the European Social Fund, which was keen to preserve and create jobs in targeted regions of the EU. One such area was the West

Midlands of the UK where a project was created to enable automotive engineering, manufacturing and business skills to be promoted to small and medium sized enterprises (SMEs) in the region. Because of the time and financial pressures on SMEs, much of the knowledge transfer was done by intense workshops and via the Internet. Learning on-line via the Internet, allowed the recipients to choose the time and place for study and he or she had complete flexibility of the pace of progress. The development and implementation of this new approach to supplier development was reported by the author in a paper published at the World Foundry Congress [Ashley *et al* (2006)] and will be described further in Chapter 7.

The physical mechanism used for computer-based supplier development was "Moodle", an open source code for managed (or virtual) learning environments, which facilitated the creation of low-cost, bespoke and generic learning platforms. These were configured both for the public and for specific organisations to enable their in-house training requirements to be met. The platform was accessed via the Internet, or an intranet, with a normal web browser. For some of the author's work, this on-line material was combined with face-to-face workshops and seminars to form, so-called, "blended learning". As a result of the project, a resource of more than 100 training courses was made available in four languages. A particularly interesting aspect of the supplier development tool was the research, creation and use by the author's team, of a split-screen lecture presentation system called "AutoView". This was developed into a leading-edge audio-visual presentation technology to bring experts to the learner's environment, on demand, to make diverse and informative lectures and presentations. During the project, research was carried out to develop a way of incorporating multilingual subtitles to facilitate transfer to non-native English-speakers. An example of this is illustrated as Figure 94 and is fully described in the paper [Ashley *et al* (2006)]. The screenshot demonstrates the unique, synchronised and multilingual subtitles developed by the author's team, which are being developed in 2007 to accommodate languages that use character sets like, for example, Chinese, Korean and Arabic.

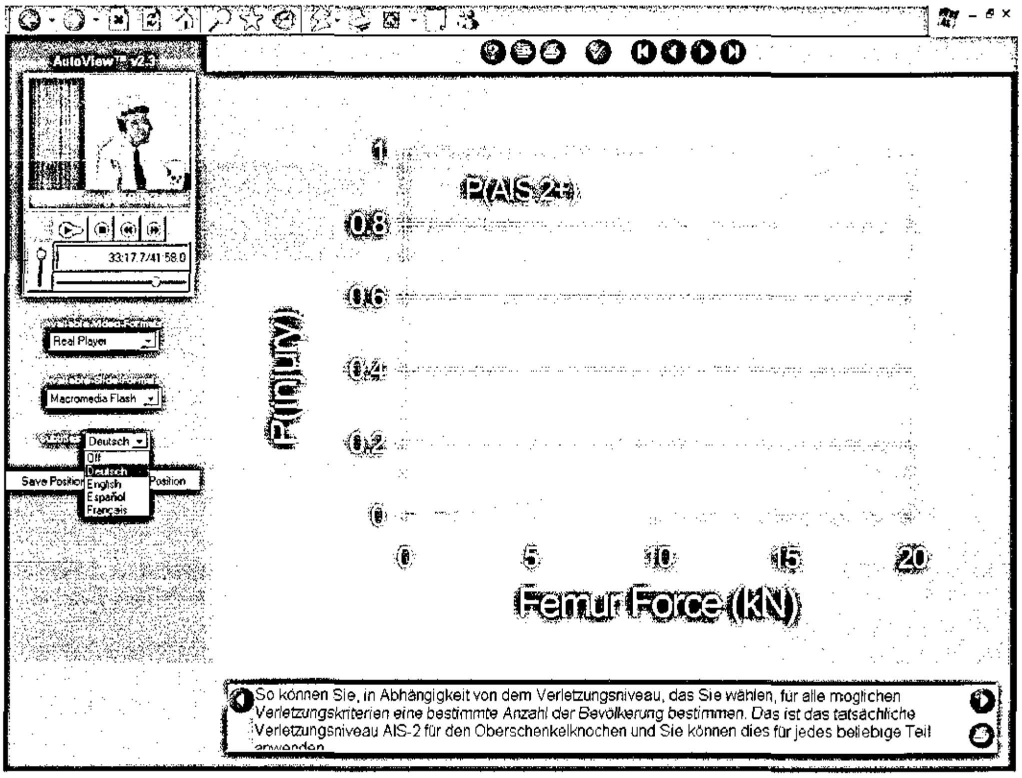



**Figure 94: Example from an "AutoView" video lecture on 'Crashworthiness'**

## 6.6 Closing Remarks

This chapter has described some of the ways in which engine engineering research and knowledge is applied to road going vehicles to assist in the achievement of good air quality from real driving conditions. It has also underlined the importance of the understanding that the engine designer and developer must have for real world applications, driving conditions and legislation when choosing the technologies for emissions compliance.

Ways of developing the supply chain have been introduced and the beneficial arrangements of a JV have been described. The innovative use of Internet-based learning has been introduced with the flexibility it brings to the learning of new skills in the supply chain.

The application of the author's research will be developed further in the next chapter, which is concerned with other aspects and contributions from his work in the field of knowledge transfer and, specifically, the dissemination of his knowledge of engines.

# Chapter 7.

# KNOWLEDGE MANAGEMENT

## 7.1 Introduction

This chapter describes more about the author's work in the important areas of knowledge transfer and engineering knowledge management. The engine R&D work described in earlier chapters would not have been as successful, or even possible, without the team working and the exchanges of knowledge and experience. In particular, this chapter will describe the dissemination of the fundamental research carried out by the author for competition and high performance passenger cars, to an emerging motorcycle manufacturer in India.

The author's career and working experience has clearly demonstrated that one of the key enabling disciplines in high performance engine development is the integration of the human factors. This includes the management of the knowledge gained during the R&D process and the transfer of successful designs, techniques and technologies from one engine to another, where they can be shown to be effective and relevant. The author's paper on this subject [Bale (2001] includes examples of good and poor practice in the use of engineers as a medium for knowledge transfer. It also compares how the industrial climate in the automotive industry both supports and destroys successful knowledge transfer, according to the methods and people philosophies adopted.

## 7.2 Knowledge Transfer

The present industrial climate of intense global competition and tight economic margins has meant that manning levels are under constant review. In former times, when the "apprentice" mentality of the craft and manual trades was adopted in engineering, young engineers, including the author, benefited from working alongside skilled designers, engineers and technicians to learn more about their subjects. This improved the understanding of the related tasks and componentry over that which even the finest university education could provide in just four years study. In the modern era, passing on one's knowledge to others is not often carried out with such enthusiasm or completeness as in former times. Economic uncertainty means that the individual never knows when cuts in manpower might be made and any unique knowledge held by a person will give them an advantage in terms of job retention, when reductions in staff are being contemplated. Hence there is less likely to be an effective and complete knowledge transfer from the experienced hand to the novice. There is a clear need for knowledge to be produced and managed in the research and everyday processes of engineering companies and for it to be retained within the business. This is equally important to adding to the wealth creation of the nation of which the company's economy forms a part [Koch (2003)].

Information management is particularly important in the field of engine development where the body of knowledge of the first 100 years of the automobile has not all been documented. That which has been, is an insurmountable mass of books, papers and articles. Beyond that, much of the original engine design and development work was conducted as an art and not as a science. Modern computational tools and precise measurement are only just identifying the factors that our forefathers in engine development seemed to adopt by instinct or by trial and error. Learning methodologies have also changed and the development of electronic media and the Internet have increased the total amount of information that is available to the learner or researcher, which can make the task of sifting the useful and the useless even more difficult.

All the ingredients are available to make ever better engine designs and developments. There is the benefit of history and the experience of those who have undertaken many experiments and developments. There is a wealth of written data and research from across the world and it is fortunate that much of the published work in engine research is in English. Modern computational tools have been developed together with an awareness, which was not always the case, that computers will process data and can produce results which look impressive but which must be validated if the tool is to be truly beneficial. In addition, advanced instrumentation and data analysis tools exist, which enable parameters to be measured more precisely than in the past and in ways that interfere less, if at all, with the conditions that are being investigated. All these benefits are unavoidably diluted by human factors which limit the amount of knowledge that can be retained and by the amount of knowledge which is transferred from person to person or from organisation to organisation.

The author has advocated more open knowledge transfer and increased awareness of the issues surrounding it [Bale (2001), Ashley *et al* (2006)]. Electronic tools for data retrieval, storage and manipulation are there to help and with operating systems changing every three years and hardware systems having, perhaps, a ten-year lifespan, even computers are not a complete solution; the social and human factors that do prevail, have to be managed effectively.

Mutual aid and reciprocated support can allow an environment for an engineer to jointly work out what is going on and what to do next. Just talking through a problem can often bring about a solution. When it comes to lateral thinking exercises, such as some parts of Potential Failure Mode and Effect Analysis (PFMEA or FMEA as so many people call it), things always work better with multiple participation in the way of team brainstorming rather than isolated contributions. Even automated safety analysis systems require a significant degree of human input in the form of annotation of fault trees and other mechanisms of describing component failure behaviour in a complex system [Papadopoulos *et al* (2004)]. This is a good example of the time and labour saving elements of computer science having to be instructed by human wisdom and experience.

It is important to remember that, for any individual engineer, the team in automotive engineering, referred to above, now includes at least the following:

- Own department
- Other technical departments
- Internal customers and suppliers
- Vendors including full service systems suppliers and engineering outsource

- Hardware and software suppliers
- Shipping and administration
- Travel department
- IT and telecommunications
- Finance
- Management
- Programme control including timing and planning
- Releasing
- Manufacturing

Some of these teams, or some of their participants, may be located anywhere in the world and cultural and language training can be most appropriate. As already mentioned, English is the universal language of most of the automotive industry, but better progress can often be made with overseas companies by at least conversing a little with them in their language. Having an appreciation of the relevant culture is almost mandatory if a successful working relationship is to be maintained. This is particularly vital in Eastern cultures (Near, Middle and Far) where body language can say more than the spoken word. This cultural awareness is especially important where the ability of the visiting English engineer in the local foreign language is little or none, as is more often the case than not.

## 7.3 Knowledge Transfer with e-Learning

The author's work developing the West Midlands automotive supply chain was introduced in the previous chapter. The work there included an element of e-learning, which was knowledge delivered wholly or partly by computer. The emergence of this method and the research into tools and techniques for delivery, have accelerated significantly in the last ten years [Cross (2004)]. The advances in the creation and delivery of video lectures by the author's team have been significant steps and support the benefits for the teaching or research-providing side of knowledge transfer [Fox and MacKeogh (2003)] as well as the receiver's gain. Lectures and presentations that had previously been delivered a number of times in the same way (such as a foundation or introduction to a new topic) could now be captured once and delivered electronically. This has freed the tutors' time for other research or personal contact time, which are arguably more useful than repetition of a standard lecture.

All areas of engineering and manufacturing are suffering intense pressure on time and budgets, with rising costs, especially energy and manpower. This pressure is made worse by rising competition from global suppliers and, as a result, both training and R&D have been squeezed for time and money within the business community. This situation has emerged contrary to the common-sense view that suppliers that do not develop will fail.

Among the answers to this economic problem are effective learning and skills development, especially those that fit into the workplace and the work-life balance. Current practice and legislation require that such training and development is fully accessible to all ages and abilities with a desirable attribute that it accommodates different learning speeds. As Fox and MacKeogh (2003) reported, there is a need for increased learning and teaching not to become over dependent on tutors and this is another reason that the author has been advancing e-learning techniques that are flexible, low-cost, and delivered on demand.

The e-learning programme created by the author, offered lectures and books by technical experts with access to the whole course at any time and anywhere with computer access. It included interactive questions and exercises and, because of its delivery by computer, remote access was possible at work and home by those without time for conventional training. This was also useful in providing knowledge and training in distant places, where the travel time of trainers could be minimised. E-learning ensured that there were uniform professional levels of presentation with only the latest version being available. This was a significant advantage over written material that may have been out of date when referred to by the learner. An excellent example of this would be the reference to a training manual and instruction course for a new machine being installed overseas where the manufacturer prepared and progressively improved the training in his factory and only the current and most up-to-date version was accessible by customers using e-learning. Such a level of document control is one of the cornerstones of operating a quality system such as ISO9001:2000 or ISO/TS 16949.

The work conducted in the author's team developing delivery techniques and materials for the engineering and manufacturing industries, resulted in it being cited in 2002 as "one of the 17 best on-line training programmes in the world" and one of only 3 such recognised programmes in the UK, with British Petroleum and the Open University [Machill and Sacconaghi (2002)].

The author was responsible for the AutoTrain "CoDesign" project from 2003 to 2005, developing the skills of small and medium sized enterprises (SMEs) in the West Midlands to meet the needs of their VM and OEM customers for more supplier input and engineering capability. Following the end of EU funding, the umbrella EuroMotor project has since become an independent limited liability partnership (LLP), a spin-out from the Research and Enterprise Services department at the University of Birmingham. The author continues to provide innovation and application for the course material and the development and promotion of the knowledge transfer techniques being applied to the automotive and related industries.

## 7.4 Case Study

An example of good practice in knowledge transfer has been demonstrated in the author's application of his knowledge of engine breathing and charge motion technology. His research was described in the earlier chapters on multi-cylinder and multi-valve engines, and has been successfully transferred to a family of current, single cylinder, 2-valves per cylinder, motorcycle engines. Fundamentals of engine design, port flow, combustion measurement and data analysis, together with simulation and experimentation, were taught by the author as part of a postgraduate level module in engine design and development. This has enabled competitive engines to be developed by TVS Motor Company (TVS) of Hosur, India, which was previously TVS Suzuki Limited, a "make to drawing" licensee of Suzuki Motor Corporation. Teaching and R&D consultancy work provided by the author to TVS since the end of 2000, has assisted with the amicable severing of the 19-year licensing arrangement between Suzuki Motor Corporation and TVS in November 2002 and the development of an independent range of motorcycles with in-house designed engines. By transferring an understanding of the function and design of engines and the previous Suzuki designs, together with instruction in the use of more specialised test apparatus, the author provided essential tools for the creation of new, improved engine designs.

The knowledge transfer programme conducted by the author was matched with strategic investment by the company, not only in the time given for training, but in the purchase of combustion analysis and fast-response gas analysis equipment. Simulation software was also purchased by TVS R&D department, in order to better understand the gas-exchange processes and to predict engine performance parameters.

After several upgrades and improvements to engines originally designed by Suzuki, including the 150 $cm^3$ Fiero engine, 110 $cm^3$ Victor engine and the 90 $cm^3$ Scooty, the first engine to have been completely and independently developed by TVS as a result of this R&D teaching programme, is that of the Centra motorcycle, illustrated in Figure 95. It has a 100 $cm^3$ engine, with 51 mm bore and 48·8 mm stroke, which delivers 5·6 kW at 7500 $rev.min^{-1}$ and 7·5 Nm at 5000 $rev.min^{-1}$. This makes it the most powerful motorcycle in its class and the maximum BMEP is 9·4 bar. Centra has a road fuel consumption of 100 $km.l^{-1}$ which is best in class by some ten per cent and makes it a genuine "1-litre" vehicle (i.e. $\leq$ 1 litre fuel per 100 km or 282·5 MPG) in the parlance of European fuel consumption.

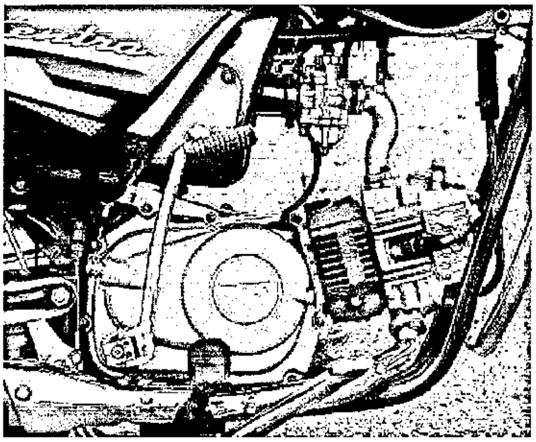


**Figure 95: TVS Centra engine**
*(Reproduced from TVS Motor Company literature)*

Having demonstrated the importance and understanding of MBT timing to engine performance, emissions and fuel economy, the author persuaded the management at TVS to invest in technology and move away from simple two-stage spark advance control and Centra is the first of the TVS engines to use a more complex ignition map. This feature is marketed by the company as 'Variable Timing Intelligent' (VTi) and it is the first time that this technology has been applied in this segment. Centra is a 4-stroke motorcycle targeted at the popular market, and advertised as the most fuel-efficient motorcycle in the country; a selling feature that is very significant in the Indian market where low purchase and running costs are much sought after attributes. The Centra engine also incorporates low friction design valve gear as well as narrow crankshaft bearings to reduce parasitic losses. The marketing campaign for the TVS Centra comes with the slogan, 'fill it once a month bike' where a full eleven litre tank of petrol will last for a month based on the average use of motorcycle customers. Urban fuel consumption as measured on the Indian Drive Cycle is 72 $km.l^{-1}$.

The Centra engine has a design based on the improved understanding of low friction features in the crankcase, valve gear designs and engine fuelling and ignition, taught by the author. The implementation of a better optimised, microprocessor-controlled, spark advance over the engine operating range, mentioned above, and the application of the author's leadership in port-induced air motion, has delivered improved combustion, performance and economy. By helping the R&D team to understand the fundamentals of port design and to recognise deficiencies in the earlier port designs, a set of effective design guidelines was drawn up by the author for TVS. The company purchased a simple air flow rig and swirl measurement system which have, with the author's teaching programme, contributed to the first successful indigenous engine developments.

The knowledge transfer plan developed by the author for TVS, consisted of an intensive 6-day teaching module with lectures, interactive tutorial periods and daily feedback sessions based on team activities. For each feedback session a different team member was required to present the team's information and additional coaching points were made by the author on content, together with some of the "soft issues" of presentation techniques, effective reporting and the use of visual aids. The course integrated the teaching of problem solving techniques in a low-key way, using real-world examples and developing team working. After each course module, there was a period of review and revision by the engineers before they sat the author's 3-hour written examination, for which answer scripts were then marked, corrected and annotated by the author. Results contributed to engineers' ongoing grading and project allocation by the TVS R&D management.

In addition to the post teaching module revision, project teams (broadly based on the teams formed during the teaching module) were assigned to product-related projects with specific objectives. These projects were the subject of a 4-5 month review and guidance by the author, followed by further work. After 9-12 months a concluding report and presentation was made to the R&D management team and the author, as well as to senior company management as available. The company's Managing Director has sent a clear message of support to the author's process, having made a point of attending these colloquium events on several occasions.

One of the post-module projects from the 2003 cohort resulted in a paper on engine gas exchange analysis and measurement, being accepted for publication and presentation at the SAE Global Small Engines Technology Conference in Graz in September 2004 [Deshmukh *et al* (2004)]. The author's teaching and mentoring contribution to the work was duly credited although it was not corporately appropriate for him to be included as a co-author. Two concluding figures from the paper demonstrate the work carried out. Figure 96 illustrates the results of performance simulation using AVL BOOST and Figure 97 demonstrates the measured results for the standard and modified engine specifications, the benefits of which have since been adopted.

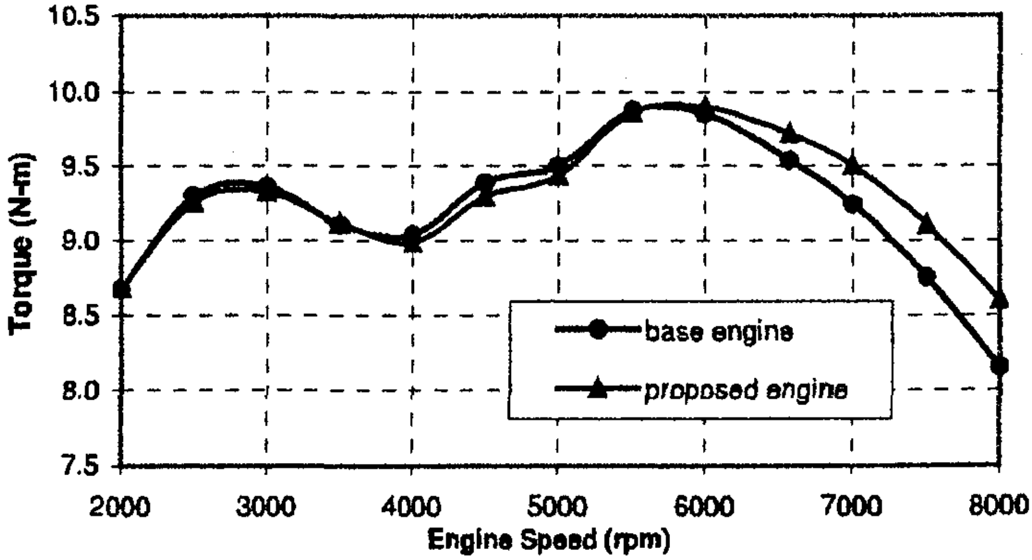



**Figure 96: Engine performance predictions for author's post-module project**
*(Reproduced from Deshmukh et al (2004))*

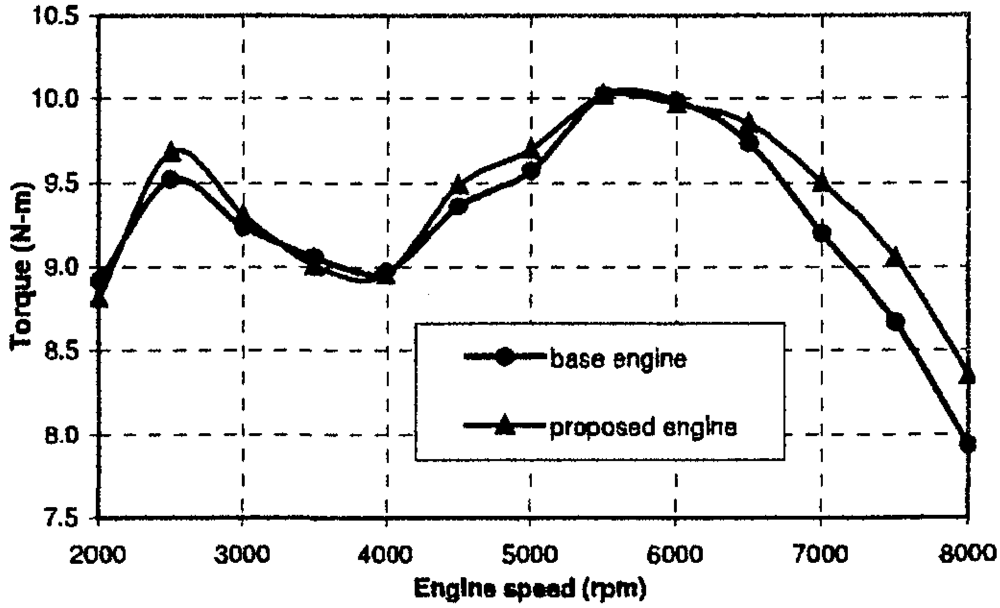



**Figure 97: Measured engine performance improvement from author's post-module project**
*(Reproduced from Deshmukh et al (2004))*

The launch of the Centra engine and the publication of an internationally refereed technical paper, signalled the growing maturity of the R&D resource at TVS. The author had successfully transferred knowledge of basic engines, design and development, testing discipline, research techniques and the appropriate use of modelling tools, sufficient to deliver the first of a very competitive new product range. This transition was extremely timely as Suzuki Motor Company entered the Indian market in January 2006 forming a new wholly owned company, Suzuki Motorcycles India Pvt Ltd., as an open competitor to TVS.

The second all-new machine in the new TVS product range was launched on 19$^{th}$ December 2005 and is called Apache. It is illustrated in Figure 98.

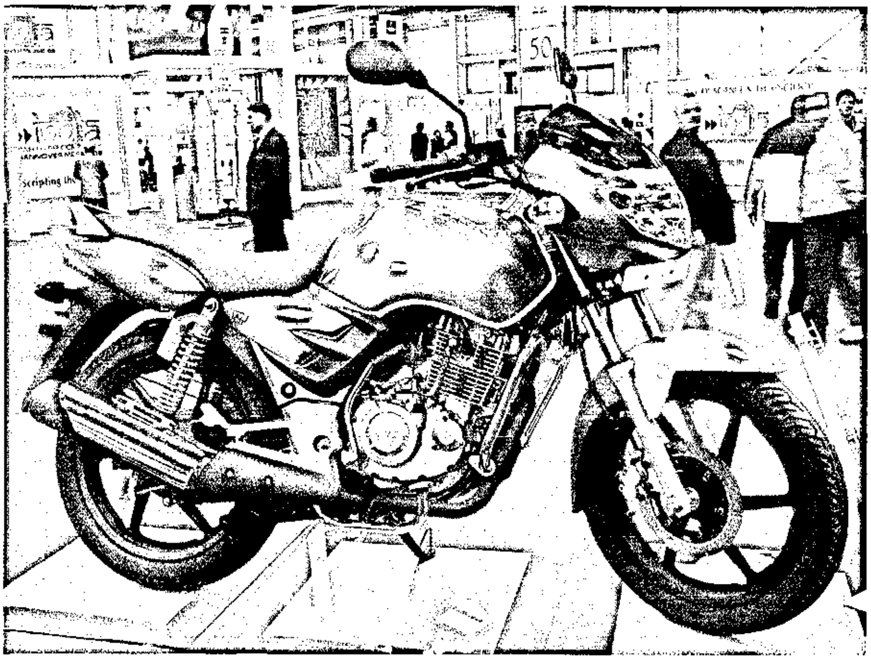


**Figure 98: TVS Apache at the Hannover Industry Fair in April 2006**
*(Reproduced courtesy of Planetary Engineering Group, Gartenau, Austria)*

Its engine builds on the mid-life upgrades to the 150 cm$^3$ "Fiero" engine by adding some of the friction reduction and gas dynamics techniques taught by the author and first applied to the smaller Centra engine. With fuel economy being a major driver in all segments of the Indian market, much of the combustion teaching and development has centred on lean burn rather than stoichiometric calibrations. By incorporating port design features to encourage charge motion and a suitably high combustion rate, the lean burn approach to fuel economy and emission control has been delivered in this new motorcycle engine. With the use of low friction features, mapped ignition and a carefully calibrated constant depression carburettor, the engine develops a competitive maximum BMEP of 10·5 bar and the highest power to weight ratio in its class, at 75·8 kW per tonne. A brief specification of the engine published by TVS, is as follows:

- 147·5 $cm^3$ single-cylinder engine with lean burn technology, developing 10·1 kW at 8500 $rev.min^{-1}$
- 12·3 Nm maximum torque at 6000 $rev.min^{-1}$
- Tuned intake and exhaust with resonators for better low speed torque, which translates to an excellent initial response and acceleration
- Inductive Digital Ignition (IDI)
- The Roller Cam Follower (RCF) technology for low engine friction providing greater fuel economy and durability

TVS continued to demonstrate its improved understanding of engine technology and its own research, with further papers in the SAE Small Engines Technical Conference held in San Antonia, Texas, 14-16 November, 2006. Of 9 papers authored or co-authored by TVS engineers, there were 5 engine papers.

## 7.5 Closing Remarks

This chapter has discussed the management of knowledge flow and has also described the author's work developing an award-winning means for knowledge transfer using the Internet. It has also described how the author's knowledge of engine research and development work, described in earlier chapters, has been effectively transferred to companies including TVS Motor Company for inclusion in a new range of engines, for which the fundamental teaching and research leadership was provided on-site by the author.

It is certain that most of the exhaust emissions and fuel consumption concerns that now affect the western automotive economies, could be avoided in India and other developing automotive markets through the implementation of such knowledge transfer. The author's teaching of engine design and development fundamentals has been accompanied by early introduction of lean-burn fuelling, digital mapped ignition, low-friction solutions and effective testing and development. This is enabling a fast-growing economy to have affordable transportation that is less damaging to the environment than its western counterparts at an equivalent stage of development. TVS is progressively launching its own range of motorcycles with class-leading power output and fuel economy, incorporating the teaching and research leadership of the author, and has been empowered to add to the body of knowledge with its own R&D.

Chapter 8 will formalise the declarations of this thesis.

# Chapter 8.

# THESIS DECLARATIONS

## 8.1 Author's Input

The invention or initial development of tumble or barrel swirl has not been attributed to any one individual or organisation. It is clear that the phenomenon was exploited for improved engine performance and combustion long before it was fully researched and understood.

As an engineering manager and director of the engines group at Tickford, the author led and was responsible for research, development, application and patents. This work included all the 5-valves per cylinder and variable geometry intake systems described in this thesis. As a senior engine engineer, the author was directly involved in the technical assessments and decision making for the direction of the projects, the resourcing and budgets for the work and technical review, steering, reporting and publishing of the outcomes. With a key business development and customer liaison role, the author was the leading interface with the clients, interpreting their product and project needs and delivering the technical results and recommendations.

The author had significant input into the development of port shapes, combustion systems and inventions described in this thesis. Where the publications included in this thesis were other than a printed technical journal paper, he presented all the submitted papers in the relevant conference or congress. He has single-handedly taught the engine research and led the application of design and development knowledge to the successful new engines now in production at TVS Motor Company.

The author has developed supply chain and knowledge transfer strategies that have been implemented in automotive companies across the world through his personal consultancy and through his teaching at universities, with students being drawn from the international automotive and manufacturing industries.

The author has contributed to the research, development and application of new techniques for delivering engineering and manufacturing skills with e-learning, particularly the development of the advanced video lecture technique, known as AutoView. This has been carried out through the AutoTrain project and subsequently through the creation of a learning development technology organisation, EuroMotor AutoTrain LLP.

This thesis and its associated publications demonstrate that the author has contributed to the body of knowledge in his field and, through his teaching and knowledge transfer, has actively disseminated and implemented that knowledge in the world automotive community.

## 8.2 Novelty and Originality

The novel research managed by the author at Tickford led to international patents being granted for 5-valves per cylinder technology and a continuously variable geometry induction system. Instead of publishing learned society or academic papers on the latter, effort was focused by the author's company on speeding up intake system development in contract R&D projects and the search for commercial application in the European motor industry with Mecaplast SAM, a leading French manufacturer of plastic automotive components.

As part of the author's work on air motion, two novel devices for evaluation of the relative design parameters were created for use with air flow bench measurements, a novel tumble chimney and a variable port configuration tool. These enabled fundamental research to be carried out into the nature of port-induced tumble. Furthermore, a novel technique for the study of air flow emerging from intake valves was developed by the author's work reported in this thesis.

As part of the author's work on the effective transfer of engineering knowledge, a novel and award-winning system was developed for the delivery of audio-visual lectures with multilingual subtitles, via the Internet.

## 8.3 Technical Significance

For those in the field of multivalve engines, and confirmed by Patent authorities, there are now only two ways in the world of incorporating 5-valves per cylinder and they relate to the patented work conducted by the author's team at Tickford and that done by Yamaha. Represented by the author, Tickford had licensing discussions with a number of major engine manufacturers regarding the design and Ford has taken the work into a "proof-of-concept" engine that has been described in this thesis.

Additional evidence of the technical significant of the author's work has come in the range of engine being produced by TVS Motor Company which are class leading in terms of performance and fuel economy in the Indian motorcycle market. These engines have only been developed since the author began teaching engine design and development to the company, which previously only built Suzuki motorcycles under licence. Through the teaching programme, post-module projects focused on product development and the author's engine research leadership, TVS Motor Company is now highly regarded in the motorcycle industry as a leading producer of successful engines. The most recent engines fitted to Centra and Apache motorcycles, have leading power and fuel economy in their respective classes. These engines have been designed and developed using the knowledge taught by the author and with his input.

The next section includes a list of the patents that have cited the work conducted by the author and his team.

## 8.4 Related Papers Since Publication

A number of Patents in the arena of multivalve engines have subsequently cited the 5-valves per cylinder work led by the author. These are summarised in Table 6 and some of the most relevant abstracts are included in the Appendices in Section 10.2:

| US Patent Number | Patent Title |
|---|---|
| 6895925 | Internal combustion engine having three valves per cylinder |
| 5960755 | Internal combustion engine with variable camshaft timing and variable duration exhaust event |
| 5957096 | Internal combustion engine with variable camshaft timing, charge motion control valve, and variable air/fuel ratio |
| 5950582 | Internal combustion engine with variable camshaft timing and intake valve masking |
| 5634444 | Intake port structure in an internal combustion engine |
| 5245964 | Intake port structure for internal combustion engine |
| 5205259 | *Modified cylinder head* |
| 5125374 | Valve actuating arrangement for engine |
| 5119785 | Intake apparatus for multi-valve engine |
| 5111791 | Cylinder head and valve train arrangement for multiple valve engine |
| 5099812 | Cylinder head for internal combustion engine |
| 5018497 | Multiple valve internal combustion engine |

**Table 6: US Patents that have cited patents created from the author's team's work**

## 8.5 Closing Remarks

This chapter has set out the author's input and the technical contribution of the engine research and technology transfer described in this thesis and its associated publications. It has also placed the author's work in context in terms of originality and the other engine research that has subsequently acknowledged it.

Chapter 9 will draw conclusions from the author's work and indicate areas where further research in the fields of his activity is assisting with the progress of development and application of the technologies and techniques described in the thesis.

# Chapter 9.

# CONCLUSIONS AND FURTHER WORK

## 9.1 Conclusions

The author's work over thirty years has delivered highly competitive road and racing engines with high performance, from engines of different origin. These include 2-valves per cylinder motorcycle engines and 2-valves, 4-valves and 5-valves per cylinder road car and racing car engines, which have the benefit of his research.

Reference has been made to benchmark 4-valve engines and to port deactivation. The latter was implemented prior to the author's work, in order to provide high levels of axial swirl when operating at part throttle. The use of tumble swirl has been successfully developed in work by the author, as well as by others working in the field. This approach provided for high levels of charge activity, and hence microturbulence at the point of ignition, in order to provide for rapid and consistent burn rates. Acceptable burn rates and CoV of IMEP were achieved with mixtures optimised for torque and those with high dilution by EGR, in different applications.

The compromise between port flow capability and port-induced air motion in the design features of port, valve and combustion chamber shape has been discussed, together with the significance of the piston crown shape in some applications. High tumble designs can create resistance to flow and are incorporated to create changes in the direction of the intake flow and hence the large scale motion of the induction charge in the cylinder. Tumble motion aids mixing, and the momentum is later broken down into microturbulence at the point of ignition to increase flame front wrinkling and increase the burn rate. The performance of these systems has been demonstrated on a 5-valves per cylinder engine running with dilute charge and with high performance from good port flow.

Airflow and motion compromises were explored in the author's work and a patented 5-valves per cylinder technology was developed that was the subject of some of the author's published work. Research into, and use of the parameters influencing tumble using a novel technique (i.e. a variable port configuration tool) proved to be an effective means of reducing the degree of compromise in engine performance and real world exhaust emissions. Further novel work was conducted in the measurement of tumble swirl and air flow. The author was instrumental in independently developing a tumble chimney for use with a Tippelmann swirl meter, which was notably different from other equipment being developed at that time. The author and his team also developed an innovative method for comparing flow fields emanating from around intake valves and used it to improve multivalve port flow performance.

Changes to engine valve gear layout and camshaft operation were developed during the author's work applying advanced combustion designs from the "no-compromise", high-cost, solution for Formula One to low cost, mass-producible, low-maintenance versions such as that created for the Ford Motor Company's "Modular" V6 engine. It was verified on this engine that tumble can be used to achieve similar emission performance and higher power output whilst eliminating port-deactivation IMRC, a costly means of generating high levels of part-load axial swirl to allow charge dilution. Further studies were conducted on other 5-valves per cylinder engines that demonstrated the benefits of the author's work at Tickford compared with that of Yamaha and its licensees.

In the course of the engine developments carried out by the author, a unique, steplessly variable induction system was designed and built. There have since been a number of production designs that adopted the major features of the work conducted by the author and his team. The novelty and innovation demonstrated by the work, created patents and engine design and development opportunities for the author's company in Europe, the US and the Far East. It was this aspect of international business development and company promotion that was a driver for the internal engine R&D spending at Tickford Limited. This sets the context for some of the author's published work.

Since the time of his early engine research publications, the author has continued to contribute to the field of engine design and development, particularly related to high performance SI engines, emission control and supply chain development, through teaching, further research and publications. The use of the Internet for knowledge transfer has been developed as part of the author's work, including a unique system for delivering audio-visual engineering and business lectures with multilingual subtitles.

The author has been responsible for transferring engine technology to the emerging Indian automotive industry having demonstrated particular success at TVS Motor Company as well as contributing to Tata Motors "Indica". Here the author led the on-site Indica powertrain launch team in its quality proving programme for three months in 2000. The value of the author's single-handed engine teaching and engine research mentoring at TVS Motor Company has been demonstrated in two indigenous production engine designs that are class leading and by the severing of the company's licensing arrangement with Suzuki Motor Corporation. The knowledge transfer and research leadership has resulted in additional contributions to the body of knowledge of engines in publications by postgraduate engineers taught by the author.

## 9.2 Further Work

If stoichiometric calibrations become mandated, as well they might, there has been sufficient ground work demonstrated in this thesis to suggest that high activity combustion systems will require less compromise than others in maintaining the optimum power output at stoichiometric calibration.

There is a whole range of TVS engines to be refreshed and new ones to be designed. The company is intent on joining the 3 and 4 wheeler markets and therefore multi-cylinder engines will be a future part of its range. Against increasingly stringent emissions regulations, including Euro IV in 2010, and the quest for ever higher fuel economy, new applications of advanced combustion systems such as those developed by the author, will be necessary.

The author's commitment to the application of his engine research knowledge and automotive engineering experience has also been delivered to some supply chain companies in the UK and to other clients in India. One of the latter is another example of a developing automotive component company benefiting from the contributions made by the author to the body of knowledge and this is enabling the fast growth of the Indian automotive economy with high performance products for a cleaner environment. There is an ongoing need for further provision of engine engineering knowledge in the developing global automotive supply chain.

Additional capabilities are being sought by users and deliverers of e-learning to add to the techniques already developed in the author's work. These include automatic synchronisation of different media, interoperable editing tools that can be used by lecturers and further bespoke information and knowledge assessment tools.

The author is already undertaking further work transferring knowledge though a number of universities where he held or holds posts as consultant, visiting lecturer and honorary research associate. By leading and contributing to ongoing engine research and knowledge transfer, the author himself is continuing to extend his field of expertise and knowledge of engines, combustion systems and emission control.

Eur Ing C.J.C. Bale

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March 2007

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# APPENDICES

# Appendix 1. Patents derived from the author's work

**European Patent EP0320233**

Multivalve Cylinder Head

Inventor; Lyle, Alastair Claude

Applicant: Aston Martin Tickford Ltd

States: AT, BE, CH, D , ES, F , IT, LI, NL, SE [1993/34]

Priority from GB Patent Application GB19870028660 19871208

Abstract

An internal combustion engine in which each cylinder has a plurality of inlet ports (17,19) and a plurality of outlet ports (14), each port being openable and closeable by means of a respective valve (12,13,11). The inlet and outlet valves ( 12,13,11) are disposed with their centre lines substantially on opposite sides of a cylinder centre plane (16) and angled generally towards said centre plane (16). There are three, or possibly more, valves on at least one of said sides, arranged with two outer valves (12) separated by one, or possibly more, inner valve (13). The angle of inclination with respect to said centre plane (16) for the outer valves (12) is less than or equal to the angle of inclination for the inner valve (13). The valves are preferably operated directly with a separate camshaft (18,20,15) being provided for each one or more valves inclined at the same angle.

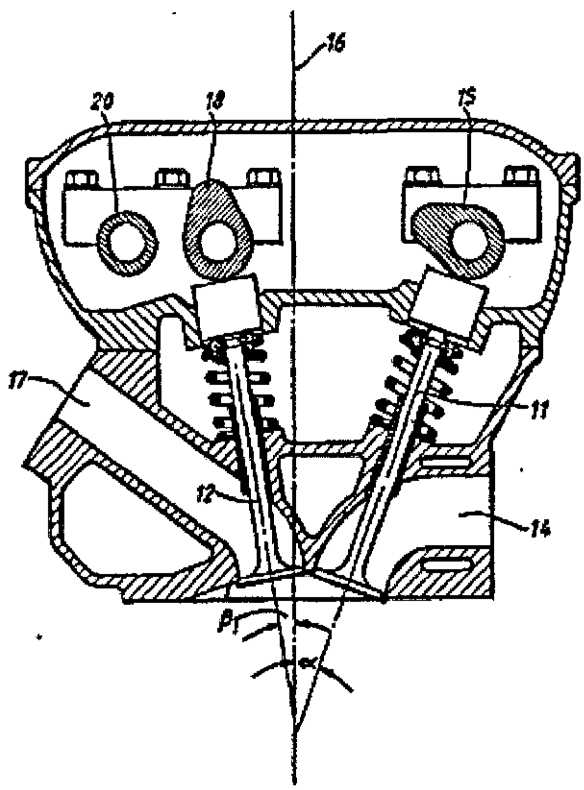



Fig.2.

**GB Patent: GB2213196**

Multivalve Cylinder Head

Inventor; Lyle, Alastair Claude

Applicant: Aston Martin Tickford Ltd

**US Patent: US4932377**

Multivalve Cylinder Head

Inventor; Lyle, Alastair Claude

Applicant: Aston Martin Tickford Ltd

Abstract:

An internal combustion engine in which each cylinder has a plurality of inlet ports (17,19) and a plurality of outlet ports (14), each port being openable and closeable by means of a respective valve (12,13,11). The inlet and outlet valves (12,13,11) are disposed with their centre lines substantially on opposite sides of a cylinder centre plane (16) and angled generally towards said centre plane (16). There are three, or possibly more, valves on at least one of said sides, arranged with two outer valves (12) separated by one, or possibly more, inner valve (13). The angle of inclination with respect to said centre plane (16) for the outer valves (12) is less than or equal to the angle of inclination for the inner valve (13). The valves are preferably operated directly with a separate camshaft (18,20,15) being provided for each one or more valves inclined at the same angle.

**Japan Patent: JP1301911**

Multivalve Cylinder Head

Inventor; Lyle, Alastair Claude

Applicant: Aston Martin Tickford Ltd

**Australia Patent: AU2671888**

Multivalve Cylinder Head

Inventor; Lyle, Alastair Claude

Applicant: Aston Martin Tickford Ltd

**Canada Patent: CA1334500**

Multivalve Cylinder Head

Inventor; Lyle, Alastair Claude

Applicant: Aston Martin Tickford Ltd

**Germany Patent: DE3883501D**

Multivalve Cylinder Head

Inventor; Lyle, Alastair Claude

Applicant: Aston Martin Tickford Ltd

**Cited documents:**
US2710602
US4667636
US4658780
GB2058919

**VARIABLE GEOMETRY I C ENGINE AIR INTAKE SYSTEM**

GB Patent GB2205609A – 1988

European Patent 0295064 - 1988

Inventors: Alastair Claude Lyle; Richard Geoffrey Sykes

Abstract:

An intake system for an internal combustion engine comprises an intake tube (5) for supply of air and a plenum chamber (8) in communication with said intake. At least one primary tube (11) communicates between the plenum chamber (8) and a respective cylinder of the engine. At least part of the or each primary tube (11) and at least part of said intake (5) are constituted by passageways (10,6) defined between cylinder and a barrel relatively rotatable therewithin. Relative rotation between cylinder and barrel in one sense causes the length of all said passageways progressively to decrease, while relative rotation in an opposite sense causes the length of said passageways progressively to increase. Hence the system can operate at optimum efficiency at different engine speeds.

# Appendix 2. Other relevant published material

Details of U.S. Patents that cite the Tickford 5-valve work (Abstract included for the most relevant citations)

PATENT NO. Title

6,895,925 Internal combustion engine having three valves per cylinder

Inventors: **Hannon; Mark S.** (South Lyon, MI); **Klotz; James R.** (Clemens, MI)

Assignee: **DaimlerChrysler Corporation** (Auburn Hills, MI)

Appl. No.: **245970**

Filed: **September 18, 2002**

5,960,755 Internal combustion engine with variable camshaft timing and variable duration exhaust event

Inventors: **Diggs; Matthew Byrne** (Farmington, MI); **Simko; Aladar Otto** (Dearborn Heights, MI); **Stein; Robert Albert** (Saline, MI)

Assignee: **Ford Global Technologies, Inc.** (Dearborn, MI)

Appl. No.: **094377**

Filed: **June 9, 1998**

5,957,096 Internal combustion engine with variable camshaft timing, charge motion control valve, and variable air/fuel ratio

Inventors: **Clarke; James Ryland** (Northville, MI); **Stein; Robert Albert** (Saline, MI)

Assignee: **Ford Global Technologies, Inc.** (Dearborn, MI)

Appl. No.: **094017**

Filed: **June 9, 1998**

Author's note: James (Jim) Clarke was the Manager of the Ford Advanced Powertrain Engineering Operations that commissioned the author and his team to create and develop 5-valves per cylinder designs for the Ford 2·5L V6 and 3·4L V8 engines in the period 1991-93

5,950,582 Internal combustion engine with variable camshaft timing and intake valve masking

Inventors: **Stein; Robert Albert** (Saline, MI)

Assignee: **Ford Global Technologies, Inc.** (Dearborn, MI)

Appl. No.: **093563**

Filed: **June 8, 1998**

5,634,444 Intake port structure in an internal combustion engine

Inventors: **Matsuki; Masato** (Saitama, JP); **Niwa; Hirosuke** (Saitama, JP); **Sugiyama; Izumi** (Saitama, JP)

Assignee: **Honda Giken Kogyo Kabushiki Kaisha** (Tokyo, JP)

Appl. No.: **429959**

Filed: **April 27, 1995**

5,245,964 Intake port structure for internal combustion engine

Inventors: **Matsuo; Syunsuke** (Kyoto, JP); **Hirako; Osamu** (Kyoto, JP); **Murakami; Nobuaki** (Kyoto, JP); **Akishino; Katsuo** (Kyoto, JP); **Furukawa; Keizo** (Kyoto, JP); **Ando; Hiromitsu** (Okazaki, JP); **Iwachido; Kinichi** (Nagoya, JP); **Motomochi; Masayuki** (Toyota, JP)

Assignee: **Mitsubishi Jidosha Kogyo Kabushiki Kaisha** (Tokyo, JP)

Appl. No.: **946476**

Filed: **November 10, 1992**

This invention relates to an intake port structure for an internal combustion engine. It is an object of the present invention to increase the strength of tumbling without lowering the maximum flow rate. An intake port (44) is constructed broader in a tumble-flow-side half (44a) than in the other half (44b) to have an intake air flow through the intake port (44) off-centered toward the side of a tumble flow, whereby an intake air flow from the intake port (44) promotes the tumble flow.

5,205,259 Modified cylinder head

Inventors: **Clarke; John M.** (Chillicothe, IL); **Faletti; James J.** (Spring Valley, IL); **Hackett; David E.** (Washington, IL)

Assignee: **Caterpillar Inc.** (Peoria, IL)

Appl. No.: **752507**

Filed: **August 30, 1991**

Multiple intake valves operatively associated in a common combustion chamber are advantageous in that the design achieves high output for an internal combustion engine. The subject modified cylinder head utilizes the advantages available in a multiple intake valve system, but further enhances the design by reducing heat rejection. In the subject modified cylinder head, three intake valves (38,40,42) having corresponding intake valve ports (26,28,30) and one exhaust valve (68) having an exhaust valve port (66) are operatively associated in a common combustion chamber. A reduction in heat rejection is achieved through a relationship between the cross-sectional areas of the intake and exhaust valve ports (26,28,30,66). The intake valve ports (26,28,30) are constructed so that their cross-sectional area is larger than about 69% of the combined cross-sectional area of the intake and the exhaust ports (26,28,30,66).

