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ANALYSIS AND DEVELOPMENT OF A
STREAM-POWERED COIL PUMP

by

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A Master's Thesis

Submitted in partial fulfilment of the requirements
for the award of Master of Science of the Loughborough
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SUMMARY

This project is concerned both with a mathematical analysis of the characteristics of the rotating coil pump and the development of a working stream-powered unit designed to be appropriate to the needs and resources of developing countries.

Many laboratory tests were carried out on various configurations of the pump's operating parameters, and the results obtained were used to formulate theories for the internal workings of the pump and its response to imposed conditions.

A small-scale working model incorporating a chevron bladed water wheel and a coil pump was constructed and tested in a laboratory flume.

A larger field test model based on this design was then constructed using a scrap oil drum and other materials considered to be readily available in developing countries. Successful field tests were carried out in a local stream and improvements made to the original design.

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1.0 INTRODUCTION

The rotating coil pump basically consists of a length of flexible piping wound on a cylindrical drum to form a continuous helix made up of a number of coils all having the same radius. One end of the piping is left open (the inlet) and the output end is connected via a sealed rotary joint to a delivery pipe which rises to a header tank (see Figure 1).

The drum is partially immersed in water with the longitudinal axis of the drum parallel to the water surface and rotated about this axis. This causes the inlet to alternatively pass through water and air. The ratio of water to air taken in at the inlet is determined by the depth of immersion of the coils.

The plugs of water remain in the bottom of the coils, and as the pump rotates, are moved from the inlet towards the outlet. After passing through the rotary valve, the plugs of water and air rise up the delivery pipe creating pressure at the bottom of the delivery pipe which is opposed by head differences formed in the coils as the plugs of water swing from the bottom of the coils and take up a position to oppose and equal the head of water in the delivery pipe. The coils in the pump are therefore acting as a cascading manometer and the sum of all the head differences in the individual coils equals the head of water in the delivery pipe.

Since alternate plugs of air and water are pumped up to the header tank, the pressure head developed by the pump is less than the height to which the water is being lifted. The ratio of lift to head developed is known as the delivery pipe ratio and this is one of the most important considerations in the design of the pump.

Also the pattern of head differences in the coils of the pump needs to be predicted to determine the number of coils needed to provide the required head.

The main aims of this investigation are, therefore, to determine, using computer-based incremental calculations, the relationship between the physical characteristics of the pump, and the expected lift:head ratio, and in particular to relate the head developed by a particular pump to the number of coils required and the levels of the water surfaces in those coils.

Following the laboratory investigation, a working field model of a rotating coil pump was to be constructed from materials readily available in developing countries and tested at an appropriate site.

1.1 Review of past work

The first reference to the rotating coil pump is a brief reference to be found in Andrew Wirtz's 'Cyclopedia of Arts and Science' of 1745, (Ref. 1). Since then, very little work has been done on the pump until recent times. Work at Loughborough University has been restricted to Final Year Projects carried out by students of the Department of Civil Engineering, (Ref. 2-5). Different approaches to the analysis of the pump have been considered, including an analysis of forces acting within the pump, and measurement of discharge pressures by means of a transducer, (Ref. 3). From this work, few definite conclusions or relationships concerning the capabilities of the pump were produced, but a good background knowledge of the pump's characteristics was obtained.