5,125,374 Valve actuating arrangement for engine

Inventors: **Saito; Tetsushi** (Iwata, JP)

Assignee: **Yamaha Hatsudoki Kabushiki Kaisha** (Iwata, JP)

Appl. No.: **550383**

Filed: **July 10, 1990**

A valve arrangement for a cylinder head assembly employing three intake valves and two exhaust valves for each cylinder. Not all of the intake valves reciprocate along parallel axes and hydraulic adjusters are operatively associated with the camshafts and rocker arms that operate the valves from the respective camshafts. The geometric relationship of the valves is chosen so as to maintain a compact and effective combustion chamber configuration and facilitate machining of the bores in which the hydraulic adjusters are positioned.

5,119,785 Intake apparatus for multi-valve engine

Inventors: **Saito; Fumihiko** (Hiroshima, JP); **Hashimoto; Noboru** (Hiroshima, JP)

Assignee: **Mazda Motor Corporation** (Hiroshima, JP)

Appl. No.: **670413**

Filed: **March 15, 1991**

An intake system for with an internal combustion engine, having a plurality of intake valves for one cylinder, includes a plurality of intake ports, opening into a combustion chamber, with openings which extend toward one side of the internal combustion engine. At least one exhaust port, opening into the combustion chamber, has an opening which extends toward another side of the engine, and is formed in the cylinder head on the other side, which is opposite to the one side of the internal combustion engine. The intake ports are arranged so that the centre intake port is inclined at an angle larger than an angle at which other, side intake ports are inclined, so as to direct fuel mixture flows, introduced through the side intake ports, toward an inner surface of the cylinder bore above a top of the piston at a lower dead point.

5,111,791 Cylinder head and valve train arrangement for multiple valve engine

Inventors: **Onodera; Hiroki** (Iwata, JP)

Assignee: **Yamaha Hatsudoki Kabushiki Kaisha** (Iwata, JP)

Appl. No.: **623698**

Filed: **December 7, 1990**

A cylinder head and valve train mechanism for an internal combustion engine having six valves per cylinder. There are provided four intake valves and two exhaust valves. In some embodiments, the size of the intake valves is varied because they are served by a common port so as to insure equal flow to the cylinder through all valves. In one embodiment, a single insert forms two of the valve seats. Also, two of the 4-valves are disposed at acute angles to both a plane containing the cylinder bore axis and a perpendicular plane passing through this axis in many embodiments. In these embodiments, the cam lobes that operate the angularly disposed valves have cam surfaces that are inclined relative to the axis of rotation of the camshaft. In some embodiments, all of the intake valves are operated by a single camshaft. In other embodiments, two camshafts operate different pairs of the intake valves. Various bearing arrangements for the camshafts are illustrated and described.

5,099,812 Cylinder head for internal combustion engine

Inventors: **Yamada; Tetsuro** (Iwata, JP)

Assignee: **Yamaha Hatsudoki Kabushiki Kaisha** (Iwata, JP)

Appl. No.: **578933**

Filed: **September 7, 1990**

cylinder head construction for a multiple valve engine that permits the use of at least three intake valves while permitting a single piece head construction and offering ease of access of the hold down fasteners for the cylinder head. The intake tappets are all slidably supported within a projection of the cylinder head with the outer tappets being positioned closer to a plane containing the cylinder bore axis so that fastener receiving bores can be formed outwardly from this point without interference from the projection.

5,018,497 Multiple valve internal combustion engine

Inventors: **Tsuchida; Naoki** (Iwata, JP)

Assignee: **Yamaha Hatsudoki Kabushika Kaisha** (Iwata, JP)

Appl. No.: **527564**

Filed: **May 23, 1990**

A multiple valve engine and more particularly an improved combustion chamber and valve arrangement for such an engine that permits high compression ratios, good breathing and a smooth combustion chamber. There are provided three intake valves and two exhaust valves. The centre intake valve is supported so that it reciprocates about an axis that lies in a plane containing the cylinder bore axis. The other valves are all inclined to each other so that adjacent edges between the valves will all lie in a common plane that extends perpendicularly to the cylinder bore axis to provide a smooth combustion chamber configuration. In addition, tapered cam lobes on the camshafts operate the angularly disposed valves. These tapered cam lobes have heel diameters that are larger than the adjacent bearing surfaces of the cams so that the cam lobes may be conveniently formed by grinding without damaging the bearing surfaces.

# Appendix 3. Author's publications


1. **Bale, C.J.C. (1984)** "Development of the Aston Martin V8 Endurance Racing Engine", Automotive Engineer, published by Mechanical Engineering Publications, Bury St Edmunds, UK, Volume 9, Number 4, August/September 1984, pp. 20-22.

2. **Bale, C.J.C. and Downing, I.C. (1990)** "The Design and Development of a Unique Five Valve Cylinder Head", Paper 905156, Proceedings of the Society of Automotive Engineers, XXIII FISITA Congress, 7-11 May 1990, Turin, Italy, Volume II, pp. 301-308.

3. **Bale, C.J.C. and Sykes, R.G. (1993)** "The Development of a High Output 2vpc SI Engine for the Ford Falcon", Proceedings of the Institution of Mechanical Engineers, Autotech 93 Conference, National Exhibition Centre, Birmingham, 16-19 November 1993, Volume C93, pp. 43-51.

4. **Bale, C.J.C. (1994)** "The Supply of Engineering Services in the Global Car Industry", Proceedings of SITEV Automotive Industry International Week, Lingotto Centre, Turin, Italy, 15-17 November 1994, published by Associazione Tecnica Dell'Automobile (ATA).

5. **Bale, C.J.C. and Sykes, R.G. (1995)** "High Specific Output, Low Emissions Spark Ignition Engines", Paper R95/122, Automotive Engineering Asia Technical Congress, Kuala Lumpur, Malaysia, 26-27 October 1995.

6. **Bale, C.J.C. and Neuhäuser, H-J. (1998)** "Technology trends in power cylinder systems", Automotive Technology International 1998, published by Stirling Publications Ltd., London, UK, ISSN 0950 4400, pp. 62-66.

7. **Bale, C.J.C. and Farnlund, J. (1999)** "Real-life Vehicle Exhaust Emission Performance Compared with Legislative Drive Cycles", Paper C575/030/99, Proceedings of the Institution of Mechanical Engineers, Volume 1999-9, Integrated Powertrain Systems for a Better Environment, pp. 27-47.

8. **Bale, C.J.C. (2001)** "High Performance Engineering", Proceedings of the Institution of Mechanical Engineers, Automobile Division 2001 Chairman's Address and Paper.

9. **Ashley, C., Bale, C.J.C., Millan, N., Williams, T.M. and Hendley, R.J. (2006)** "Latest Trends in Industrial Skills Development Techniques", Paper 197, Proceedings of the Institute of Cast Metal Engineers, World Foundry Congress, Harrogate, UK, 4-7 June 2006.

# Paper 1

# "Development of the Aston Martin V8 Endurance Racing Engine"

**Bale, C.J.C. (1984)**

Automotive Engineer

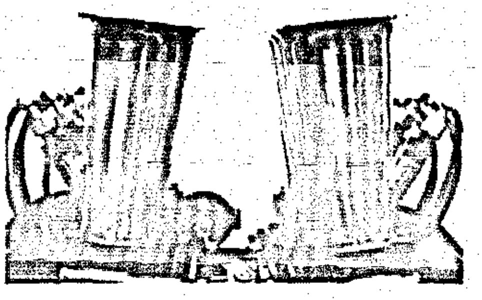
Published by Mechanical Engineering Publications,
Bury St Edmunds, UK.

Volume 9, Number 4, August/September 1984,

Pages 20-22

# Development of the Aston Martin V8 Endurance racing engine

C J C Bale, BTech, CEng, MIMechE, Senior Development Engineer, Aston Martin Tickford Ltd, lists the limitations of the standard engine and describes the changes made for racing, relating the development from analysis of component failure, through design to proving of new parts. Performance in actual races is described, with analysis of early defects.



*Induction system and petrol injection unit (Fig 4)*

The very existance of Aston Martin as a marque is the result of motor sport. Since the enthusiastic Lionel Martin first modified one of his dealership Singers for sprints and hill-climbs, the seeds were sown. Martin and his partner Bamford put together the highly rated Isotta Fraschini chassis and the new Coventry Simplex 1.4 litre engine in 1913 and competed most successfully at events such as the Aston Clinton Hill Climb. This in turn led to a demand for ready made cars from enthusiasts keen to follow Martin's success. In 1920 it was decided that the Aston Martin (combining the names of the founder and the site of his great success) should be a thoroughbred, being designed, developed, engineered and built as an individual car; this is a principle which still exists today.

The Aston Martin DBS model was first introduced in 1967 and 2 years later became available with a new 5.4 litre V8 engine with mechanical fuel injection (Fig 1). Later the V8 was changed from fuel injection to four twin barrel Weber carburettors for improved drivability and fuel economy. Further engine refinements, notably the introduction of the high performance 'Vantage' variant in 1977 and the revision of all engines in 1980, using polynomial camshaft profiles with revised ignition and carburation, bring the basic unit to where it is today.

## Engine construction

The engine is of all aluminium alloy contruction and is entirely built at the Newport Pagnell site. The block is mounted with centrifugally cast, chrome vanadium steel wet cylinder liners, being top-mounted and in direct contact with the water jacket. The nitrided crankshaft is of two-plane design carrying four big-end journals each bearing two connecting rods. The rods themselves are machined from bought-out forgings, are balanced, graded and matched with pistons into evenly weighted engine sets prior to assembly. The cylinder heads incorporate fully machined hemispherical combustion chambers with two large valves inclined at 64 deg. The valve guides are in bronze with the exhaust side in direct contact with the engine coolant. Steel valve seat inserts are used, both being assembled with a differential expansion technique of heated aluminium head and chilled guides and seats.

Four overhead camshafts run directly in the heads with five caps for each cam. The drive is by two-stage duplex chains from the crankshaft nose with manual and automatic tensioners. The cam lobes operate on hardened nickle molylodenum steel bucket tappets with shim-adjustment on the end of the stellite tipped valve stem. The four twin barrel carburettors feed fuel mixture to the engine from an electric fuel pump and remote wing-mounted air filters. The type and jetting of the carburettors differs according to engine variant and selling market. The production lubrication system draws from an 11.4 litre sump via a front mounted, chain driven oil pump.

Each engine (Fig 2) is assembled by one man on his engine stand with only the routine task such as fitting of camshafts to heads by repeated blue-ing and scraping, being entrusted to apprentice engine builders. The engine is run-in on the dynamometer where final leak and noise detection can be carried out. Each engine is subject to spot checking of power at various engine speeds, not to ensure repeatability as each builders engine performs differently, but to check that a minimum output level is reached. The engine is passed off with the builders name on a brass plate on the cam-cover and a fresh fill of oil, before moving to the assembly station to meet its chassis.

## Feasibility for racing

With the introduction recently of fuel capacity and refuelling regulations (capacity 100 litres falling to 80 litres in 1984 and 50 litres per minute refuelling rate), the emphasis was beginning to change from outright power to weight ratio to a combination of power to weight and fuel efficiency ratio in sports car racing. In April 1981 the possibility of using the Aston Martin V8 engine for racing in Group C was investigated as the inherent characteristics of the engine made it ideally suited; especially with these new targets in mind. The rationale was as follows:

- The engine is low revving allowing a higher top gear ratio giving lower transmission efficiency losses due to friction and oil churning in the gearbox.
- It has a wide torque range giving the ability to drop into a higher gear without going 'off the cam'. This is obviously beneficial to fuel consumption.
- The immensely strong and sturdy engine block and cylinder head assembly is well suited to use as a structural member, to enable chassis constructors to build a car down to the weight limit and employ 'ground effect' bodywork which in itself carries a weight penalty.
- The excellent combustion chamber design gives good specific fuel consumption throughout the range (Fig 3).

## Engine speed-effects on design

Fuel injected engines produce considerably more than 390 kW targeted at 7000 rev/min and also have better specific fuel con-

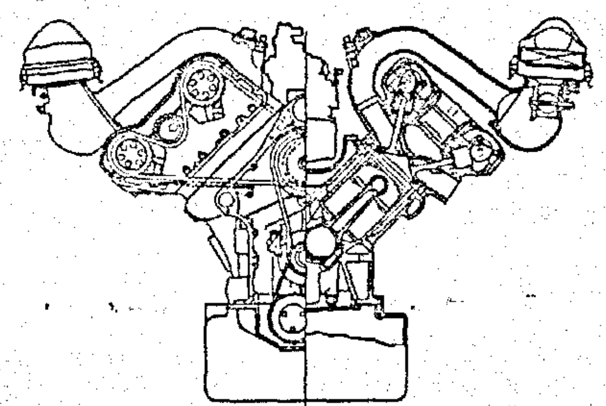


*Fig 1. Part section through original production engine with Bosch mechanical petrol injection and wet sump*

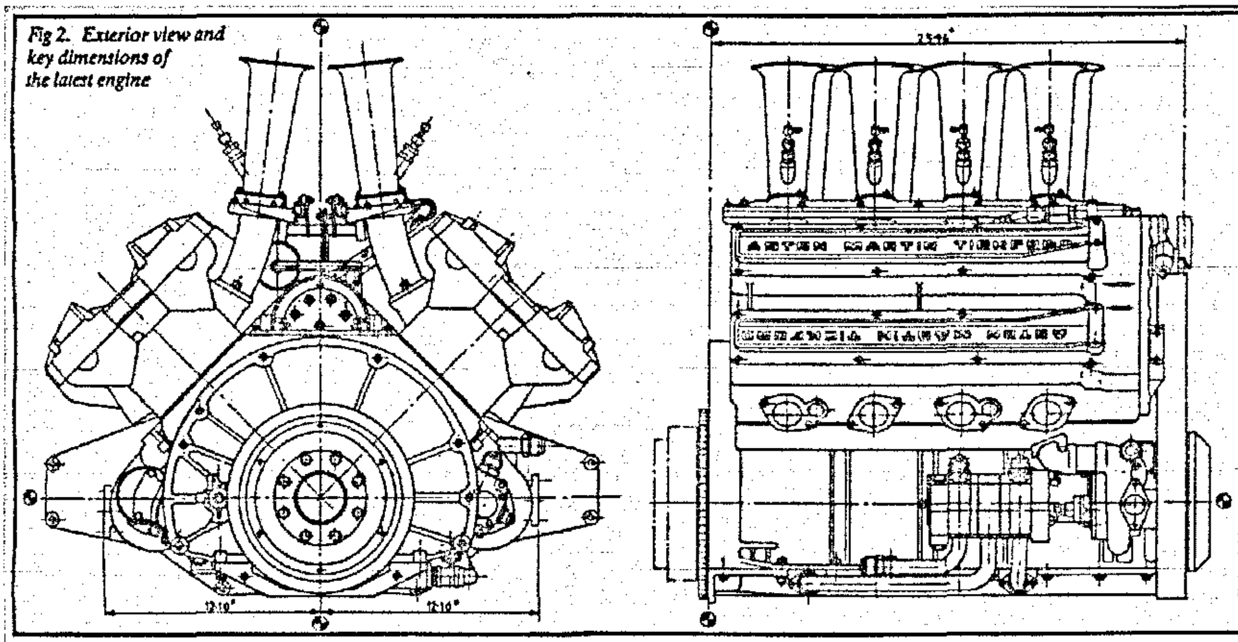


Fig 2. Exterior view and key dimensions of the latest engine

sumption than expected. The ceiling of 7000 rev/min was imposed because:

- The faster an engine runs, the lower its mechanical efficiency and also the greater the difficulty of maintaining good specific fuel consumption. Also the larger the cylinder volume is, then the harder it is to adequately fill the cylinders at high speed. The Aston Martin V8 has large cylinders.
- Since the engine only has two valves per cylinder it was deemed unlikely that it would be possible to maintain the necessary volumetric and thermal efficiency above 7000 rev/min. This is because the existing valve sizes cannot be increased due to space limitations, and the current racing camshafts are just about at the feasible limit of life without jeopardising the reliability of the engines valves gear.
- Crankshaft torsional vibration tests undertaken have shown that above 7000 rev/min the amplitude of the crankshaft deflection begins to rise quite rapidly, which will lead to a reduced crankshaft fatigue life should this limit be regularly exceeded.
- The original connecting rod assembly was also considered to be a potential weak point of the engine should 7000 rev/min be exceeded by any significant amount. The latest forgings are an improvement and provided that there are no hidden flaws under the surface of the material which cannot be seen under crack detection, then the new forged rods and steel caps should make the engine very reliable when running to a ceiling of 7000 rev/min. It was also considered that the gudgeon pin, which is 23.8 mm diameter in the standard engine, could be beneficially increased in diameter to 25.4 mm. The new rod forgings and latest pistons have 25.4 mm diameter pins in the interests of reliability.

## Track development

The petrol injection system chosen was the mechanical racing system from Lucas (Fig 4). The major advantages of the system are that the introduction of fuel is timed to the inlet valve opening period, also the slide throttle gives a clean tract, which is free from the venturi fundamental to carburettors, and as the fuel is pressure-fed there is no fuel surge under high cornering forces which might starve the engine of mixture. The fuel is supplied from a high pressure electric pump for starting, thereafter a mechanical pump on the left hand front cover provides fuel. That front cover is a new unit cast in magnesium alloy in lieu of the standard aluminium part. The throttle slide covers are also of magnesium alloy and these directly carry the steel trumpets with integral injector mountings. New intake manifolds carry the charge the short distance to the inlet port. The mechanical metering and distribution unit is housed in the engine vee, driven off the intermediate timing gear sprocket.

The ignition system is the Lucas Capacitive Discharge racing system which is crankshaft-triggered. This system means very accurate ignition timing unaffected by scatter and keeps the electronics off the engine in the control unit. The front cover and torsional vibration dampers are modified to carry the pick up and triggering slots respectively. The ignition timing is fixed, with an electronic 10 degrees retard during cranking. The distributor is gutted and carries the high-tension ignition voltage only. Racing sparking plugs — of the surface discharge type — are sometimes used, depending on conditions.

The race pistons are forged, not die cast like production units, and have a new crown shape which improves combustion and eliminates detonation. The valve pockets are modified to accommodate the new valve lift. Very quickly a second series was produced to carry the 25.4 mm gudgeon pins. Early engines fitted with 23.8 mm diameter gudgeon pins and flat Seager circlips had shown signs of distress in the pin bores and also raised a burr on the outsides of the circlip grooves. Later pistons and rods therefore used a 25.4 mm diameter pin to improve the pin boss bearing pressures, and also round wire circlips which are less prone to knock-out than flat Seager circlips. After one engine managed to knock-out four round wire clips whilst on test at Silverstone, the design has been further refined so that the 25.4 mm diameter pins have been shortened slightly, and a Nylatron thrust pad has been inserted between each end of the gudgeon pin and the round wire circlips. This later set-up has eliminated the problem, but the severity of the gudgeon pin knocking in the failed engine suggested that there might have been a vibration in the car which was aggravating the problem, this was especially so since another engine which had completed over 2400 km on the early Seager circlips, had shown only quite mild evidence of circlip hammering.

During testing a connecting rod cap failure occurred, and it was decided that new caps machined from solid EN19T were needed.

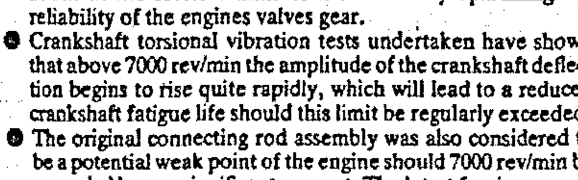


Fig 3. Cylinder head section

These were designed and a set manufactured in one week. This failure was certainly contributed to by running the engine on the speed limiter which makes the engine operate as if it is on overrun, since this is the only way that it is possible to generate a suitable tensile stress in the area of failure. The failure initiated in the fillet radius around the bolt boss spotface and proceeded inwards through the weakest cap section until it reached the big end bearing bore.

The standard production dual valve springs incur valve crash at 7500 rev/min, but were used on all the carburated racing engines. For the fuel injected engines new springs were obtained which had a new rate and a degree of coil interference to damp surge. These changes raised the new crash limit to 8200 rev/min which provided a greater safety margin. An early standard valve spring failure during a track test was investigated, and showed minute particles of surface oxide on the spring towards the bottom of the shot peening indentations, due to the springs being from an old batch and being stored under production, less than ideal conditions for racing components! Later springs were made from Oteva 82 which is less prone to sag, than other materials, so that strict control of the shimming of valve spring fitted length should eliminate all but the occasional random failure that occurs with all valve springs.

The valves themselves are hollowed less than standard to make the heads stiffer but they are run on thinner seats, and are made lighter overal, raising the valve crash limit to 8400 rev/min. The racing valves are, like production valves, Tuftrided to ensure trouble free life where the exhaust valves are inclined at only 13 deg from horizontal when installed. Investigation of the distributor rotor arm and cap from the first engine to race at Silverstone, showed that the rotor arm runs very close to some of the segments in the cap. In fact, vibrations had caused it to hit three of them. Hence the latest rotor arms have the brass portion shortened by 0.75 mm radially and are fixed to the distributor shaft with Sealastic RTV as a means of damping out vibrations.

For the 24 hour du Mans race, the engine was rebuilt (Fig 5) and upgraded to steel big end caps and 25,4 mm gudgeon pins, with latest specifications giving 418 kW at 7000 rev/min and 597 Nm torque at 5000 rev/min. That engine did a further demonstration run at Silverstone before removal form the chassis, duration being an amazing total of 37 hours running.

## Further development

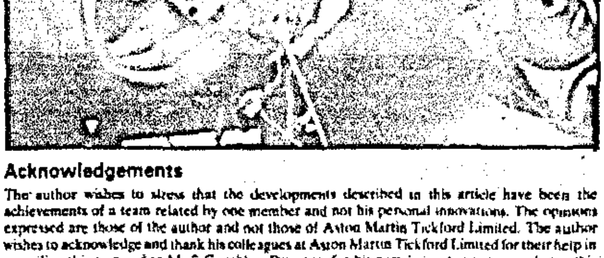
For the 1983 engine several revisions were incorporated, with the aim of reducing the weight of the power unit whilst consolidating its reliability. The front of the engine was re-designed with simpler auxiliary drives and the centre vee changed to incorporate the distributor and alternator alongside the mechanical fuel metering unit. The exisiting timing case was replaced by a new magnesium casting to accept the new auxiliaries and their drive system (Fig 6), via a toothed belt from the front end of the compound half speed timing sprocket. The existing water pump was replaced by two smaller pumps mounted each side of the engine. The oil pumps assembly being driven via an Oldham coupling from the rear of the right hand water pump. A take off from the left hand water pump drives the mechanical fuel pump. The magnesium timing case itself acts as the principal means by which the engine is bolted to the monocoque tub of the latest chassis.

The starter installation is further revised with the motor now integral with the engine. The remote bendix is driven by an angle-jointed shaft. The inclusion of even an alternator on the engine means the unit is completely self contained. All engine covers are now in magnesium alloy with the necessary mountings for the car's rear suspension on the cam covers. The engine is now 125 mm shorter than the series production unit, reflecting the need for a compact installation. The outcome of development meant revised intake trumpet length and exhaust system dimensions, with valve timing changed to increase the spread of the torque curve, whilst minimising the reduction in top end power. A weight saving of 22.7 kg on the engine has been made by substituting aluminium alloy cylinder liners for steel, magnesium alloy engine covers for aluminium ones and revising the auxiliary drive system. The liners have a coating of Nicasil and the new pistons by Mahle have a compound top ring of stainless steel with molybdenum infill. This prevents the rings from picking up in the bore whilst maintaining sufficient support to prevent the ring collapsing.

Further work had been done in the valve spring area, with a change to progressive rate springs with special surface treatment. This change removes the need for coil interference and has extended spring life under racing conditions. The valve spring top retainers have had to be strengthened to survive our valve spring rig test, and the race engines have adopted the new type for durability. The valves themselves have only had minor changes, the main one being the use of Nimonic 80 exhaust valves, less prone to fracture than the earlier type. They cope with high temperature much better and reduce valve seat wear. A final small weight saving has been achieved by reverting to aluminium intake trumpets, which had earlier been prone to fracture, by the development of a flexible mounting, much lighter than its predecessor, having lighter trumpets to carry.

During 1984 the engine would have been changed from mechanical to electronic petrol injection had the regulations not changed. This will permit optimised engine mapping date to fuel the engine giving even better fuel efficiency. The resultant fuelling accuracy will yield the best output from the engine where it was previously misfuelled by the less precise mechanical system. There are also schemes for digital ignition mapping and for incorporating four valves per cylinder.

*Fig 5. Engine dynamometer test*



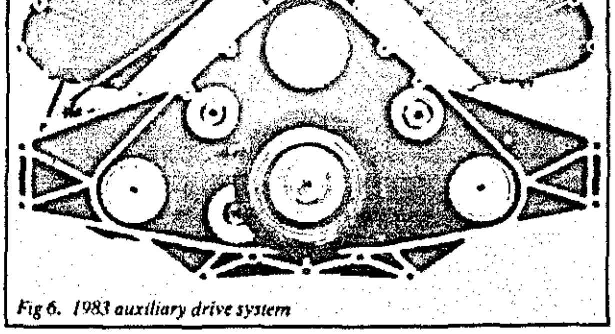


*Fig 6. 1983 auxiliary drive system*


### Acknowledgements

The author wishes to stress that the developments described in this article have been the achievements of a team related by one member and not his personal innovations. The opinions expressed are those of the author and not those of Aston Martin Tickford Limited. The author wishes to acknowledge and thank his colleagues at Aston Martin Tickford Limited for their help in compiling this text and to Mr S Coughlin, Director, for his permission to prepare and give this article. In particular the autor is endebted to Mr A C Lyle, Chief Engine Designer, Mr D B Morgan, Chief Development Engineer and Mr A G Wilson, Development Engineer, for their detailed information and race reports.

# Paper 2

## “The Design and Development of a Unique Five Valve Cylinder Head”

**Bale, C.J.C. and Downing, I.C. (1990)**

Proceedings of the Society of Automotive Engineers

XXIII FISITA Congress

Turin, Italy

7-11 May 1990

Paper 905156

Volume II, Pages 301-308

# XXIII FISITA CONGRESS

## TORINO (ITALY) 7-11 MAY 1990

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905156

# The Design and Development of a Unique Five Valve Cylinder Head

**I. C. Downing, C. J. C. Bale,** Tickford


ABSTRACT

The paper details the design and development of a unique five valve cylinder head. The new cylinder head design is significant as it advances the performance boundary for the naturally aspirated petrol engine.

The performance limitations of current four and five valve cylinder head designs are discussed with reference to air mass flow, combustion chamber shape and valve train loads.

The layout of the new design is detailed and the improvements of the design relative to existing five valve layouts are highlighted.

The development of the design is discussed with particular attention to inlet valve flow shrouding and inlet camshaft profile development.

The anticipated performance increase of the five valve design relative to a four valve design for the same bore diameter was demonstrated on two engine types.


THE LIMITING FACTOR FOR ENGINE PERFORMANCE is ultimately the mass of air flowed by the engine and hence the mass of fuel burned at the corresponding air/fuel ratio. The principal limitation of an engine to achieve the maximum airflow is inlet valve area which in turn dictates inlet port area. The search for more power has lead to the development of the four valve per cylinder engine to its current levels. However, the ultimate performance of the four valve design is geometrically limited by the inlet valve area obtainable from normal circular bores. This has lead to the investigation of various five valve per cylinder layouts which offer the potential of greater performance by utilising a greater percentage of the cylinder bore area as inlet valve area. This paper discusses the design and development of a unique five valve per cylinder design, demonstrates the performance advantage achieved to date and indicates those areas where further work is required to exploit the full potential of the design.

FIVE VALVE CYLINDER HEAD DESIGN

The five valve cylinder head design described in this paper was originally designed for use in high performance racing engines. The layout of the chosen five valve design was based on a review of existing four and five valve designs. The limitations of these current designs can be summarised as follows:-

Four Valve

a. Inlet valve area is limited to approximately 28 % of bore area for typical inlet to exhaust area ratios and material thicknesses between valves.

b. Inlet valve lift is limited by effective valve mass and resulting operating stresses in the valve drive train.

Five Valve

The combustion chamber shape is compromised by the requirement to use a single inlet camshaft with direct operation of the valves. This requires the tops of the valve stems to be closer together than the valve heads. This results in a combustion chamber that narrows in the region of the third, or central inlet valve and can result in poor cylinder scavenging and a reduced detonation limit.

However, the five valve layout has the following advantages:-

Five Valve Advantages

a. For a given bore diameter, the five valve layout provides the maximum achievable inlet valve area as a percentage of bore area (32 %), based on typical inlet to exhaust area ratio and material thicknesses between valves. This characteristic has been investigated and reported in Reference 1.

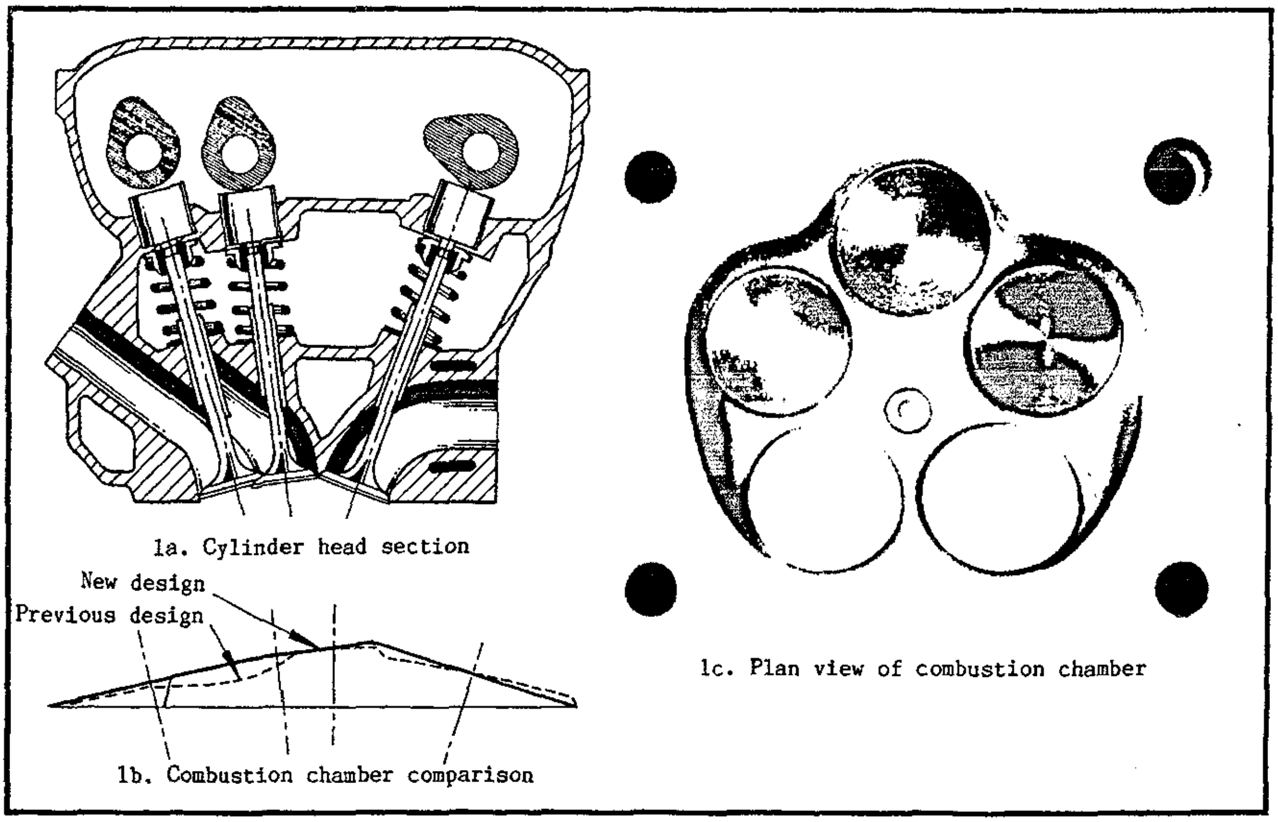



FIG. 1 FIVE VALVE DESIGN DETAILS

b. The five valve layout provides an increase in inlet valve curtain area of approximately 25% relative to a four valve layout for the same bore diameter.

c. Reduced effective individual inlet valve mass.

d. Reduced inlet valve lift to achieve the same airflow as a four valve design for the same bore diameter.

e. Lower valve train loads than an equivalent four valve design for the same performance and bore diameter.

A new five valve design was investigated to take advantage of the benefits of the five valve layout and to remove the compromise caused by the use of a single inlet camshaft. The layout of the new design is shown in Figure 1. Figure 1a shows the inclination of the inlet valves which has allowed the combustion chamber to be opened up, thus avoiding the thinning of the combustion chamber in the region of the central inlet valve which occurs with other five valve designs. For the racing version of the engine, two inlet camshafts with direct action have been used. One camshaft operates two inlet valves per cylinder, the second camshaft operates one inlet valve per cylinder. The camshafts are connected by synchronisation gears which ensure simultaneous valve motion for all three inlet valves.

An alternative single inlet camshaft layout has also been detailed for passenger car engine applications where cost and simplicity is of greater importance. This layout features direct actuation of the two inner inlet valves with hydraulic followers and actuation of the third inlet valve by a rocker arm with hydraulic adjustment.

The design also features inlet ports which are all at similar angles relative to the valve head to optimise air flow into the cylinder. The outer ports are angled in to converge at the inlet manifold face, this reduces the total port volume and improves the dynamic response of the gas column in the inlet ports.

Figure 1b compares a cross section of the combustion chamber for the new design with that of a typical five valve design. The illustration clearly shows the improvement in combustion chamber shape offered by the new design in the region of the central inlet valve. The improved combustion chamber shape allows greater scope for the compression ratio to be raised without forming an excessive area of thin section at the outer edge of the chamber which can promote detonation at the high levels of performance achievable with a five valve design.

Figure 1c shows a plan view of the combustion chamber for the 90mm bore diameter engine.

Patent application has been made for the design in the following countries :- Great Britain (2213196A), Europe (88311584.2), United States (07/281,228), Japan (310997/88), Australia (26718/88), Canada (T.B.A).

DEVELOPMENT OF THE DESIGN

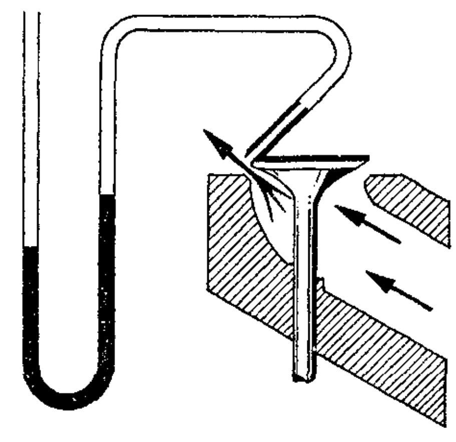
Two engines have been developed with the new five valve cylinder head design. The first engine was developed to demonstrate the performance improvement achievable with the five valve design relative to a comparable four valve design. This engine was a 3.5L V8 Formula 1 engine with a bore diameter of 90mm and a stroke of 68.6 mm. The second engine was also a 3.5L V8 Formula 1 engine but with a bore diameter of 94mm and a stroke of 63 mm for use in the 1989 Formula 1 series.

CYLINDER HEAD DEVELOPMENT — A five valve design geometrically offers a gain of approximately 25% in flow at low valve lifts compared to a four valve design due to the increased curtain area which results from the greater inlet valve perimeter of the five valve design. At higher valve lifts the flow becomes influenced by valve throat area and in this region the five valve design offers an increase of approximately 14% over a comparable four valve design due to its increased usage of bore area.

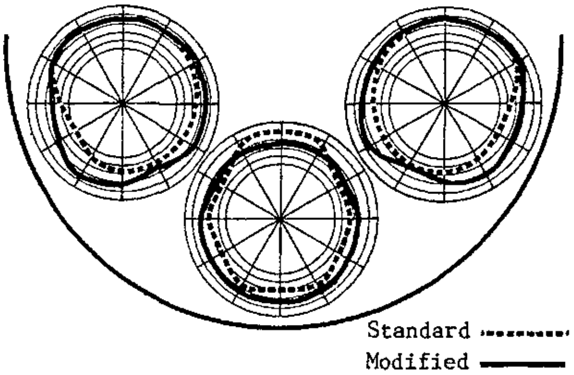
Extensive air flow rig testing was performed on the cylinder head design to realise the theoretical air flow improvement of the five valve design compared to an equivalent four valve design. The air flow rig used a viscous flow element to measure mass airflow and use of a suite of computer programmes allowed the calculation of terms such as flow coefficient, Mach number etc..

With all cylinder heads, flow out of the inlet valves is reduced due to shrouding by the combustion chamber walls and interference between the flows from adjacent valves. These effects are particularly significant on a five valve design due to the closeness of the combustion chamber wall to the outer inlet valves and the interference that occurs between the flow from the central inlet valve and the two outer inlet valves.

Therefore, in addition to basic air mass flow measurements, studies were made of the air flow profile around each valve using the apparatus shown in Figure 2a. The apparatus consists of an inlet valve with a small hole through the valve seat and at right angles to the seat. The hole is connected to a low density fluid manometer capable of resolving small pressures. The gap between the valve head and valve insert seat acts as a venturi when air flows and the greater the air velocity at a given point the lower the static pressure at that point due to Bernoulli's relationship. Therefore, high flow corresponds to a lower static pressure as measured by the manometer. Tests were made by measuring the valve seat static pressure profile for each inlet valve at 1mm lift steps up to 10mm with all the inlet valves being opened simultaneously. The pressure profile was measured by rotating the valve through a complete rotation in 30 degree increments and measuring the static pressure at each point. Analysis of the pressure profiles showed regions where flow was reduced due to flow interference between adjacent valves or shrouding due to the combustion chamber walls.



2a. Pressure measurement apparatus



Standard

Modified

2b. Inlet valve pressure profiles

FIG.2 INLET VALVE PRESSURE PROFILE MEASUREMENT

Figure 3 shows flow coefficient (Cv) against L/D ratio for the five valve cylinder head at several stages of development, this non dimensional presentation of flow data allows comparison with other designs regardless of size or valve layout. The significant stages were as follows:-

Stage 1. Baseline measurement of original design. The flow coefficient (Cv) of the initial design was found to be lower than a comparable four valve design but mass flow was still higher due to the greater valve area of the five valve design. The low flow coefficient was found by pressure measurement to be due primarily to shrouding of the flow from the outer inlet valves by the combustion chamber walls.

Stage 2. The outer inlet valves were unshrouded by machining a clearance cut in the combustion chamber at a radius centered on the valve centreline and inclined at the valve guide angle relative to the cylinder head. This gave an improvement in flow at all valve lifts. Further unshrouding of the combustion chamber around the outer valves was not possible within the confines of the bore.

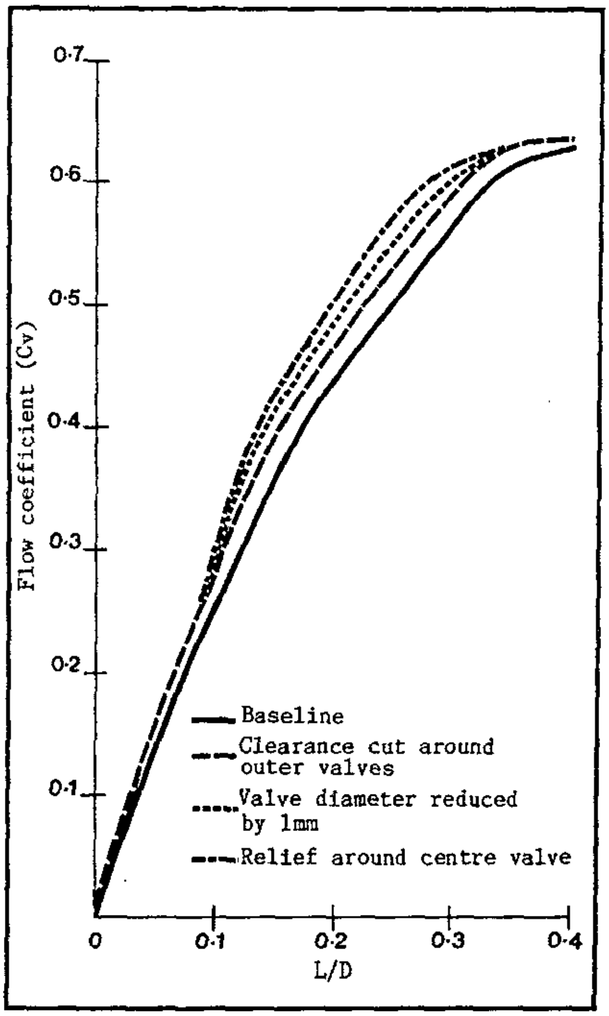



*FIG. 3 EFFECT OF AIRFLOW MODIFICATIONS*

Stage 3. The diameter of all the inlet valve heads were reduced by 1 mm, this improved clearance around and between the valves and gave substantial improvements at L/D ratios of 0.1 to 0.35.

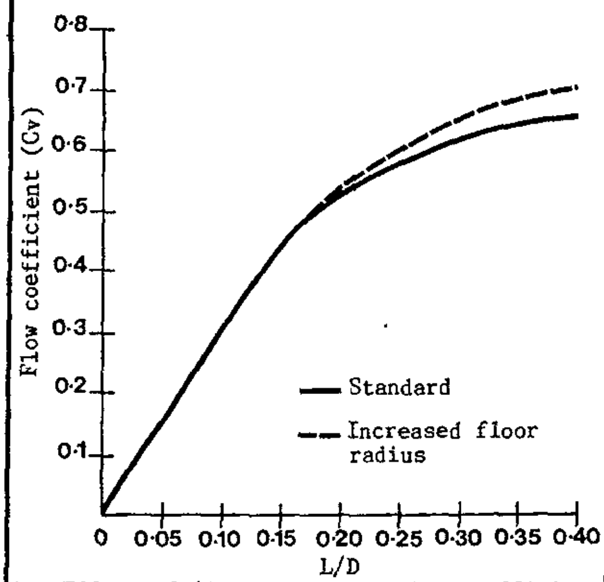
Stage 4. Pressure profile measurements showed that shrouding of the outer valves by the chamber had been reduced as far as was practical within the constraints of the bore diameter. However, these measurements showed that flow interference between the centre and outer inlet valves was still significant and that the flow out of the centre inlet valve was being restricted at the sides and rear. This was causing extra air to be forced out of the front segment of the valve resulting in interference of the flows from the outside valves. The chamber was relieved around the centre valve to give a minimum clearance of 3mm between the outer edge of the valve and the combustion chamber, this gave a further improvement in flow coefficient.

Figure 2b shows the effect of the reduction in chamber shrouding and flow interference developed during stages 1 to 4 on the pressure profile around the inlet valves compared to the original design.

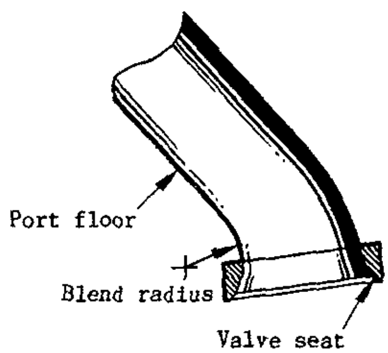
Additional testing showed that further material removal from the combustion chamber around the centre inlet valve did not give any significant improvement in air flow but would lead to loss of compression ratio.

Stage 5. The inlet ports to each valve were flowed individually by closing the valves of the other ports, filling the inoperative ports with modelling clay and streamlining the entrance to the port at the junction of the three inlet ports in the cylinder head. Previous testing had shown that results were inaccurate if this procedure was not followed.

The flow performance of the centre port is shown in Figure 4a. The port performance up to a L/D ratio of 0.21 was good, but at higher ratios did not achieve that measured for high performance four valve ports. Further investigation showed that the port geometry was similar to that of the high performance four valve port except for the radius between the port floor and the valve seat as detailed in Figure 4b. This radius was increased and the effect of this change is shown in Figure 4a. The efficiency of the centre port now equalled that of the four valve ports investigated.




4a. Effect of floor radius on flow coefficient




4b. Schematic diagram of floor radius

FIG. 4 EFFECT OF PORT FLOOR RADIUS

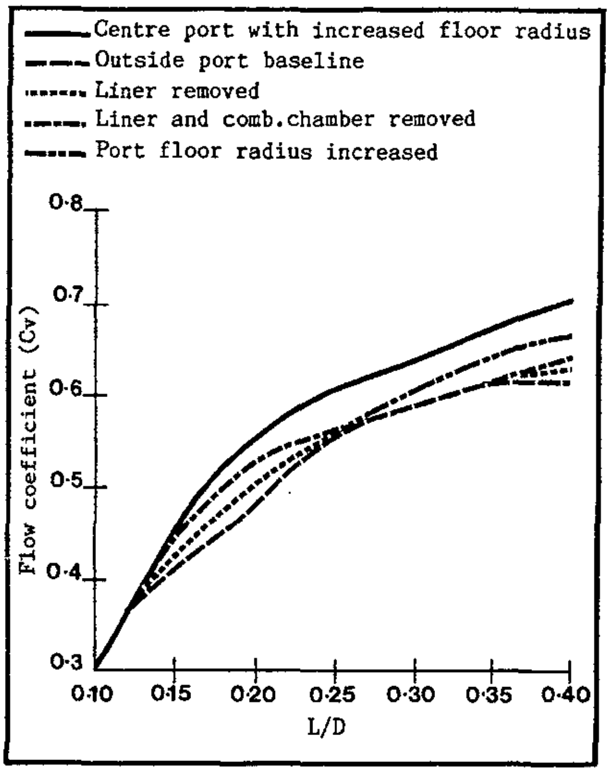



FIG. 5 EFFECT OF LINER AND CHAMBER SHROUDING

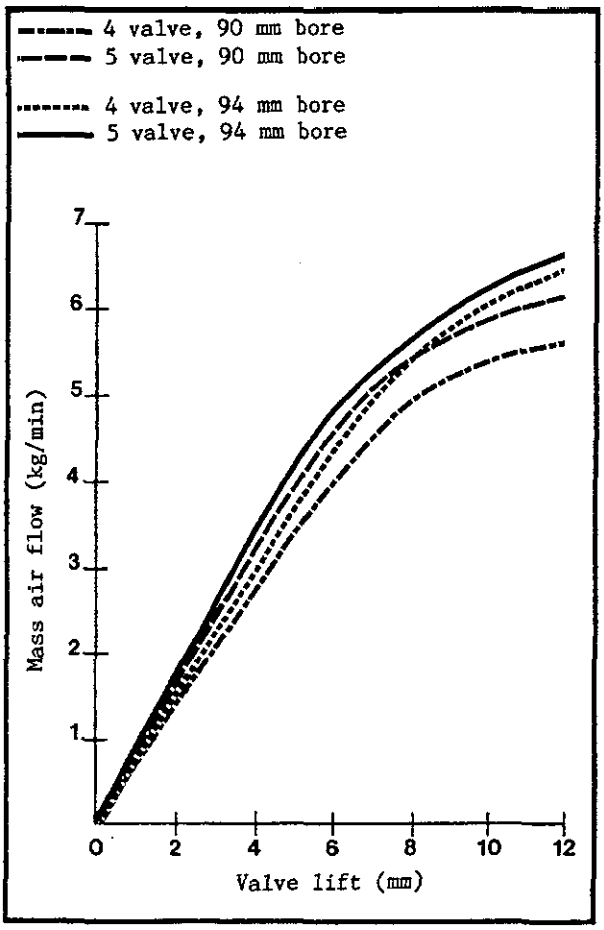



FIG. 6 INLET MASS AIR FLOW

The flow performance of the outer inlet port is shown in Figure 5. The outer inlet ports were less efficient than the centre port, particularly in the L/D range of 0.125 to 0.25. This was believed to be due to the angle of approach in a horizontal plane of the port to the valve or shrouding of the valve by the combustion chamber and cylinder liner. A series of tests was made to investigate the effect of each factor. Figure 5 shows the results of these tests. Firstly *the cylinder liner was removed, this gave a* small improvement in port efficiency in the L/D range of 0.125 to 0.25. The combustion chamber was then completely removed around the valve to give an unrestricted exit from the valve, this gave a further increase in port efficiency in the L/D range of 0.125 to 0.25 but no improvement elsewhere. The port floor to valve seat radius was then increased as for the centre valve, this gave a similar improvement in the L/D range above 0.25. These tests showed that the lower efficiency of the outer inlet ports was due to the geometry of the port relative to the valve seat. It has not been possible on the two versions of the design developed to date to change the angle of the outer ports. It is intended to design *and develop a further version with an improved* angle of approach for the outer inlet ports.

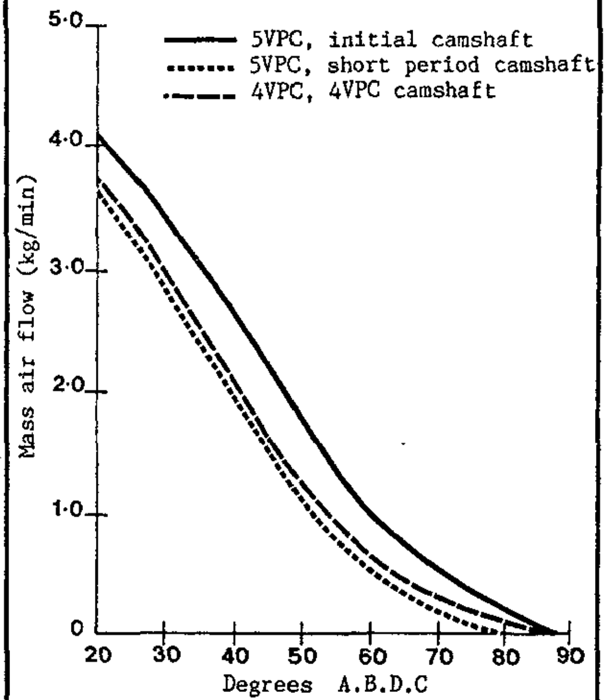
Air mass flow for the fully developed versions of the two five valve cylinder heads are compared to the corresponding four valve designs of the same bore diameter in Figure 6. This shows that the five valve layout is capable of flowing significantly more air.

## INLET CAMSHAFT PROFILE DEVELOPMENT

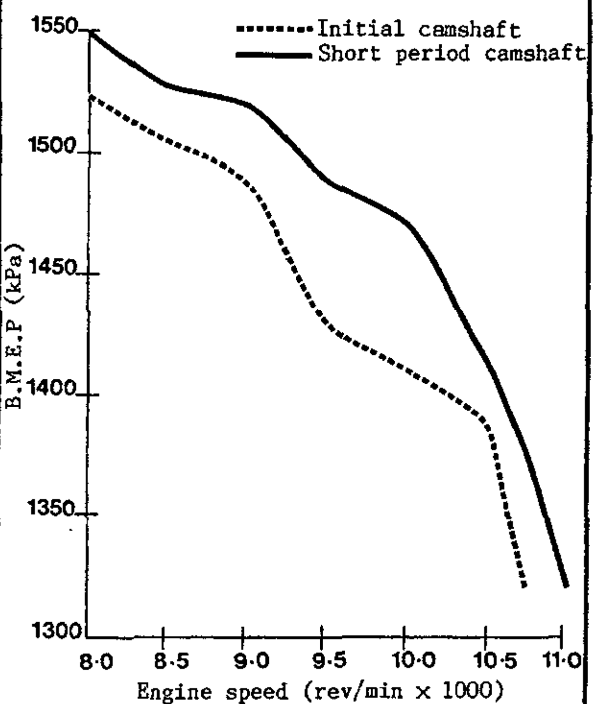
INLET CAMSHAFT FLOW REQUIREMENT -- High performance racing engines use a combination of wave inertia and resonant effects to generate volumetric efficiencies significantly above 100%. Inlet inertia ram is used in conjunction with late inlet valve closing to give cylinder filling after bottom dead centre by converting the kinetic energy of the inlet gas into pressure energy behind the inlet valve. This allows flow to continue up to the point at which cylinder pressure rises above port pressure and the flow direction reverses. Empirical values are used for inlet port sizing which relate inlet port gas velocity to the volume flow expected at the maximum power speed of the engine. The values are well established within a band of velocities which allow the engine to produce an acceptable spread of torque and flow sufficient air to produce its maximum power speed.

The closing point of the inlet valves also has a significant effect on performance. If the valves are closed too early, full benefit is not gained from the inertia ram effect after bottom dead centre. Alternatively, if the inlet valve is closed too late, the pressure due to inertia ram is overcome by cylinder pressure and charge is forced out of the cylinder by the piston.

Initially, the inlet camshaft profile for the five valve design was designed based on four valve experience. Engine performance was disappointing and the engine exhibited severe blow back of inlet charge and a narrow spread of torque.




7a. Effect of camshaft profile on mass air flow




7b. Effect of camshaft profile on engine performance

FIG. 7 EFFECT OF INLET CAMSHAFT PERIOD

Curves of mass flow against camshaft angle were calculated for the five valve design and the base four valve design. These are compared in Figure 7a. The curves were aligned by matching their maximum lift points. The mass flow curves for each configuration were measured using the air flow rig with a constant pressure differential across the valves. Assuming that both engine types are designed with a similar port velocity at their maximum power speed and have identical piston movements, then the pressure differential across the inlet valve will be similar at a given point in the cycle. Therefore, the flow into or out of the cylinder must be due to the inlet valve area present at each point in the intake cycle. Therefore it appeared that the blow back was due to the increased curtain area of the five valve design giving greater flow for a given pressure differential near the point of inlet valve closing. A new inlet camshaft profile was selected by calculating the inlet valve lift profile required to match the flow through the valves to that of the four valve design over the region before valve closing. The effect of this new inlet camshaft profile on engine performance is shown in Figure 7b.

INLET CAMSHAFT DRIVE LOADS -- A significant problem with Formula 1 type engines is the high loads generated in the valve drive train due to the combination of large valves, high valve lift and large valve acceleration required to achieve the maximum performance. These loads are normally expressed in terms of an instantaneous 'stab' torque which acts as a shock load through the valve train and can lead to failure of valve train components. Although the effective weight of an individual inlet valve assembly for a five valve design is lower than that of an equivalent four valve design, the combined effective weight of three inlet valve assemblies is greater than the two inlet valve assemblies of an equivalent four valve design, based on similar materials being used for both. In addition, as previously explained, the five valve design requires an inlet camshaft having a shorter period to prevent blowback. To accommodate the requirement for a shorter camshaft period and to remain inside the stab torque limit, either a reduction in the effective mass of the valve train or a reduction in maximum valve lift is required. It was not possible within the constraints of the project to reduce the valve train mass and therefore the inlet camshaft profile was designed to give maximum performance within a stab torque limit.

Figure 8 shows harmonic diagrams for the selected five valve inlet camshaft profile and for the four valve inlet camshaft profile used on the 94mm bore diameter engine. The magnitude of excitation produced by the camshaft profile is plotted against orders of engine speed. The harmonic diagram was generated by performing a Fourier Analysis on the acceleration diagrams of the camshaft profiles. The five valve inlet camshaft profile produces significantly lower excitation than the four valve inlet profile for the same engine. This was achieved by a reduction in maximum valve lift of 6% for the five valve camshaft profile relative to the four valve and detailed analysis of the profile acceleration diagram. Because of the airflow increase of the five valve design relative to the four valve design, it has been possible to use lower valve lifts without a significant reduction in performance. However, the use of lighter materials and a reduction in valve train component mass is an objective of future developments. This will allow valve lift to be increased within the stab torque limit and for further gains to be made in performance.

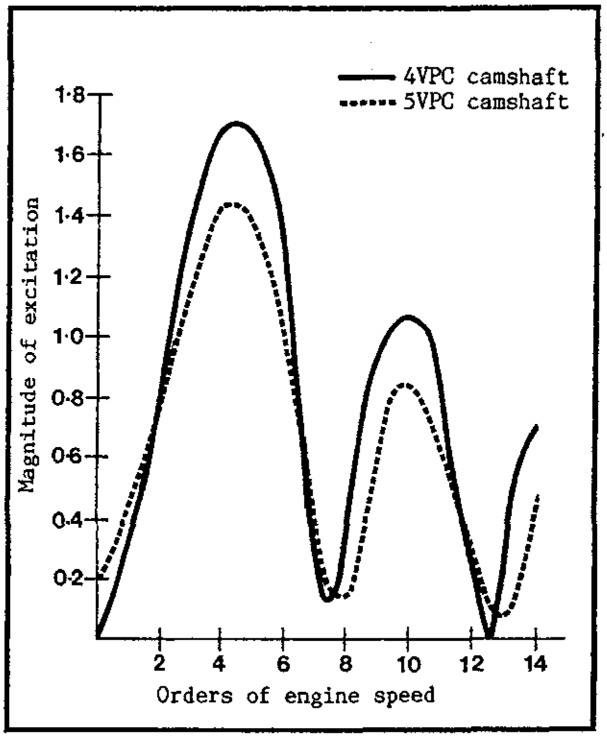



FIG. 8 INLET CAMSHAFT HARMONIC CURVES

PERFORMANCE SUMMARY

The previous sections describe the development of the main features of the inlet system of the engine. Obviously, work was performed on other features of the engine to obtain the maximum performance but this work was similar to that carried out for four valve units and is not considered to demonstrate any new technology. Figure 9 shows a summary of engine performance for the two five valve engines developed against their respective base four valve engines. Both five valve units produced maximum power at higher engine speeds than their four valve equivalents. For comparison purposes the engine speed points of the power curve have been plotted on a percentage of the maximum power speed basis. It can be seen that both five valve units produce significantly more power than their four valve equivalents, the increase in maximum power being 4.3% and 4.2% for the 90mm and 94mm bore diameter versions respectively.

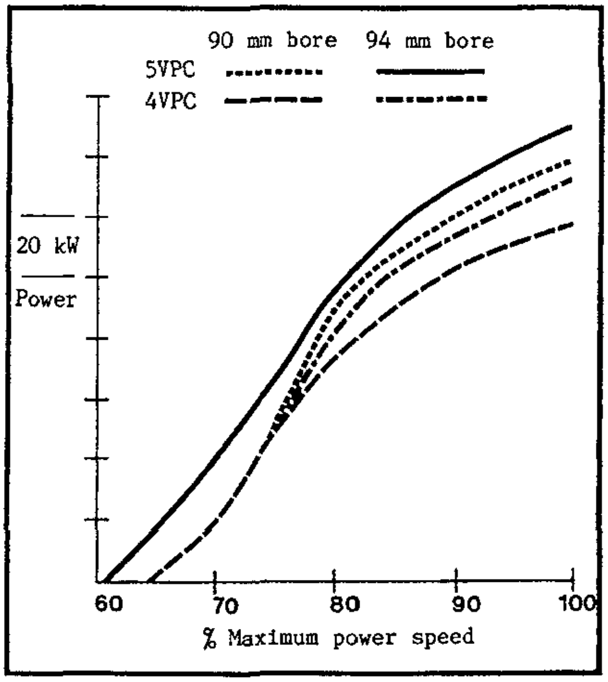



FIG. 9 PERFORMANCE SUMMARY

ADDITIONAL DEVELOPMENTS

SWIRL CONTROL -- Both axial and barrel (or tumble) swirl were measured using a Tippleman swirl meter and suitable adaptors for the 94mm bore diameter five valve cylinder head, for a number of valve opening arrangements.

Axial swirl is defined as rotation of the air charge in the cylinder about a vertical axis located on the bore centreline. Barrel swirl is defined as rotation of the air charge about a horizontal axis across the cylinder and parallel to the inlet valves.

Figures 10a, 10b and 10c show measurements of air mass flow, barrel swirl and axial swirl respectively. Figure 10a shows that airflow increases almost linearly with the number of inlet valves open for the five valve design, although the flow with three valves open is lower than the theoretical value of the sum of the three individual valves due to flow interference between the valves. Figure 10b shows barrel swirl for the two symmetrical combinations of inlet valve opening which produce purely barrel swirl on the five valve design. These combinations are a) all inlet valves open, b) the outer two inlet valves open. Also shown is barrel swirl for the base four valve design. It can be seen that the barrel swirl produced with all three inlet valves open is significantly higher than the four valve design and is similar with two inlet valves open. Figure 10c shows axial

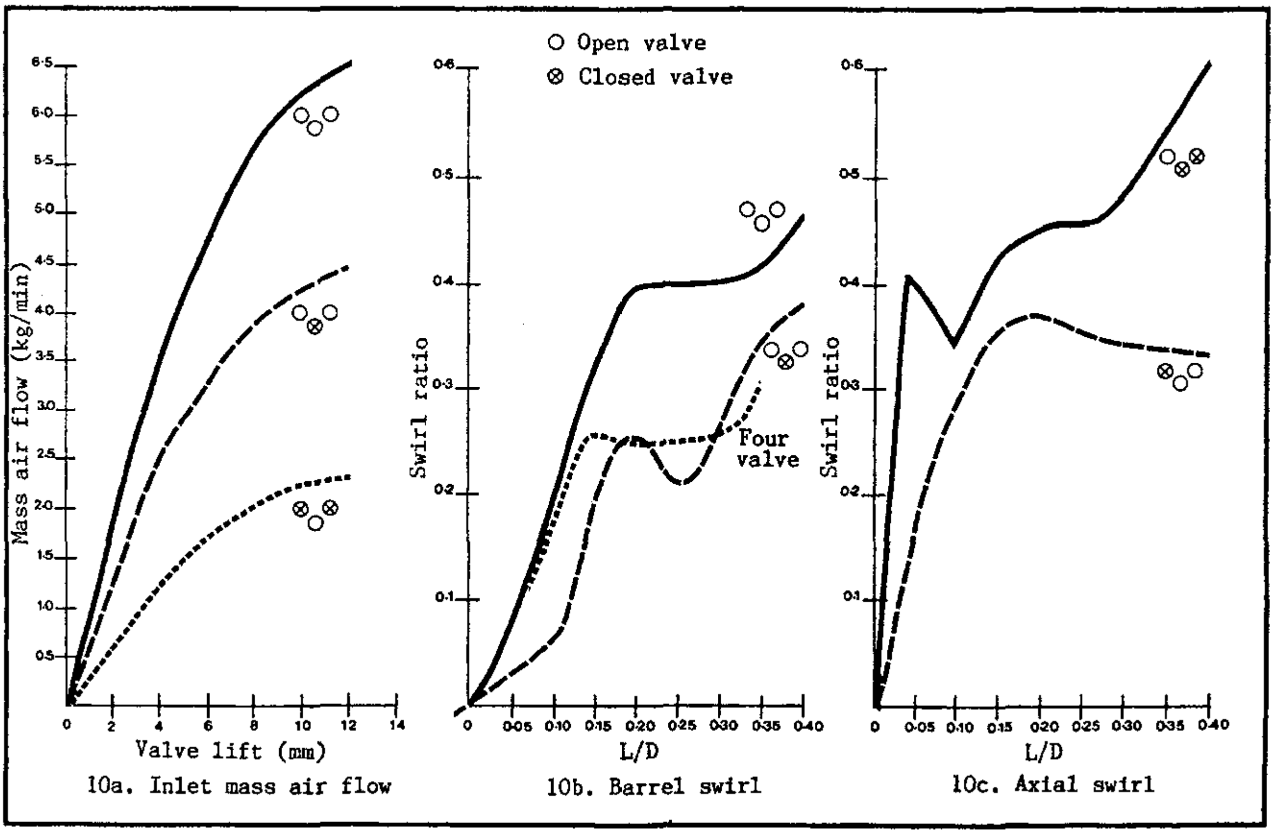



FIG. 10 BARREL AND AXIAL SWIRL RESULTS

swirl for the five valve head for the one and two inlet valve open cases which primarily produce axial swirl.

At the time of writing, insufficient engine testing has been done to ascertain the effects of these changes in swirl characteristics on engine performance. However, we believe that the increased flow region over which swirl control is available from a five valve design combined with the increase in maximum airflow, offers the potential for an engine with port or valve de-activation to have an exceptional torque spread and good emissions control.

## CONCLUSIONS

1. For a given bore diameter the five valve design gives significantly greater air flow than a four valve design.

2. The present five valve design has produced approximately 4.3% more power than the equivalent four valve design.

3. The mass flow versus crankshaft angle curve before the point of inlet valve closing, is similar for five and four valve versions of the same engine.

4. Valve train loads for five and four valve versions are similar.

5. The anticipated potential for high specific output engines combined with useful air motion has been identified.

## NOMENCLATURE

| | |
|---|---|
| B.M.E.P | Brake mean effective pressure |
| Cv | Flow coefficient based on inner seat diameter |
| D | Effective valve seat diameter |
| L | Valve lift |

## ACKNOWLEDGEMENT

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# Paper 3

# "The Development of a High Output 2vpc SI Engine for the Ford Falcon"

**Bale, C.J.C. and Sykes, R.G. (1993)**

Proceedings of the Institution of Mechanical Engineers

Autotech 93 Congress

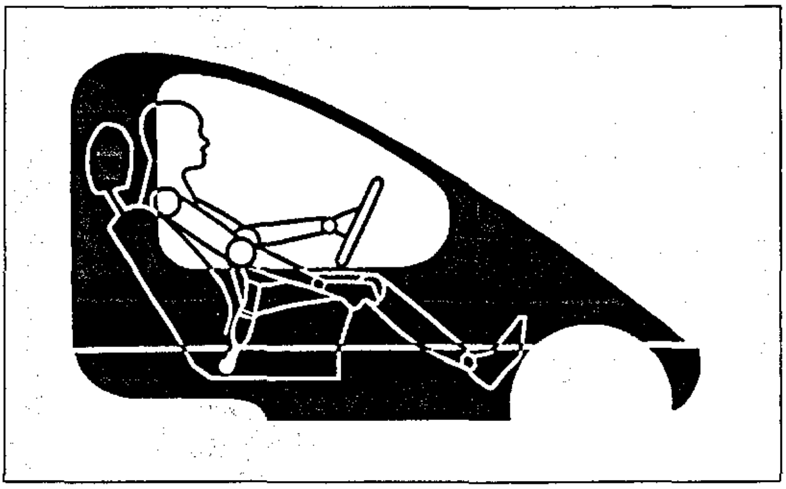
National Exhibition Centre, Birmingham, UK

16-19 November 1993

Volume C93, Pages 43-51

I MECH E

# Automobile emissions and combustion



Selected papers from AUTOTECH '93

MEP

# The Development of a High Output 2vpc SI Engine for the Ford Falcon XR6

by C J C Bale, Eur Ing, BTech, CEng, MIMechE, MSAE, and R G Sykes, *Tickford Ltd*

SYNOPSIS A higher performance derivative of a 4 litre I6 petrol engine was developed with particular regard to minimising tooling and production costs. The resulting package offers very good value for money and has proved to be outstandingly successful in the marketplace.

## INTRODUCTION

There are many examples of the application of new (and currently expensive) technology to motor vehicles. It is always the hope that ways will be found to incorporate such techniques at acceptable cost. This paper describes a different but very typical aspect of automotive engineering, that is to provide the customer with what he or she wants at the minimum cost.

In 1990 the senior management of Ford of Australia realised the need for a facility to develop and manufacture low volume special versions of their products.

It was finally decided to set up a separate joint venture operation with another company already experienced in this type of work.

After a thorough world-wide search Tickford was selected as the partner and in 1991 Tickford Vehicle Engineering Pty (TVE) was established close to Ford of Australia's Broadmeadows plant on the outskirts of Melbourne.

TVE started by carrying out tasks such as the fitting of optional accessories which were inconvenient to fit in the main factory. Hitherto some of this work had been carried out by dealers on a one at a time basis resulting in high labour costs and inconsistent quality.

In parallel with these activities projects were started to provide Ford of Australia with some derivative sportier models of the Falcon to improve the image of the range.

The first uprated model to enter production was the Falcon XR6, announced in Autumn 1992, and it is the engine of that vehicle that is the subject of this paper.

## THE DESIGN BRIEF

The design brief for the XR6 was classically simple. To increase performance at minimum cost as a means of producing a value for money "Gentleman's Express".

It was essential that there should be minimum change to the manufacturing process as the engine had to be built down the same line as the standard power unit.

The finished product should have very similar fuel economy to the base vehicle and it had to meet the same drive by noise and exhaust emission regulations. It was always the intention that this vehicle should be sold complete as a fully homologated vehicle when new and not rely in a loophole in the regulations which allow anti-social modifications to be carried out to a vehicle after it has been registered.

Finally the vehicle had to carry the same warranty as the standard car so all modifications had to be thoroughly proven to normal Ford standards.

## SPECIFIC PERFORMANCE TARGETS

Production 4.0 litre engines in the Falcon EB2 are quoted as producing 148kW at 4500 rev/min. and a maximum torque of 348Nm at 3750 rev/min. The target performance was set at 160kW at no higher than 4750 rev/min with at least the same torque at 3500 rev/min as the base engine.

It was known that this increase in performance could be achieved easily given a no expense spared approach but the development exercise was concentrated on selecting those components which would provide the desired performance improvement at minimum production cost.

## DESCRIPTION OF THE ENGINE

The Ford of Australia in line 6 engine has a bore and stroke of 92.26 x 99.31mm giving a total swept volume of $3984cm^3$. It has been developed to be admirably suited to typical Australia driving where high torque is appreciated more than high rotational speed. It follows that if the engine was to be asked to run at speeds higher than originally intended a fundamental re-design would be required.

The combustion chambers are part spherical and contain one inlet and one exhaust valve each set at approximately 20 degrees from the cylinder axis.

The valvegear is quite cheap, compact and effective. A single camshaft mounted along the centre of the head operates the valves through forged aluminium rockers fitted with roller followers at the inner ends and hydraulic lash adjusters above each valve.

The inlet manifold has a single plenum chamber with individual runners or primary pipes leading to each port. Top feed solenoid injectors are carried in the manifold flange and spray petrol down each port. The fuelling and ignition control is by a variant of the Ford EEC IV system using speed and inlet manifold pressure as the basic input parameters.

## COMPONENTS THAT WERE EVALUATED OR CHANGED

### Inlet port and Valve

The proximity of the edge of the inlet valve seat to the edge of the bore prevented any increase in valve diameter without offsetting the valve axis. Although this sort of action may be taken when preparing special engines for competition it is totally out of the question for mass production. To do so would have required a massive resetting of the head machining line for each batch of special heads. Not surprisingly it was decided to continue with the standard valve and seat insert which provide an inner seat diameter of 43.2mm. The baseline inlet port had been well developed on the flow bench and under steady flow conditions showed characteristics which would be difficult to improve upon. The main bore from the inlet manifold face was of 39mm diameter bulging out to approximately 46mm diameter just downstream of the valve guide then narrowing in to 41mm diameter through the throat of the seat insert. Tickford considered that the inlet gas velocity in the main bore of the port was slightly lower than optimum and thus reduced the port size to 37mm. It was known that a bulge in the port just upstream of the valve seat was detrimental to the dynamic breathing therefore this region was reduced in diameter to provide a gentle increase in cross sectional area from the straight section to the seat insert.

In fact there were two stages of experimental inlet port designed to these guidelines and which are outlined in fig 1. The accompanying flowbench results of these and the standard port are shown in fig 2. The initial modified port with a 50mm radius between the floor and the seat insert showed satisfactory results on the flowbench in that it achieved the same flow as the standard from a smaller port. Unfortunately it would have required modifications to the water jacket core to ensure adequate metal thickness.

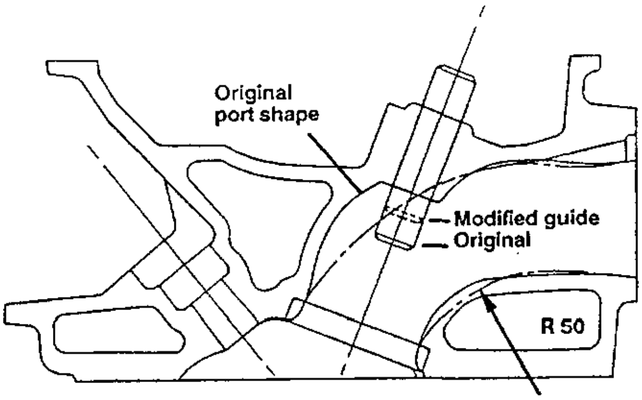



Fig 1a Low Line gradual expansion inlet port superimposed on the original shape.

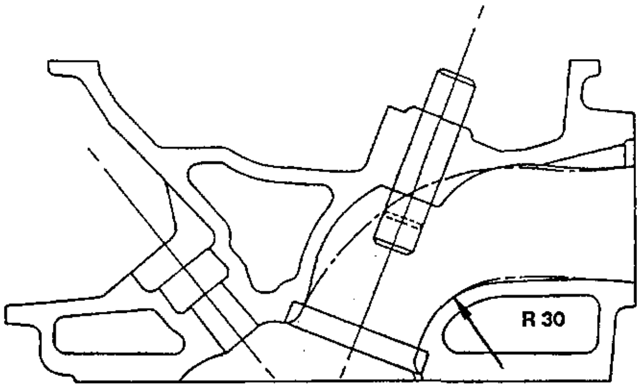



Fig 1b *Second version of improved inlet port which avoided changes to the coolant jacket*

A decrease of floor radius to 30mm vastly reduced the tooling costs but resulted in a flowbench performance marginally inferior to the standard port. Despite this the second version of the modified port was incorporated into an engine where it showed a worthwhile performance improvement over the standard design. This experience is one of many which have confirmed that port development carried out only on a steady state air flow bench can sometimes lead to imperfect results.

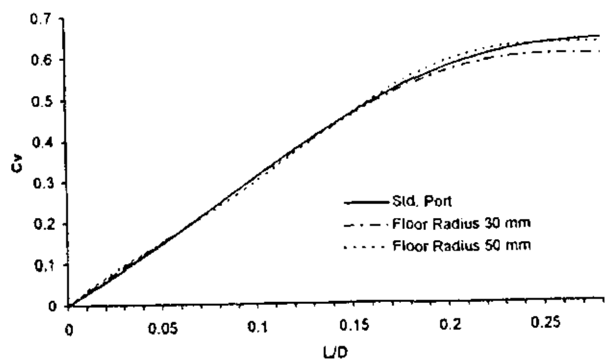



*Fig 2* *Inlet Port Flow Results*

The modifications were relatively inexpensive to incorporate as only the port core shapes were changed within the standard head casting equipment. It was also necessary to make small changes to the port throat machining just upstream of the valve seat insert but this was easily achieved by a change of tool within the standard holder. To ensure that the desired shape could be achieved in production the Tickford liaison engineer temporarily resident in Melbourne was asked to find the producers of the tools used for this operation at the Geelong engine plant. From the other side of the world came the slightly surprising answer that the tool tips originated from a factory only 3 miles away from Tickford in Milton Keynes, England.

**Exhaust port**

Minor changes to the exhaust port made a useful improvement to the flow. The tendency for flow separation between the valve seat and the port floor was cured by raising the floor at the valve end and providing a larger radius. It was not possible to raise the roof by a similar amount without an expensive change to the water jacket so an unconventional means had to be adopted to retain the necessary cross sectional area. At the valve end of the port the cross section was locally distorted towards a D shape with the flat downwards resulting in a good combination of cross sectional area and smooth flow.

Removal of the intrusion of the valve guide, and its boss, from the top of the port improved flow considerably but inevitably reduced the support of the valve. A short but very interesting investigation was undertaken into the effects of valve guide length on valve gear noise and on exhaust valve temperature. As a result of this work it was possible to choose a suitable guide length which intruded only slightly into the port yet supported the valve adequately.

**Exhaust manifold**

The standard exhaust manifold consists of two castings each collecting gas from three cylinders. An interconnection allows the exhaust gas from the front three to flow into the back casting where it joins the flow from the rear three cylinders. As a result the secondary pipe lengths are very short.

There are various aftermarket tubular exhaust systems available for this engine and many tests were run on the dynamometer with different combinations of pipe diameter, primary and secondary pipe lengths.

It was proved that some, but certainly not all, configurations produced a significant increase in torque or power. Unfortunately there were severe problems in packaging these tubular systems whilst retaining the required clearances, providing a suitable catalytic converter position and providing for adequate heat shielding of adjacent components. There were also indications that the reduced mixing of the exhaust gases could cause erroneous readings from the HEGO sensor. Finally it was apparent that some development work would be required to reduce the direct radiated noise from the thin wall tubular exhaust manifold.

Solutions to these problems were schemed but it became obvious that the time and cost involved could not be accommodated within the programme. As an example the shortening of the catalytic converter entry pipe which connects to the end of the exhaust manifold would have incurred approximately $Aus 200,000 in tooling charges for the pipe, converter entry cone and heat shield plus an increase of around $Aus 50 in the piece part.

With these problems and the need to avoid any high risk it was decided, with a little regret, to continue with the standard exhaust manifold.

This was a good example of the need for a project management team to take a firm line to ensure good value in terms of performance increase for time and money.

**Exhaust Pipe, including Catalyst and Silencers**

Initial running on the engine dynamometer was carried out with a back pressure valve in the exhaust system. Tickford are very aware of the limitations of such devices in simulating real exhaust system gas dynamics but it does provide a quick way of evaluating the effects of varying back pressure. As an aside it is interesting to note that a complete vehicle exhaust system generally produces a significantly higher back pressure on the engine dynamometer than it does in the vehicle. This is attributed to the large amount of exhaust system cooling that occurs when the vehicle is travelling at speed and which is very difficult to reproduce in a test cell.

It was rapidly apparent that significant performance improvements could be made by reducing exhaust back pressures to levels more typical of those experienced by engines in European cars. Investigations were made into ways in which the catalytic converter and the remainder of the exhaust system could be made less restrictive.

At this time there was great excitement about the potential for metallic substrate catalytic converters especially as a suitable sized unit, already in production for another vehicle, was identified. The direct extra cost per vehicle would have been of the order of $Aus 100 but there was very little experience of such units within Ford of Australia (FoA) and no data on which to base confidence of emissions durability.

Clearly it would not be possible to claim that the emissions deterioration factors could be accurately predicted and thus justify a shortened road durability programme. The anticipated engineering costs for development and proof of durability plus the potential problem of not achieving the desired results prevented the use of metallic substrate catalytic converters.

As an alternative, ceramic substrates with lower cell densities than standard were investigated. It was known that a suitably sized brick was being used with success on another vehicle and this route looked like an attractive proposition, with a lower risk than metallic substrates, until the costs of tooling the catalyst "can" specifically to fit the Falcon were obtained. The projected tooling costs were such that it became apparent that it would only be viable for a mass production volume mainstream product. As with many other possibilities the potential performance improvements were outweighed by other factors and it was decided to continue with the standard catalytic converter.

The exhaust pipe and silencer offered more scope for changes without incurring high tooling costs. The pipes are bent by a programmed tube bender and, apart from normal limitations on the closeness of individual bends and the maximum bend angles, it is possible to have any shape and many diameters of pipe with minimal set up charge. Existing silencer casings were utilised to avoid the high tooling costs associated with pressings but inside these it was possible to make useful changes to the baffles and return tubes without great expense.

Typically heat shields are also pressed and can incur unexpectedly high tooling costs but by careful design it was possible to use carry over heat shield components from the base model vehicle.

The exhaust system finally adopted for the XR6 is the same as a standard Falcon from the engine to the downstream end of the catalytic converter. From there the pipe splits into two parallel 57mm bore pipes running either side of the drive shaft from the gearbox to the rear axle. In each leg there is a silencer box utilising the standard casing with different internals which include bell mouthed return pipes to further reduce the pressure drop.

Over the rear axle the two pipes rejoin into one 63mm pipe which leads to the single rear silencer. Once again a standard casing is used with subtly changed internals to reduce the pressure drop.

Although this was a very quick and effective method of achieving the desired low back pressure, in conjunction with meeting the legal drive-by noise levels, the system requires careful alignment to maintain the clearances and is therefore not the perfect solution for a mass production vehicle.

(It has to be suggested that given more time and funding it might have been possible to develop a single pipe system which meets the same noise and back pressure targets, which would have been lighter and also easier to fit.)

***Camshafts***

The standard camshaft has an inlet cam with a 252 degree period and 12mm valve lift. The exhaust cam has a 256 period with a maximum lift of 11.5mm. It was anticipated that longer period cams would be required to improve the maximum power so a series of cam profiles varying from 260 degree to 280 degree period were designed for experimental purposes. Valve lifts greater than 12mm might have been beneficial from the performance aspect but would have required special valve springs and modifications to the machined surface in the head on which the spring sits. In line with the philosophy of minimising the changes to the engine production facility it was decided to maintain the valve lift at 12mm.

Very little information was available on the dynamic behaviour of the single camshaft and aluminium rocker valvegear although it was relatively easy to measure the stiffness and equivalent masses of the valve train. From these measurements we came to the conclusion that caution was necessary in the cam design parameters. Some time was spent in attempting to locate the source of the initial design calculations and the computer simulation model which was rumoured to exist at a university in Germany.

When these detective efforts came to nothing it was proposed to set up a computer model for our valvegear dynamic simulation program. Unfortunately the particular valve train did not relate to any existing model and it would have required considerable time to set up, refine and validate correlations with measured results.

Meanwhile the engine development team were desperate for some experimental cams so something had to be done quickly.

As a quick way of achieving longer period cams with a high likelihood of satisfactory operation the standard cams were stretched as shown in fig 3. The ramps and maximum flank acceleration were identical to the standard inlet cam but the positive acceleration period was very slightly shortened to reduce the maximum velocity points and maintain the 12mm lift. The stretching of the cam nose period had the additional benefit of reducing the nose acceleration and allowing the same valve spring load cover factor even though the maximum power speed had risen from 4500 to 4750 rev/min.

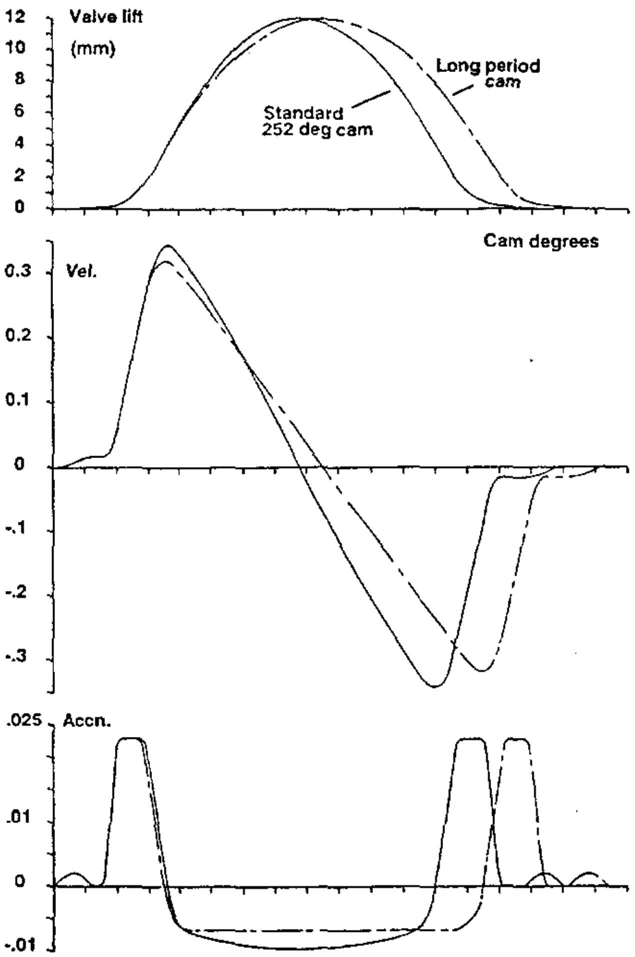



Fig 3 Simple stretching of original cams to retain the same lift. The ramps and maximum flank acceleration are identical.

Engine tests were made with many different camshafts having different combinations of inlet and exhaust period with various phasings between them. Some of the longer period camshafts produced maximum power at engine speeds beyond the safe limit for the bottom end and were rapidly discarded. In fact very quickly it was possible to home in on the range of interest and a considerable number of tests were made with quite minor differences to find the camshaft which produced the desired performance curve with an idle stability as good or better than the standard engine. The idle stability was evaluated by a method developed at Tickford in the which engine speed is measured from the starter ring teeth. This fluctuating speed trace is then processed and analysed in a particular manner which may be equated to Coefficient of Variation of BMEP.

**Valve springs**

Although the camshaft had been designed to use the standard valve springs an intermittent problem did show up during durability testing when occasionally a valve spring would lose up to 5% of its initial compression load. Investigation showed that this was caused by a combination of manufacturing tolerance of material specification in combination with cumulative temperature and time effects sometimes encountered during intensive durability running.

The simple but effective solution was to increase the nominal preload by 5% so that even if a spring relaxed slightly it would remain within specification.

**Compression ratio**

Test bed work established that the optimum compression ratio on 91 RON fuel was 9.0:1 which is very close to the standard compression ratio of 8.8:1.

Typically it would not be worth bothering with this small change but fortunately Ford of Australia seem able to hold production tolerance of compression ratio much closer than many other manufacturers. Taking advantage of this the Compression Ratio was raised by a very cheap extra skimming on the gas face of the cylinder head which for some unexplained reason consistently provided a performance improvement of slightly more than the expected 0.8% and made the small change worthwhile.

**Inlet manifold runner lengths**

It was quite obvious from the start that it would not be cost effective to undertake a major change to the inlet manifold. However it is always very tempting for a development team to find out how much they could have gained if only they were allowed to. As a compromise experiments were carried out using the same plenum chamber and throttle body as standard but with the length of the tracts to the inlet ports shortened by just 40mm by cutting and welding.

By chance this new length brought together beneficial wave ram and inertia ram at the maximum power speed and would have allowed the maximum advertised power to be increased by 4%. This single figure does not reflect the complete story for the area under the power curve, which is of more significance on vehicle acceleration times, was not very different.

**Fuel system**

The increased air breathing capacity had to be matched with an increased fuel supply but ECU recalibration alone would not have been sufficient as the standard injectors were already very close to their flow limit. The simple solution was to specify a fuel pressure regulator running at 3 bar above inlet manifold pressure instead of the standard 2.5 bar unit. This provided the necessary 10% increase in maximum fuel flow with the standard injectors.

**Air inlet system**

The air cleaner is a moulded plastic box, containing a flat paper element, which nestles somewhere behind the left hand headlight. It draws air from a swan necked scoop with a very narrow entry slit which is squeezed between the top of the radiator and the front edge of the bonnet. Any normal engineer would look at this and decide it must be restrictive and should be improved. The usual progressive component removal tests were carried out and additional experiments were made using a similar but larger inlet system developed for the 5 litre Vee8 engined car. Quite incredibly there was very little to be gained by changing any of the components which, if nothing else, showed the value of a thorough test programme.

**Calibration of the Fuelling and Ignition**

Tickford undertook the complete calibration of the EEC IV system to ensure the vehicle produced the correct performance, fuel consumption and exhaust emissions. The base calibration and the cold start work was carried out in the UK but the emissions calibration, driveability refinement and hot environment testing was carried out in Australia by Tickford UK engineers. The use of Ford of Australia's facilities and their normal locations for hot testing somewhere near Alice Springs gave them confidence that the vehicle would survive their local conditions. It was during this Hot test trip that fuel rail temperatures of 80 deg C were observed at the end of a hot soak prior to a start.

**FINAL PERFORMANCE AND COMPARISON WITH OTHER ENGINES**

The performance curve of the engine as it went into production is shown in Fig 4. where it is compared with a typical standard engine.

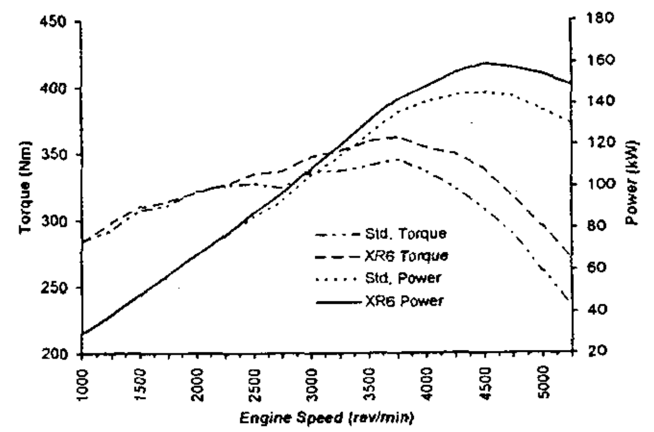



Fig 4 Performance Comparison

The average maximum power of three engines was a genuine 161 kW when measured and corrected to DIN70020 (Net) rating. The torque of 335 Nm at 2500 rev/min with a maximum of 365 Nm at 3650 rev/min is particularly useful to the average Australian driver.

This engine performance, when allied to subtle chassis and gearing modifications, resulted in a vehicle which was received exceptionally well by the motoring press. They were surprised to find it faster than any of the Australian made competitors including any of the larger Vee8 engined models available at that time.

On the Automatic gearbox version one of the improvements most noticed and appreciated by customers cost virtually nothing. A minor recalibration of the gearshift points provided kick down at lower throttle angles when in the P (Performance) mode and resulted in a much more responsive feel.

It is interesting to compare the engine performance with established state of the art.

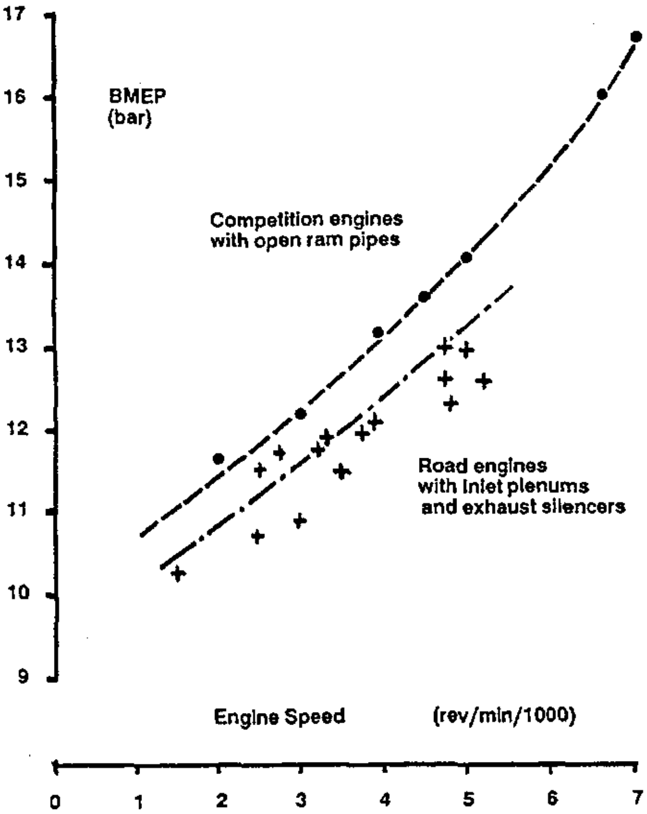



Fig 5 Significant BMEP figures from a variety of road vehicle and competition engines tested at Tickford

Fig 5. shows significantly high BMEP spot points taken from many different naturally aspirated petrol engines tested by Tickford. The implication of the general curve is that if an engine could have totally variable ports, valves, camshaft, induction and exhaust systems the torque curve would gradually rise with engine speed. This corresponds to the known behaviour of induction ram which is related to the mass entrained in the ram pipes and the velocity of that gas. Based on this admittedly empirical data it is possible to draw an envelope of the maximum torque that could reasonably be expected from any 4 litre road going engine. This is shown in Fig 6. on which is superimposed the performance of the XR6 Falcon. Given that it was never intended to produce maximum torque at speeds above 3750 rev/min the overall performance of the XR6 is very acceptable.

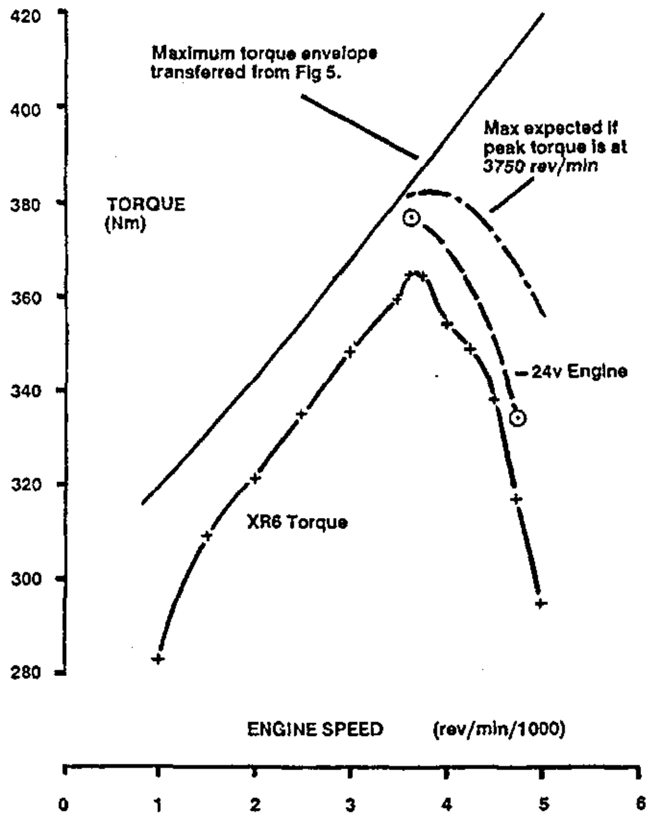



Fig 6 Comparison of XR6 Performance against the likely maximum.

Also shown on this graph is a small section of the torque curve of another manufacturer's four litre, in line six, engine. The maximum torque is some 3% higher than the XR6 but the complexity and cost of the engine is far greater as it has twin overhead camshafts and four valves per cylinder. It is estimated that if all other factors were equal this sophisticated engine might cost almost twice as much to produce as the FoA XR6.

## CONCLUSION

The Tickford developed engine for the Ford Falcon XR6 has increased the already considerable performance of the standard engine by 8.8% in power and 4.9% in maximum torque. These quite modest percentage improvements have lifted the performance curve to approximately 95% of the maximum envelope that could be expected from a road legal engine of this size and speed range.

Most significantly the improvements have been made at very little extra production cost thereby allowing Ford of Australia to offer a very good value vehicle in terms of performance for money.

This is reflected in the sales of the car which exceeded the initial estimates by a factor of four times, helping Ford of Australia to beat their main rival for the first time for many years.