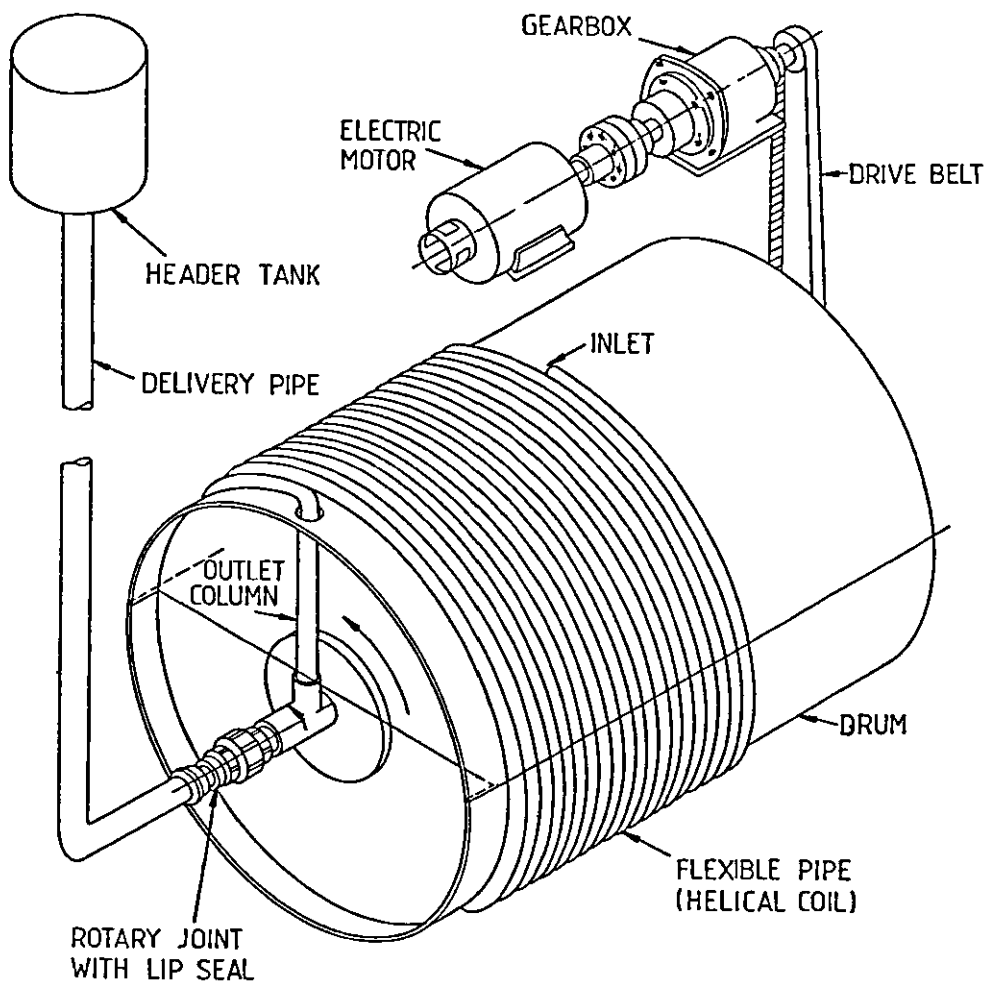
A mathematical analysis of the pump was carried out by Alexander Weir, (Ref. 9). This analysis, however, was restricted to a pump with a small number of coils, which correspond to the first three or four

coils of the type of pump studied in this project, and the application of Weir's analysis is therefore limited.

Work on a stream powered version of the coil pump has been carried out by Stuckey and Wilson at Salford University. The analysis of the pump's operation centred around the grouping of variables into dimensionless groups which were then related graphically.

Tests on a small stream powered pump were carried out, although practical difficulties, primarily concerned with the rotary joint, were encountered. The seals within the joint were found to create a large resistance to turning if an air tight joint were to be obtained.

Slackening of the seals reduced the torque requirement but resulted in leakage and loss of performance, (Kef. 10).



Laboratory arrangement of pump

Figure 1

2.0 DESCRIPTION OF PROJECT OUTLINE

The five parameters which are considered most important in determining the pump's performance are depth of immersion, speed of rotation, bore of delivery pipe, bore of coils and the diameter of the coils. Work has previously been done on varying the number of coils and the effect this has on the operation of the pump. Each of the five parameters were varied within a set range whilst the other four were kept at a standard value. These standard values were based on data from earlier tests and taken as the 'middle of a useful range'.

The range of tests are given in the following table.

	Max.	Min.	Standard value
Depth of immersion (Proportion of diameter submerged)	0.7	0.3	0.5
Speed of rotation (rpm)	16	8	12
Bore of delivery pipe (mm) (Two sizes)	25	12	25
Bore of coils (mm) (Three sizes)	38	12	25
Diameter of coils (mm) (Two sizes)	911	487	487

For each different configuration of these main parameters the height of the delivery tank was raised from 4 m, through increments of 1 metre, until the pump could no longer produce the required lift. This usually involved five or six increases in lift; for the larger

sized pump the initial lift was 8 m. For each test (i.e. for each separate lift) measurements were taken of the quantity pumped, the head developed in the delivery pipe, the power absorbed by the electric motor, and the water levels in each coil were measured.