**ACKNOWLEDGEMENTS**
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# Paper 4

## "The Supply of Engineering Services in the Global Car Industry"

**Bale, C.J.C. (1994)**

SITEV Technical Congress, Automotive Industry Week

Lingotto Centre, Torino, Italy

15-17 November 1994

Tickford

# The Supply of Engineering Services in the Global Car Industry - Adventure or Joint Venture?

by

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## ABSTRACT

The way Engineering services are purchased has changed dramatically in recent years in a similar way to buying production parts. However, such services are not like production commodities and this leads to different relationships between the customer and supplier ranging from old, adversarial, arm's length relationships to true partnerships and joint ventures. Most changes are clearly for the better but there are advantages and disadvantages to the "Tier" system being applied by many vehicle manufacturers. Good, trusting, long-term relationships have real benefits to both parties and the best embodiment of these is true Joint Venture. This paper details Tickford experience of all types of relationship and picks out the best to be followed and the worst to be avoided.

## INTRODUCTION

One of the advantages of working with many different customers around the World is that the opportunity exists to compare how each of them operates with its suppliers. This paper relates experiences of Tickford customer to supplier arrangements that cover the extremes of the spectrum from the old "arm's length" relationship to very close partnership.

Tickford is a broad based automotive engineering consultancy and supplier of engineering services to the vehicle manufacturers, the component companies and the petroleum industry. The company has a number of peaks of excellence, for example: in engine design and development and whole vehicle engineering, particularly for niche and expanding markets. The company was formed in 1981 out of the engineering department of the Aston Martin Lagonda sports car company and became independent 3 years later within the CH Industrials public group. In 1991, Tickford Directors made a successful Management Buy-Out from the diversified CHI group, complete with all Tickford purchase orders and contracts. The company has now grown to some 350 employees in 7 locations with operations in UK, Australia, USA and Germany and supplies a range of consultancy and engineering services to the global automotive community.

The way such services are supplied and purchased has developed in recent years and several new "styles" of relationship now exist. However, despite what many manufacturers may suggest, the old adversarial type with lowest price being the only way of being selected, still dominates the purchasing process. The service supplier's attributes of quality, technical competence, efficiency and delivery are hopefully becoming more significant considerations to most customers as a way of reducing total programme cost, and the suppliers benchmarking their own capabilities against World class standards is increasingly important.

## ADVENTURE?

The first type of relationship that I will discuss is the one that covers most instances today and that is the "preferred supplier" or "Tier 1 supplier" status. Despite hard work from the suppliers and firm assurances from the customers, entering into such a relationship can still be likened to embarking upon a business "adventure". Are today's suppliers being forced into "unhappy marriages"? Are commitments to their customers becoming so demanding that they prevent the normal rules of good practice to be followed? Do some suppliers yearn for the old days when more flexible relationships were the norm and they were free to develop customer relations in their own style?

The answer to these questions is that no-one should wish to turn the clock back. We have made so much progress in developing and understanding the new and closer working relationships between the customer and supplier that we can only go forward. There are some real benefits to be had from better understanding of each others business:

### Benchmarking

Since the manufacturer usually has access to a wider supply base, this can be used to compare the supplier operations across a wide field and the best working practices can be noted. This highlights those aspects of suppliers that are currently World Class. It will also highlight those that are not there yet but that are advancing rapidly in the right direction and, if given encouragement, will soon make it.

From these comparisons, providing the manufacturer freely and openly helps the supplier, both can benefit from Continuous Improvement Programmes aimed at the deficiencies. It is vital that the benefits are shared as they arise, between customer and supplier for their mutual gain. However, we the suppliers will not give up our processes and skills that are a competitive advantage otherwise the discrimination between suppliers would vanish. The areas of pre-competitive waste reduction must continue to be the real target from which all will ultimately benefit.

### Cost management

With true "open book" relationships the main cost factors are obvious to both the supplier and the customer, and hence the control of costs at both the design phase and the volume production phase becomes easier. Thus when, in a nominally fixed price project for example, there is a large movements in fuel prices due to global supply or politics, the problem can be relayed more easily to the customer and joint action planned to control the overall programme cost impact. The manufacturer must see that the supplier makes no additional margin but also that he is not unreasonably penalised by circumstances genuinely and unpredictable out of his influence.

During the design phase, the cost impact of changes can be truly seen by all parties and informed judgements made rather than solely engineering-led decisions that, in many cases in the past, produced wonderful but unprofitable products. Similarly an open partnership enables all hidden costs such as line rejects through handling damage, warranty costs, etc., to be exposed to the benefit of both parties and rapid action can be taken to resolve concerns. In many cases such benefits pass on down through the manufacturer to the vehicle end-user.

**Systems simplification**

Some manufacturing customers have drastically reduced their number of suppliers in product and engineering areas and the "Tier 1", "Tier 2", etc., structure has emerged as a means of passing administration and risk to the Tier 1 supplier. This is an ideal opportunity for the manufacturer to simplify its purchasing processes and to concentrate more on the best suppliers with the most capability. They, in turn, can buy-in the incremental services that may be required by the vehicle manufacturer but which the Tier 1 does not have in-house. The smaller suppliers will feed off the big ones.

This is all very fine in theory but in practice it needs to be very carefully thought through:

- The "Tier 2" suppliers will feel rejected and will then work hard at securing business from other customers. Useful skills and resources may become unavailable to the prior customer who de-selected that supplier, as he himself becomes a "non-preferred" or Tier 2 customer.

- The "Tier 1" supplier should be happy at being amongst the chosen few but the customer's buyer must be wary of thinking that he has bought the suppliers soul - he should not now put on unreasonable pressure for price reductions, only those that can be justified from economies of scale. Such economies are real <u>only if the volume of business placed by the vehicle manufacturer increases to a point where the supplier's overhead can be reduced as a result of less account of low utilisation of staff and facilities.</u> Without such real economies, the suppliers prices must actually <u>rise</u> to accommodate the Tier 2 administration and risk.

- The "Tier 1" supplier may use "Tier 2", former suppliers, to reinforce his capability and resources. This must be done with caution as some of today's Tier 2 are former Tier 1 competitors with other customers and loyalties. What the Tier 1 cannot afford to do is become a purchase order channel for the vehicle manufacturer's buyer, just to put orders onto former "Tier 2" sources. Naturally some product engineers' favourite suppliers will have been removed from the supply list and there is a natural instinct to find a way to use them. The Tier 1 cannot afford to pass significant orders through to Tier 2 without being correctly recompensed for the effort and responsibility. Quotations, orders, invoices, quality, supplier management, payments, etc. all have a cost to the service supplier. Even if this is paid for by the customer, jobs going through the Tier 1 system with costs (from Tier 2) of some 95% of revenue, do nothing but distort the company financial performance.

**Convergence of business objectives**

Some customers are moving towards a better understanding of suppliers needs and closer business relationships result, with benefits for both parties. By removing the adversarial factors and replacing them with mutual trust, the customer and supplier can work together and both can make savings and benefits.

As a supplier, Tickford has often had a privileged view of the hidden inefficiencies in vehicle manufacturer operations, for example: the time it takes to process documentation or hardware from A to B, the delivery van that will arrive with a test engine but is not allowed to collect a similar engine ready for return, the buyer who treats time and materials as separate, fixed budgets (not allowing the flexibility that can save time by doing, in-house, a task that was quoted as a sub-contract), etc.. These are the areas that are wasteful of time as well as effort and can be beneficially removed. Time is a most valuable commodity in the quest for shorter product development cycles and we all know that short programmes cost less than long ones.

Most suppliers and customers want to be the best. They typically share or include all or some of the following mission statements:

- Be the World's No 1
- Achieve World-wide growth
- Lead in customer satisfaction
- Achieve World-wide excellence
- Empower their staff
- continuously improve efficiency

Mutual recognition of the ways to achieve these sound objectives can be constructive. The customer recognising that the supplier needs a profit to feed investment and training, is a fundamental step forward. The supplier recognising that the customer needs to reduce Product Engineering costs is similarly important and the two can work together to produce progress.

Some steps can be startlingly simple. For example: a manufacturer requires his supplier to use a particular CAD package that could be new in the market. Why does he not invite the supplier to share in an in-house training session for the vehicle manufacturer's own designers? This has to be a better cost per man than a separate course and, one way or another whether directly or in project prices, the vehicle manufacturer will eventually pay for the training.

Having a group of competitive suppliers all ready to carry out project work but not actually active, creates another inefficiency. The customer wants the supplier to be constantly ready and at a low cost. The supplier is expected to hold resource available and prepare countless competitive quotations for projects he may never carry out. The cost of the standby labour and the cost of proposals becomes part of the supplier's overheads and the customers pay for it in higher charges for paid contracts. Is it too naive to suggest that customers outline their resource requirements well in advance and short-list 2, or at most 3, suppliers for each project and tell them so. That way the supplier only bids for fewer jobs and, if he is capable and competitive, he will get 1 in 2 or 3 as contracts. Also, if he is not chosen for the first job, let everyone know that he is in "pole position" for the next similar job in X months time. That supplier will keep resource available and competitors will submit abbreviated quotations, demanding sensible effort, that ensure that the favoured position is not exploited with a high price. Less wasted effort by all, less unused resource, more efficient operation, lower overheads, more competitive prices, happier customer and happier suppliers. Should this not be our shared objective?

## OLD ADVENTURES

The old way of dealing with suppliers of engineering services was to extract a set of hourly rates for different types of labour and facilities, then squeeze these rates down on a year by year basis. The essence being that the buyer would have a comprehensive breakdown of the resources estimated by the supplier (hours and grades) on the pretext that the quotation could then be verified for the right amount of the right skills. The real reason is to play off one supplier against the other and look for the lowest unit cost. Any reductions would be taken by the buyer towards his cost savings for the year. Alternatively, refusing any request for a rate rise would also be a cost saving.

The inevitable consequence of this approach to variable contracts (or fixed price ones with a complete breakdown) is a squeeze of suppliers margins. In a production environment, a supplier can look for ways of reducing unit price on raw materials or more efficient processes to lessen the price of finished product. The same cannot be said for selling human resource in a quality company. Wage inflation affects all businesses and if a contract is framed on the basis of a number of hours of a person whose cost is rising, rate fixing or reduction lessens margins on the work done and hence profitability and investment funds will suffer. Overheads have to be controlled of course, and some efficiencies can be made where the organisation is not already slick. However, as a result of the last 5 years, most, like Tickford, are lean machines. Quality shortcuts become a temptation and, inevitably, a less capable (and less costly) person will be supplied to fill the low price role allowed by the buyer.

Building a dependency on just a few customers can lead to intimidation of a supplier. Firstly, the supplier is vulnerable to the economic fortunes of his dominant customer and his politics. By building up the market share that the buyer has, the supplier becomes reliant on a continued flow of business. Once this has been established, the buyer can squeeze the supplier to take work at a marginal cost, just to keep going. This is a supplier who will not improve! Furthermore, the pressurised, low cost supplier can be used to intimidate any alternative supplier who will be shown that "Company X sells me this service for this low price - you must be lower to get the business".

The other reaction to such adversarial purchasing tactics is to bid the very minimum to match the requested quotation, even if the supplier knows that the job cannot be completed as such. In this way he can be sure that the engineering customer will require additional work and the order can then be extended. This helps the supplier to ease his pressured margins but leaves both the engineering customer and the buyer with a sour taste.

The other way in which this uncertainty could manifest itself is the "not to exceed" order. In this case the supplier submits a quotation for the scope of work specified and receives a purchase order for an amount "not to be exceeded". The buyer reserves the right to audit the use of resources and if the task is completed in less time than the total order, the payments stop and the contract ends with a lower cost to the buyer. However, should the supplier have underestimated the task, the buyer will expect the work to be delivered at the maximum contract price (not to be exceeded) regardless of any additional work required at the suppliers cost. This supplier is then left with a loss or below-budget profit and that same sour taste.

This complicated cost and deliverables estimation leads to uncertainty on both sides and the classical adversarial relations between a buyer who wants the most work for the least cost and the seller who wants the most revenue for the least effort. One readily recalls the dark days of labour disputes, strikes and "us versus them" that plagued European and US industry. We must acknowledge that the Far East industrial culture of partnership, trust and fair profit, has taught us all a number of lessons, particularly the improvement of quality plus savings in time and total cost.

The increased openness of buyers and suppliers is a move for the better. The supplier is under pressure for increased quality (especially quality customised to one client), increased investment in new tools and technology, increased efficiency, faster response (more speculative resource?). The buyer still has his target for cost reduction but the word COST is now the key-word, not RATE. In many cases the supplier can offer a lower COST by supplying the resource with a higher RATE.

e.g.

1. An expert at £100 per hour could do a job in less than half the time of a man charged out at £50/h. The price of the job would therefore be less.

2. It is no benefit to the buyer to place a CAD contract with a supplier offering seats at £30 per hour if the person designing is poorly qualified and with no experience. The job will take a lot of £30 hours to complete (assuming that it can be done at all!). Again it is more efficient to pay a higher rate to get the job done well, right first time and quickly.

Buyers are increasingly turning to fixed price contracts against agreed deliverables leaving the resourcing to the supplier. If he can beat his estimated use of resources, some extra profit may accrue: this must be allowed since, if the supplier has under-estimated the job, he is committed to allocating additional resource at his own cost in order to fulfil his contract. On balance over the year he will win on some jobs and lose on others. This is a contrast to the "not to exceed" method where only the buyer was allowed to win. (Does not the very fact that I am drawn to quote winners and losers, indicate a shortcoming in that relationship?)

## NEW ADVENTURES

As the World's vehicle manufacturers get more efficient, more and more of the product development responsibility is being passed to the component suppliers. The days are mostly gone when a vehicle manufacturer passes the component supplier a completed drawing of a proven part ready to be manufactured in volume for production. Instead the supplier is given a space and a function for the component (or more likely now, a system) together with a commitment for at least a share in the production volume, and the design and validation responsibility is his.

This is a scenario welcomed by most component suppliers but it is one for which they were not initially well-equipped. Engineering service companies are playing a useful role in assisting component suppliers with projects while all adjust to the new way of working. Investment in new skills, new staff and new engineering tools are all things that take time. The vehicle manufacturers, of course, like to do things as a step change and companies like Tickford are there to provide the capability and resources to achieve the short term solutions for vehicle manufacturers and component suppliers. Some component suppliers, recognising that there is a peak of activity during the engineering phase of a new product or application that then diminishes during the product cycle, continue to use Engineering Service companies as their engineering department and choose not to build their own. This is a matter of policy and careful review of the cyclic nature of each product life.

In the same way for both vehicle manufacturers and component suppliers, Tickford and its competitors can help at the end as well as the beginning of a product life cycle. Imagine the situation:

- New product cycle identified; engineering effort starts on revised or replacement model.

- Manufacturer allocates major resources to achieving Job 1.
- Legislation or sales and marketing input requires effort on existing product to keep it going or to sell economically significant volumes in a different market.

What is the manufacturer to do? Re-allocate some of his main resources and jeopardise the Job 1 for the new product? Try and recruit and train for a short term need? Both unlikely in most cases; the better solution is to engage a suitable experienced engineering service company to carry out what can be a well defined piece of work, while the mainstream team continues un-deflected on the new high volume product.

The above scenario is typical of the opportunities facing vehicle manufacturers, and hence component suppliers, in expanding and emerging markets in the World. Significant incremental volumes of sales are available but territory requirements such as legislation, local fuel quality, climate, road surfaces and terrain, customer preferences, etc. make the current mainstream product unsuitable. What is required is a "derivative" vehicle adjusted for the new market needs, but without re-engineering the whole car. In the cases where fuel quality or composition is very different (such as is found in some South American countries), the re-engineering is multi-faceted; that is the fuel system needs attention as well the engine management mapping, the exhaust system may need changes, etc.. The suspension, the interior trim and equipment may also need adapting to local needs so a broad range of capability is needed in the engineering source. In the vehicle manufacturer, so many different departments may be implicated that the programme management becomes a nightmare. This is another good reason why vehicle manufacturers use Tickford; because we are used to managing fast-track, multi-functional projects, we understand the customer's needs and constraints, and we execute the programme with the minimum load on in-house resources.

The essence of a good relationship in the "New Adventure" scenario is one in which the supplier endeavours to help the vehicle manufacturer to deliver his wishes where the in-house culture does not allow them to do it themselves.

**JOINT VENTURES**

The ultimate manifestation of a committed customer/supplier relationship has to be the formal "Joint Venture" (JV) and Tickford has useful experience to share. The operational needs of a joint company are underpinned by the financial and resource commitment of both parties. The result is commonality of purpose and roles for both shareholders that, in turn, gives rise to fast response to market needs without risk of damage to either "parent" or constant recourse to guidance.

Tickford operates such a company in Australia. It is called Tickford Vehicle Engineering (TVE) and was formed five years ago in conjunction with Ford

Motor Company of Australia. Over 40,000 cars have passed through the TVE facility in Melbourne to date and production is currently running at a rate of 11,500 per annum. An example of one of the models produced is shown in Figures # and #.

The establishment of a JV as a separate and self-standing legal entity company brings many benefits to the manufacturer and supplier relationship. To ensure this success, the JV must operate to a well derived set of rules that are built into its formation agreement. The aim of the operating agreement is to ensure a common purpose and an integrated financial structure so that the company has a clear and rewarding role to play for both its shareholders.

A considerable initial effort has to be made by both parties to plan all the interfaces between the three entities (the two parent companies and the JV itself). All processes and responsibilities must be covered in great detail. However, the long term nature of a JV makes this up-front effort justifiable and well worthwhile. A well planned and well operated JV will perform better that the sum of its parts and both sides will want a JV to succeed because both sides will benefit when it does.

In between sharing the work and the rewards comes the inevitable need to share the risks. But there is the (not always so obvious) need to share in the excitement also. In our case with the type of product manufactured, there is indeed some excitement to be had. The JV must be fully integrated into the product policy and planning processes, right through to the launch of the product into the marketplace. The manufacturing data systems must be linked and information freely exchanged.

In our case, the optimum route for creating the special vehicles is planned and agreed between the main factory and the JV manufacturing teams well ahead of the production introduction date. Cars are built as far as possible down the main production line, but often to a "delete" specification or with some essential items replaced by slave components. This avoids wastage later when new components are substituted. This situation means that much of the engineering of the product is targeted for line manufacture and not for low volume. Such things as the derivative engine must be machined and assembled down the track even if the castings are unique in incorporating new ports, etc..

Based on our own experience, a properly formed Joint Venture Company is the best relationship for which to aim. But the will of both parties to want it to work, and the importance of that initial set of ground rules should not be underestimated.

## CONCLUSION

The ultimate aim of an engineering supplier/customer relationship is like any other purchase - that both parties are satisfied with the services and funds exchanged. In a good relationship this can be described as a "win, win" situation. In a joint venture this becomes "win, win, win" as all three managements are satisfied. Like any good marriage, the outcome is most effective when both parties contribute and recognise that each must take out good value for what they put in, not only for today but also for the future.

The supplier who does not invest in his people, facilities and quality systems, is bound to fail in the present tide of change. The customer who does not recognise that these value-adding features have a price is deluding himself and risks becoming a non-preferred customer with the best-in-class suppliers. The inevitable knock-on effect on his own business will be the true price of false economy.

The most efficient solution in engineering services has to take a lesson from world class production suppliers:

- The customer shares enough of his plan to enable the supplier to invest in the right skills and facilities.
- The supplier has the temerity to have core resources available to meet the timetable
- The customer honours the suppliers' investments by using the programme resources in a flexible framework of fair price and delivery
- The supplier honours the customer's commitment by delivering the project on time and on budget without seeking to exploit every minor deviation in the engineering programme.

As already stated, engineering services are <u>not</u> like product. An engineering supplier cannot absorb tooling and investment against the commitment to future product volumes for years to come. There is no engineering service supplied during the life of a car once it is launched successfully; it is this fundamental that makes external sourcing advantageous to the manufacturer in controlling his costs.

In summary, customer/supplier relationships can cover a very wide range of "togetherness". Customers can be very distant from their suppliers and vice versa. Tickford has proved that this need not be a problem in the geographical sense although geography is often used as a excuse for poor relationships when it is not the real cause. True partner relationships allow both purchaser and supplier to become more efficient and prosper at lower cost. The main point is that no matter where the relationship currently stands, it can always be made better by each party <u>willingly accepting product ownership</u> and making "Continuous Improvement" with the aim of attaining and maintaining a World Class performance from which the manufacturer, the supplier and, most importantly, the end-user, will all benefit.

Figure 1 - The way it was

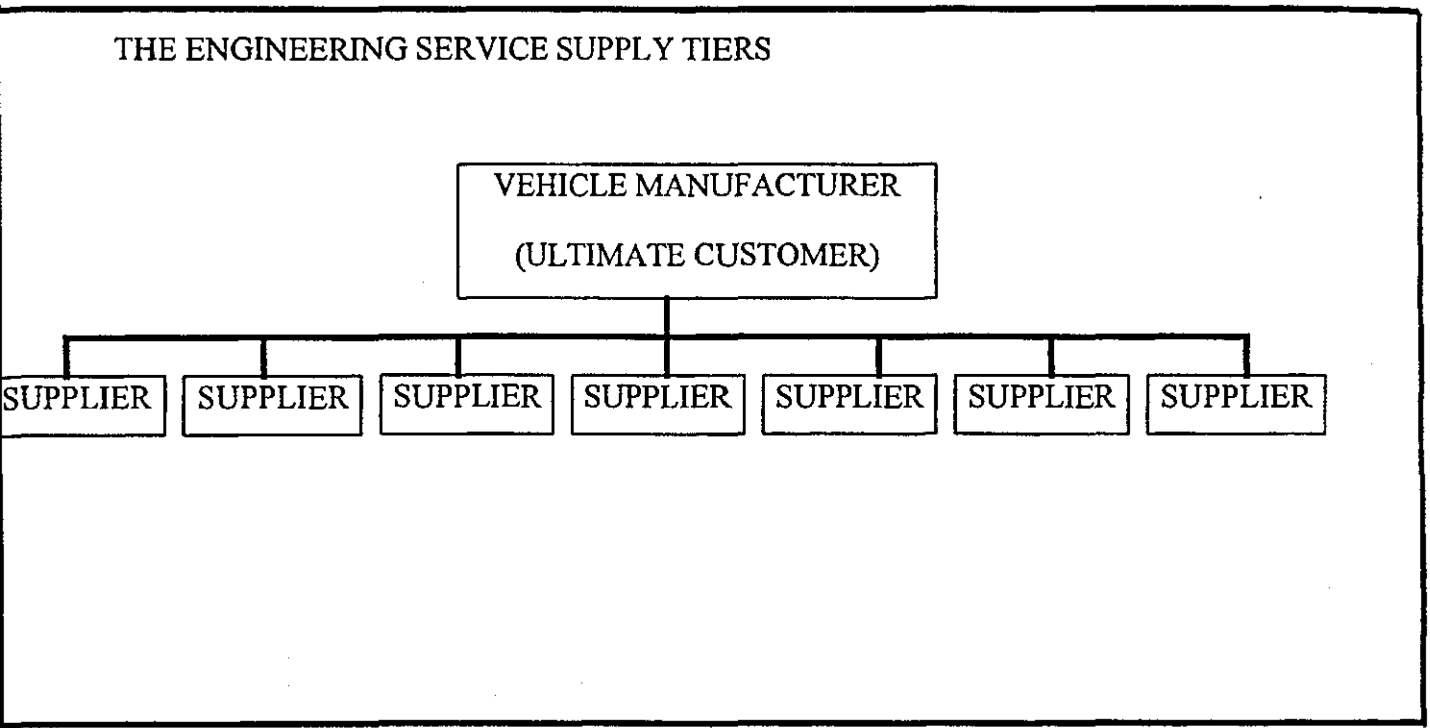



Figure 2 - The way it becomes

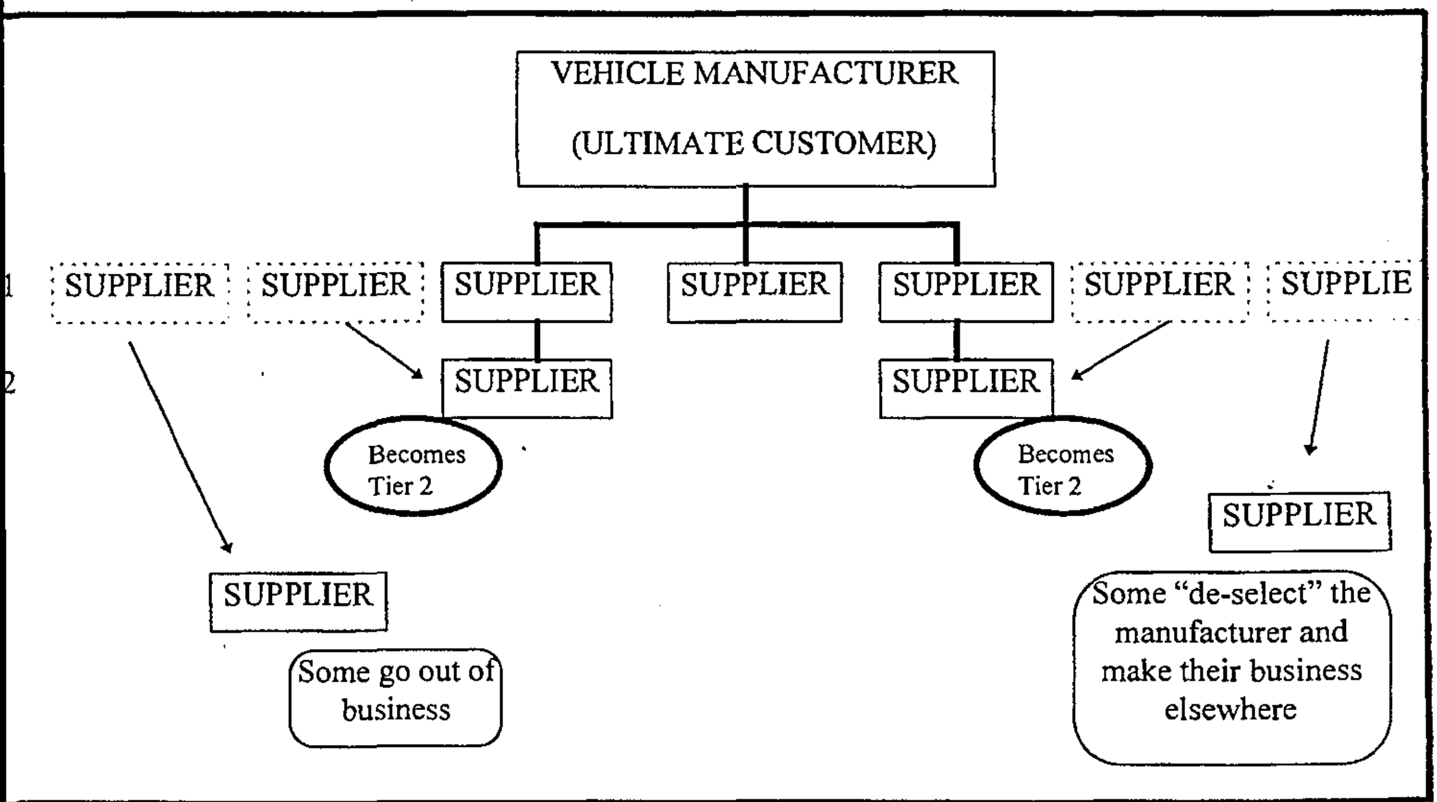


Figure 3. The Joint Venture

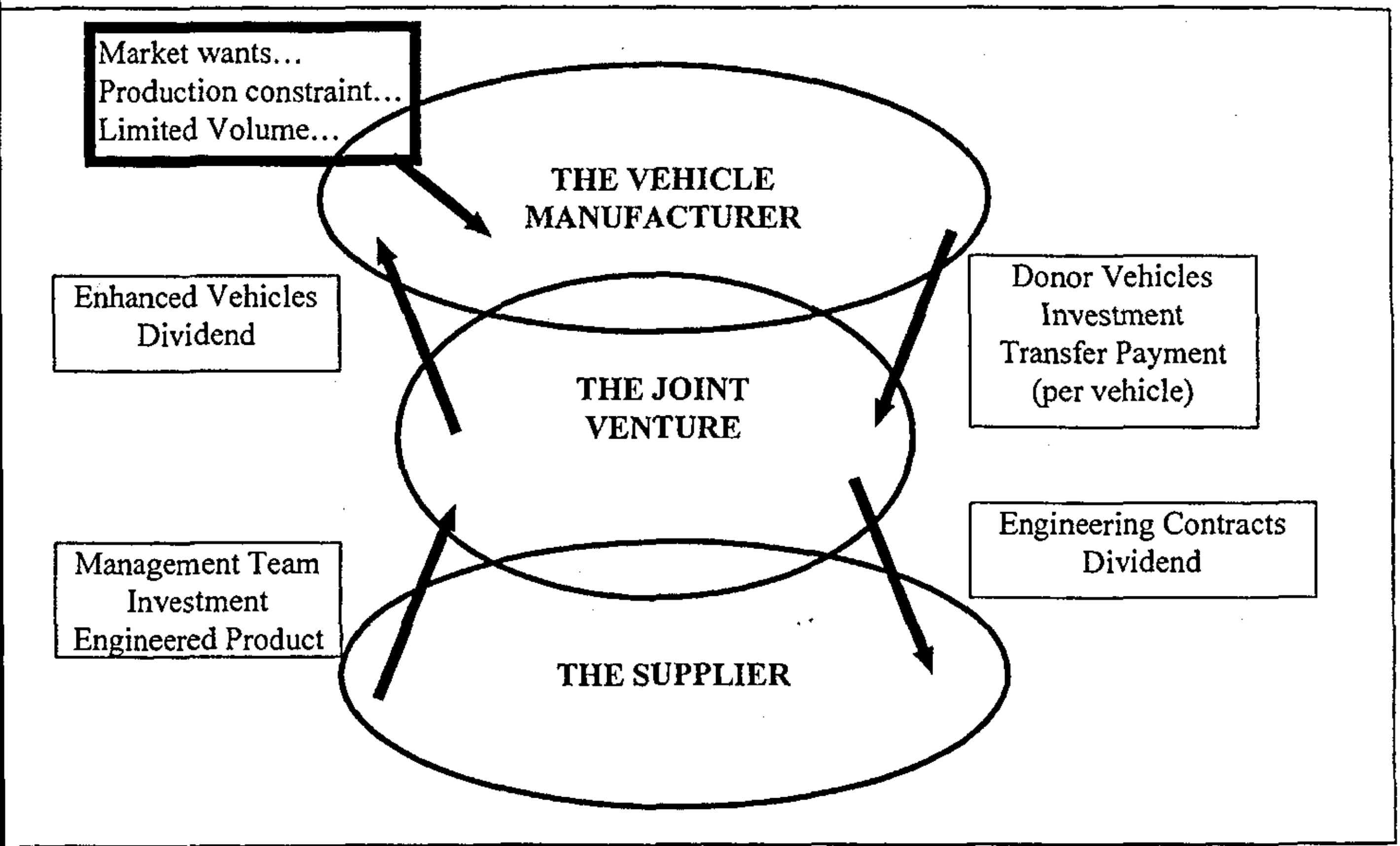
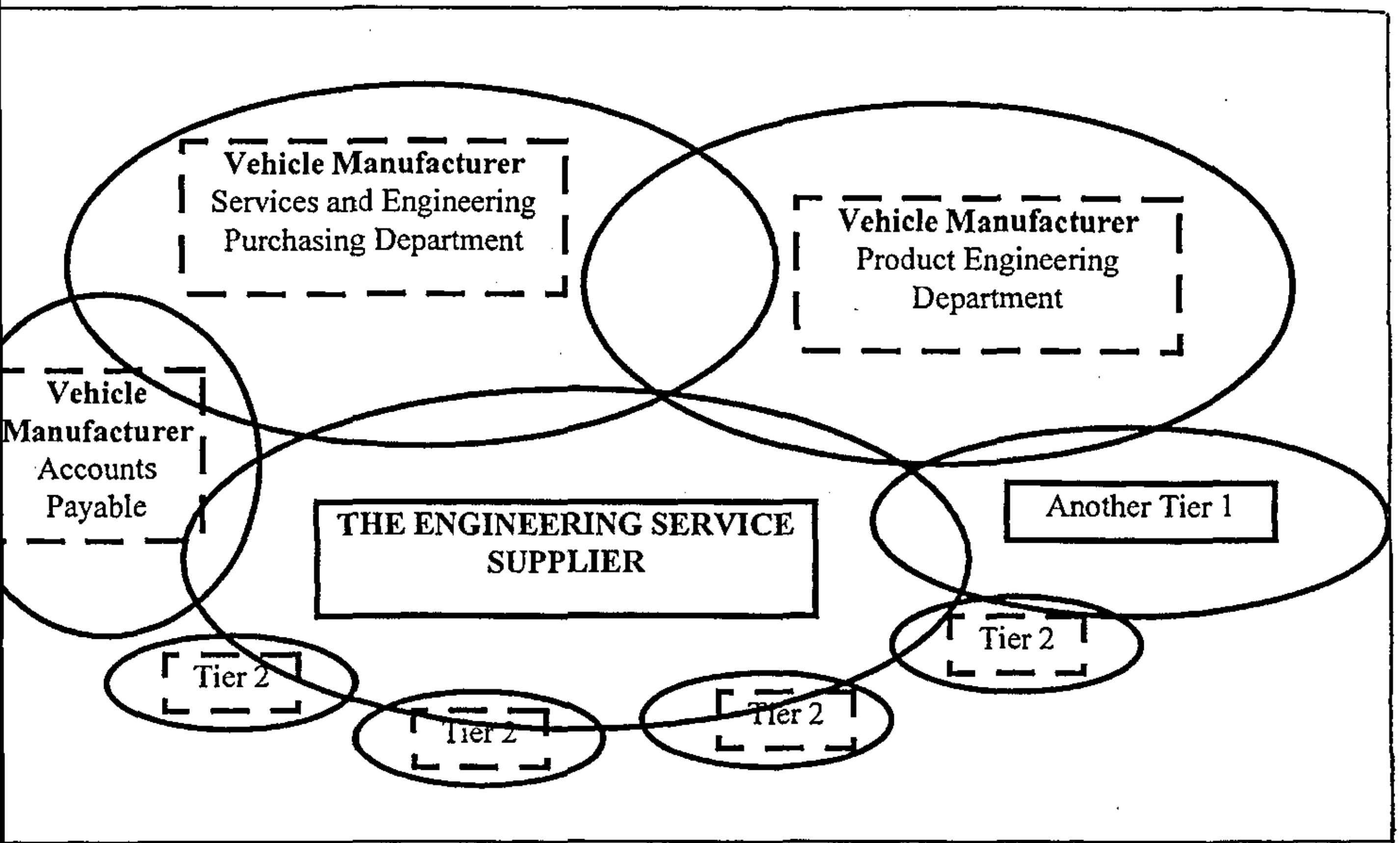


Figure 4. The Supplier's relationships



# Paper 5

# "High Specific Output, Low Emissions Spark Ignition Engines"

**Bale, C.J.C. and Sykes, R.G. (1995)**

Automotive Engineering Asia Technical Congress

Kuala Lumpur, Malaysia

26-27 October 1995

Paper R95/122

Technical Paper R95/122

# High Specific Output, Low Emissions, Spark Ignition Engines

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AE ASIA 95
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*Kuala Lumpur, 26-27 October, 1995*

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## ABSTRACT

Knowledge of port flow and port induced air motion together with an understanding of the consequent combustion and emissions in spark ignition engines has been applied to two very different projects. A large 2 valve per cylinder engine has been improved in performance with minimum on-cost and a five valve per cylinder version of an already competent four valve per cylinder engine has been developed applying tumble technology. The resultant engines are particularly suitable for passenger cars that require low fuel consumption and exhaust emissions with good performance.

## INTRODUCTION

There are many newly invented or resurrected ideas which are claimed to be beneficial to the performance of engines. Whether or not these features are of real benefit is often the subject of lively debate within the engineering community and even devices which offer a measurable advantage are sometimes difficult to justify. Two features applicable to engines which are still being debated and researched are the number of valves per cylinder and the relative merits of tumble or axial swirl motion of the incoming charge.

TUMBLE OR AXIAL SWIRL?- It is very well established that motion of the inlet charge is beneficial for combustion, especially at light load. Typically it results in more stable running with a higher tolerance to EGR and lean mixtures. What is not so readily agreed is whether it is best to employ axial or tumble swirl.

Full time axial swirl is relatively easy to achieve using one inlet valve per cylinder but is slightly more difficult with multiple inlet valves. In some diesel engines both vertical inlet valves in each cylinder are positioned towards one end of the engine so that incoming air enters the cylinder at a tangent. There have also been a few gasoline engines in which the two inlet valves are diagonally positioned and the fresh charge enters from both sides of the cylinder head. Neither of these is a totally satisfactory layout for a compact and tidy installation with a good combination of volumetric efficiency and combustion chamber shape.

The typical solution is to employ part time axial swirl by de-activation of either one inlet valve or port in each pair at light load when the charge motion is most needed. This system allows both inlet valves to be operational at full load to maximise performance and also has the potential to avoid excessive charge motion at high engine speeds.

Such valve de-activation or inlet manifold runner control (IMRC) is not without its problems nor cost penalty and thus an alternative such as potentially offered by tumbling charge motion is still of interest.

Tumble can be created in engines with two or three inlet valves per cylinder when the inlet valves are disposed in a conventional way, that is symmetrically about the cylinder centreline and all on one side of the cylinder head. Appropriate shaping of the inlet ports and the combustion chamber can cause the incoming air to rotate around a horizontal axis parallel to the crankshaft. Except for those cases where the inlet ports are fully machined, or hand finished, there is typically a significant trade-off between maximum air flow and amount of tumble.

It is possible to generate some measure of tumble component in two valve per cylinder engines especially where the axis of the incoming charge can be directed across the cylinder centre line. However, research by Tickford on the airflow bench has identified the overall compound motion that results and the relative contribution of the axial and tumble components. It is important to state that it is NOT the motion of the incoming charge that is significant but rather the motion of the charge on a micro scale, at the point of ignition. The gross charge motion has benefit in mixing the air, fuel and EGR components prior to ignition but it is the breakdown of this motion into micro-turbulence at the point of ignition which enhances the flame propagation through the mixture and the igniting of the leaner areas of the charge. The burn rates thus achieved are faster than those of a stagnant charge where the speed of advance of the flame front is determined by combustion kinetics.

The benefit of such a combustion system is the resultant ability to burn lean or dilute or poorly prepared mixtures giving fast heat release and rates of pressure rise in the cylinder, coupled with the potential for complete combustion. This is a means to enhance the specific output of the engine while retaining a complete burn with good emission levels. The engine development and calibration team therefore has more scope to optimise the performance, economy and emissions balance.

PORT PERFORMANCE - It has been known from many years of design study and flow bench testing on a wide variety of cylinder heads and flow blocks that both the inlet port and combustion chamber shapes can significantly effect the flow and tumble. The mean flow and mean tumble numbers for every port are entered into a continuously updated database and chart to judge the performance in relation to others. A selection of results is shown in Figure 1, where it can be seen that there are many cylinder head designs which are nowhere near the optimum on either scale. However, the upper edge of the envelope confirms the logical characteristic that is as tumble increases, the maximum airflow potential decreases.

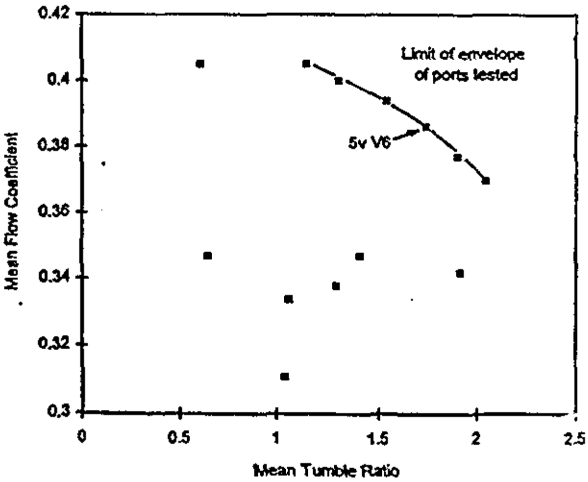


Figure 1. Port Flow/Tumble Trade-off

Tickford has undertaken a program of work to determine some fundamental guidelines for the design of inlet ports to generate high tumble ratios with the maximum possible airflow. A single port model was made in with the port to valve angle θ, the intersection height **h** and the inner radius of the port, **r**, (all shown in Figure 2) could be varied.

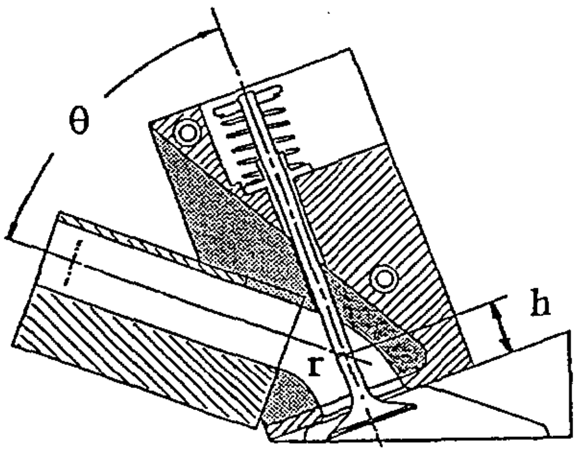


Figure 2. Variable Inlet Port Model

HOW MANY VALVES - Two valves per cylinder were the norm for many years but as piston speed have risen, the limited valve area has become restrictive on volumetric efficiency for some engines. As a result the four valve layout has been developed to the point where, in conjunction with long period cams, it can be made to provide adequate airflow for most highly rated engines. So initially there seems to be no point in trying to increase the percentage of the bore occupied by valves as provided by the three inlet, two exhaust layout that give the largest valve area and inlet to exhaust area ratio of all possible layouts for circular bores. However, road going cars need other important attributes other than maximum power. They should have a good spread of torque to aid driveability, they must be as fuel efficient as possible and pass exhaust emissions regulations. They should provide the lowest manufacturing cost and have superb reliability. When these things are considered, there are good reasons why and increase in valve area may be desirable.

The body of this paper describes two projects involving engine design and development where Tickford's understanding of cylinder head design, ports and combustion was applied to a two valve per cylinder production engine and an advanced five valve per cylinder unit from opposite sides of the World.

TWO VALVE PER CYLINDER CASE STUDY - New and expensive technology is often featured in technical literature and in manufacturers brochures. It is always the hope that ways can be found to incorporate such techniques into production engines and vehicles at acceptable cost but the challenge is to optimise the performance and cost benefit. Tickford recently undertook such a program with the objective of increasing performance at minimum cost as a means of producing a value-for-money "Gentlemen's Express". It was essential that there should be minimum changes to the manufacturing process as the engine was to be built down the same line as the standard power unit. The finished product was to have similar fuel economy to the base vehicle, meet the same drive-by noise and exhaust emission regulations and be thoroughly proven to normal manufacturer standards.

The Ford of Australia in-line 6 cylinder engine has a bore and stroke of 92.26 x 99.31 mm giving a total swept volume of 3984 $cm^3$. It has been developed to be admirably suited to typical Australian driving conditions where high torque is appreciated more than high rotational speed. The combustion chamber is part spherical with one inlet and one exhaust valve each set at approximately 20° from the cylinder axis. The valvegear is quite cheap, compact and effective. A single camshaft mounted along the centre of the head operates the valves through forged aluminium rockers fitted with roller followers at the inner ends and hydraulic adjusters above each valve.

**Cylinder head and ports**

The proximity of the edge of the inlet valve seat to the edge of the bore prevented any increase in valve diameter without offsetting the valve axis. The standard port had been well developed and, under steady state flow conditions, showed characteristics upon which it would be hard to improve. The main bore from the inlet manifold face was of 39mm diameter bulging out to approximately 46mm just downstream of the valve guide and then narrowing to 41mm though the throat of the valve insert. Tickford considered that the inlet gas velocity in the main bore was lower than optimum and reduced the port size to 37mm. The bulge before the valve seat was detrimental to dynamic breathing and this region was reduced in diameter to provide a gentle increase in cross sectional area from the straight portion to the seat insert as shown in Figure 3.

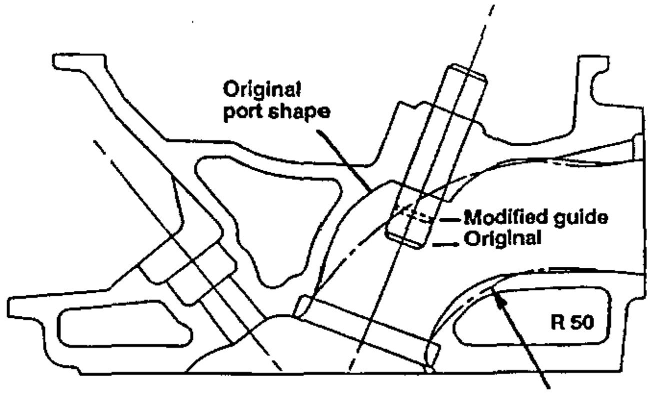


Figure 3. Low Line gradual expansion inlet port superimposed on the original shape.

In fact there were two iterations of this design. The first modified design with a 50mm port floor radius showed satisfactory flow bench results and matched the same flow as standard with the smaller port. However, it would have been necessary to modify the water jacket core to ensure adequate metal thickness and the high tooling charge justified further development of the design as shown in Figure 4.

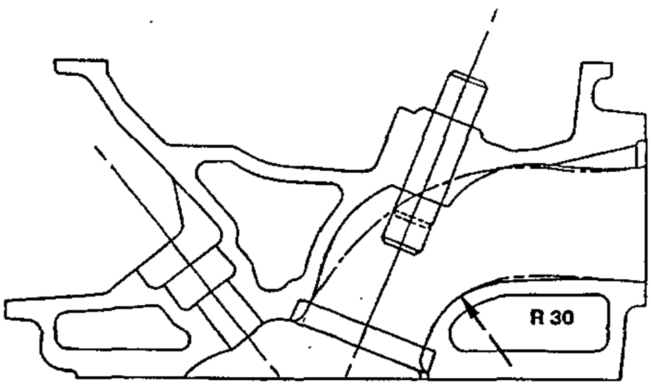


Figure 4. Second version of improved inlet port which avoided changes to the coolant jacket

A decrease of the floor radius to 30mm resulted in marginally inferior flow bench performance of the port but this design was incorporated into a test engine where is actually showed a performance improvement over the standard design. This is only one of our experiences that has shown that port development carried out only on a steady state airflow bench can lead to imperfect results compared with a fired multi-cylinder engine running under dynamic flow conditions.

The modifications were relatively inexpensive to incorporate as only the port core shapes were changed within the standard head casting equipment. Minor changes were also made to the exhaust port to prevent flow separation between the valve seat and the port floor by raising the floor at the valve end and providing a larger radius. Cross sectional areas was retained by making the port locally distorted to a D shape resulting in a good combination of cross sectional area and smooth flow without any changes to the water jacket. Removal of the valve guide and its boss from the top of the port improved flow considerably but inevitably reduced the support of the valve. A short but detailed review of the effect of valve guide length on valve gear noise and exhaust valve temperature was undertaken. As a result of this work it was possible to choose a valve guide that intruded only slightly into the port and yet supported the valve adequately.

**Camshaft**

The standard camshaft has a an inlet profile of 252 degree period and 12mm lift. The exhaust profile is 256 degree and 11.5mm lift. Computational performance prediction indicated that a longer periods would improve performance and so a number of profile from 260 to 280 degrees were designed. The analysis also indicated that increased valve lift would be beneficial but the consequences on valve springs and the cylinder head machining of the spring seats meant that such improvements would not be cost effective and they were therefore eliminated from the program.

At that time there was no computer model available for the valvegear simulation of the Falcon arrangement and time was, as usual, of the essence. The engine development team was pressed for progress and an innovative solution was applied. The development camshafts were made with the same ramps and flank acceleration as the standard camshaft but with the positive acceleration period slightly shortened to reduce the maximum velocity points and maintain the 12mm lift. This is shown in Figure 5. The stretching of the cam nose had the additional benefit of reducing the nose acceleration and allowing the same spring load cover factor even though the maximum power speed had risen from 4,500 to 4,750 rev/min.

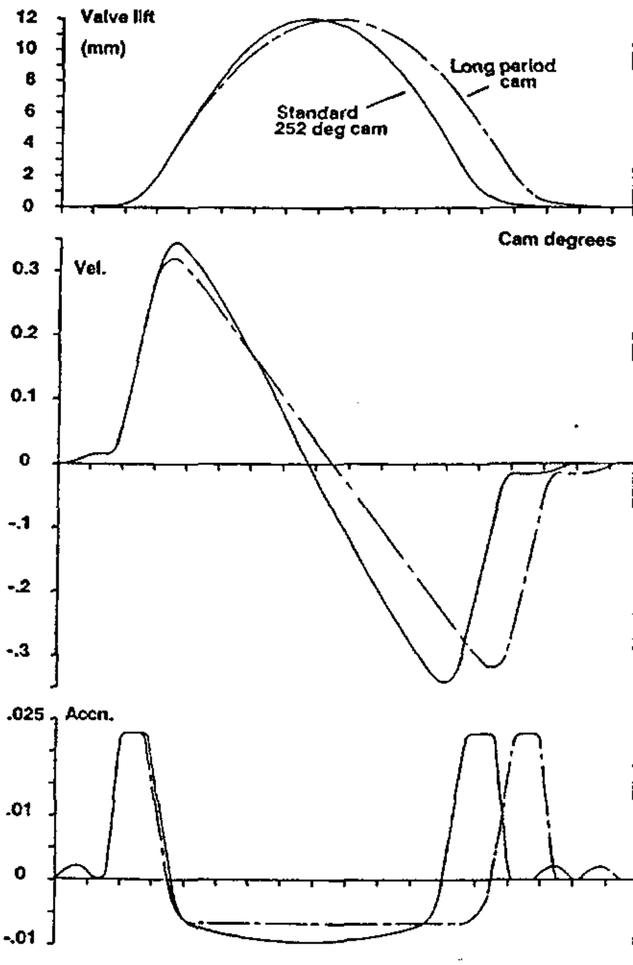


Figure 5. Simple stretching of original cams to retain the same lift. The ramps and maximum flank acceleration are identical.

**2vpc Project Conclusion**

Refinement to valve spring pre-load to ensure consistent compression in service and an increase in compression ratio from 8·8 to 9·0:1 (made worthwhile due to the close tolerance in Ford of Australia manufacturing) completed the mechanical package. Experimental changes to inlet system and exhaust manifold, though effective in function, would have been expensive to implement and were left out of the specification.

The project was completed with full engine calibration and a validation program. The final performance is shown in Figure 6 in comparison to the standard engine. An average maximum power from three engines was a genuine 161 kW when measured and corrected to DIN70020 (Net) rating. The torque of 335 Nm at 2,500 rev/min with a maximum of 365 Nm is particularly useful to most drivers.

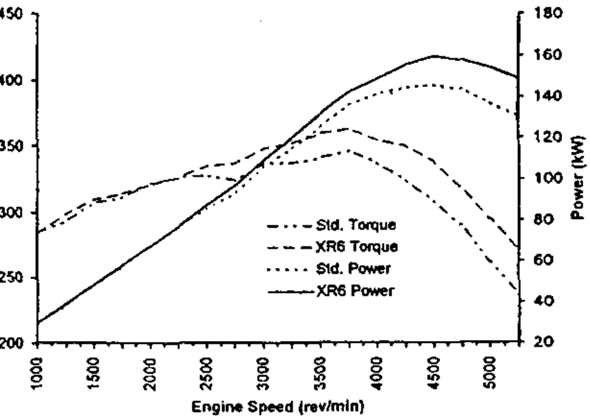


Figure 6. WOT Performance

The Tickford designed and developed engine produced 8·8% more power and 4·9% more torque than the standard engine, lifting the curve to approximately 95% of the maximum that could be expected from a road legal production engine. It is interesting to compare this with the state of the art. As engineering consultants, Tickford is often called upon to perform benchmark testing of one manufacturers vehicle and engine versus another. This adds to the company database of real information and measurements from production and other engines, an extract of which is shown in Figure 7.

Overlaying the performance of this cost effective two valve per cylinder engine onto the typical "best in class" line give the Figure 8. Given that it was never intended to produce maximum torque at speeds above 3,750 rev/min the overall performance is very acceptable and only 3% short of another manufacturers engine ( also 4·0L and six cylinder) with far greater cost due to having twin overhead camshafts and four valves per cylinder.

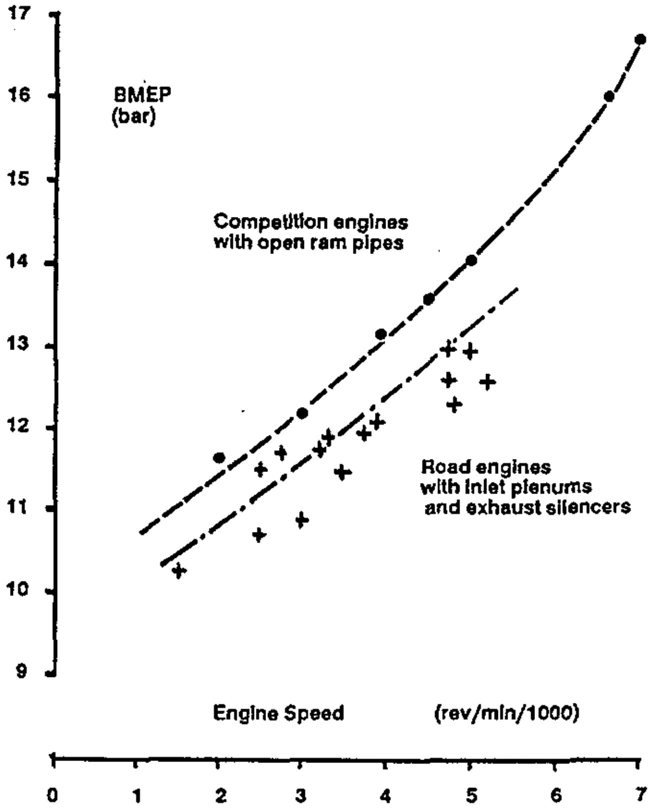


Figure 7. Significant BMEP figures from a variety of road vehicle and competition engines tested at Tickford

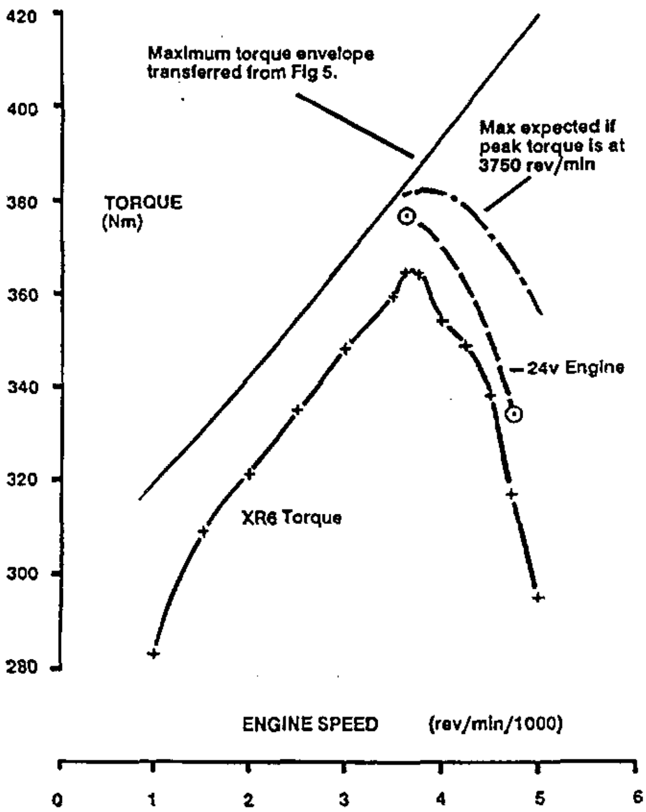


Figure 8. Comparison of XR6 Performance against the likely maximum.

It has been demonstrated through performance data and, more importantly, through customer car sales, that innovative engineering and attention to detail can bring very cost effective results.

FIVE VALVE PER CYLINDER CASE STUDY - There is a clear trend towards compact, lightweight yet powerful engines that can power mid sized passenger cars and provide driver satisfaction while improving on the fuel consumption of the larger swept volume engines that they replace. These engines are typically multi-valve designs with good maximum power but often have difficulty matching the low speed torque of their larger predecessors. The objective of the Tickford project was to produce a 5 valve per cylinder variant of the standard four valve per cylinder engine with superior torque and equal or greater power at no more than 6,500 rev/min. Combustion efficiency and Exhaust Gas Recirculation (EGR) tolerance was to be maintained but without the use of Inlet Manifold Runner Control (IMRC). A separate wish was for a 40mm reduction in the height of the cylinder head to suit packaging constraints. This latter requirement explains the unusual mix of valvegear types shown in Figure 9.

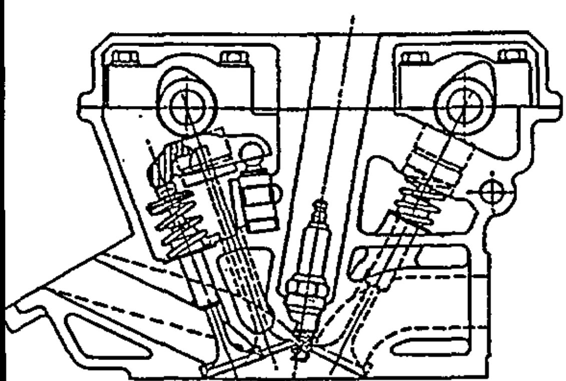

Figure 9. Valvegear Layout of 5v Head

The chosen engine has a bore of 82.4mm with a stroke of 79.5mm and the standard head provides commendably large valves for a four valve layout. With such a good baseline it is difficult to make improvements but finally an inlet valve seat area increase of just 3.5% was achieved with the five valve head. The benefit of the five valve per cylinder design was not the absolute increase in inlet valve area but rather the rate at which valve area becomes available. This in turn makes it possible to use shorter valve periods with consequential benefits in low speed performance.

Calculations of valve spring requirement indicated that the lower individual inertia of the small light valves could be controlled by lower load springs than used in the standard engine. Full advantage was taken of this to minimise friction losses and reduce the contact stress between cam nose and follower.

**Inlet flow**

The inlet valve mean flow coefficient of the high tumble ports was inevitably lower than the exceptionally free flowing ports of the base four valve head, in fact by 7 % which is more than the increase in inlet valve area. Consequently the total steady state flow capability, as measured on a flow bench, was actually less than the standard engine. This fact was not of great concern for two reasons. Firstly both four and five valve versions offered adequate inlet valve area. Using the long established calculation of Inlet Valve Mach Number or Gulp Factor, as originally defined by Livengood and colleagues in the mid 1940's, it showed that on both versions the Gulp factor was on the plateau of highest volumetric efficiency. Secondly, as already noted, it was known that five valve heads offered improvements in dynamic breathing capability which are not shown on the steady state flow bench.

ENGINE TEST BED DEVELOPMENT

In parallel with the design and manufacture of the modified engine the opportunity was taken to fit a standard engine with in-cylinder pressure transducers and run baseline tests to provide data that would be directly comparable with forthcoming measurements on the five valve version.

This work provided some very interesting results which showed how very effective the IMRC is in modifying the charge motion with its consequential effect on burn rate. In fact the low load combustion was such that it set a very difficult target to equal.

**Performance development**

The five valve per cylinder version suffered very few of the typical teething problems during initial break-in and soon produced the target power from the very first build configuration.

The primary runners (or ram pipes) were changed slightly to optimise the breathing and produce a small increase in performance at lower engine speeds. Subsequently more work was undertaken on the induction system to modify the internal volumes of the twin plenum chambers and to provide an interconnection between them. This was aimed at smoothing the mid speed torque curve by moving the commonly experienced adverse resonance trough between the two beneficial peaks.

Some of the configurations tested have produced a BMEP as high as 13.2 bar but generally these high figures have been associated with unacceptably peaky torque curves. A typical example of some more useful curves measured to date is shown in Figure 10. Here it can be seen that the five valve version produces around 10-12% more torque than the standard engine over the important mid speed range and a maximum power increase of approximately 8%.

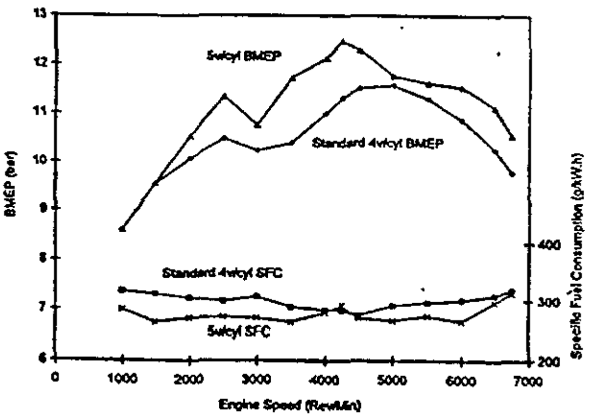


Figure 10. Maximum Performance and SFC

Interestingly the specific fuel consumption of the five valve version at WOT is approximately 10% better than the baseline four valve engine when each was set to optimised fuelling and spark timing. The test results gathered to date indicate that combustion chamber activity affects the degree of enrichment beyond stoichiometric AFR at which maximum performance is attained. Intense activity which promotes mixing of the fuel and air, as in this tumbling five valve configuration, minimises the power losses resulting from running at stoichiometric AFR rather than leanest for best torque (LBT) fuelling. This is of great interest for the forthcoming emission regulations where the three way catalytic converter will need to be active right up to full load, yet manufacturers will wish to be able to quote the highest maximum power figures. Further research work is being carried out investigate this matter in much greater detail and thus quantify the potential advantages of high tumble chambers.

**Rates of pressure rise**

The downside of a high activity chamber running at WOT is the high rate of pressure rise in the cylinder which results from the fast combustion. Typically this causes a harsh sound and feel to the engine which is difficult to totally insulate from the driver. This undesirable characteristic may prove to be a real problem with this, or indeed any type of engine, which cannot modulate the amount of activity within the chamber.

The standard four valve V6 has rates of around 3 bar per degree whereas the five valve version attains up to 6 bar per degree. In Europe it is commonly considered that rates of pressure rise up to 4 bar per crankshaft degree are acceptable but this is, to some extent, an arbitrary limit arrived at from experience with existing in line four cylinder engines, although individual cylinders have been measure with excursions up to 6 bar per degree.

A very simple subjective assessment made by those persons who have been close to this and other fast burn engines, when running power curves, suggests that different engine configurations and structures respond in different ways. The maximum rate of pressure rise suitable for an in-line four cylinder of 1.6 to 2 litres capacity may not be appropriate to a larger V6 or V8.

This is another subject worthy of further research in which it is planned to discover whether such high rates of pressure rise can be accommodated in practice.

**Fuel injector effects**

Much of the test work concerned the investigation of fuel injector position and spray pattern. To minimise the wall wetting of the inlet tract in four valve engines, i.e. with twin inlet valves, it is becoming more common to employ twin spray injectors having one beam aimed at each valve.

Triple spray injectors are even more difficult to make consistent and, being very aware of the considerable demands that this would put on the injector supplier, several simpler configurations were tested to determine if the engine would work satisfactorily.

Initial break-in and power testing was carried out with a single, fairly narrow, spray aimed at the central inlet valve. With this arrangement the power output was a little disappointing at normal Air Fuel ratios. Study of the specific air consumption figures indicated that not all of the air was being combined with fuel and burnt. The stratification of the intake charge was confirmed by the exhaust emissions where an unusually large combination of Oxygen and Carbon Monoxide was present at the same time.

To establish a best case baseline for steady state conditions some wider angle spray injectors were installed mid way along the inlet ram pipes to give more time for the fuel and air to mix thoroughly before entering the cylinder.

This modification produced an immediate improvement in specific air consumption of approximately 3 to 4% over most of the speed range, but the most outstanding change was in the emissions of unburned Hydrocarbons as shown in Figure 11.

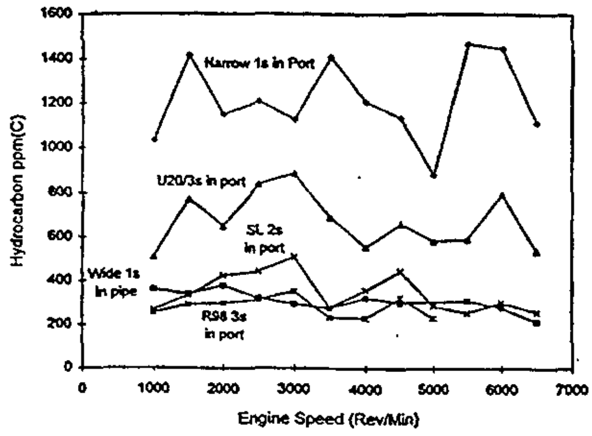


Figure 11. HC Emissions with various injectors

Unusually high HC emissions were recorded with the single narrow beam injector in the inlet port. The wider cone injector installed back up the ram pipe reduced this by 75% i.e. to one quarter of the original, in fact significantly lower than the baseline four valve HC emissions which are influenced by the necessity of aiming the single injector at just one of the two inlet valves of each cylinder.

A current production twin spray unit with claimed good atomisation lowered the HC towards the in-pipe results but once again was not considered suitable for transient conditions in a vehicle application.

The first attempt at making a tri-spray injector produced three separate sprays but with some imbalance of the individual beam flows and with rather poor atomisation. On the power curve this produced HC emissions around half that of the original single narrow spray cone injectors but still twice that of the in-pipe installation.

The second version tri-spray injector had a much better atomisation and beam to beam fuel distribution than the first. The HC emissions at wide open throttle were generally slightly lower than the in-pipe injector.

This proved that it is feasible to design and make an injector that will spray evenly over the back of all three inlet valves and enable the tumbling multi-valve engine to achieve good fuel mixing at wide open throttle. In doing so the HC emissions are considerably lower than can be achieved when inlet manifold runner control or valve deactivation necessitate injection down only one inlet port of a pair.

**Part load burn rate and EGR tolerance**

At the Ford World-wide mapping point the burn rates achieved by the tumbling five valve engine matched or exceeded those of the base four valve with IMRC activated to produce axial swirl.

Figures 12 and 13 show further results from the EGR loops run at this condition. Figure 12 shows that the tumbling five valve version closely matched the 0-10% burn angles of the base four valve engine. The 10-90% burn has a different response rate to EGR concentration but was slightly faster up to 20% EGR.

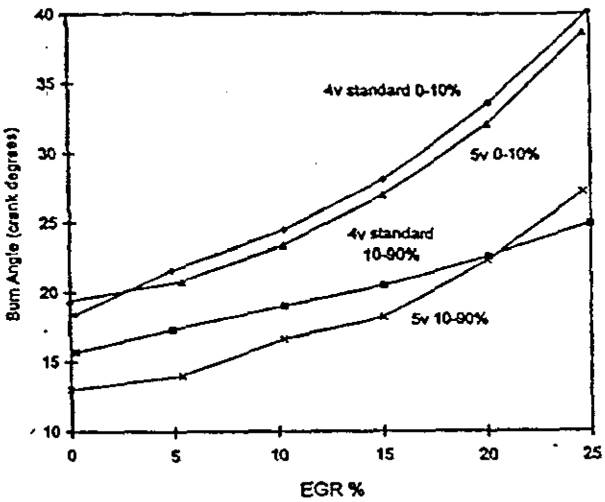


Figure 12. Burn rate at low load point.

Figure 13 shows the comparative specific emissions of unburned HC and NOx for the same tests. The tumbling five valve produces slightly more HC but lower NOx.

Although not shown here both the specific fuel consumption and the stability, as measured by the Coefficient of Variation of IMEP, were almost identical up to 20% EGR flow.

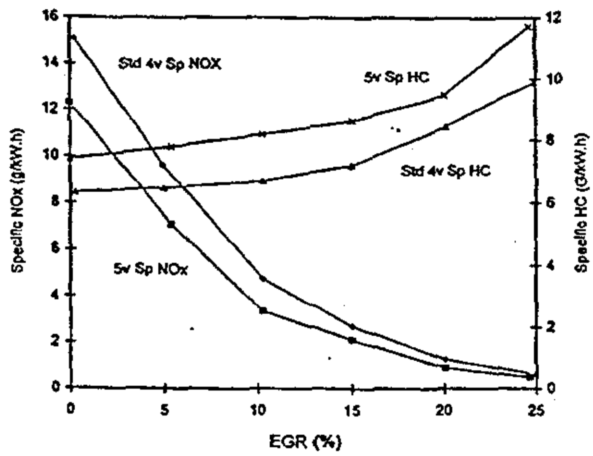


Figure 13. HC and NOx at low load

**5vpc Project Conclusions**

A five valve per cylinder version of an existing four valve per cylinder V6 engine has been made which produces significantly more performance and better fuel consumption in conjunction with lower HC emissions at Wide Open Throttle.

Initial tests indicate that lean best torque fuelling is closer to stoichiometric AFR than in combustion chambers with less activity. This is clearly worthy of further investigation including testing various tumble ratios to determine the trade off against rates of cylinder pressure rise and their effect on NVH.

The part load burn rates and EGR tolerance are very similar to the four valve baseline engine which utilises IMRC to achieve the desired in-cylinder charge activity.

The combination of the volumetric efficiency advantages of a five valve per cylinder layout with vigorous tumbling charge motion has resulted in an engine with the potential for matching or bettering the base engine without the complication, cost and potential problems of the IMRC and its control systems.

## CONCLUSIONS

In two very different projects, Tickford has demonstrated its thorough knowledge and understanding of port flows and combustion. The prime objective of meeting the client's production requirements have been combined with the quest for optimum performance in a way that delivers a quality product.

Tickford currently has engine projects on engines with 2, 3, 4 and 5 valves per cylinder and in a range of 800 $cm^3$ to 6·3L for passenger cars and light commercial vehicles. The skills demonstrated in the work of this paper are applicable to a wide range of applications and also to the different fuels used in the engines. These include gasoline of various qualities from round the World, Compressed Natural Gas (CNG), Liquefied Petroleum Gas (LPG) and special gasoline / alcohol mixes used in some countries.

The fundamental principles of port flow, charge motion and combustion efficiency demonstrated by the two case studies of this paper are being applied to customer projects World-wide. These in turn provide Tickford clients with high specific output, low emissions engines delivered to the market in class leading cycle times.

## ACKNOWLEDGEMENTS

The authors are grateful to Tickford Vehicle Engineering Pty of Melbourne, Australia, and the Powertrain Department of Advanced Vehicle Technology, Ford Motor Company, Dearborn for providing financial and technical support for these projects.

Thanks are also due to the many colleagues at Tickford who carried out the design and development work described in the paper.

# Paper 6

## "Technology trends in power cylinder systems"

**Bale, C.J.C. and Neuhäuser, H-J. (1998)**

Automotive Technology International 1998

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Pages 62-66

# Technology trends in power cylinder systems

Eur Ing CJC Bale, Dr HJ Neuhäuser, AE Goetze Advanced Technology Centre, T&N Piston Products Group

*The well-known pressures on the automobile from society relative to air and noise pollution, fuel consumption, durability and price are exactly mirrored by the pressures on the power cylinder system (piston, rings, pin and cylinder liner).*

The cylinder system is fundamental to the engine and is at the heart of the energy conversion process from fuel to motion. The design of particular components has an immediate effect on the net work extracted, the amount of fuel used and the emissions produced both as pollutants and noise.

The development of cylinder, or piston systems is also subject to industry pressures to reduce defects to levels measured in a few parts per million, reduce lead time and price to the engine builder, extend durability and reduce lifetime cost to the owner or operator. All of these factors can be summarised in a table of the parameters that govern cylinder system design (Table 1).

The piston designer has to accommodate a series of conflicting demands. Performance improvements are achieved by an increase in engine speed or brake mean effective pressure. This requires a more robust piston, yet higher speeds demand a lower reciprocating mass in order to contain bearing loads and skirt side forces (Figure 1).

Engine designers require pistons that are shorter either in skirt or compression height to achieve a shorter connecting rod and lower cylinder block height. This height reduction requires rings with low axial widths for all piston grooves, but with similar or increased expectations for strength, durability and overall performance.

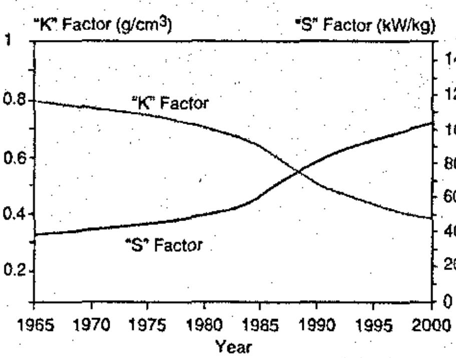



Figure 1. Comparison of piston mass and engine ratings.

Shorter skirts pose challenges in terms of guidance in the cylinder bore and a potential adverse effect on piston mechanical noise and ring pack performance.

Traditional measures for comparison of different designs have involved 'apparent density' or 'K' factor, expressed as weight divided by cube of diameter. However, this takes no account of engine output and other factors. In response to this, AE Goetze has adopted an alternative 'S' factor which takes into account how pistons have become lighter as ratings have increased.

$$\text{'S' Factor(kW/kg)} = \frac{4 \times \text{Power(kW)} \times \text{Bore (mm)}}{\pi \times \text{Stroke(mm)} \times \text{N° of Cylinders} \times \text{Piston Mass (kg)}}$$

Reductions in the skirt contact area have been taken a stage further by AE Goetze through its 'AEconoguide' design and the 'X-Piston' (Figure 2), both of which offer demonstrable improvements in specific fuel consumption (Figure 3). Development of both these concepts continues in the pursuit of even lower friction and reduced engine noise.

Another key area of interest is the piston material, on whose properties the success of the design depends. Improved properties allow the designer to reduce the quantity of material required, with benefits in the areas of weight, cost and process. AE Goetze has developed a new family of aluminium-based alloys for both gasoline and diesel applications. For example, the high copper content alloy AE135 has resulted in major benefits for high top ring design gasoline pistons due to its superior strength and wear characteristics.

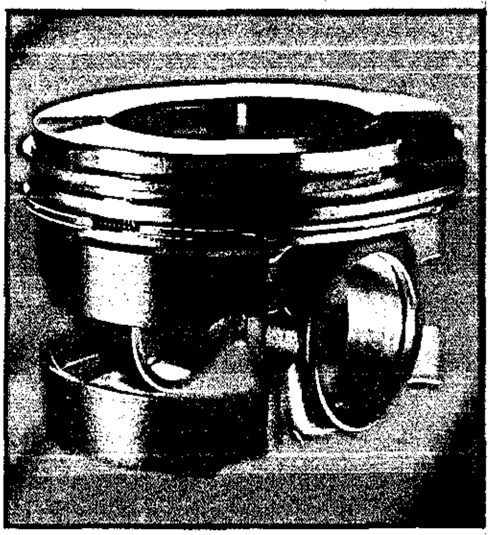
AE135 has also achieved major improvements in damage resistance at the ring and ring groove interface which is subject to high temperatures and marginal lubrication. This improvement is particularly effective when combined with AE Goetze's proprietary phosphate treatment (AE082) which provides a cost-effective alternative to anodising. In order to achieve the optimum piston solution through gravity die casting and other processes, alloy development is an ongoing process.

Materials development is not confined to conventional aluminium. AE Goetze has produced pistons using Metal Matrix Composites utilising their

| Direction of Change | What Parameter | Effect on piston system |
|---|---|---|
| REDUCED | EMISSIONS | Higher top ring<br>Reduced oil consumption<br>Low noise designs |
| | FUEL CONSUMPTION | DI Gasoline designs<br>Lean burn<br>Higher specific power<br>Low friction |
| | NVH | Reduced clearances<br>Innovative coatings |
| | COST (of product, development and ownership) | Improved processes and innovative manufacture |
| | LEAD TIME | More available technology<br>Validated CAE tools<br>No iterations<br>Reduced testing |
| INCREASED | PERFORMANCE | Increased cylinder pressure<br>Increased crown temperatures |
| | DURABILITY | Higher fatigue life materials<br>Better life modelling |
| | RECYCLING | Eliminate heavy metals |

Table 1. Parameters governing cylinder system design.

Figure 2. AEconoguide (above) and 'X- Piston' (below).



high strength-to-weight ratio. Magnesium pistons, aluminium-based powder metallurgy materials and carbon-reinforced carbon pistons are also in development at the company's Advanced Technology Centre in Bradford. Currently, the cost of these alternative materials remains an issue, but their advantageous properties may lead to specialist applications in the future.

The large number of factors governing the design of the piston, together with the complex interactions of the piston with other components, have accelerated the movement towards the integrated design of complete power cylinder systems. The primary roles of transmitting power and forming part of the combustion chamber must not overshadow the function of the piston as a complex bearing which interacts with the rings, pin and cylinder bore. To meet this challenge, AE Goetze provides a complete range of piston ring materials and surface coatings.

As ring axial widths are reduced, especially for gasoline engines, to give extremely lightweight pistons with reduced top land height, AE Goetze ring technology is moving forward in base materials of both steel and cast iron, together with sophisticated coatings. Compression rings as low as 0·8mm axial width, second rings of 1mm and oil control rings of 2mm width have all been successfully developed by AE Goetze. This trend will progress to the successful development of two-ring pistons with very low axial widths.

Depending on the application, in gasoline or

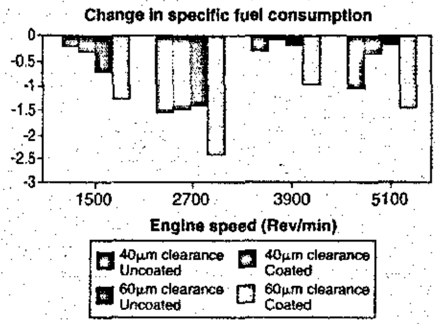
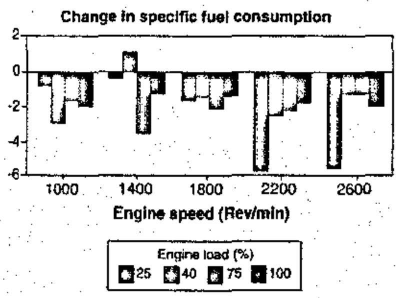



Figure 3. Improvement in SFC due to 'AEconoguide' and 'X -Piston'.



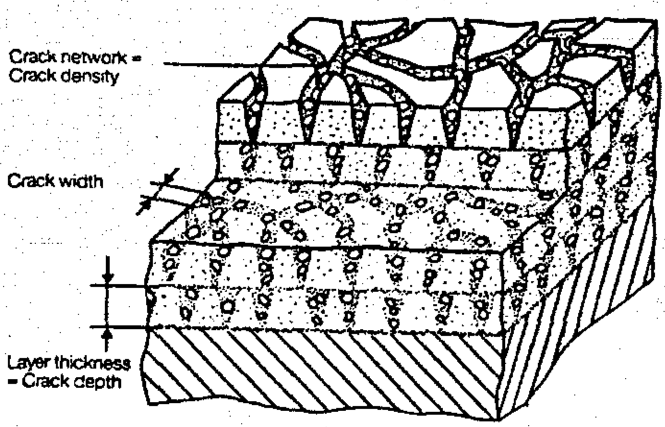




Figure 4. Chromium ceramic coating CKS36.

diesel engines, AE Goetze has four different surface treatment technologies for piston rings, which allow them to be tailored to offer optimum levels of scuff and wear resistance.

## Piston ring surface treatment technologies:

- Nitriding
- Hard chromium coatings, especially AE Goetze's successful chromium ceramic coating CKS36 (Figure 4)
- Flame and plasma-sprayed coatings with high velocity oxygen flame (HVOF) technology as the latest stage of development, and
- Physical vapour deposition (PVD) coatings.

All of these improvements in the design and materials are brought together in pistons and rings which are designed and evaluated with the use of advanced computer tools. The basic design rules can be applied to the known dimensions provided by the customer to produce initial designs in a matter of hours rather than days (Figure 5). The solid models generated are then subjected to stress, temperature and fatigue simulations to ensure that the design is robust (Figure 6).

At this stage the considerable knowledge base

Figure 5. Piston solid modelling.

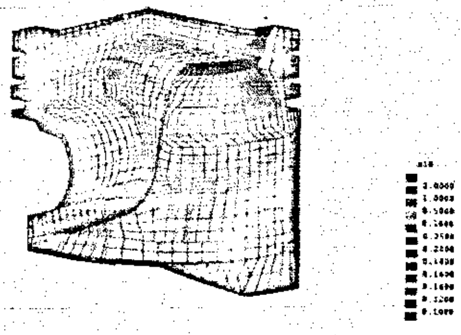
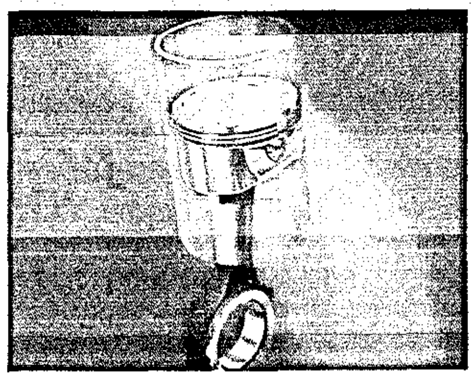


Figure 6. FE result.



System graphic.

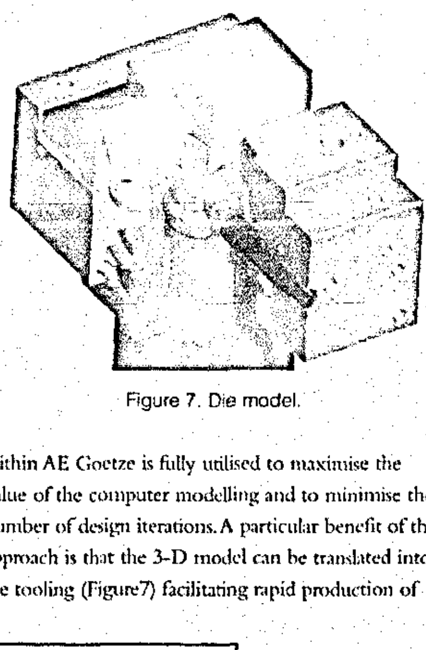


Figure 7. Die model.

within AE Goetze is fully utilised to maximise the value of the computer modelling and to minimise the number of design iterations. A particular benefit of this approach is that the 3-D model can be translated into die tooling (Figure7) facilitating rapid production of prototypes suitable for engine development. The final testing of the piston can be confined to validating the development done in the previous analytical stage.

Ring and liner development also benefit from the company's advanced predictive engineering resources, predicting ring dynamics (movement, blow by, friction and oil film), ring conformability and shape (including twisting and pressure pattern throughout the engine cycle) and bore distortion. Prediction of bore distortion, which has been validated on over 250 engine designs, is proving extremely useful for customers as it takes into account the influence of engine design under cold and hot running conditions.

Industry trends in engine block construction (Figure 8) are having a significant effect on cylinder system engineering. More alloy blocks and, in the interim, more linered engines, offer new challenges. AE Goetze has developed innovative methods of liner production to meet both tribological (inside surfaces) and structural needs (outer areas).

The integration of key competencies and expertise in tribology, gas dynamics and power transmission, together with an in-depth understanding of the dynamic engine structure, enables AE Goetze to provide customers with the evolutions and revolutions in piston system technology into the next millennium.●

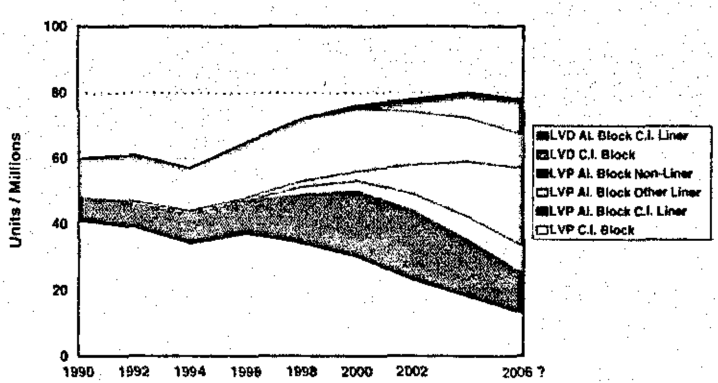



Figure 8. Trends in cylinder block construction.
LVP: Light Vehicle Petrol, LVD: Light Vehicle Diesel.

For furthe[illegible]

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# Paper 7

# "Real-life Vehicle Exhaust Emission Performance Compared with Legislative Drive Cycles"

**Bale, C.J.C. and Farnlund, J. (1999)**

Proceedings of the Institution of Mechanical Engineers

Conference Transactions 1999-9

Integrated Powertrain Systems for a Better Environment,

Paper C575/030/99

Pages 27-47

C575/030/99

# Real-life vehicle exhaust emissions performance compared with legislative drive cycles

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## ABSTRACT

Progress in improved emissions performance of vehicles to date has largely been driven by legislation and not by market demand. Although all prospective buyers, whether private or corporate, are likely to say, when asked, that they want "green" vehicles, few are prepared to pay more for the privilege. Currently, individual vehicle manufacturers only have to ensure that their products perform to the legislated standards when measured for certain emissions, that are common to all, and will specify their lowest cost solution to meet those standards. These legislated emissions are based on test cycles that the industry accepts but which, as is well known, do not reflect actual driving conditions accurately. This paper sets out some assessments of the emissions performance of various passenger cars over a range of speeds and loads and relates their performance relative to the stated legislative performance. It shows large variations in real-life emissions and fuel consumption performance of a range of vehicles that comply with the legislation

## 1. INTRODUCTION

During the last two decades, massive changes have taken place in the automotive industry and environmental awareness; the industry has invested heavily to meet ever increasing demands from the legislators for environmental and safety improvements. Of necessity, the test standards include tight specification of the drive cycles during which the emissions and fuel consumption of the test vehicle are measured, and the temperature conditions of operation. Furthermore, the testing is done on a chassis dynamometer whose load characteristics are derived from further fieldwork that is monitored by legislative bodies or their representatives. All these factors are laid down for compliance of vehicles sold over a large geographical area for which the test and driving conditions may be more or less indicative of the operation of the vehicles in public hands. Deriving the drive cycles has taken years of testing, negotiation and implementation; a task that should not be underestimated when dealing with a variety of languages, cultures and individuals representing different countries. It was inevitable that the current solution would be a compromise but it is one that drives the manufacturers' final engineering solution.

In the course of developing a range of test equipment for vehicles, which enable performance and emissions measurement over a wide operating range and for a large variety of vehicles, Rototest AB has recorded a large amount of data from vehicles. Knibb, Gormezano and Partners (KGP) is a specialist consultancy focusing on management and technology in the automotive industry. It engages in worldwide, single and multi client studies for vehicles, components and service companies as well as writing reports on the sector. The company's work in the field of Vehicle Environmental Performance (1) has given it a good understanding of the techniques and hardware used to make clean powertrains around the world.

This paper reviews data from the present market situation and highlights alternative techniques that are used. Data taken in legislative and non-legislative trials has already come into the public domain and this is certain to influence increasing numbers of the increasingly environmentally aware public. Data on vehicle emissions is increasingly available in Europe (2) where legislative bodes, consumer magazines and specialist publications like Automotive Environment Analyst (3) now report on results and states on knowledge in manufacturers, dealers and the public at large.

Rating schemes all have their limitations with regard to what vehicles are tested and how many examples of each specification can be accommodated. Not all models available in the market are assessed and each of the bodies carrying out the testing has its own agenda. The selection of test conditions under which to rate a vehicle, will re-open many of the arguments faced by legislators who derive the type-approval test conditions. Nevertheless, the variety of data being produced is increasing and although some will see it as "industry bashing" in the hope of increasing circulation of the publications, that is clearly not the objective of this paper. On the contrary, by drawing attention to the type and quantity of work being done in this field, manufacturers and systems suppliers stand to gain competitive advantage from a measured response to the knowledge base that is growing rapidly in the public domain.

To illustrate the points in the paper, it is primarily written with spark ignition, gasoline terminology. However, other fuels and diesel engines are similarly affected.

## 2. DRIVE CYCLES

The present European legislative drive cycle requires a vehicle to be started and run from a preparation temperature of between 20 and 30 degrees Celsius. The vehicle is then driven on an "urban cycle" for some 4 km at speeds up to 50 km/h, followed by an "extra urban" portion of the test that runs above urban speeds and includes a short excursion to 120 km/h (later referred to and shown in Figure 6). The US Federal test procedure uses the same start temperatures, interestingly, and then proceeds to follow a fully transient, stop-start driving cycle at speeds up to 57 miles per hour. There is an additional "Highway Fuel Economy Test" that is used to determine fuel consumption for comparative purposes where a check on NOx emissions is also included. Broadly speaking, most other cycles in the world are based on one or other of the above, perhaps with the exception of Japan and India that have their own drive schedules.

It is therefore easy to be sympathetic with those in Scandinavia and North Germany who feel that the proving of vehicle emissions compliance under these conditions, is barely relevant to the impact of vehicles on air quality in their locality. On the other hand, those in Italy or Spain probably feel that the temperature conditions are on the cool side. It is factors like this that mean compromise is inevitable.

The other key factor is the range of engine speeds and loads that are used during the emissions cycle. Any cycle will have a defined set of accelerations and speeds that, when combined with an individual vehicle weight and power, mean different speed and load combinations relative to the whole output envelope.

A recent paper gave an example of the typical load requirements of a vehicle undertaking the US FTP drive cycle; this is shown in Figure 1.

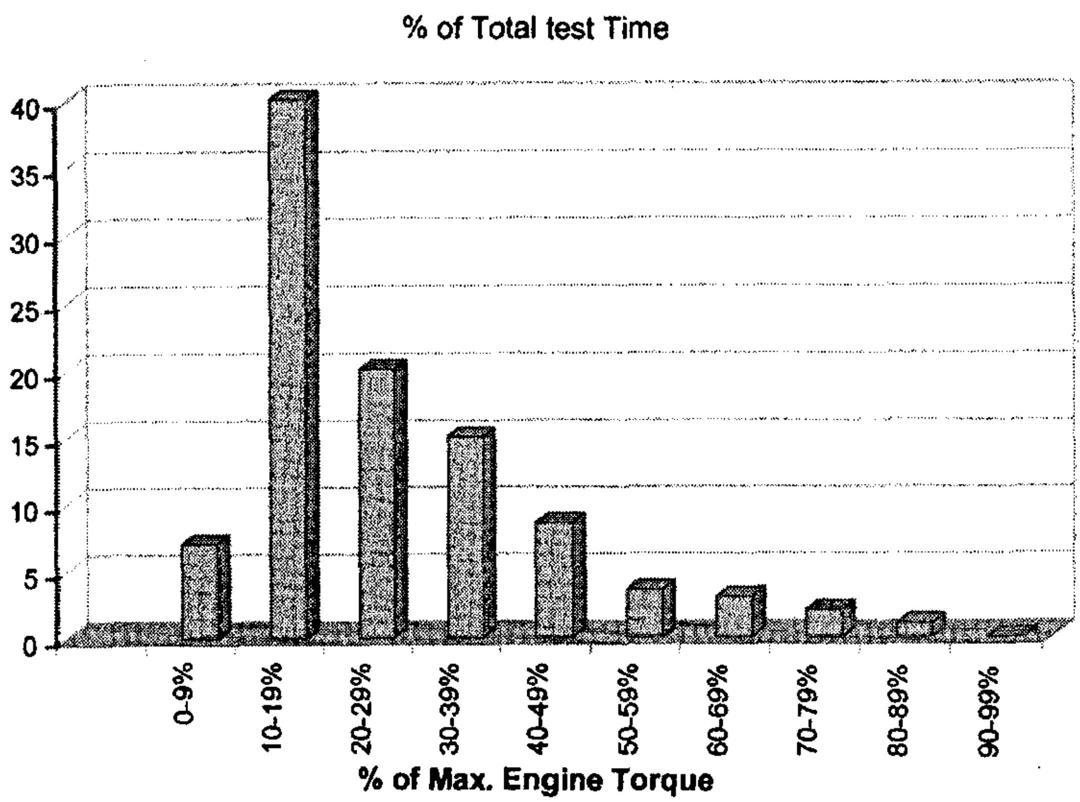



Figure 1: Proportion of engine loads used on U.S. FTP drive Cycle
Source: Akenhurst et al, 7th EAEC Congress, Paper STA99P407

## 3. VEHICLE CALIBRATION

Manufactures must make a calibration of the engine management systems for fuel amount, fuel delivery timing and ignition timing to ensure proper operation throughout the engine operating conditions. This is essentially based on past data and theoretical values (desktop calibration) refined during extensive engine dynamometer mapping and drive simulation, followed by final refinement in a fleet of vehicles using chassis dynamometers, road testing and climatic development. During the engine dynamometer and chassis dynamometer stages, the emissions output can be monitored closely to optimise the engine and minimise pollution. Additionally, the temperatures of the exhaust system and catalyst system is monitored to develop an optimum conversion temperature as quickly as possible whist preventing damage due to overheating. During the road and climatic test periods, the accent is on ensuring proper starting, operation and derivability for the customer.

Some engine management systems and calibrations maintain closed loop control throughout the operating range of the engine. This ensures that three-way catalyst operation is maintained but it is then necessary to cap the performance of the powertrain at the limits of temperature of the catalyst system. Figure 2 shows differing engine output comparisons for the freely mapped (solid line) and closed loop (dotted line) conditions. This clearly indicates an element of choice on the part of the manufacturer and the consumer regarding the emissions and output performance of the vehicle. When it was first suggested that vehicles run closed-loop throughout the operating range, one manufacturer adapted a current engine by fitting a smaller throttle body in order to limit air delivery to the equivalent of the closed loop power line in the figure. The customer was offered a lower power but cleaner alternative.