Due to the difficulty of measuring the water levels in the coils as the pump is rotating, these measurements were taken for the static condition existing after the pump had been brought to a stop. Previous work done (Ref. 2) using a video camera had shown that this method was justified and that, although the water levels oscillate slightly during pumping (due to the varying pumping head), the levels remained stationary once pumping ceased, at an elevation equivalent to the mean of the dynamic oscillation.

The method of analysis used in this project resulted from consideration of the methods previously employed. It was thought that modelling the internal dynamic mechanisms, whilst being an exhaustive procedure, was too complex an approach to the work. It has also been found from previous work (Ref. 2-4) that the depth of such an analysis required to produce useful guidelines to the pump's behaviour would be excessive, and could not be accommodated within the framework of the project.

Past work (Ref. 5) indicated that considering the pump as a rotary cascading manometer could be a viable means of analysis, and gave a means of investigating the pump's working mechanisms by measurement of static water levels within the coils. A requisite of any developed theory was that it could be represented in a form that could be used and understood by the engineer in a practical situation. For these reasons, equations were derived that related to a practical range of pump designs, with prime consideration being given to agreement of theoretical results and experimental data. By formulating computer

programs which included all of the parameters of pump design, it was then possible to predict the characteristics of one particular design, or produce design charts which could be used for a wide range of differing pumps.

The bulk of the project is concerned with this analysis of the pump's behaviour; further work on a waterwheel power source led to the construction and testing of a working stream-powered pump suitable for use in a developing country.

3.0 LABORATORY TESTING AND ANALYSIS OF THE ROTATING COIL PUMP

3.1 Design of Test Rig

The size of the tank in which the pump would be housed was based on the largest size of pump to be used. The biggest arrangement envisaged consisted of 20 coils of 38 mm bore piping wound around 1 m diameter drum which called for a tank measuring 1.7 m by 1.5 m by 1.4 m deep.

Because of previous difficulties in examining and measuring the lower water levels in the coils, the sides of the tank were made of thick perspex to allow an unrestricted view of the complete pump. A 1.5 kW electric motor was supported directly above the pump by two rectangular section steel beams which ran the length of the tank. The drive was via a 31.5:1 ratio gearbox and toothed belt which, with the relative size of the two cogs considered, gave a final gearing ratio of 63:1. Connected in series with the motor were a digital voltmeter and ammeter and a freeze button to hold the readout at a constant value whilst readings were taken.

The whole tank was supported on legs to bring the pump up to eye level for easier observation. A two inch diameter pipe was connected to a nearby pipe bank for filling the tank and a drainage pipe was fixed to the floor of the tank. Calibrated tape was affixed to the perspex sides of the tank and used to measure the water level in the tank. A depth gauge was fixed to a length of angle section steel which was in turn clamped to the top of the tank.

Water from the pump was pumped up the inside of a water tower and a variety of delivery pipes were cut to ensure a straight vertical

lift to the header tank which drained back into the pump's tank to ensure that the water level in that tank remained constant.

3.2 Measurements Taken and Errors

As mentioned in section 2.0, for each separate configuration of the variables, the pump was made to lift to five or six different heights, the total number of individual tests finally performed being 64. For each test (i.e. a particular configuration lifting to a given height) the following measurements were taken:-

- Speed of rotation
- Water level in tank
- Power absorbed by motor
- Quantity of water pumped
- Levels of water in coils
- Height of water in outlet column
- Head developed by pump.

The speed of rotation and the water level in the tank had to be set at the desired values with the pump lifting to the test height, and it was ensured that a steady state had been reached before any other readings were taken.

The speed of rotation was measured by pressing a tachometer against the rotating shaft of the motor. A variation of ± 4 r.p.m. was the greatest accuracy that could be achieved with the rheostat which controlled the power supply to the motor. When divided by the gearing ratio of 61:1, the variation in the rotational speed of the pump is ± 0.066 r.p.m., an error of 0.55% for a speed of 12 r.p.m. The drive between motor and pump was direct, ensuring no slippage.

The water level in the sump tank was measured by means of graduated tape on the perspex wall. As the level had to be set whilst

the pump was in motion, a certain amount of movement was caused on the surface of the water. However, it was still possible to set this level with an accuracy of ± 1 mm, or 0.25% of the standard coil diameter.