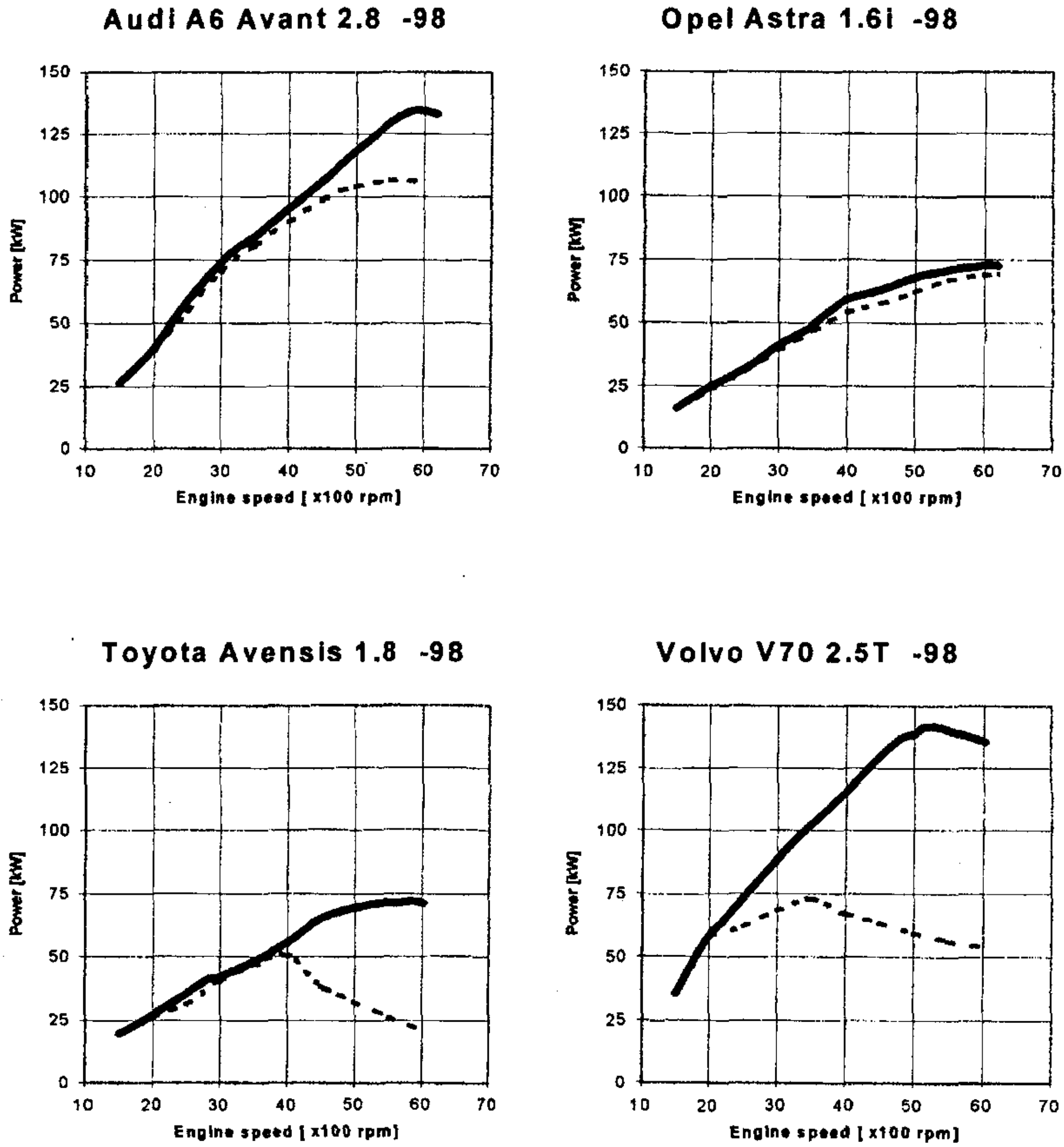



Figure 2: Closed loop areas

Source: Rototest AB, 1999 ©

## 4. ENVIRONMENTAL PERFORMANCE

The requirement to have different mapping criteria for different operating regimes of the engine result in varying degrees of emissions control. In most vehicles it is necessary to move outside the optimum conversion area of the catalyst system (i.e. into open loop operation) in order to preserve the life of the system so that it works well and for a long time when in its controlled area (i.e. closed loop operation). The impact on emissions of hydrocarbons for a typical engine is illustrated in Figure 3 from which the closed loop area can be inferred.

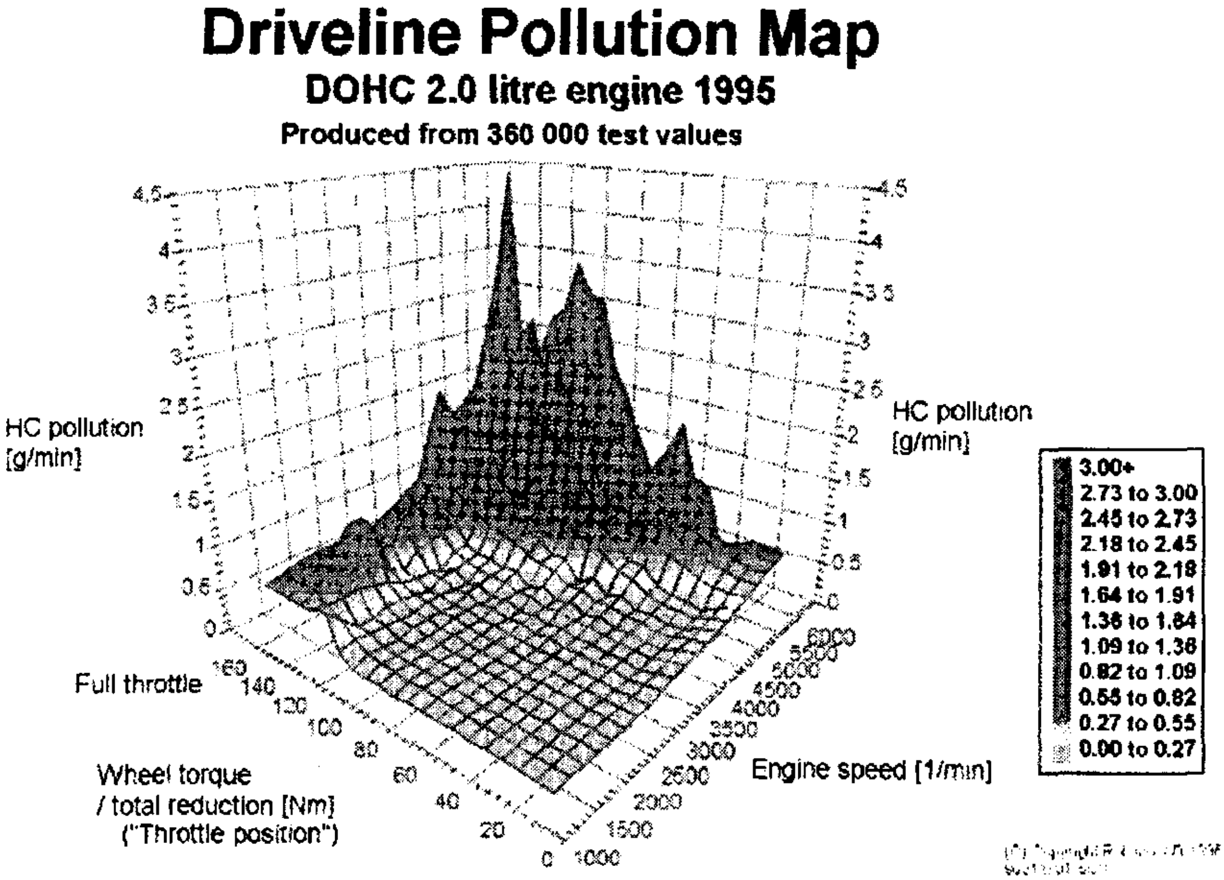



Figure 3: Example of Hydrocarbon emissions versus operating condition
Source: Rototest Data. © Copyright 1998-1999 Rototest AB

It is this engineering necessity that causes real life emissions over a wide range of operating conditions to be a different set of measurements from those achieved solely over the legislated driving cycle.

By combining the various emissions measured during testing, Rototest AB has developed a combined measure for emissions output call Environmental Performance Index (EPI) for which the calculation is given in Appendix 1. Extracts from these rankings has been published in 'Teknikens Varld'[#] in Sweden and 'What Car?' in the UK and, while the results can make sensational journalism, the sound data clearly highlights the difference between vehicles EPI and the quoted or certified emissions performance. The published data is reproduced in Figures 4 and 5 showing a spread of some gasoline vehicles tested.

For example, taking a random selection of vehicles from different positions in the "Green Car Ranking List", the quoted vehicle emissions (2) can be compared in Table 1.

[#] It is worth noting that it was Teknikens Varld that initiated the Mercedes 'A' Class crisis with its now famous 'Moose Test'. This amply demonstrated the impact that media reports directed at the consumer can have.

| | | | Quoted Emissions over European Drive Cycle | | | |
|---|---|---|---|---|---|---|
| Vehicle | Position in Ranking List | Rototest EPI | THC g/km | CO g/km | NOx g/km | CO2 g/km |
| VW Lupo 1·4 16v Manual | 1 | 64 | 0,157 | 0,053 | 0,044 | 73 |
| Audi A4 1·8T Manual | 25 | 95 | 0,798 | 0,143 | 0,121 | 71 |
| Peugeot 306 XS 1·6 Manual | 54 | 124 | 1,049 | 0,151 | 0,048 | 74 |
| Mitsubishi Space Wagon 2·4 GDI Manual | 80 | 196 | 0,070 | 0,040 | 0,050 | 70 |

Table 1: Quoted Emissions and EPI for a Selection of Vehicles (2)

Source: Kraftfahrt-Bundesamt 1999 (Kraftstoffverbrauchs- und Emissions-Typprüfwerte von Kraftfahrzeugen mit Allgemeiner Betriebserlaubnis oder EG-Typgenehmigung)

Using the calculated EPI (ref. Appendix 1) Rototest has collected data on many vehicles and some of the combined results are shown in the following charts, Figures 4 and 5.

The data in Table 1 clearly shows that the Rototest "Green Ranking" position does not actually reflect the certified emissions performance quoted. The fact that a vehicle performs well over the emissions cycle does not necessarily represent its overall emissions for a wide range of operating conditions.

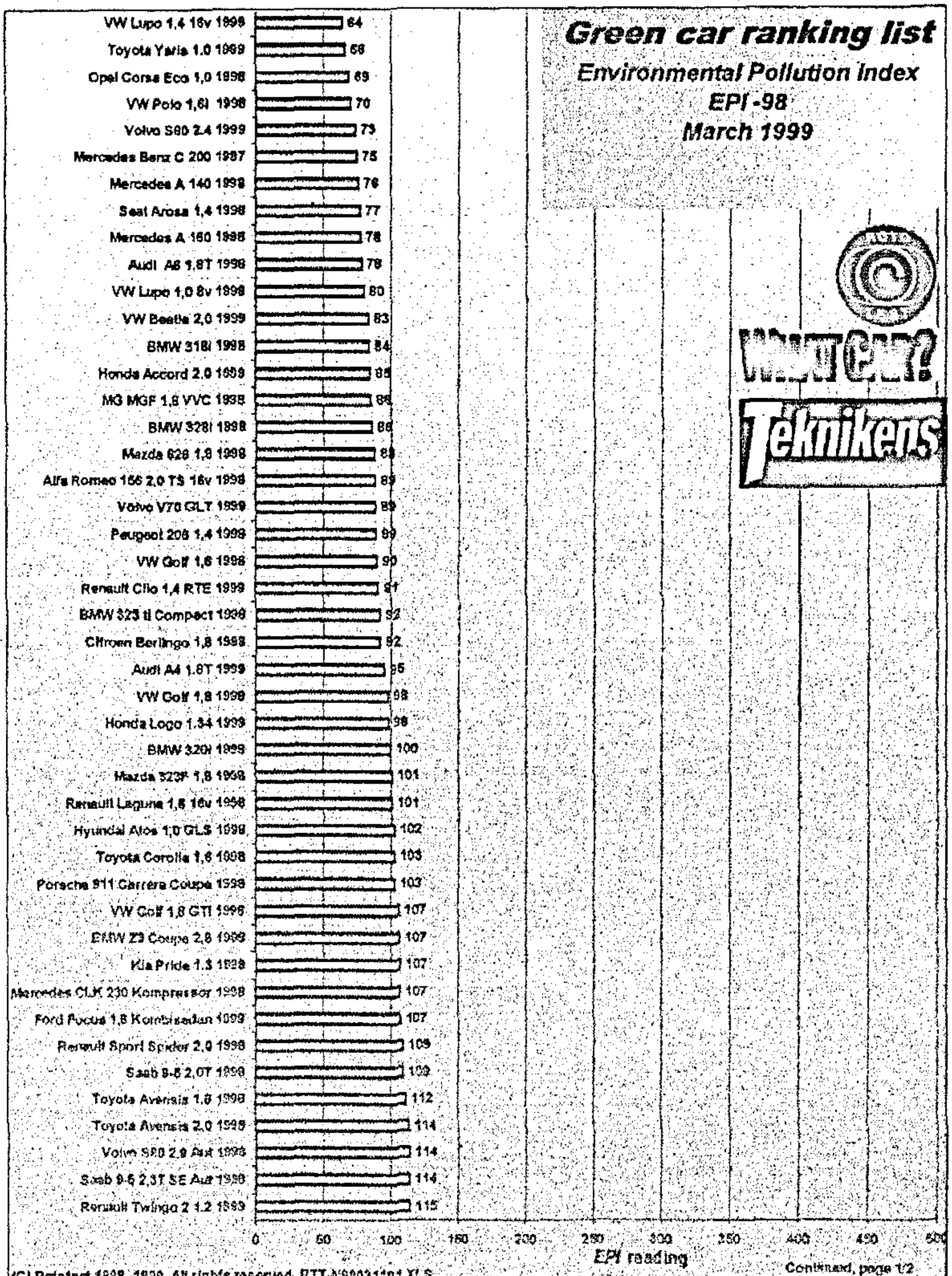



Figure 4: Environmental Performance Index Examples (Lower Values)

This figure shows the upper part of the table of examples and this can be referred to as the clean half or the "green" half of the data set.

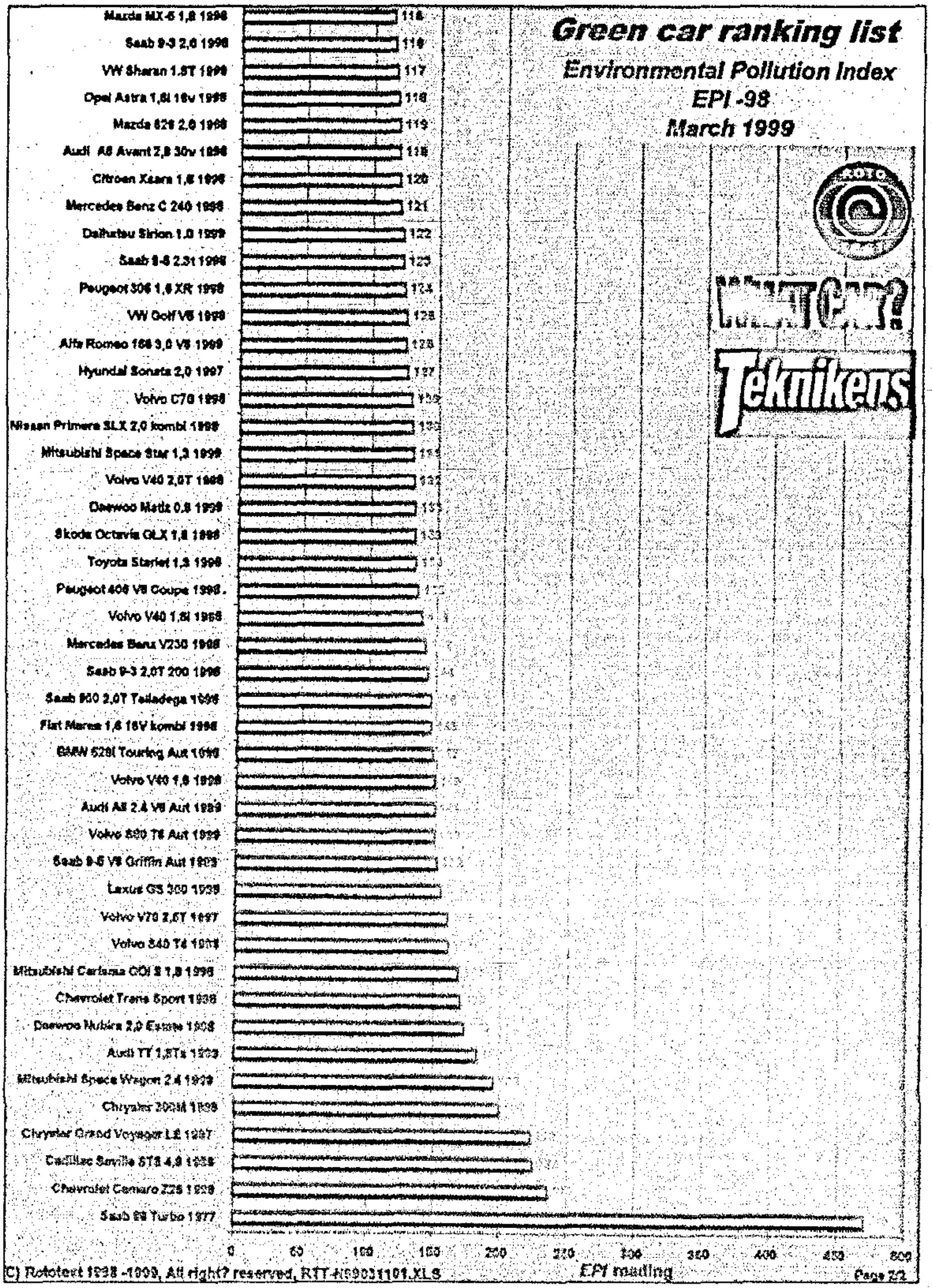



Figure 5: Environmental Performance Index Examples (Higher Values)

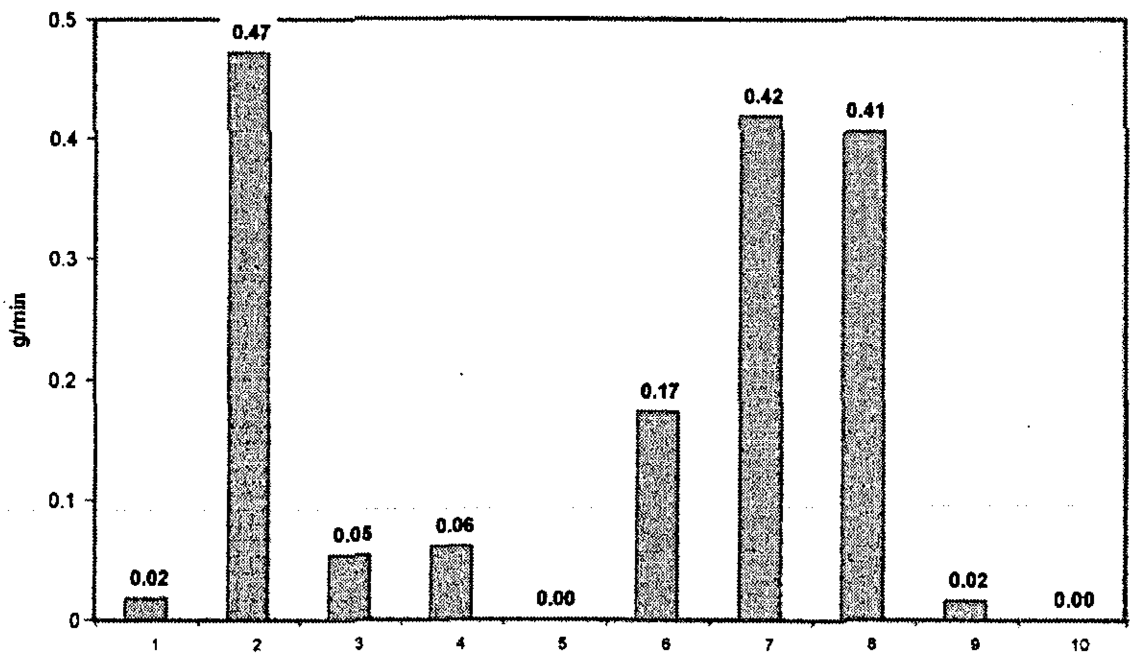
It is important to note that the comparison made in table 1 is not exactly like with like. The legislative test cycle is derived from countless studies of driving behaviour in urban and extra urban areas. The maximum acceleration used is 0·53 m/s$^2$ from 0 to 50 km/h, i.e. it will take 26 seconds to reach 50 km/h. This compares with the typically quoted, full load, 0-100km/h times in the sales literature of about 10 seconds, which is an average of 2·78 m/s$^2$. It is safe to say that the emissions drive cycle is biased towards idle and low load operations and will therefore favour cars with small engines, even in bigger cars. The smaller engine will be run at loads where it will have higher efficiencies. The cycle also requires all cars, whether it is a Porsche 911 or a Nissan Micra, to be driven with the same accelerations and make the gearshifts at the same speeds. In real driving this will not be the case. In some cases, the gear ratios in the transmission are optimised to suit the test cycle instead of the cars real use.

The intention of the EPI is, first of all, to compare cars and to give an answer to which cars are the better ones for overall environmental performance. This is also one reason for using an index as comparison. The EPI-test tries to cover most of the different types of driving that an average driver uses. To begin with, one has to accept that cars are driven differently depending on their performance and should, to some extent, be tested differently. An example of this is the selection of gear at 50 km/h. This is specified to be the highest gear where the car still has a minimum acceleration capacity of 1·0 m/s$^2$. Most of the cars will have to use third gear but some cars will be able to use fourth and, in some cases, even fifth gear.

The EPI-test covers engine outputs from 3-4 kW, at 50 km/h, up to the maximum power output. The reason for including high loads is that the pollution levels can be a 1000 times higher, or more, compared to steady speed and will therefore give a substantial contribution to the total amount of pollution, even if the time for which it is used may be short. The high loads in real driving stems from accelerations, going uphill and towing a trailer just to name a few. Overtaking other cars is a case where it is easy to understand the result of differences in performance. A more powerful car can have higher pollution levels but during a shorter time. The EPI takes this into consideration (Ref. Appendix 1).

As an example of the differences in emissions with closed-loop control compared to open-loop, figure 6 shows the CO-emissions at a constant speed of 90 km/h (55 mph). Some of the cars have levels near the detection limit and others up to nearly 0.5 g/min.

Figure 6: CO Emissions at 90km/h Constant Speed Source: Rototest Data. © 1998-1999 Rototest AB



In figure 7, the same cars are shown at their maximum torque. In this case the levels are reaching, in the worst case, nearly 1 kg/min!

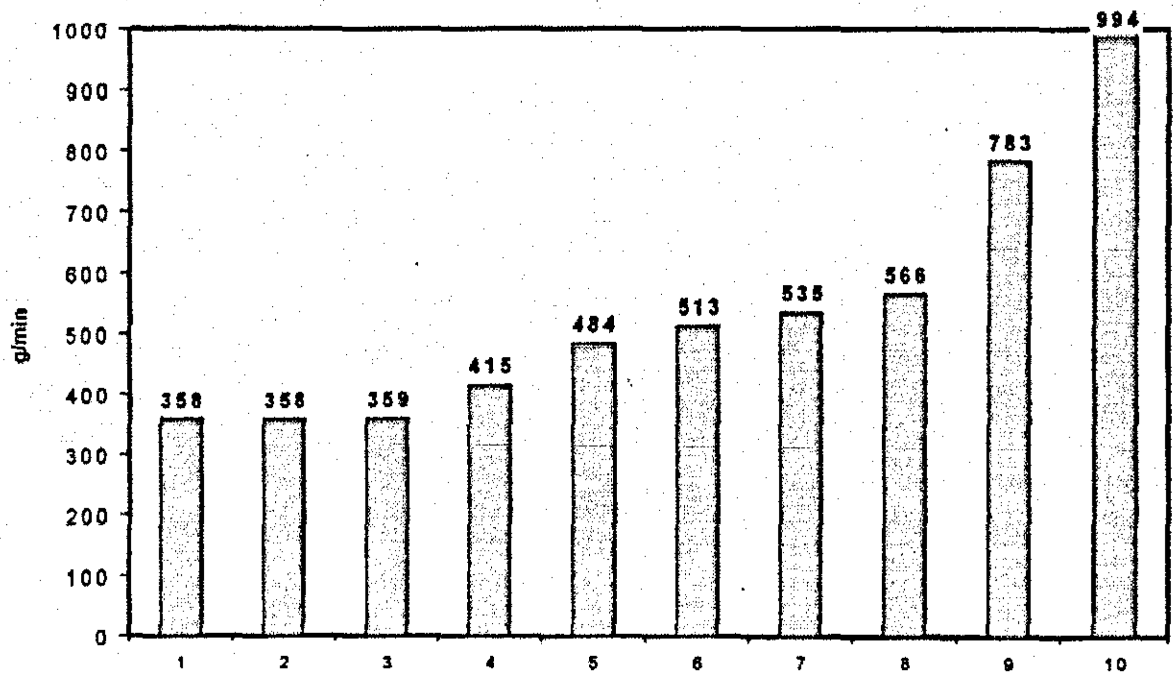



Figure 7: CO-emissions at Maximum Torque
Source: Rototest Data. © Copyright 1998-1999 Rototest AB

Depending on the power to weight ratio of the test vehicle, a small minority do need full throttle to make the final acceleration to 120 km/h in the extra urban cycle. There is even a dispensation for lower powered cars that cannot keep to the speed trace for this acceleration, to accelerate at full throttle until the driving speed and the cycle speed can be coincident. This is illustrated in figure 8.

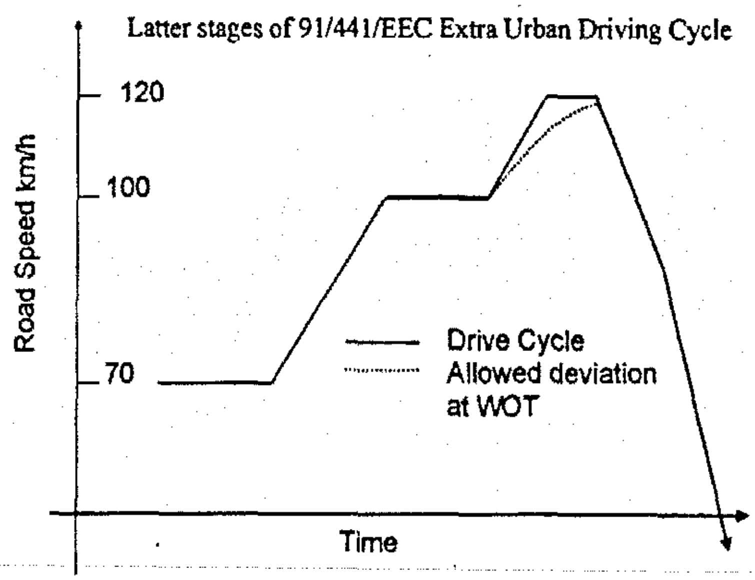



Figure 8. Example of Allowable Deviation from EUDC for Low Powered Vehicles

Such vehicles are using areas of their calibration at the edge of the the operating map. These vehicles therefore have to have emissions systems that maintain adequate control at full load, for the drive cycle. It could therefore be argued that these are cleaner cars by virtue of being better controlled but, as low powered vehicles they spend more time at high load (and perhaps high engine speed) and could alternatively be regarded as relativley high polluters and high fuel consumers

Further work on real-world emissions measurement is being carried out by Rototest AB in the field of particulate emssions and unregulated emissions using similar test technquies to those outlined above and in Appendix 1. Early indications on the subject of particulate emissions are shown in Figure 9 and will be reported during the presentation of this paper. The example in Figure 9 compares a normal SI-engine (VW 1·6L) with the 1·8L direct injected gasoline engine from Mitsubishi and a direct injection diesel engine from SAAB. It is clear that the particulate emissions that have been regarded as being a diesel phenomenon, are present in all engines and the nature of the combustion of direct injection gasoline engines results in more particulates than indirect or port fuel injection.

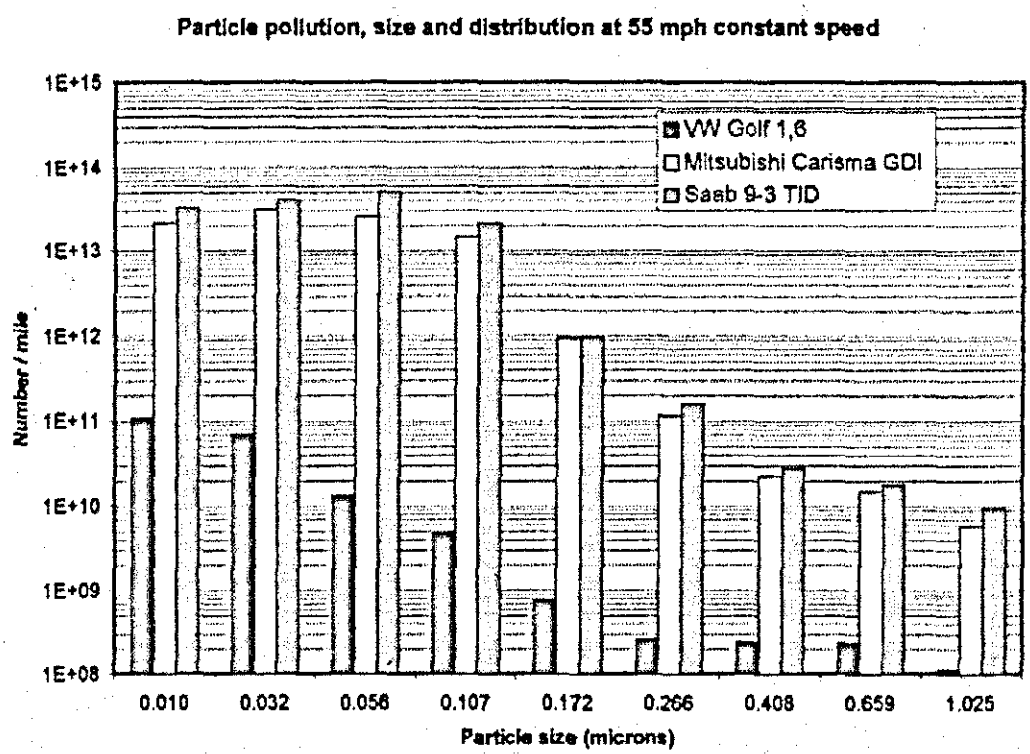



Figure 9: Examples of Particulate Emissions at Constant Speed
Source: Rototest Data. © Copyright 1998-1999 Rototest AB

## 5. FUEL ECONOMY

The differences outlined above with respect to exhaust emissions represent the same dichotomy that makes it necessary for manufacturers to display clear notice on new vehicles regarding the fuel consumption figures. Those displayed at the point of sale do not relate to the vehicle on which they are shown, nor can they be guaranteed since driving conditions and driver behaviour vary. This situation only helps the customer to compare one vehicle with another on the basis that one example of each has been carefully prepared and driven over a standard drive cycle on a dynamometer. The moment the driver launches himself off the forecourt onto the motorway, any suggestion that he is going to return the consumption figures on the label in the showroom, disappear faster than the precious fuel in the tank. The reason is again the low loads used in the drive cycle, which will favour cars with small engines, even in bigger cars. The smaller engines will be run at loads where it will have higher efficiencies. In real life, when the speed and loads are higher, too small an engine will lose in efficiency. This means that in some cases it will be nearly impossible to reach the advertised figures. Some of the, so called, 3-litre cars (3 l/100 km) are more or less tailor-made to the drive cycles, and stand a big risk of showing just small improvements in real driving compared to the same car with a bigger engine.

An interesting example is the comparison between the VW Lupo 1·0 and 1·4 and the Opel Corsa 1·0 Eco at different constant speeds is shown in figure 10.

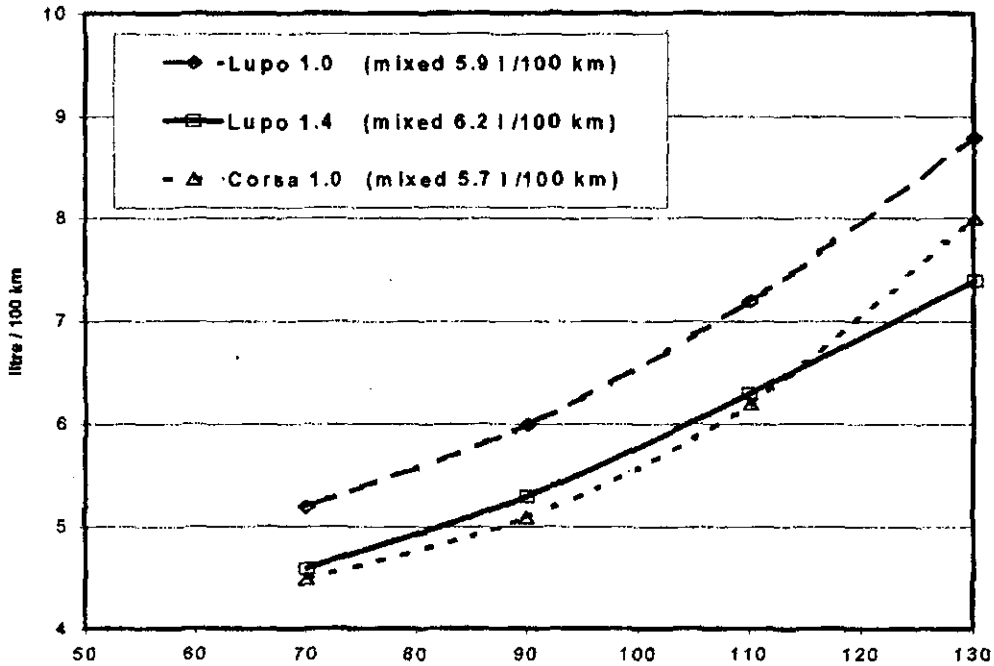



Figure 10: Fuel Consumption at Constant Speed (*mixed* are official figures)
Source: Rototest Data. © Copyright 1998-1999 Rototest AB

The 1·0-litre Lupo has a lower mixed fuel consumption compared to the 1·4-litre, according to the official figures. However, when the cars are driven at highway speeds, there is a huge difference in favour of the 1·4-litre. Now, it could be argued that the 1·0-litre engine might be of an old design and therefore not a fair comparison. If it is compared with the Opel Corsa 1·0 Eco instead, which is a modern engine, the performance is very close. This engine actually has a small benefit, being a 3-cylinder engine, since the internal friction should be slightly better.

Rototest AB has also been involved in measurements of, among other things, the instant fuel consumption on cars driven in real traffic. Some of the results from one of the cars are presented in figure 11.

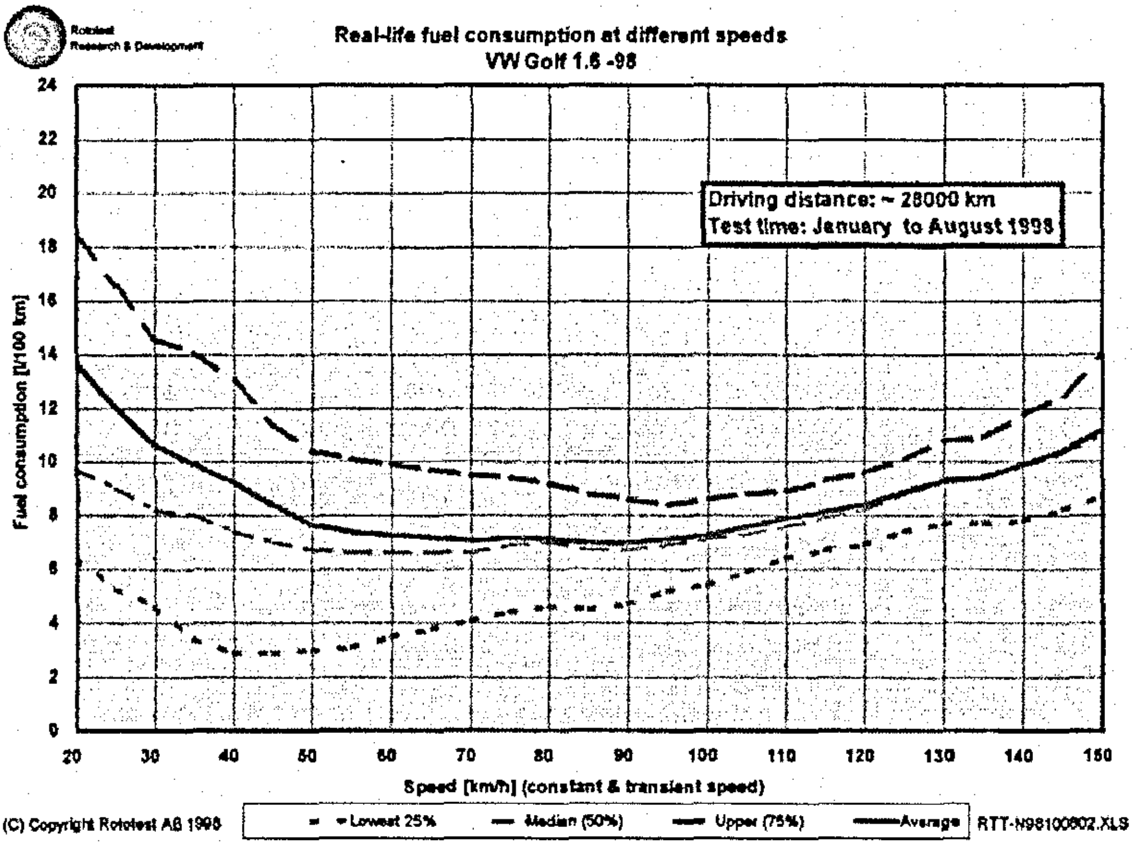



Figure 11: Real Life Fuel Consumption of a VW Golf 1·6
Source: Rototest Data. © Copyright 1998-1999 Rototest AB

The test illustrated in the figure was conducted over a period of 8 months with a total driving distance of 28000 km (17000 miles) using some 20 different drivers.

The different lines illustrate the upper quartile, median, lower quartile and overall average of the fuel consumption at different speeds.

The variation of the fuel consumption is due to different gear selections in the lower speed region and, of course, if the car is accelerating or decelerating. The median line shows, more or less, the actual fuel consumption at constant speed. One of the reasons the value rises at lower speeds is that the efficiency of the engine decreases. Another reason is the use of lower gears, which reduces the efficiency even more.

The official fuel consumption figures are 7·6 l/100 km (mixed) and 5·9 l/100 km (highway). For the reasons stated earlier, it is obvious that it is impossible to present one figure of the fuel consumption and state that this is the true figure; it will always be a comparison at some specific test. The figures would need more information to indicate how sensitive the car is to different types of driving, if a more meaningful representation were to be made to the customer.

The data was recorded using an in-house developed on-board data acquisition system.

## 6. THE COST OF EMISSIONS COMPLIANCE

A recently published FT report (1), which included benchmarking studies aimed at measuring best practice in environmental achievement, shows that there are wide differences in the methods applied and the corresponding costs of achievement for the same limits. Figure 12 shows the results of a benchmarking project by KGP that examined the cost of achieving Euro 2 emissions regulations model by model for 40 cars across the European spectrum. It is clear that relative costs vary by over fifty per cent.

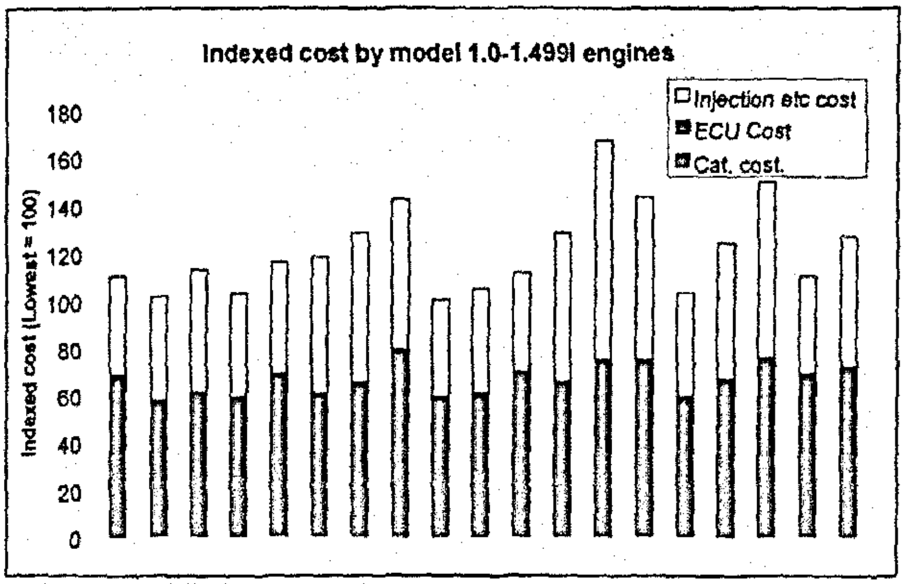



Figure 12: Index Cost of Emissions Equipment
Source: KGP Research

The wide variation in the achievement level of vehicles can be attributed to many factors including vehicle size, engine age, fuel and transmission type etc. However one of the largest factors influencing current attainment levels is the vehicle manufacturers' requirement to reduce cost, and development time. Whilst some individual tuning and calibration is carried out for each vehicle/engine combination, many parts of the emission control system are common to a number of models. KGP has investigated the cost of compliance, breaking emission control systems down into a number of parts, which shows that performance measured under the current test method, published by the Kraftfahrt-Bundesamt (KBA), can vary by over 100% for vehicles with the same basic equipment. Furthermore, the cost of compliance for constant levels of performance, can similarly vary by over 100%.

Figures 13 and 14 illustrate the exhaust emissions results quoted by KBA, against the evaluated cost of the control system. The index is calculated on the basis of the minimum cost system being valued at 100%.

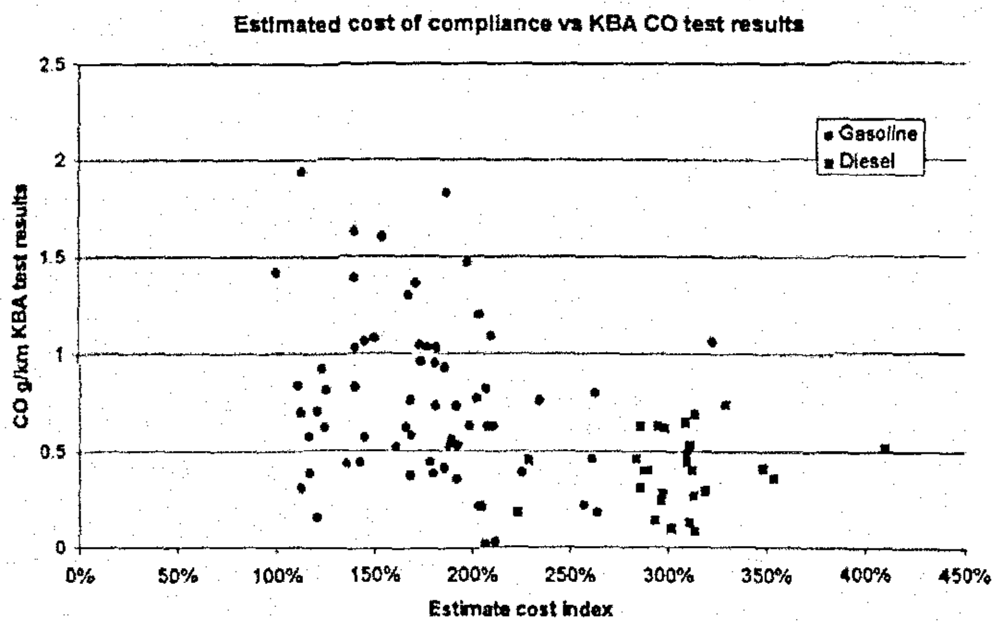



Figure 13: Cost of Compliance Plotted Against Quoted CO Emission
Source: KGP Analysis using KBA Data (2)

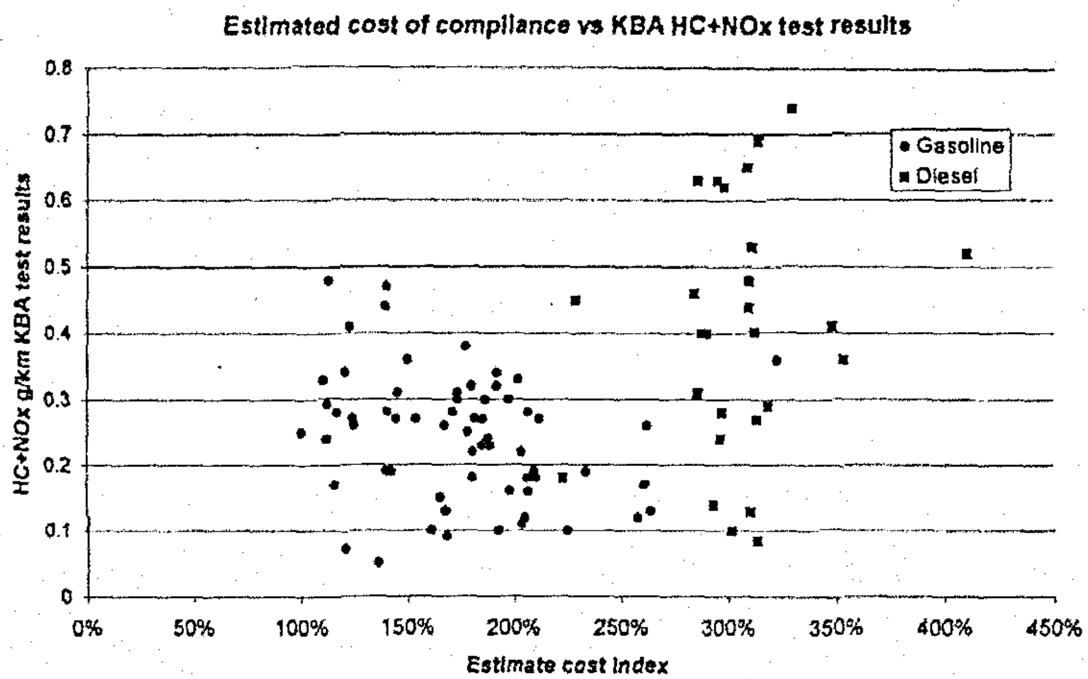



Figure 14: Cost of Compliance Plotted Against Quoted HC+NOx Emission
Source: KGP Analysis using KBA Data (2)

Optimisation of the emissions performance of each individual vehicle will require a realistic measurement of performance, and rationalisation of vehicle manufacturers' engine/transmission combinations. Figure 15 shows the results for seven vehicles split by fuel and transmissions type. This clearly shows how emissions capability is shown to vary very significantly, not only between models, but also between model variants. Difference in fuel, body type (and weight) and transmission type (from effect on engine load by the gear ratio), give rise to some of these variations. It can be seen that there are very big differences in the results obtained from vehicles tested against the Euro II and Euro D3 and D4 limits.

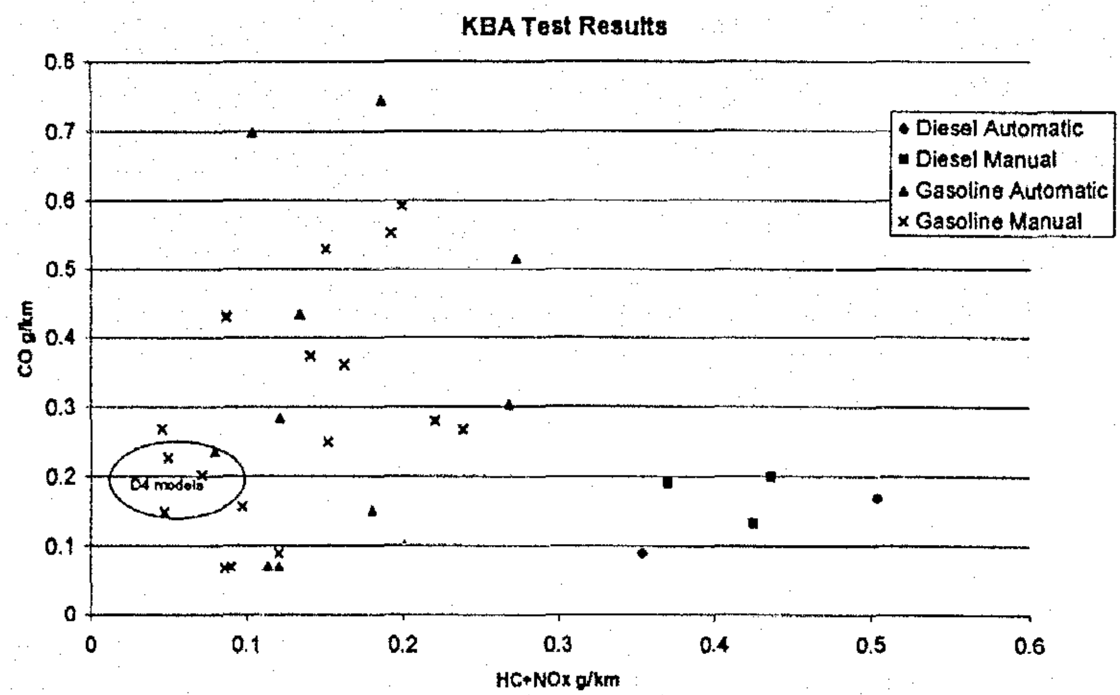



Figure 15: Emissions Performance of 7 Common Models with Differences Between Engine/Transmission Variants
Source: KGP Analysis using KBA Data (2)

On the subject of tax incentives, it has been shown that there is no clear correlation between vehicle performance and cost of compliance, largely due to lack of optimisation across the large number of models available. Similarly, current tax incentives are applied on a blanket basis, which does not reflect actual performance or cost of compliance. In effect, current legislation encourages practices that could be considered artificial, not only by the vehicle manufacturer but also the consumer and the legislators. Until there is a test that recognises a real life test, tax incentives will not be directed to the best environmental vehicles, only to those that meet a minimum standard on the test cycle. The European Commission is currently investigating an in-use cycle for vehicle testing that will monitor the way emissions performance is maintained in vehicles (4).

## 7. EFFECT OF TEMPERATURE

On the basis of the concern raised in the introduction, other work has been carried out and reported (5) from Finland to address the apparent discrepancy between the legislated emissions test conditions and real ambient temperatures in service. The study with in-use vehicles investigated the effects of temperature and the effect of total vehicle mileage of test vehicles. The findings were of some comfort regarding the effect of the temperature but the report found wider variation in performance by vehicle type that supports the findings of the present paper. The main conclusions of the Finnish work were:

- The overall in-use emissions performance was not unduly affected by the cold Finnish driving conditions but followed the average trend as expressed in the assigned deterioration factors in the regulations.
- The emissions of CO and HC recorded at low ambient temperature had almost no dependence on vehicle mileage, but depended much more on vehicle type
- The emissions of CO and HC in cold-start, low ambient temperature conditions were mainly a measure of direct engine-out emissions rather than a function of catalyst performance.
- The emissions of NOx were largely dependent on total vehicle mileage and, even in low ambient temperature conditions, closely following the trend in performance assessed at normal temperature.

It is worth noting that despite not being engineered for the US market, most of the vehicles tested passed the US limit of 6.2 g/km of CO for the emissions test carried out at –7° C, some achieving less than 50% of the limit value. This shows that improved emissions can be obtained by supplying the right amount of fuel for starting rather than excess, as used to be the practice. The benefit on fuel economy as well as emissions must be significant.

The increasing use of direct injection gasoline engines also has an impact on the warm-up rate of the engine on the new European drive cycle (NEDC). Work published by AVL (6) has shown that the fuelling strategy markedly affects the coolant and lubricant temperatures of an engine. A summary of the results of this aspect is shown in figure 16.

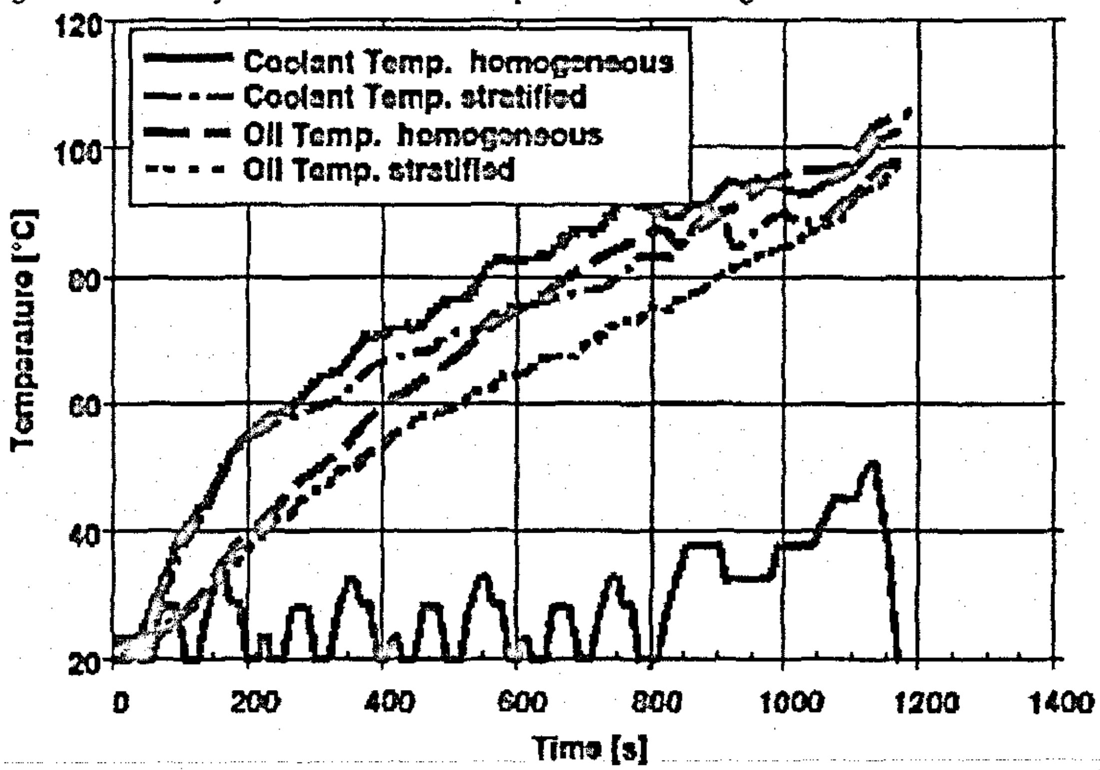



Figure 16: Comparison of Oil and Coolant Warm-up Characteristics on NEDC

This leads the authors to conclude that some cold-start emissions development and evaluation will need to be reviewed to evaluate the comparative warm-up rates of direct-injected engines on emissions cycles and in real-life driving. The importance of the cold and warm-up phase on tailpipe emissions (and hence air quality) is well known and although direct injection is set to give benefits in warmed up fuel consumption, the emissions trade-off in real life should not be overlooked.

## 8. CONCLUSIONS

8.1. Environmental rating schemes for cars are now published in Austria, Finland, Germany, Sweden, Switzerland and the UK.

8.2. For the development of proactive strategies in the automotive industry, it is now vital for the industry to be aware of developments in car rating schemes and examples have been presented in this paper.

8.3. Comparisons have been made between the rankings of vehicles according to their certification emissions results and those measured during higher load and speed operation, arguably more representative of in-use driving.

8.4. Increasing numbers of studies are being carried out to establish the real world emissions and fuel consumption of vehicles. Tests have been reported on cold temperature and warm-up studies, particulate emissions and on-road fuel consumption.

8.5. The relative cost of emission control systems has been evaluated and compared with the level of performance as measured by the legislated emissions results. The findings show wide variation in tailpipe performance 'value-for-money'.

8.6. Further work is being done on non-regulated emissions at normal and cold temperatures together with more investigations into particulate emissions of diesel, port-injected gasoline and direct-injected gasoline engines.

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**APPENDIX**

Calculation of Environmental Pollution Index (EPI) 

$$EPI_{98} = 40.23 \cdot wm_{NO} + 321.84 \cdot wm_{HC} + 0.47 \cdot wm_{CO} + 0.23 \cdot wm_{CO_2}$$

where:

| | | |
|---|---|---|
| $EPI_{98}$ | = | environmental pollution index –98; |
| $wm_i$ | = | weighted mass emissions for the pollutant i, in grams per minute; |

$$wm_i = m_i^{50} \cdot 25\% + m_i^{70} \cdot 24\% + m_i^{90} \cdot 17\% + m_i^{110} \cdot 13\%$$
$$+ m_i^{130} \cdot 6\% + w\overline{m}_i^{HL} \cdot 15\%$$

where:

| | | |
|---|---|---|
| $wm_i$ | = | weighted mass for pollutants i in grams per minute; |
| $m_i^x$ | = | mass for pollutants i in grams per minute at load case x; |
| $w\overline{m}_i^{HL}$ | = | weighted mean mass of pollutant i in grams per minute; |

## High load part

Four engine speeds are measured, at WOT and at 80 % of measured WOT power at each engine speed, i.e. eight load cases in total.

Engine speeds are chosen with the formula:

$$n_1 = n_{idle} \cdot 1.6; \quad n_1 \geq 1000 \text{ min}^{-1}$$
$$n_2 = n_{mpower}$$
$$n_3 = n_{mtorque}$$
$$n_4 = n_1 \cdot 40\% + n_2 \cdot 20\% + n_3 \cdot 40\%$$

*Engine speeds are rounded to nearest 100 min$^{-1}$*

where:

| | | |
|---|---|---|
| $n_i$ | = | engine speed point; |
| $n_{idle}$ | = | idle engine speed; |
| $n_{mpower}$ | = | engine speed for measured max power; |

$n_{mtorque}$ = engine speed for measured max torque;

## Weighting of high load emissions

$$w\overline{m}_i^{HL} = \overline{m}_i^{HL} \cdot \sqrt{\frac{t_{theor}}{c_1}}$$

where:

| | | |
|---|---|---|
| $t_{theor}$ | = | theoretical acceleration time; |
| $c_1$ | = | factor $1.55 \cdot 10^1$; |
| $\overline{m}_i^{HL}$ | = | mean mass of pollutant i in grams per minute; |
| $w\overline{m}_i^{HL}$ | = | weighted mean mass of pollutant i in grams per minute; |

$$t_{theor} = \frac{m_{vehicle}(v_2^2 - v_1^2)}{2(\overline{P}^{HL} - \overline{P}^{80-120})}$$

where:

| | | |
|---|---|---|
| $m_{vehicle}$ | = | vehicle mass (kerb weight increased by a mass of 150 kg); |
| $v_1$ | = | start speed – 80 km/h (≈ 22.2 m/s); |
| $v_2$ | = | end speed – 120 km/h (≈ 33.3 m/s); |
| $\overline{P}^{HL}$ | = | mean vehicle power; |
| $\overline{P}^{80-120}$ | = | mean running resistance power between 80 km/h and 120 km/h in a 2% slope; |
| $t_{theor}$ | = | theoretical acceleration time; |

# Paper 8

## "High Performance Engineering"

**Bale, C.J.C. (2001)**

Proceedings of the Institution of Mechanical Engineers

Automobile Division 2001 Chairman's Address
and nationwide lecture tour

*Nominated as the*
*IMechE-SAE Exchange Lecture for 2001-02*

I MECH E

# Automobile Division
# 2001 Chairman's Address

High Performance Engineer-ing

Institution of Mechanical Engineers
Headquarters, London

Thursday 4th October 2001

Eur Ing Chris J C Bale BTech, CEng, FIMechE, MSAE

The lecture was presented at an ordinary meeting of the Automobile Division

# High Performance Engineer-ing

By

Eur Ing Chris J.C. Bale
BTech CEng FIMechE MSAE

Prepared as the Chairman's Address for the Automobile Division of the Institution of Mechanical Engineers, 2001-2002.

First presented at an ordinary meeting of the Automobile Division held at 1 Birdcage Walk on 4 October 2001

## Abstract

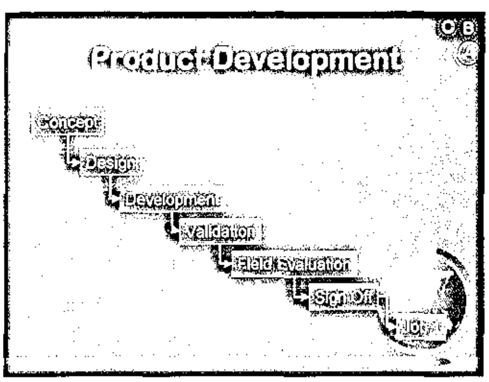
The paper draws parallels between the process of developing a highly competent automotive engineer with that of an engineering product. The factors that effect the concept, specification, training, development and proving of an engineer are discussed and the enablers and detractors are explored. The need and methods for continuing improvement and development are reviewed and some of the issues surrounding the management of knowledge are raised.

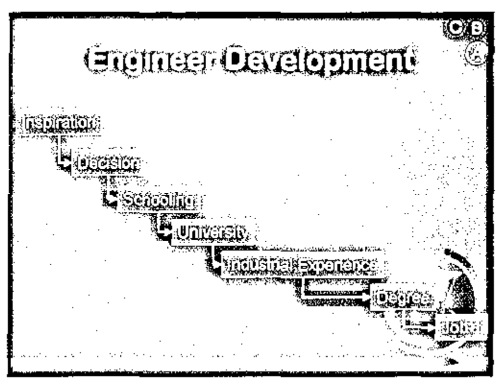
The paper is not about engineering education per se, nor does it seek to lay down a rigid template, but through exploring the process, the awareness of all those giving and receiving an automotive engineering training will be raised and the quality of engineers can be improved.

## Introduction

This is not an autobiography, nor is it another career history as many AD Chairman's addresses used to be. The content of this paper does draw on my personal experiences, however, as does every project and assignment I carry out as a consultant, and it would be a useless and idealised text if that were not so. Given that there are many routes to becoming a competent and capable chartered automotive engineer, the model cannot be definitive but the intention is to inspire and motivate all those considering being, or intending to be, a high-performing chartered engineer, and those from whom they learn on the route.

The pattern process I have chosen to adopt is a simple product development plan; one with which we can all associate and one which highlights the key stages of the process. The opportunities for simultaneous engineering, simulation and fast tracking are there just as they would be for a new vehicle or powertrain. I find likenesses in the major milestones and activities.




## Applying the Product Development Process

***Concept*** *– sowing the seeds of the new engineer*

***Design*** *– specifying what skills are needed for what discipline*

***Development*** – *Academic training and "formation"*

***FEU*** – *Into action*

***Validation*** – *The route to Charter*

***Job 1*** – *The Engineer is launched*

***Continuous Improvement Process*** – *CPD, etc*

***End of Use Recycling*** – *Feedback loops*

## Concept

### *Inspiration*

As with combustion, something has to light the fire regardless of how much fuel or kindling is present. The great inventors from history and the present play their part but it is incumbent on all to lead by example and promote the skill of engineering and correct awareness of what it is.

Patiently explaining how things work to children and adults alike adds to their knowledge and understanding of what engineering is and what it does for mankind. New telephone numbers and URL have updated it but I still carry the IMechE car sticker that rightly declares "Nothing Moves Without Mechanical Engineers".

### *Encouragement*

There are many pressures on young people, including peer pressure, computer entertainment, attractive careers in the City and financial circles and so on. Add to the this the popular misconceptions about engineers and the scant details that circulate about real engineering salaries and it is clear that any spark of enthusiasm for engineering has lots of mechanisms by which it can wither and die. I am not suggesting dishonest promotion of engineering careers, far from it, I would prescribe openness and honesty about the interesting and positive sides that rarely get a mention. Tell them that engineering means high-tech, travel, testing, investigation and problem solving, teamwork, etc.. The tools of modern automotive engineering are not a boiler suit and a set of spanners but a laptop and a passport.

### *National Curriculum*

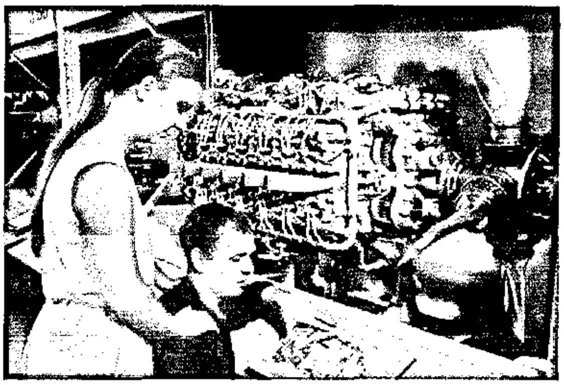
It is a sad indictment, as I write this in the year 2000, that the term "Engineering" cannot be found in the National Curriculum that describes the core knowledge that the future generation has to have. The basics are there, of course, the design, make and test projects that embody many principles of engineering are grouped under the term "Technology". The "appliance of science" is a term reserved for a washing machine commercial rather than a description of engineering. This is doubly sad, as any reference to that apt phrase may only prolong the misguided thoughts that engineers mend washing machines.

### *Teaching staff*

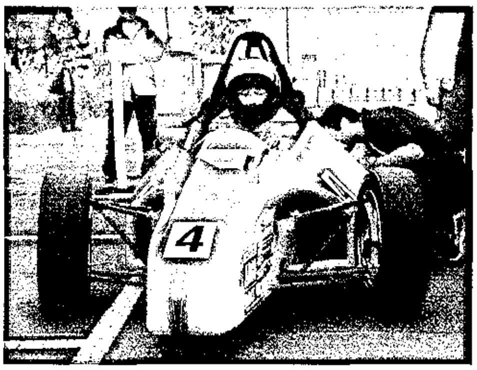
If the national curriculum does not yet contain an adequate description of engineering, let us all try and influence the state of awareness of the profession amongst teachers. In my youth, I participated in the Engineering Council's initiative "Opening Windows on Engineering". My role was to go into classes of 13-14 year olds and describe what I do in my job and hence inform about the role of an engineer. The Engineering Council coached participants into making these links and broadening ones own experiences into the greater world of engineering. I encourage all engineers to support similar activities today. Things will not get better by themselves and who else would you leave it to?

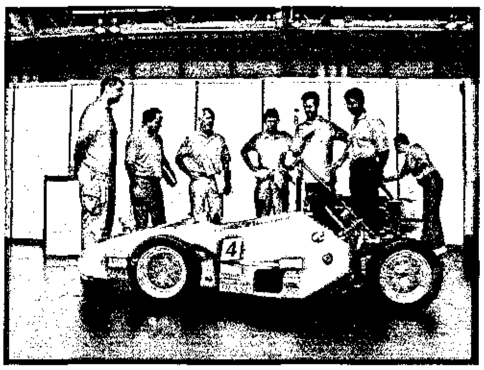
### *Other support*

IMechE and other Institutions have roles to play as well (if it can get through). There are many engineering related events that go on throughout the year; these need to be available to aspiring engineers to inform and encourage them. Evening lectures are part of this together with the organised initiatives for getting engineers into schools (Neighbourhood Engineers, Schools Liaison Officers, etc.). Even parental and visitor contact with schools and colleges can add to knowledge. I would also suggest that one should never forget that it is also important to give a potential candidate enough information to allow them to choose NOT to pursue such a career if it is not right for them.



The design and build car projects that exist are amongst the best means of promoting an active interest in automotive engineering. Remember what has been achieved by Formula Student [1] and the high calibre of young engineer that emerges from that program as a most desirable and employable person. Note too those potential students for entry to Universities now asking them if they participate in Formula Student. It has become a criterion for selection of university by the students. We need to translate that drive into schools.





## Design

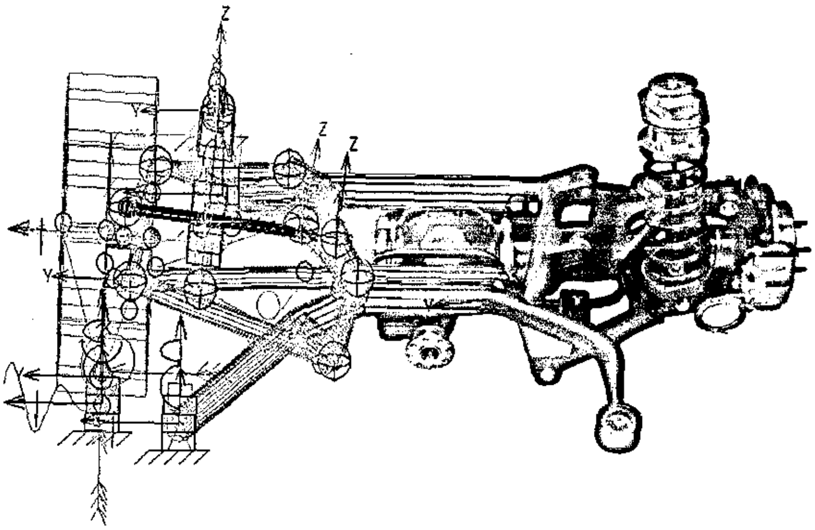
This is where academe, the institution and industry must work together. Each can blame the other for not delivering what is required but, as in the product development process, specification is everything. The scope of the function, the tolerances allowed and the quality of the base material must be laid out. Having said that, where there is room for manoeuvre or innovation, the specification must not be restrictive. For example, the output quality and quantity of a university course is far more important the precise number of A-level points that a candidate has at entry. We all have our bad days and, despite there not being a really effective alternative, school examinations do leave room for error in absolute terms. Even continuous assessment and coursework has its drawbacks as the supervisor or teacher has little way of knowing how much of the work is the student's own and how much has been "researched" (to be polite) or assisted.

Likewise, the maintenance of quality through universities is a headache in a system where the number of warm bodies on the course can dictate revenues. In such a regime, there is a strong financial incentive for a poor student to be propped up or allowed to continue when the output created is likely to be poor. Such a poor output is not good for the University, the employer of the graduate nor, I suggest, for the graduate himself or herself if they end up with an unsatisfactory role or in the wrong career.

Industry has to be in contact with the process in order to "write" the specification and keep it up to date with changing needs. There are clearly pockets of good practice, some of which are written about, (2) (3) (4) (5) (6) but theses need active support and participation from engineers in industry to succeed and must be expanded.

***New technologies***

Part of the value to industry of having new blood, in the form of graduate engineers, is the knowledge they bring from the research environment of their university and other sources of information gathering. In this way, tools and techniques can enter the company without being searched for, together with a breed of mind that is used to software tools, simulation and new technologies.



**Understanding the basics**

The university degree course must endow the engineer with sufficient knowledge to understand most first principles and the fundamentals of operation of things mechanical (or whatever the discipline is). The foundation must also be laid for the student to be able to grasp relevant concepts and to find out what he/she needs but does not know. The hardest message to get across is that "you don't know what you don't know" at that stage of life.

**Want to learn**

The learning course must be designed to keep inspiring the student to gather knowledge and must also provide the means or the linkage to making such learning possible. Extra curricula engineering such as motor club activities, car maintenance groups, visiting lecturers, factory visits, etc., are all important potential aids to developing knowledge and enthusiasm for the subject. Seeing theory put into practice always motivates one to pursue the theory with more commitment and interest.

## Development

The specialists in academe are experts in balancing the depth and breadth of the subjects to be covered by any course. It has to be recognised that, within a confined timescale and timetable, not everything can be done to full depth and across all subjects. What is important is giving the students sufficient grounding to master the basics and sufficient method to let them dig deeper where necessary. The "research skills" involving the use of libraries, publication and the Internet should be programmed in from the beginning and the students encouraged to learn rather than "being taught". This was always regarded as the cultural difference between school and university but in these days of tuition fees and the like, students still give the impression that they expect to be given all they need to pass the course with, sometimes, little or no effort on their part.

Another key element that needs to be sharpened in university training, is a sense of reality to balance the sense of wonder. The deep science and all the new innovation can seem all to attractive but must be placed in the context of real application cost and time constraints. It is the "art" of engineering to optimise a solution between conflicting constraints – this message has to be made clear to engineering students from the start and it is vital that they have this before any contact with the real world of manufacturing lest they lose all credibility on day one by offering theoretical solutions that can not be made economically or in a sensible time. Any periods of practical experience that can be incorporated into courses, will teach the student about these realities. The time it takes to carry out a laboratory test or computer simulation can be scaled up to indicate the resources and time needed to complete the engineering and manufacturing process. Practical activities will also support my earlier point regarding the reality of engineering parameters. Concepts such as repeatability and reproducibility can usefully be introduced at this stage through laboratory sessions and the like.

With these enhanced tools, a student can undertake collaborative work with or in industry. Better courses (in my opinion) make this activity mandatory in the form of a project, or a semester or year "out". Some more enterprising universities make this placement available in another country, which is another key building block for the global automotive industry into which the graduate will eventually emerge.

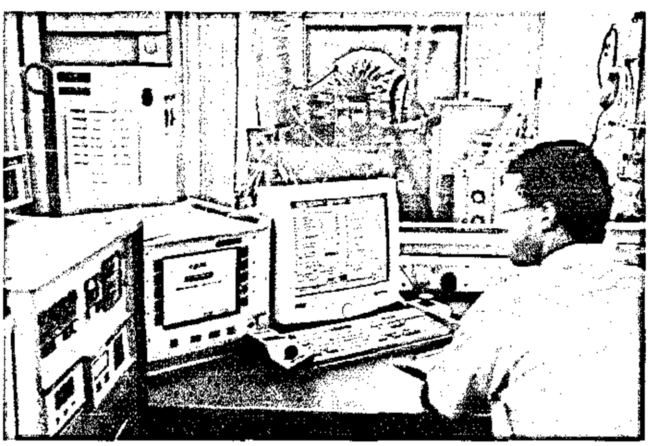
### *The basic degree*

With a degree in hand, an aspiring engineer has proven what? That he or she has a certain level of intelligence and application that can steer them through a degree course. It is often, and should always be, more than that but, in the limit, that is what you get. There is clearly room for a graduate in one subject to become proficient in another and the multidisciplinary nature of automotive engineering is a classic example. In modern engineering we need co-workers of at least the following types: mechanical, electronics, electrical, chemists, physicists, telecommunications experts, production engineers, computer programmers, software writers, polymer scientists, metallurgists, etc., etc. In delivering a product to the customer and making commercial margins for the company, who is to say which is most important. Andrew Ives, a former AD

Chairman, made this point very eloquently in his address on the electronic mechanical engineer.

***Industrial training and experience***

In my experience, topping up the inspiration for an engineer at regular intervals along the course, is a positive influence on the outcome. Maybe not in terms of the class of degree that is awarded (though this may be the case), but in providing a more rounded and competent individual for further roles, be that in industry, research or academe. Frequent reminders of the purpose of understanding the theory can make add relevance and motivation to the academic study. Conversely, and true for later life, a thorough understanding of theory can help to understand or explain some of the vagaries that one frequently finds in engineering, especially engines. The idea that engines have a mind of their own is not totally surprising giving the number of variables, the number of components and the speed at which things have to happen. In a computer simulation, feeding in the same data will (almost) invariably produce the same output no matter how many times you repeat the analysis. Sending the same injector pulse width (at the same time) and spark advance angle to an engine running against a fixed resistance at a fixed throttle opening, will not only produce a variable results in terms of output, fuel consumption and exhaust emissions over time (as recorded by data acquisition or chart recorder), but also in terms of rate and amount of heat released, work done, etc., on a cycle to cycle and cylinder to cylinder basis. Try explaining that to a novice student without seeming to endow the unit with a mind of its own.



***Learn from the voice of experience***

Every opportunity for knowledge transfer should be taken, with the aim of equipping the engineer with shortcuts to "learning the hard way". This has to be balanced with the need to develop one's own experience and profit from the experience of making a mistake to learn from it. Some input in the form of typical values for various things can come from experienced co-workers, giving the learning engineer a better feel for the accuracy or appropriateness of their own deductions before getting too far. This connection between what is expected and what happens is a very important to a young engineer. For example, when an engine test result is obtained, the engineer has to examine that the basic data is sound, the calculated values have been manipulated correctly and that the results is in line with expectations. If it is not then test conditions and the engine health should be checked.

***Try something new***

Unlike the USA, where first grade students are seemingly quite happy to take centre stage and tell the class who they are and, with some pride, where they were born and raised, young British-taught graduates are, for the most part, reluctant to present themselves or any technical topic. I have learned over many years that there is an art to delivering a speech or paper well and there is no harder one than the first. However, there are definitely a few basics that, if learned early, can make the whole process easier, less stressful and more successful as perceived by the recipients. This is a skill for life in engineering and business and well as a benefit to many human relations.

The same can be said for report writing, including the correct use of spelling, grammar and punctuation. Call me old fashioned if you like (and some modern day teaching seems to encourage children to get the words down no matter what) but I can distinguish between a well-written piece of work and a poor one; a well structured document is always more convincing than a poor one. Again the fundamentals are so simple and cardinal rules can be laid down from the first coursework assignment the undergraduate does (or could such lessons be embodied into A level studies?). Teach the basic structure of report writing and it is a lesson learned for life, no matter how complex the subject becomes, the basic requirements of such things as the abstract, discussion and well-reasoned conclusions, can never be learned too early.

Other key issues for engineers today are:

- Sustainability
- Ethics and values
- Incentives and costs
- Legislation and litigation
- Sourcing and vendor management
- Systems engineering
- Etc.

Somebody along the line has to identify the issues and expose the new engineer to them together with some guidance to work out an approach to dealing with these areas on an individual, corporate and "humankind" basis.

***Learn by your mistakes***

You all know the adages but they bear repeating:

*"I hear and I forget, I see and I remember, I do and I understand"*

*"The man that never made a mistake never made anything"*

These are certainly true and while not the only way, certainly help. It is a hard lesson for a supervisor or manager to learn that it sometimes pays to let someone make a mistake that you can easily see. Only by having a painful or embarrassing failure can difficult lessons sometimes be learned. The clever part of project management and mentoring is to be able to allow such indiscretions at times and places in the program where damage is limited and end dates are not compromised.

***Learn to take responsibility***

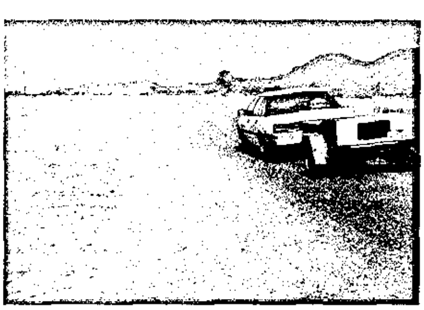
Increasingly, with project-based assignments and such wonderful initiatives as Formula Student, young engineers and engineering students are taking responsibility for their own actions and working as a team. This is all to be encouraged and it is no surprise that those graduates who have passed through Formula Student are regarded as being years ahead of their age group in terms of project management and engineering delivery within time and financial limits.

Even through the academic part of their formation, would-be engineers must become accountable for meeting deadlines and quality standards. If these basics are not programmed in at this stage, the realities of life will come as all the more of a culture shock. Thus, meeting hand-in deadlines is important and, although the academic system has to take account of genuine extenuating circumstances, the basic practice of "doing what you have to do, when you should do it" should become well established.

## FEU (Field Evaluation Unit)

***First real jobs***

Sending a young engineer out into industry is a crucial step in his development and this requires co-operation and planning on the part of the company, the academic institution and, dare I suggest, the professional Institutions who set the guidelines for the route to charter. This is the first chance the engineer has to really understand the pressures of budget control, the working week, maintenance, labour relations, and so on. All the fine ideas of optimum solutions using the latest tools and techniques can get blown away very quickly but this must be done in a way that does not destroy the creative spark and the thirst for improvement.



***Learning to work with superiors***

So far the young engineer has had to deal with authority in the form of parenting, teaching, instructors (of practical skills) and coaches (if sporting). Now they must learn to understand and deal with a management structure, departmental boundaries, working practices and the discipline that goes with the workplace. Ideally this is taught in a systematic way during the induction phase but it is worth reminding ourselves, those of us that fulfil this role, what we are achieving.

***Learning to work with subordinates***

The process outlined above works in the other direction too. Learning to instruct technicians and other working colleagues in an appropriate way that gets the job done and builds trust and respect is no mean undertaking. We should not forget that this too is a skill for life and somebody has to teach it. It is no surprise that sporting icons like Sally Gunnel, Will Carling and Sir Steve Redgrave can now command considerable fees for motivational talks and team building. Those who play sports have to learn the importance of interactions and team play. A former employer of mine once said that that his company was not a gentlemen's club where we all have to like each other and have a jolly time, it is a mix of skilled "players" who can produce great results technically and financially. He speculated that a certain, somewhat selfish, football midfielder may not be liked for his individual ball-hungry play but on the basis that he created an above average number of goals for the team, his presence was unquestionably valued.

***Learning to work with suppliers***

The suppliers are a key element in automotive engineering today. They have capability, tools, commercial interest and, in many cases, partnership or a joint commitment with vehicle manufactures (assemblers) for production of systems and vehicles. The suppliers in turn have tier two and three sub-system suppliers to manage and the new engineer may find himself anywhere in the chain of expertise that builds into a vehicle. The engineer may also find himself a place in, or working with, an engineering service supplier. All these relationships can work well at a technical level but they often fall apart in a commercial or administrative way. Learning to deal with this and being forearmed against the pitfalls is a must for the modern automotive industry engineer. This is being attacked in a collaborative program between the SMM&T and the DTI and needs to be expanded [7].

***Finding your place in the industry***

The first keen lunge into a company can put a young engineer in a strange situation. Some, I'm sorry to say, still make the tea and do the odd jobs as they trawl round a "graduate scheme" from department to department. Some companies' graduate schemes are outstanding and others are not. We should all be trying to positively influence the process in the companies where we have some sway. Getting a flavour of all the different roles that go to make up an engineering unit or a factory unit can unleash unfound potential in a young engineer, doing the odd jobs will not.

### *Final Calibration – knowledge management*

Knowledge is like a currency and has value in a similar way. It is an unfortunate fact that the present competitive culture in industry and the mobility of labour (both by choice and imposed by employers' streamlining processes) has tended to limit the cascading of knowledge. It used to be that the senior engineer would take the junior "under his wing" and teach them the ins and outs of the job, passing on knowledge and experience in the process. Unfortunately, if I told you everything I know and you told me nothing, you would know all you know and all I know so that when the industrial cycle calls for cuts, I am far more vulnerable than you. Such a culture works against the development of young engineers and I would petition companies and individuals to be aware of this. From the company point of view, although a young person may have gleaned knowledge from an older mentor, but real experience takes time to acquire and the substitution of the old with the young will not work all the time.

## Validation

### *Gaining experience*

There is no substitute for time but the environment in which a young engineer works can make a difference. More knowledge transfer is needed for the same reasons as I have already explained and despite the fact that knowledge is power, we must try and contribute since, only then, can one delegate some of one's work to the upcoming subordinate, with confidence.

Seeing some of their own ideas and projects come to a result, be it a success or a failure, validates the engineers' knowledge and updates it. This single act of "seeing the job through" can instil pride and motivation to a young engineer that would be unmatched by someone who just does their part and "throws it over the wall", never seeing the impact that their work has had on the final product. Bear in mind too, that failure management is something that has to be handled well with a young engineer. The emphasis has to be on lessons learnt and corrective action (short and long term if appropriate). Retribution, punishment, etc., should be meted out carefully.

### *Learning and remembering*

This is where the young engineer must take the initiative and make sufficient notes (mentally and physically) to be able to recall the benefit of experiences. This is best done in a supportive atmosphere with encouragement from those around who will either explain or tell the engineer where to look for more information.

### *Understanding*

Through knowledge and experience comes knowledge and it is also worthwhile involving the young engineer in using their knowledge as soon as possible. There is no better test of whether you understand something than to have to explain it to someone else. This could be in the form of a report or presentation or even informally to a technician or machine operator.

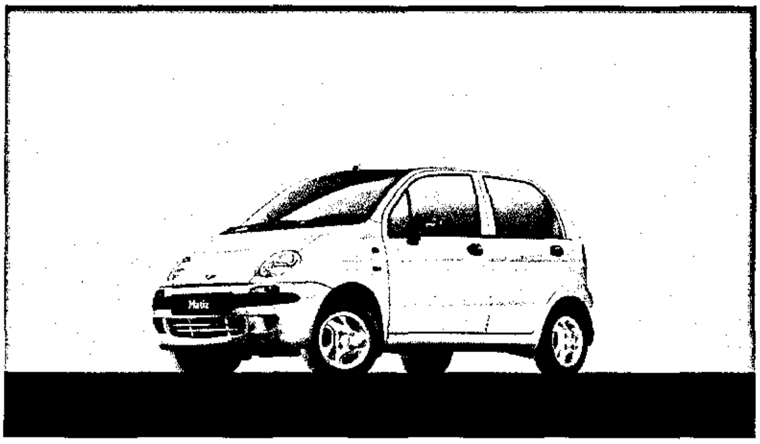
### *Team working*

"A problem shared is a problem halved". Another old adage that holds true. Mutual aid and mutual support can allow an environment for an engineer to jointly work out what is going on and what to do next. As above, just talking a problem through can often bring about a solution and when it comes to lateral thinking exercises such as some parts of Potential Failure Mode and Effect Analysis (PFMEA or FMEA as so many people call it) always work better with multiple participation in the way of brainstorming rather than isolated contributions.

It is important to remember that the team in automotive engineering, referred to above, now includes:

- Own department
- Other technical departments
- Internal customers and suppliers
- Vendors including full service systems suppliers and engineering outsource
- Hardware and software suppliers
- Shipping and administration
- Travel department
- IT and telecommunications
- Finance
- Management
- Program control including timing and planning
- Releasing
- Manufacturing
- Etc., etc.

Let it not be forgotten that some of these teams or their participants may be located anywhere in the world and cultural and language training can be most appropriate. Although English is the universal language of the automotive industry, better progress can often be made with overseas companies by at least conversing with them in their language and an appreciation of the relevant culture is almost mandatory if a successful working relationship is to be maintained. This is particularly vital in Eastern cultures (Near, Middle and Far) where the body language can say more than the spoken, especially where language ability is little or none, as is most often the case.



### *Taking responsibility*

The engineer has to learn progressively how to take right decisions and with the support of co-workers, subordinates and superiors, this can be done without disaster. Taking more and more responsibility comes with experience, more project responsibility and better knowledge. Going to your boss with a problem, accompanied by potential solutions and a recommended course of action, will soon result in respect and autonomy when the recommendations become the action of choice.

## Job 1 - CEng

### *Passed your driving test? Now learn to drive*

Another adage with parallels in engineering. Reaching the level of competence to be made a chartered engineer is not the final deliverable. In the same way, passing your driving test confirms that you have reached the level of MINIMUM competence to be let out on the road unsupervised. A Chartered Engineer has reached at least the minimum standard but must, and will, go on learning. By definition, getting this far has taken time. A Chartered Engineer has to recognise that they can now go on learning from their elders, their peers and from the younger engineers who work around them. Knowledge transfer at work again and the different strengths of different individuals makes well-formed project teams into effective groups and each individual can learn something from the others.

Job 1 means that the first full production vehicle (or engine) has been built. This is cause for some celebration but remember that it also leaves space for job 2 to come down the line and Job 1 itself goes on to serve its purpose. Life goes on and so does the process of engineering evolution.

### *Take the Advanced test*

Taking the driving analogy one step further should encourage the engineer to look for career and knowledge progression that can be recognised within their organisation or work outlets. Within the circles of professional engineering, this is continuous professional development (CPD) a commitment made by the achievement of Fellow grade in their chosen institution. Again no cause to stop progressing and learning but a recognition of having reached a higher level of competence and experience. An advanced driver has a badge on the outside of the vehicle but the award recognises the state of mind, incorporating interest, attitude, awareness and the constant will to improve oneself and to accommodate the less skilful. Fellows should keep that in mind too as they move into the next phase and all Institution members must monitor and improve their skills. We expect our medical doctors to read their journals and papers to keep up with new diseases and treatments; can we not reasonably expect every engineer to do the same?

## Continuous improvement

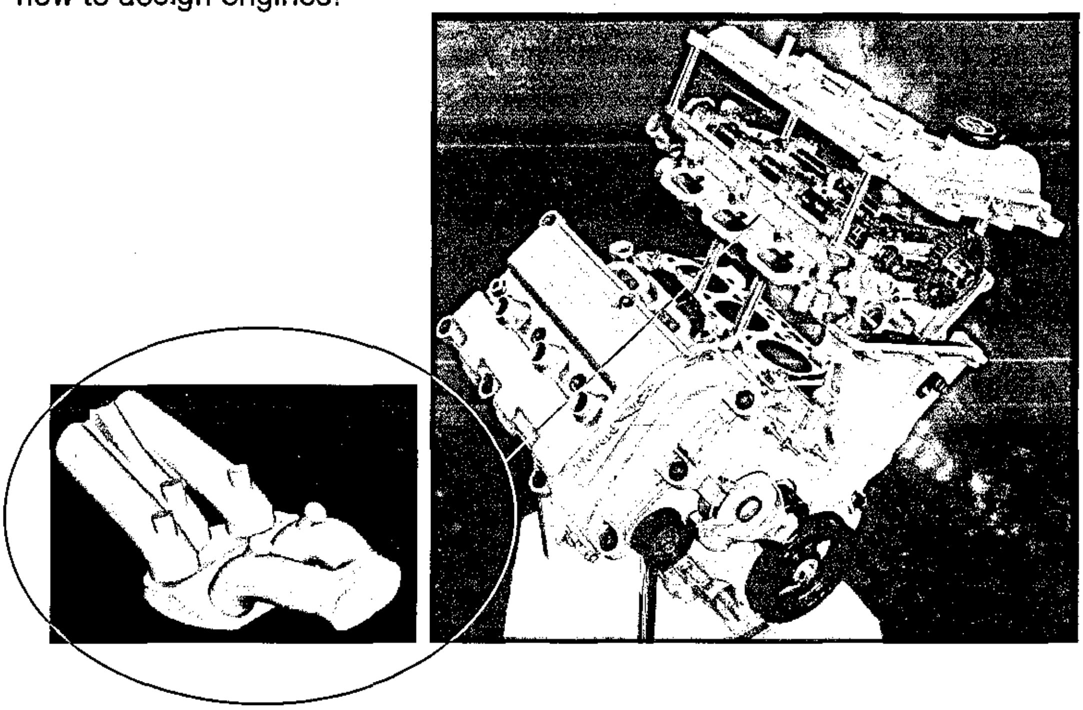
Just as a product has a life cycle and gets refreshed or facelifted, gets endowed with more features, etc., as it goes through time, so the engineer can and must add new skills and knowledge. This can be at the instigation of the company or the individual and I do not subscribe to the saying that you cannot teach an old dog new tricks. You can if he is prepared to learn and you are prepared to teach in a way that recognises that the new subject may not be instinctive to a more mature engineer.

***On the job training***

Learning a new skill or fact can be very informal as well as the result of "going on a course". These less structured sources of knowledge should also be recorded, as they will become part of the sum of Continued Professional Development of the engineer.

***New qualifications and skills (IT, languages, etc.)***

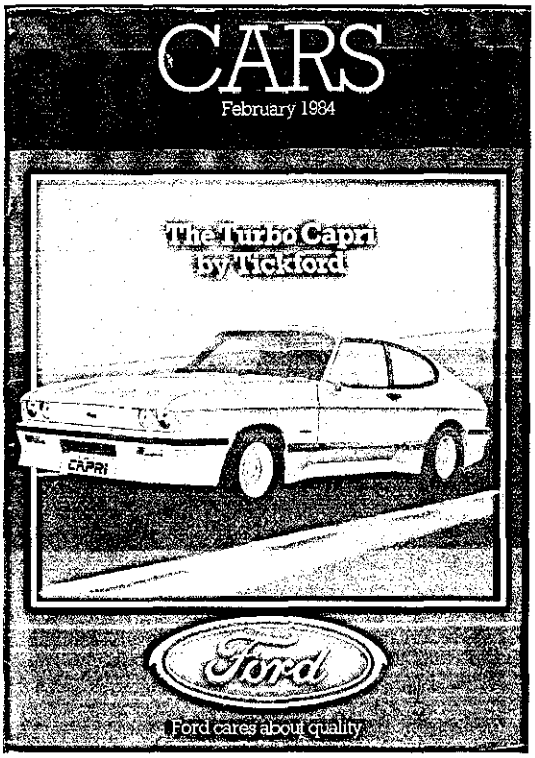
More formal training can be provided through one's employer or from external courses. These relate to new competencies or improved skills in existing competencies. When you teach old dogs new tricks and you get a more capable dog as a result. In my era it was a question of teaching CAD to engine designers and, hard though it was to use different techniques at first, it produced a far better and faster result tan trying to teach a keyboard operator how to design engines!



***New disciplines (commercial and management)***

Many engineers move onto the project and people management ladder as a means of career progression. In some companies, this is the only way to develop a career in terms of job grading. In all the jobs that I have had where I have had influence over the hierarchy, I have advocated technical specialist and project management routes as equally valuable, equally rewarded, options to progression for a good engineer.
Such new skills also need to be learned and a combination of formal education, peer group working and experience usually combine to achieve the

best end result. For example, in an engineering product environment, engineer input is essential to service and sales information.





***Learning through experience***

With a receptive attitude and commitment to improvement, the closed loop of experience continues to develop the engineer. New tasks, maybe those never handled before, stretch the ability beyond the existing envelope (comfort zone) and, as before, failures and problems become learning points and those around in supervisory or other roles should be there to minimise impact on the business and the customers (internal and external) involved.

As an independent consultant, I have learned to operate on the edge of my comfort zone and to make learning of new skills and technologies just a routine part of my year. I have had the pleasure of passing on some of that knowledge, both existing and new, to postgraduate level engineers both in the UK and abroad. It is disappointing sometimes to see how much more the engineers in other countries are keen to learn new skills. Having a mentor, guide or company ethos behind you makes a lot of difference. Management please note!

## Recycling

Rather than "scrapping" an engineer late in his career, it is important to use the strengths and skills to improve the next generation. The old engineer can be taken apart (metaphorically) and the useful parts used to put into the new models.

### *Inspiration*

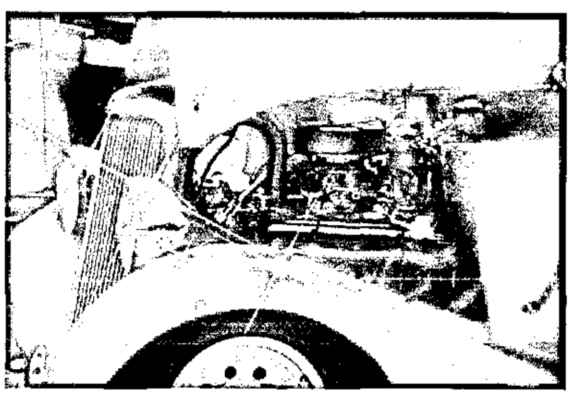
Put the experienced engineer with the younger one to demonstrate what can be done, what can be achieved and what has been achieved.

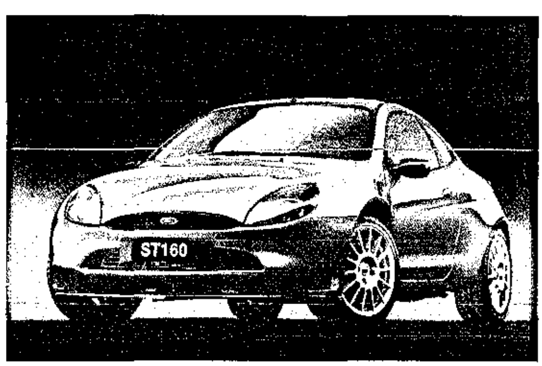
### *Training and Education*

Why not use the experienced engineer as the mentor to lead formalised training for younger engineers? There is a potential downside to be taken into account, that of lack of respect where age differences are too large and the ideas portrayed may be seen as being old-fashioned. This can be managed.

### *More knowledge management*

The twilight of an engineer's career is a time when they are less likely to feel threatened from the potential dangers of giving away their knowledge. The downside that I introduced earlier of an engineer being disadvantaged in job-security if one shares his or her knowledge, holds little or no fear for someone approaching retirement anyway. What better combination of having a willing and unreserved teacher to coach the up and coming generation? I personally think that the employer who exploits the power of knowledge to the disadvantage of his senior engineers will soon find the younger generation moving on earlier than they otherwise would. Just as loyalty breeds loyalty, disloyalty breeds disloyalty, or am I being a little old fashioned?