When a steady state had been achieved, measurements of motor power and pumping rate were taken. The outputs given by the digital voltmeter and ammeter which were connected in series with the motor could be 'frozen' to facilitate reading. Although these readings were accurate, the resulting power measurements suffered from great variation (up to 75% in cases) as the power varied during the pumping cycle, and the results were therefore not used in the analysis of the pump's performance.

The pumping rate was determined by placing a measuring cylinder under the return pipe to record the quantity of water flowing from the header tank. A stilling tank ensured that the characteristic fluctuations due to the water/air pumping cycle were smoothed out before the quantity measurements were taken. Three separate readings were taken and in the rare event of the variation in these readings being more than 5%, a further three measurements were taken to give a more representative sample. The average variation was more commonly of the order of 2%.

Before the measurement of water levels in the coils was taken, it was necessary to stop the pump. This is the most accurate method available because such a measurement is virtually impossible whilst the pump is rotating. The dynamic water levels are preserved in the static case although such effects as those due to friction, surface tension, etc., are absent, but calculations suggested these were small.

Measurement was by means of a depth gauge with a hooked pointer which was mounted on a beam fixed across the top of the test tank.

The depth gauge was fitted with a Vernier scale giving an accuracy of ± 0.1 mm, i.e. a total error of ± 4 mm in the sum of head differences on a 20 coil pump.

The head of water in the delivery pipe was also taken with the pump at rest. The side of the tower up which the pipe ran was marked at 1 m intervals and the height of water above an interval was measured with a metre rule. Due to the occasional difficulty of access to such measurements, an error of ± 10 mm was experienced.

3.3 Analysis of Delivery Pipe

The situation in the delivery pipe is a complex one. Alternate plugs of water and compressed air enter the bottom of the pipe from the outlet of the pump. As these plugs rise up the delivery pipe the air plugs become less compressed as the head of water above them diminishes. The air plugs also rise through the water plugs above them as water simultaneously runs back down the inside of the delivery pipe.

Assumptions and Equations Used

- i) It is assumed that all variations are linear over each time increment.
- ii) Effects of friction and surface tension have not been considered, but calculations suggest this incurs errors of less than 1%.
- iii) The delivery pipe is assumed to be vertical and of constant cross section.
- iv) The weight of an air plug remains constant as it passes through the pump.
- v) Air does not begin to rise through a water plug until the whole of the water plug has entered the delivery pipe.
- vi) The relative velocity is constant along the pipe.

vii) The discharge from the delivery pipe is open to the atmosphere.

Consider first a simplified case in which the lengths of the plugs of air and water as they enter the delivery pipe are the original lengths that occupy the inlet coil of the pump. Each air plug will be compressed by the head of water acting on it, and this compression is governed by the equation:-

$$P_1 V_1 = P_0 V_0 \quad (\text{See Appendix I})$$

where P_1 = Total pressure acting on air plug

V_1 = Compressed volume of air plug

P_0 = Original (atmospheric) pressure

V_0 = Original volume of air plug (at inlet)

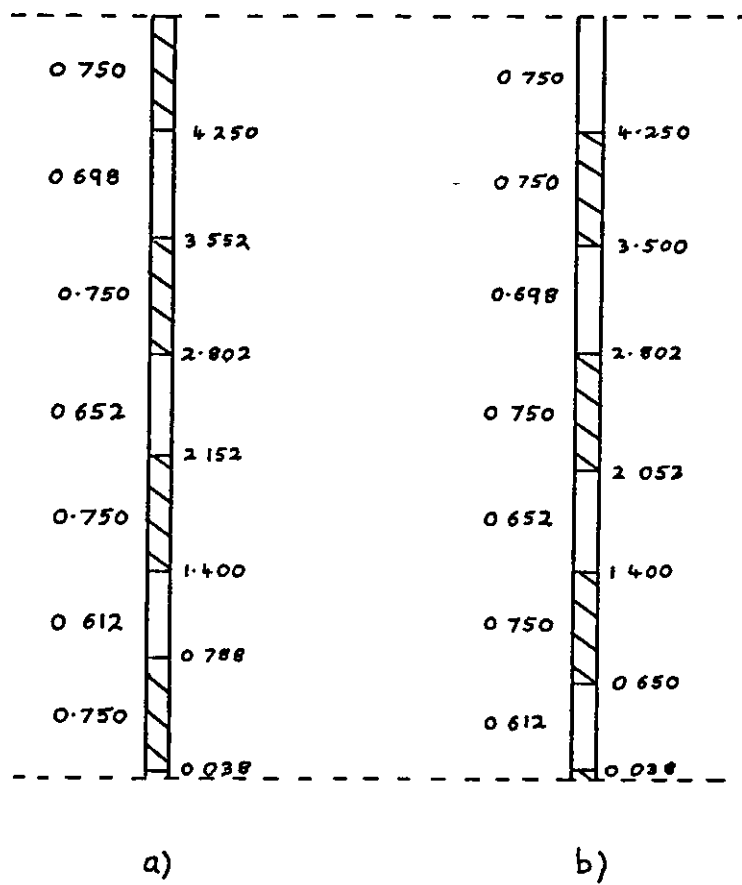
Since the cross-sectional area of the delivery pipe is constant then volumes can be replaced by lengths; it is also convenient to express pressure in terms of metres head of water, so the equation now becomes:-

$$H_1 L_1 = H_0 V_0 = H_{\text{Atm}} V_0$$

Assume that the coils on the pump are of the same bore as the delivery pipe and that, immersed to half its diameter, the inlet is taking in plugs of water both of length 0.75 m. Let the atmospheric pressure be equivalent to 10 m head of water.

Figure 2 shows two possible arrangements of these plugs in a delivery pipe used for a 5 m lift: the values on the left of each pipe being the length of the individual plugs, those on the right the heights of each water/air interface above the bottom of the delivery pipes.

For case (a) the head being developed by the pump (i.e. the sum of the lengths of the water plugs) is 3 m, for case (b) 2.288 m.



Two different positions of a set water/air sequence in a delivery pipe .

Fig. 2

This demonstrates two important points:-

- (i) that the lift achieved by a pump is greater than the head developed by that pump;
- (ii) that the head developed for a given lift can vary depending upon the positioning of the water/air sequence in the delivery pipe.

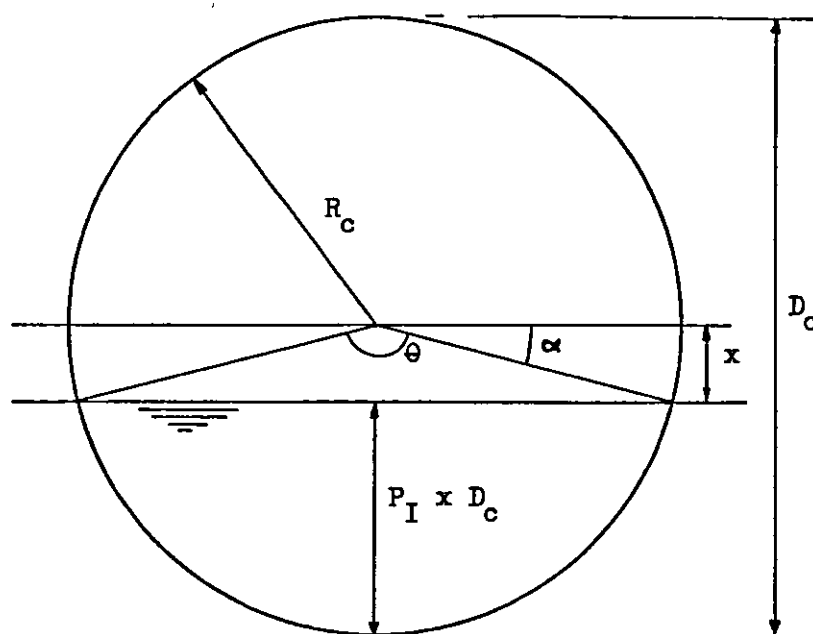
The ratio of the lift to the head developed is termed the delivery pipe ratio (R_D) and it is the determination of this ratio for all the tests performed that is the aim of this analysis.

In this simplified example, no account was taken of the air rising through the water, and the two cases shown were hypothetical ones used to illustrate the situation in the delivery pipe. In the full analysis, the situation is examined over a series of short time intervals, the position and length of each individual air and water plug being calculated for each time interval.