Puma ST160 Concept

## Conclusions

The objective of this paper was to inspire and remind all involved in the development of the next generation automobile engineers. It was not intended to lay down a particular route or blueprint, but rather to highlight some of the ways in which better, higher performing, engineers can be brought into the profession, kept in it and made better while they are in it.

A conscious effort is required on the part of mentors and managers to facilitate the most effective outcome, in the form of a high performance engineer, and the basic steps of a product development process are applicable reminders of the stages that are passed.

I make no specific recommendations other than to increase awareness in mentors and the learners alike, and to appeal to business managers to provide an environment that allows the process to happen.

## Acknowledgements

This paper is dedicated to some individuals to whom I owe some of my success in life. Amongst them are lifelong friends, one who has passed on knowing only part of the story and one has gone without ever knowing what he did for the world of engineering.

I admire the perception of my first form teacher at Corby Grammar School, the late Charlie Stevens, who was an ex Merchant Marine Chief engineer who later turned to teaching of metalwork. As a fast-tracked primary school pupil who took his 11+ at 10 and scraped into the E stream, he wrote in my first subject report for metalwork: "Christopher demonstrates a flair for this subject and in the absence of other aspirations, I commend a career in engineering".

More subtle technical inspirations came from various members of my family: a chauffeur-turned-garage-mechanic grandfather, a motorcycle-racing uncle, a GT-loving aircrew Squadron Leader for a godfather and a father who started his working life as a RAF wireless technician.

For longer lasting inspiration, I will always be grateful to David Morgan, who was Chief Development Engineer at Aston Martin when I first worked for him as an undergraduate trainee. I worked for David again at AML as a development engineer four years later and continued through my 16 years at Tickford. David's commitment, engine experience and instinct, and sheer hard work made up for any early stop in academic qualifications. He is responsible for growing much of my practical knowledge and working ethos.

Others who deserve special mention are Albert Tingey of Lucas, who developed an automotive mechanical engineer into an electronically-literate engine management calibrator with a genuine passion for emission control, and Steve Coughlin who guided my transition into program management and allowed me the scope to learn from my mistakes.

I hope each can be proud of their contribution to what I have achieved so far and the way I have done it. I seek to inspire others in the same way.

For assistance in the preparation of this paper my grateful thanks also go to Prodrive, for access to the Prodrive Tickford image library, and to the University of Hertfordshire, for use of pictures of its 2001 Formula Student car.

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# Paper 9

## "Latest Trends in Industrial Skills Development Techniques"

**Ashley, C., Bale, C.J.C., Millan, N., Williams, T.M.**
**and Hendley, R.J. (2006)**

Proceedings of the Institute of Cast Metal Engineers

World Foundry Congress

Harrogate, UK, 4-7 June 2006

Paper 197

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# Latest Trends in Industrial Skills Development Techniques

Dr. C. Ashley*, Eur Ing C.J.C. Bale*, N. Millan*, T.M. Williams * and R.J. Hendley**.

*AutoTrain LLP, UK, ** School of Computer Science, University of Birmingham, UK.

***Abstract***
This paper describes the application of Internet-based technologies to improve skills within manufacturing industries. Employees need continued training and e-learning offers a more flexible method to complement and replace conventional training. The methodologies and advantages of e-learning are described, with its freedom from time, travel and accessibility constraints, as developed by one of the world leaders in e-learning techniques. Supported by the EC Leonardo da Vinci programme, a multidisciplined engineering and software team has developed innovative ways to deliver vocational education and training to companies and individuals via Managed Learning Environments. Interactive learning material and multilingual video lectures are included in the presentation and the written paper, with references from the international companies and organisations that contribute to, and make use of, this modern approach to knowledge management and transfer.

***Key words***
Training, e-learning, lean, manufacturing, quality

***Introduction***

Automotive, engineering and manufacturing industries have had to come to terms with increasing global competition and price pressures that make it possible for only the most skilled companies to remain competitive. Under such circumstances, training time and budgets can be difficult to obtain, especially for small companies. The growth, or even survival, of businesses in the automotive, manufacturing and engineering sectors is threatened by a lack of skills that is perceived as one of the main obstacles for generating a competitive offering against global competition. The AutoTrain and AutoTrain-Europe projects, run jointly through the School of Computer Science and EuroMotor at the University of Birmingham (UK), have developed a friendly system of online training courses, primarily for the benefit of small and medium size companies (SMEs) within the manufacturing, engineering and automotive sectors. The resulting Internet-based technologies are used to generate online learning systems that learners can use to improve skills and to implement company training policies in their required areas in a customised way, at a time to suit the individual, at their own pace and at minimum cost for the business.

***Online training***

Online training is a type of training system where learners "attend" classes that are delivered through the Internet. EuroMotor (www.euromotor.org) was formed in the nineties to improve the knowledge base of the European motor industry through high-level technical collaborative training, as a network and partnership of European universities, car manufacturers and suppliers, in order to compensate for the increased difficulty in gathering attendees to conference and training events. In the last two years the AutoTrain-Europe programme has acted as a virtual college to bring online training to the European automotive industry and AutoTrain LLP has been established to create Managed Learning Environments (MLE) for individual European institutions, both academic and commercial whilst delivering further value from over one hundred courses available via the Internet, which were created by previous EU-funded projects.

To illustrate the practicalities of online learning, one of the recent projects can be examined in more detail. The AutoTrain-Europe project was set up with its own website at www.autotrain-europe.com (Figure 1) that is used as a portal for the delivery of its online training resources. The website was designed to be both functional and easy to access, even by novice users. The entire system is interactive, user-friendly and easy to navigate even for inexperienced web-users. Furthermore, when it comes to functionality, the design solution chosen by AutoTrain has generated a system that is able to work both effectively and efficiently through a web-browser, going to the project's main page and clicking on the *"Training Centre"* link to gain access to all the resources available.

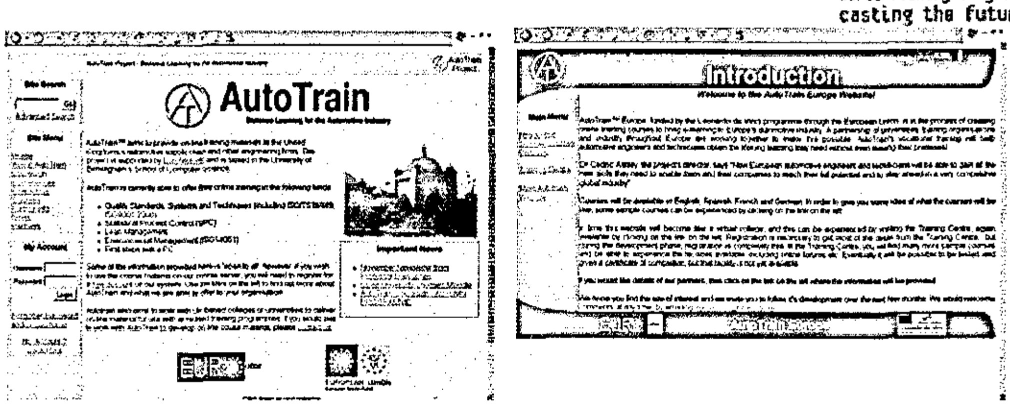



*Figure 1. Home pages of the AutoTrain and AutoTrain-Europe Websites*

In order to be able to access any of the resources available from the online learning system, users are required to register for an account in the system. Under EU-funded projects, the registration is available at no cost to the end user and it is a quick and simple three-step process that involves the input of some personal and company details in return for a username and a password, which entitle the user to unlimited access to all the online resources available from the site. Outside of EU funding, the access is either provided by the company or institution that hosts the MLE or by subscription payable through the online system.

After initial development work using WebCT, the current training materials are delivered through an "Open Source" MLE named *Moodle*. *Moodle* is a user-friendly virtual learning environment that facilitates the delivery and presentation of courses in an easy-to-browse way and allows educators to create quality online teaching materials. *The Moodle* platform is also used by learners to access courses, to keep track of their progress, to assess their knowledge and to interact with their tutors or fellow learners through the various forums. Key advantages of the *Moodle* MLE are its low cost and the ability of the AutoTrain developers to configure the system to suit different needs.

In its AutoTrain-Europe format, a list with all the materials is available from the "*Training Centre"* link on the home page. Clicking on the titles of any of the training courses gives the user a description of the course contents to enable them to decide whether they want to add a particular course to their account. Once the user has registered for an account, he can add or remove as many resources as he needs, as frequently as he needs, with no time restriction or expiry date. The "My Courses" function enables the learner to access his chosen courses more quickly while leaving all the other available material in the background in a way that does not clutter his learning environment with irrelevant material.

The online training courses produced by AutoTrain (www.autotrain.org) (Figure 1) presently come in two main formats: **e-books** and **video presentations**. E-books are text-based courses that present several levels of interaction. Video presentations are video lectures easily identifiable from the general list of courses by a special icon. Both types of resources are easy to access just by clicking on the title and adding them to each user's account. Nevertheless, in order to be able to watch a video presentation displayed on the computer, the user will need to have additional software such as *RealPlayer* ™ or *QuickTime*™ installed on their machine. For those users who do not have any of these plug-ins installed on their computers, AutoTrain specifies links to the sites where the programs can be legally downloaded at no cost.

***e-books***

AutoTrain e-books are standard text-based courses that have been digitised (Figures 2 and 3). They are delivered through the Internet so that individuals or companies that require training in a particular area can have the training resources that they need available, on demand, at their convenience. E-books are easy to access and easy to navigate and offer different levels of interaction for the users based on a choice of quizzes and exercises that users can access while they are learning, in order to help them monitor their own progress.

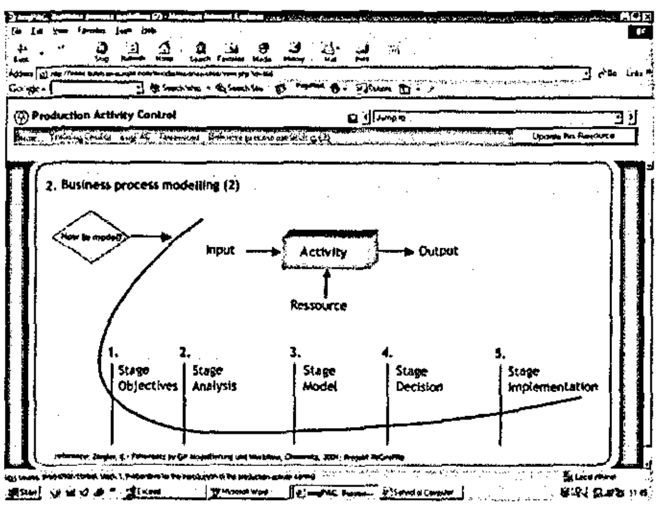



*Figure 2. Example of e-book course on Production Activity Control developed by ATB.*

Amongst the advantages of the *Moodle* MLE is the way in which the resources are organised, most of which are structured around a single course page. That page can also be used by teachers to add and edit additional activities and features. Learners and teachers can adopt the system very quickly, and, in terms of maintenance and development, there is an international network of developers and users supporting it.

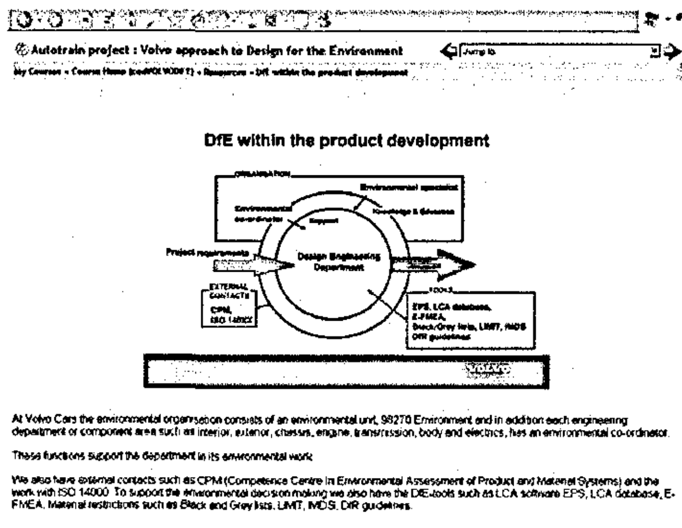
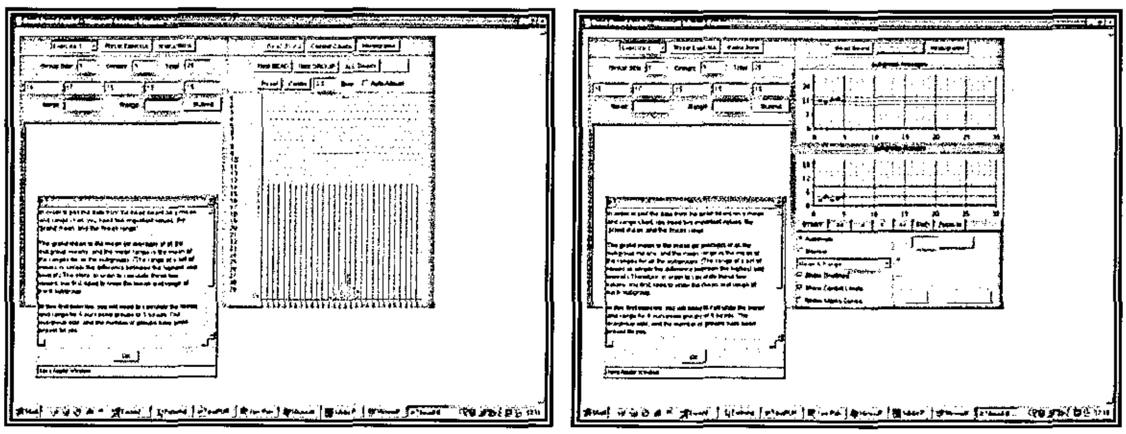



*Figure 3. Example of e-book course Design for the Environment by Volvo Cars.*

AutoTrain courses delivered through *Moodle* have been provided with additional functionality to encourage learning in a collaborative environment. The main page of each course is organised in sections to give the user access to a range of learning tools. For instance, under the *Activities* section, the user can find access to resources such as glossaries, quizzes or course fora just by clicking on the relevant link. The *Search* function allows running searches on the fora or the course contents, to look for a particular topic or piece of information. The *Administration* link lets the user monitor their grades in the various assessments. The link labelled *My Courses* can be used to help keep track of all the courses that the user is registered for. *Course Tools* includes links to news pages and a "*resume*" function that the learner can use to bookmark the last visited page and start future sessions at the point where he finished the last time he was using a course.

The duration of e-book courses varies depending on the nature of the training. However, a learner may not be required to complete an entire course, since the information that he may need to acquire may be included in just one module. To help customise the information that learners need, courses are divided into modules. Each module contains smaller sections and, ultimately, each of these sections is divided into pages. The purpose of this fragmentation is to help maximise the learner's time and to give them the flexibility and the capacity to choose the specific units of knowledge that are relevant for what they want to learn, helping them in this way to discriminate all that information that may be redundant or unnecessary. To assist the learner with the navigation through the different pages and modules that make up a course, AutoTrain has incorporated a system of scroll bars, navigation arrows and drop-down boxes that are available to use to move quickly to a particular part of the course.

Most courses are interactive (Figures 4 and 5) and some modules include visual aids and demonstrations in order to illustrate the course content and enable the learners to carry out investigations and exercises to help them reinforce the knowledge that they will acquire throughout the module. A good example of this is the animated "Bead Board" which helps to explain the concept of variation and the factors that influence it, whilst learning the subject of Statistical Process Control and the various types of charts that are used (e.g. histograms and control charts). This is shown in figures 4 and 5. Progress can be monitored via the self-assessment quizzes.



*Figures 4 and 5. Examples of interactive applications in 'The Pursuit of Excellence' course.*

### *Video Lectures*

The other main type of training material developed by AutoTrain is the video lecture, also known as a video presentation or, informally "talking heads". Video lectures use cutting edge video-streaming technology generated by AutoTrain, to bring the user the expertise of leading figures in the engineering and manufacturing industries in the shape of movies that they can watch from their computers (Figures 6 and 7). From the website, video courses are easily identifiable by a dark icon or by the *RealPlayer*™ logo. The user can choose the video lecture of his interest and add it to his account following the same procedure for the standard e-book courses.

Video lectures are available in different video player formats in terms of the plug-ins that the user needs to have installed in his computer in order to be able to see them. AutoTrain has developed video lectures in two formats. The standard format uses RealPlayer and a combined format that uses other programs like QuickTime or Windows Media Player to display the presentations. Relevant information on how to download and use these tools is available from the AutoTrain website. All the courses are created in a form that will work with different Internet connections. In the worst-case, this is a 56 kB/sec modem but, with systems such as Real Player, it is possible for the learner's computer to recognise the connection speed and use the best available definition for the video format. The software is also capable of pre-loading the slides so that they are readily synchronised with the presentation and are not dependent upon download time during the training session.

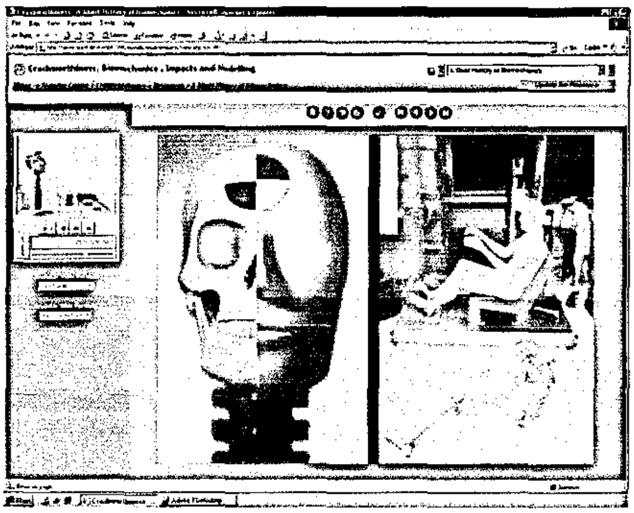


*Figure 6. Detail of Video Lecture on 'Crashworthiness'.*

The look-and-feel of the video presentations remains uniform regardless of the player format that may be chosen to access them. When the user opens a link to a video presentation, the screen splits into two main frames: the frame on the left side of the screen contains a window to display the movie and a number of control buttons that add functionality to rewind, pause, fast-forward or stop the movie at a particular time. The rest of the screen is dominated by a bigger frame that contains slides to reinforce the contents of the topics that are being explained throughout each of the presentations. Users are given the choice to watch the video presentation and slides in a synchronised or asynchronous way. They can also watch a particular slide or a particular fragment of the presentation. In some cases, some of the presentations include subtitles to highlight the main concepts that are being explained in order to make them easier to follow. Additional functionality includes a *help* function that contains additional information on how to access the video presentations and some troubleshooting information, a *slide menu* that gives an overview of the slides used for a particular presentation, a *print* function that can be used to print a paper copy of all the slides in one presentation, *navigation arrows* to move straight to the beginning or the end of a presentation or to the previous or next slides. There are no log-out buttons when a user is accessing a video lecture. In order to end up a session after watching a video presentation, the user simply has to close the browser, or alternatively, if he wishes to continue using the site, a *home* button is provided that he can use to go back to his course index page.

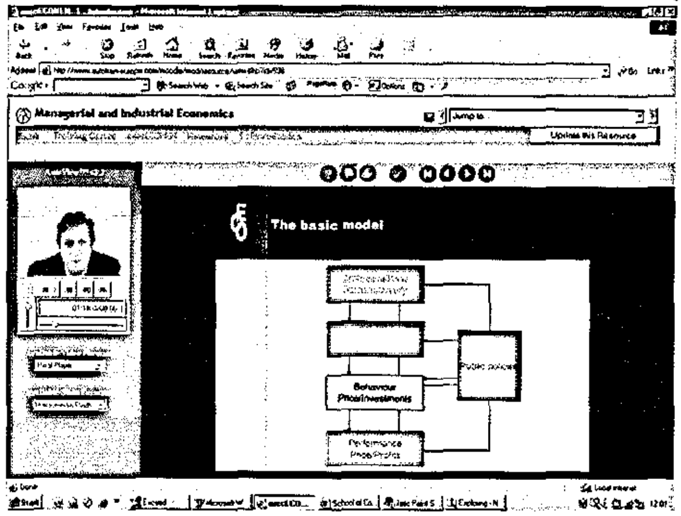



*Figure 7. Detail of Video Lecture on' Managerial and Industrial Economics'.*

The most recent development to the AutoTrain video lecture is the incorporation of selectable multilingual subtitles (Figure 8). Subtitles were first introduced in order to make training material more available to the aurally disadvantaged with switchable (on-off) subtitles running at the foot of the screen and the functionality to allow printing of the script. However, in the light of the international audience for Internet training, additional functionality was added in 2005 in order to incorporate subtitles in different languages so that lectures delivered by experts who were not of the same nationality as the learner, could be used. In the context of the automotive industry, English is widely spoken but since this is not the native tongue of many, to have subtitles in ones own language can aid effective acquisition of knowledge from the lecture whilst also improving language skills. The subtitles can be switched between languages (presently English, French, German and Spanish) without losing synchronisation and there is no technical reason why many more different languages cannot be available.

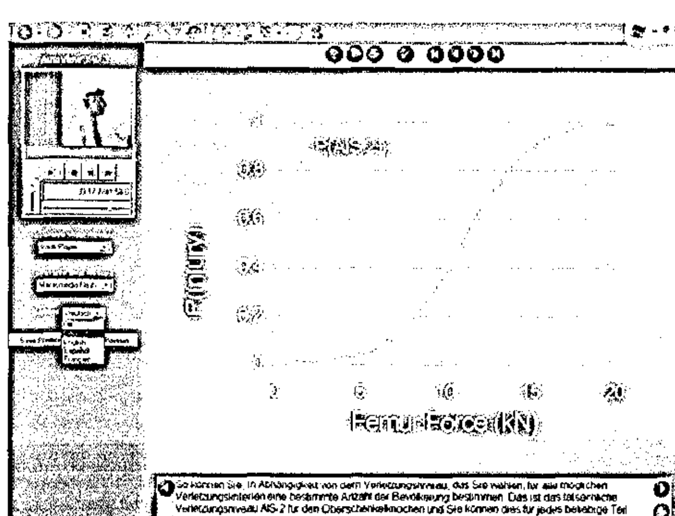



*Figure 8. Example of Video Lecture with multilingual subtitles (choice in left screen pane).*

***Advantages and disadvantages of online learning technologies***

Using distance learning technologies for the delivery of training materials has its advantages and disadvantages for both individuals and corporations [1]. The major advantages are that online learning enables the learner to develop a study environment suitable to his needs, online learning allows the user to study at his own pace, at any time and from any location where there is a computer that has access to the Internet.

Organisations can also benefit from the use of online training systems to implement in-house training policies and programmes at very low cost [2]. In its EU programmes AutoTrain delivered online training materials for SMEs completely free of charge, which is extremely beneficial for small companies that otherwise would not be able to afford to send part of their workforce away on training, both in terms of cost and time. Registration fees for conventional training courses tend to be very expensive and often unaffordable for small businesses. More importantly, in most cases as far as time is concerned, training is often beyond what a small company can afford as they may have to close down elements of their operations, even a production line, for the duration of the training course while their key staff are away.

Online training brings the training to the user's desk at low cost. Managing Directors can improve their own skills and those of their of their workforce, as well as the performance of the company, by encouraging online training at times when staff can be available. Staff interested in training may also find it useful that they can start a particular course while they are at work and continue or go back to it from their computers at home, while they are away or while they are travelling to work. Besides, by using online training systems like those developed by AutoTrain, companies can benefit from the flexibility of having customised training programs to meet their specific needs.

A further key advantage to companies and institutions, is the possibility of enabling a low-cost, dedicated learning environment to deliver a structured training programme that is individual to that institution. This can be delivered via the Internet or, more commonly, via the institution's own intranet. Such systems use the company's own training material and can feed user information and assessments directly into the company or academic evaluation area. The online systems can not mitigate the time cost of the employee or individual but it can at least reduce travel time and use the medium to arrange training time around the workplace and home life priorities rather than the other way round.

One of the main disadvantages of online learning is that users may find the discipline of self-learning is difficult. Without a "teacher", answers to their questions may be difficult to obtain. Studying online requires a great deal of discipline and self-control in order to be effective and one of the key courses from AutoTrain is a course to teach some effective techniques to study online.

Other disadvantages of online learning include the lack of human contact, the intimidation that some users may feel when they sit in front of a computer, the lack of skills in IT, and, at a more technical level, some disadvantages which are a result of the computer infrastructure used by the learner. As an example, a user that is accessing video lectures through a modem will find the process less efficient than a user using a Broadband connection, as the amount of time required to download the material will be significantly different. Further disadvantages of e-learning systems [3] are generated by the conformity or non-conformity of the learning resources with appropriate standards and the difficulties of arranging accreditation and testing of the learner against different awards or standards [4]. This latter disadvantage actually has more impact on the developers than on the end users themselves.

In order to alleviate some of these pitfalls some solutions have been developed. To minimise the isolation of the user as a result of a lack of personal contact, human support has been provided through a telephone line and through regular seminars, conferences and workshops. Also, in terms of the compatibility with the system, AutoTrain learning resources are produced in a wide range of formats to match the specifications of the computer system of the end-user and technical help is also available from the development team or the institution's IT department.

***Types of courses***

Online training uses the Internet as a channel for the delivery of training resources. Courses can be developed in a vast number of areas. AutoTrain-Europe and its sister project AutoTrain provide specialised training in areas that are primarily relevant to small and medium size business, principally within the automotive, engineering and manufacturing sectors. These areas include quality systems, environmental management, lean manufacturing, co-design, crashworthiness and automotive engineering amongst others.

***Quality Standards***

One of AutoTrain's areas of specialisation is that of quality standards. It offers training courses on the international standards ISO 9001:2000, ISO/TS 16949:2002, the technical specification for the automotive sector and AS/EN9100, the international standard for the aerospace sector. These are standards that have now become accepted around the world as the benchmark for all quality management systems. AutoTrain offers courses on Statistical Process Control, Visual Management, Overall Equipment Effectiveness, Six Sigma and variety of Japanese style production techniques such as Kaizen, Kanban, 5C or 5S and Total Productive Maintenance (Figure 9), with the aim of encouraging small business to implement quality systems that will promote consistency in production and reliability in delivery, as these are essential requirements to keep the business competitive in today's marketplace. The courses also aim to raise awareness of the importance of meeting customers' expectations all the time, every time, to keep them satisfied and loyal, since this is the key to prevent an organisation's customers from going elsewhere.

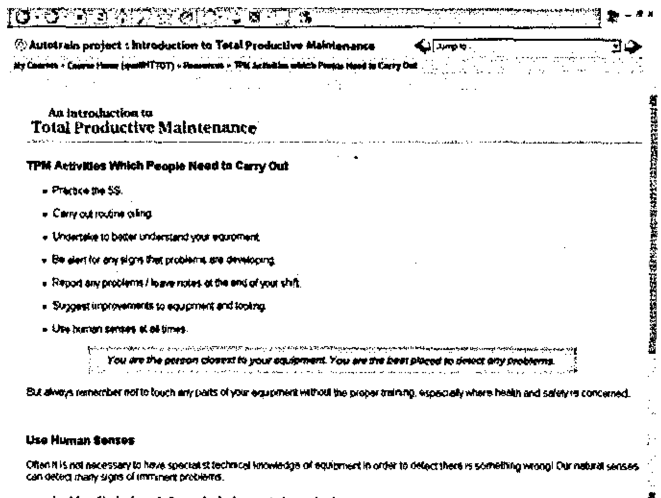



*Figure 9. Example of e-book course on Total Productive Maintenance*

Some of AutoTrain's training courses in quality standards are focused on the requirements that need to be met by the companies in order to conform to the standards. Other courses concentrate on the auditing process, since the ability to be audited by an independent, third party organisation is the foundation of the worldwide acceptance of these international quality standards. AutoTrain online training resources are an easy and cost effective way to encourage companies to adopt these quality standards that are becoming crucial for the survival of their businesses and to assist companies in the process of becoming compliant. Compliance with the ISO 9001 and ISO/TS 16949 standards can help companies attract more business as customers, both new and old, will have increased confidence in the organisation's ability to meet their expectations.

***Environmental Training***

AutoTrain also provides online training courses to assist businesses in complying with the environmental standard ISO 14001 (Figure 10). AutoTrain online training courses are essentially focused on environmental management, environmental law and techniques for waste management and pollution control in order to help businesses meet the existing legislation, reduce costs and help them gain a competitive advantage. A course in the AutoTrain CoDesign project, guided learners in designing and assembling products in environmentally favourable ways.

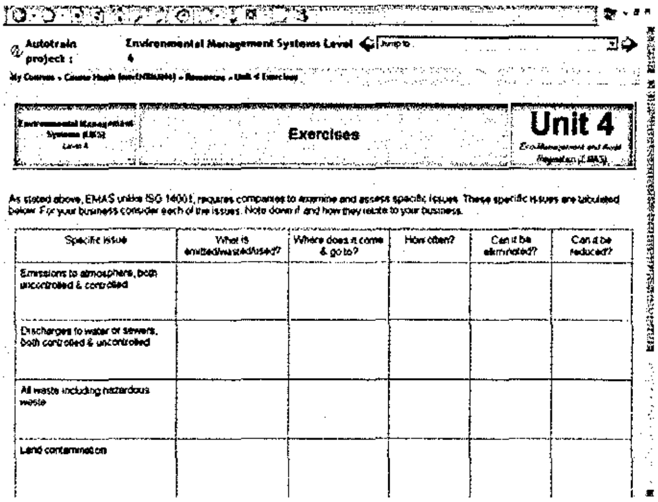



*Figure 10. Example of e-book exercise from Level 4 Environmental Management Course*

### *Six Sigma*

Six Sigma is a data-driven approach and methodology to eliminate defects in any process, but essentially manufacturing processes. The fundamental objective of the Six Sigma methodology is the implementation of a measurement-based strategy that focuses on process improvement and variation reduction through the application of Six Sigma principles based on defining, measuring, analysing, improving and controlling the process. Both AutoTrain-Europe and AutoTrain have offered several courses on Six Sigma (Figure 11) that include case studies in implementation on the shop floor and bring together the expertise of the leading figures in the field. In recent years these learning materials have been reinforced by face-to-face courses to fast-track learners to recognised standards such as "Green Belt" against supply chain and independent assessor criteria.

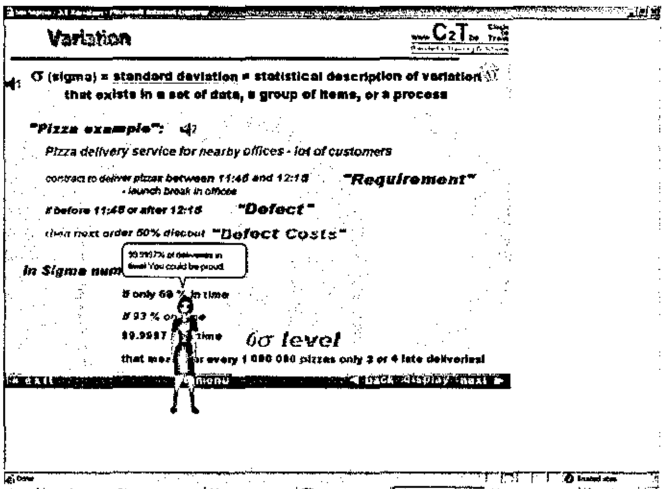



*Figure 11. Example of animated e-book course on Six Sigma*

***Lean Manufacturing***

AutoTrain-Europe and AutoTrain have been delivering courses and video presentations for a number of years on Statistical Process Control, Total Quality Management, The Toyota Production System and Six Sigma to highlight the importance of lean manufacturing in specific areas such as organisational infrastructure, analysis tools and methodologies for process improvement, project identification strategy, project management and project review. For those just embarking on the process, concise video case studies in 5C (5S) have been prepared.

***CoDesign***

AutoTrain also delivered online training aimed at improving the design and manufacturing processes using joint design through the Automotive supply chain with the purpose of sharing the design process between suppliers and customers (Figure 12). All too often "make-to-drawing" operations have very good ideas of how to improve the quality, cost and delivery of the product and are not able to input or are left to resolve manufacturing issues against predetermined tolerances and processes. CoDesign promoted the idea of training driven downwards from Vehicle Manufacturers and Original Equipment Manufacturers (VMs and OEMs) and First Tier Suppliers who want their suppliers to take more responsibility for design in order to improve quality and reduce cost.

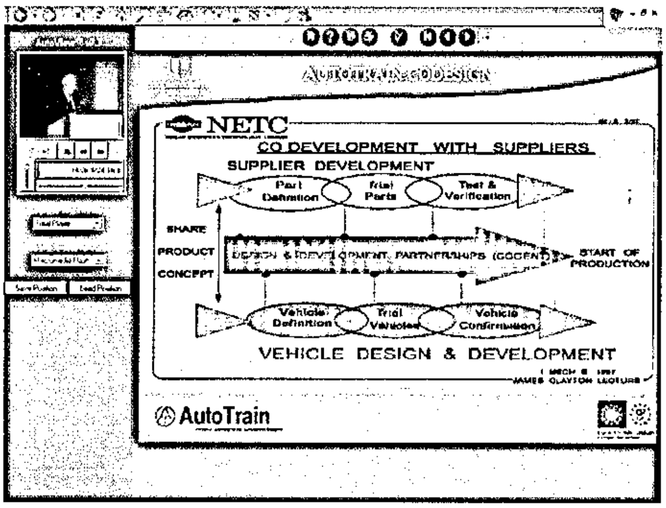



*Figure 12. Example of video lecture on CoDesign*

***Basic IT Skills***

There is a dichotomy that online training has to manage. In order to study on-line, you need a degree of IT skill or training. This is becoming less and less of an issue as more and more people from toddlers to pensioners become more proficient with computers and the Internet. AutoTrain has developed Internet training solutions to improve skills amongst the manufacturing and engineering industries. Even if the core of the online training is focused around quality and environmental standards and the implementation of techniques that help improve the manufacturing process, AutoTrain also offers a limited amount of training courses on basic computers skills for those who believe in the potential of online training solutions but cannot make the most of it due to limited knowledge.

***Crashworthiness***

AutoTrain-Europe has developed a video training programme course that consists of edited recording of a three-day course on crashworthiness held at Aston University (Birmingham, UK) in April 2003. The course offers more than fifteen hours of video material through 28 different video presentations featuring the slide presentations and the recoded delivery of world-class experts in this field.

***Automotive Engineering***

The Masters degree level course, developed in Germany by IKA Aachen, one of AutoTrain's German partners, provides detailed theoretical knowledge on the areas of power and energy demand, drive-train components and layout, vehicle dynamics and driving performance (Figure 13).

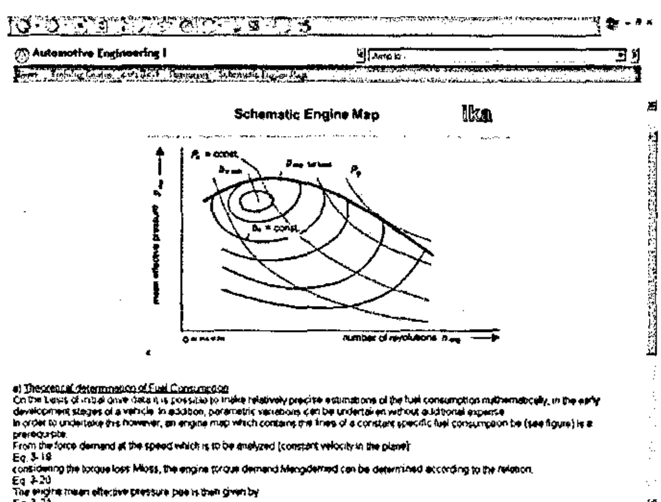



*Figure 13. Example of e-book course on 'Powertrain and Vehicle Dynamics'*

**Training validation**

Learning online can be isolating and learners may, at times, feel disoriented and unable to measure the progress that they are making. To help them to get an idea of the nature of their improvements, AutoTrain has developed courses that are interactive and provide the learner with as much feedback as possible. The learner can monitor and validate the progress made through interactive exercises and online quizzes. Other more formal accreditation is also possible by a number of different mechanisms, some of which are being developed further, but often the most effective measure of the success of the learning is demonstrated by production quality, volume, delivery or, most emphatically, by improvements to the company bottom-line earnings.

Some courses are provided with the support of a University or College that can provide workbooks and coursework assessments, which will lead to an accredited qualification. Implementation in the workplace and demonstration of real improvements can lead to the award of National Vocational Qualifications at a number of levels.

An interesting "hybrid" approach from one of AutoTrain's Spanish partners involves remote study of the course (in Mechanics) followed by conventional examination assessment at different places in different countries at the same time.

An issue of online assessment and verification that needs further development, is providing certainty that the registered learner is the same person providing the assessment answers (or that there is not a small committee sitting at the computer during the assessment!). The increased use of biometrics in passport and identification documents, allows scope for the use of these technologies in in-line assessment. This is an area of further development for the AutoTrain team and everyone else involved in the field.

It must be said that one of the main measures of effective training is the "bottom line" performance of a company. Many of the companies that have engaged in AutoTrain programmes in recent years have been less interested in certificates and accreditation than in improved technical and business performance. In fact there are examples of part-modules being studied to deliver a step change in the business.

***Interactive exercises***

AutoTrain-Europe training resources present various levels of interaction depending on the nature of the course. As an example, each module on a course on Statistical Process Control contains a variety of exercises to reinforce the knowledge that is transmitted through each of the sections in the course (Figure 14). The nature of the exercises varies depending on the topic. In some cases an exercise may require from the learner just the input of a single word to complete a sentence or a definition. In other cases, learners are required to perform a calculation. Some exercises are more sophisticated and may involve printing and plotting a certain type of chart or printing a table a monitor the performance of a certain process within a company. For most of these exercises the system developed by AutoTrain provides immediate feedback and opportunities to go back and revise the knowledge areas where the learner may feel there is a weakness.

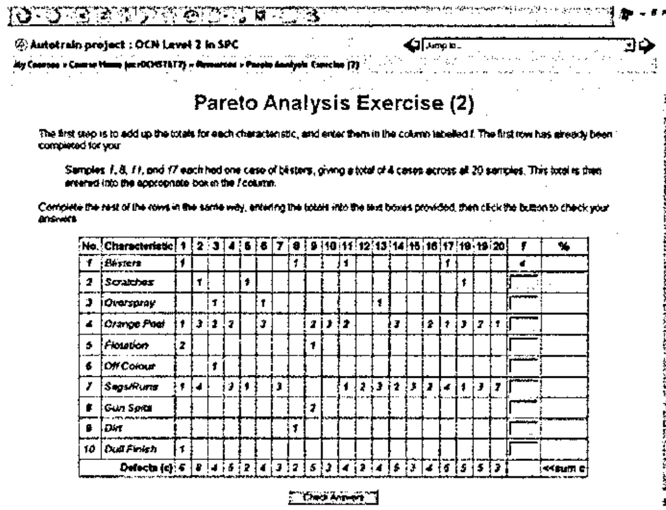



*Figure 14. Example of interactive e-book course on SPC*

***Online Quizzes***

Another way to help learners monitor their progress is through the use of online quizzes. Most AutoTrain courses include quizzes at the end of each module that are designed to measure the improvement that the learner has made for each of the sections in the course. The nature of the questions in a quiz varies from multiple choice formats to formats where the learner is required to feed back into the system a short answer or two find matches between related pairs of concepts. The aim of the quiz is to *reassure the learner's knowledge. Marks are assigned for every quiz and* feedback is provided for every answer. Also the system enables the learner to keep records with the results and time scales from all the courses that he has attempted for a particular module to give him a clearer idea of his performance and achievements (Figure 15).

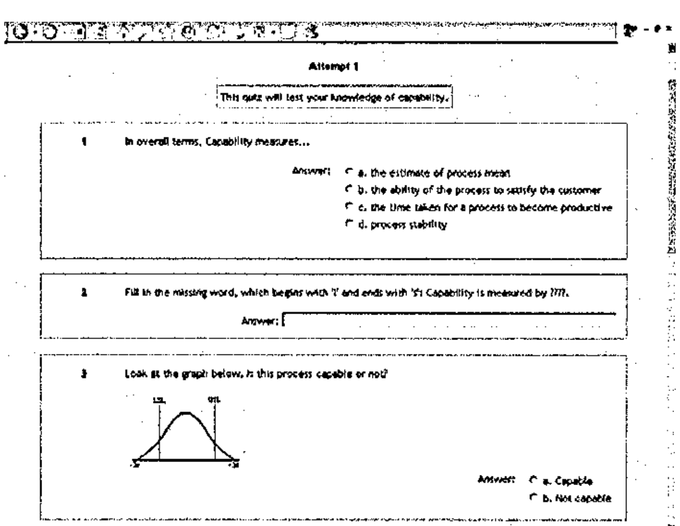


*Figure 15. Example of self-assessment quiz from e-book course " Introduction to SPC"*

### *Assessment*

AutoTrain has made evaluation forms available via the Internet to assess the status of companies for introduction of quality standards (Figure 16). The completed self-assessment can be interpreted (using standards established with experts) in order that the company gets a judgement of its current status together with a summary of appropriate strengths and weaknesses that can be fed into a management or action plan. Similar assessments against other standard or criteria could be developed.

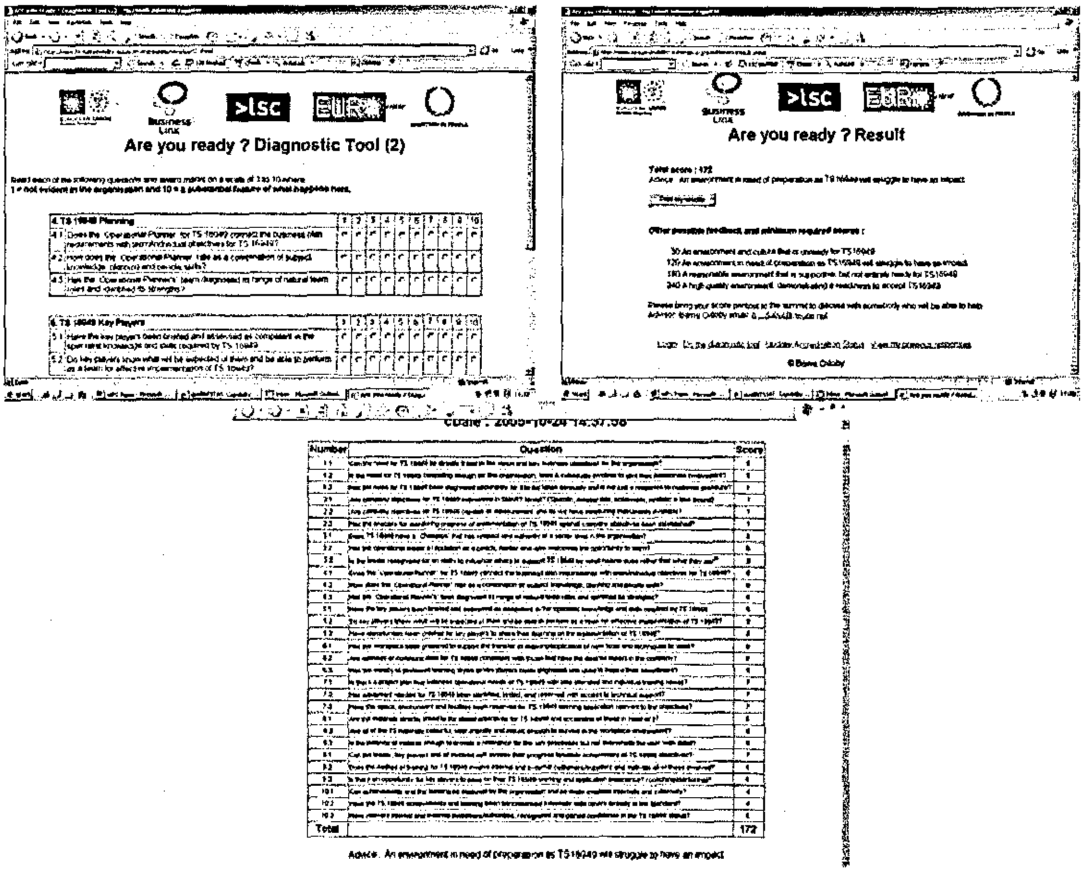



*Figure 16. Example of online self-assessment and outputs*

### Future development

In terms of future developments, the customisation to individual differences in learning and to company-specific training needs will be as critical for AutoTrain as the search for more innovative ways of incorporating interactivity and multimedia capabilities.

The team will continue researching and developing technologies that will provide the user with more responsive systems. One way to achieve this will be by improving the current video and audio technologies that AutoTrain uses for the delivery of training materials. AutoTrain will also concentrate on the development of improved technologies to generate simulations and interactions, especially for self-testing feedback. The use of video conferencing and mobile technologies such as wireless or PDA, leaves an open door for AutoTrain to work on new channels to distribute its courses and to configure learning environments for clients. Adaptability and the search for tools that will identify user requirements and relate them to users' needs is another field where AutoTrain will increase its research activity.

An obvious benefit of an MLE that holds training material and training needs information for a company or group of individuals, is directing the appropriate courses to individuals and groups. Development of the AutoTrain on-line method of assessment of knowledge and training needs can link the outcome evaluation to suitable course material

***Conclusions***

- AutoTrain has been established for over 13 years as a distance learning resource for the automotive and manufacturing industries. Despite being targeted at specific audiences in Europe and the regional initiative of the UK, the online learning material has been adopted by thousands of users from over 40 countries of the world.

- The implementation of *Moodle*, an open source code for managed learning environments, has facilitated the creation of low-cost, bespoke and generic learning platforms for the public and for specific organisations to enable their in-house training requirements to be met.

- The current resource of more than 100 training courses is being increased through the incorporation of new training material made available to the developers. Technologies are being adopted to make these courses economically available for both the learning community and for AutoTrain to meet the cost on support and ongoing development.

- The leading-edge AutoTrain video presentation technology continues to be developed and to take advantage of the increasing availability of powerful PCs and high-speed data links. The software package and its editing package are being shaped for commercial sale so that others can combine video footage and presentations, such as Microsoft PowerPoint, for their own training purposes. Class-leading presentation and subtitling capability can be increased to incorporate more of the worlds languages and to capture expert lectures from many more nations without loss of clarity or teaching effectiveness.

- The proven expertise of the AutoTrain team is available to the global industrial and academic community to provide MLE implementation and online learning, an important facet of knowledge management and knowledge transfer.

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***www.autotrain.org (Courses Developed as part of AutoTrain)***

- Statistical Process Control
- ISO 9001:2000
- ISO/TS 16494:2002
- Six Sigma
- 5S or 5C
- Lean Management
- Value Analysis and Value Engineering
- Problem Solving Tools and Techniques
- Environmental Management ISO 14001
- Electromagnetic compatibility
- CoDesign (including CAD compatibility, IPR, Project Management, etc.)
- Etc.

***www.autotrain-europe.com (Courses Developed as part of AutoTrain-Europe )***

- Aerodynamics *(FKFS Stuttgart, Germany)*
- Production Control *(ATB GmbH, Germany)*
- Managerial Economics *(ACES, Grenoble, France)*
- Analysis of Structures *(CIMNE, Spain)*
- Crashworthiness, Biomechanics, Impacts and Modelling *(EuroMotor, UK)*
- Automotive Engineering *(RWTH IKA Aachen)*
- Product Design *(CIDAUT, Spain)*
- Kinematics of Rigid Bodies *(INSA, Lyon, France)*
- Potential Failure Modes and Effects Analysis *(IAA, Belgium)*

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