The iterative calculation process starts at time $t = 0$ with the first full plug of water in position at the bottom of the delivery pipe, the trailing end of the plug being at the bottom of the pipe and the leading end being at a height w_0 . The length w_0 is the original length of the water plug at the inlet and is calculated thus:-

(see diagram overleaf)

3.3.1 Calculation of Original Plug Length



Vertical distance from centre of coils to water level:-

$$x = P_I \times D_C - R_C$$

where P_I = proportional depth of immersion of pump

D_C = diameter of coils

R_C = radius of coils

$$\begin{aligned} \therefore x &= 2P_I R_C - R_C \\ &= R_C(2P_I - 1) \end{aligned}$$

$$\begin{aligned} \text{Now } \alpha &= \sin^{-1} \left(\frac{x}{R_C} \right) \\ &= \sin^{-1} \frac{R_C(2P_I - 1)}{R_C} \end{aligned}$$

$$\therefore \alpha = \sin^{-1} (2P_I - 1)$$

Angle subtended at centre of coils by water plug = θ

$$\begin{aligned} \text{where } \theta &= \pi + 2\alpha \\ &= \pi + 2 \sin^{-1} (2P_I - 1) \end{aligned}$$

Length of water plug $w_o = R_c \cdot \theta$

$$\therefore w_o = R_c \left[\pi + 2 \sin^{-1} (2P_I - 1) \right] \frac{d_D^2}{d_P^2}$$

where d_D and d_P are the bores of the delivery pipe and the coils respectively.

Similarly, the original air plug length at the inlet, a_o is given by:-

$$a_o = R_c \left[\pi - 2 \sin^{-1} (2P_I - 1) \right] \frac{d_D^2}{d_P^2}$$

The next stage of the process involves determining the position of this first plug after a suitable time interval Δt . For this to be possible the velocity of the plug up the delivery pipe (v_D) needs to be known along with the relative velocity of the air plug through the water plug (v_A). If the piping used for both the coils and delivery pipe is of the same cross-sectional area then $v_D = v_P$ (peripheral velocity of coils).

Therefore after time interval Δt , the position of the top of this first (uppermost) water plug is given by

$$TOP(J) = w_o + v_D \cdot \Delta t$$

N.B. The subscript J is used to denote the uppermost water plug in the delivery pipe.

Since the air plug is moving through the water plug, the height of the bottom of the water plug is given by

$$BOT(J) = (v_D + v_A) \cdot \Delta t$$

∴ Length of water plug

$$\begin{aligned} PLUG(J) &= TOP(J) - BOT(J) \\ &= w_o + v_D \cdot \Delta t - (v_D + v_A) \cdot \Delta t \\ &= w_o - v_A \cdot \Delta t \end{aligned}$$

Hence the plug has reduced in length by an amount $v_A \cdot \Delta t$ and this water has been lost to the plug of water in the last coil of the pump. This is referred to as plug $(N + 1)$, plug (N) being the lowest plug of water in the pipe. It is assumed that the bottom of the delivery pipe is connected directly to the outlet of the pump.

The compressed length of the air plug is given by:-

$$\begin{aligned} a_c &= a_o \left(\frac{H_{Atm}}{H_{Atm} + H} \right) \\ &= a_o \left(\frac{H_{Atm}}{H_{Atm} + w_o - v_A \cdot \Delta t} \right) \end{aligned}$$

The position of this air plug then needs to be determined to ascertain whether plug $(N + 1)$ has begun to travel up the delivery pipe.

It is easiest to demonstrate this and subsequent calculations by means of an example as shown in Figures 3 to 9. In this example the initial condition is taken as the time when the first plug of water is about to enter the delivery pipe, i.e. $TOP(J) = 0.0$

The values of other relevant parameters are:-

$$w_o = 0.75 \text{ m}$$

$$a_o = 0.75 \text{ m}$$

$$H_{\text{Atm}} = 10.329 \text{ m } (\equiv 760 \text{ mm Hg})$$

$$\Delta t = 1 \text{ sec.}$$

$$v_D = 0.3 \text{ m/s}$$

$$v_A = 0.1 \text{ m/s}$$

$$H_t = 5 \text{ m (height of delivery)}$$

$$d_D = 0.025 \text{ m}$$

$$d_P = 0.025 \text{ m}$$

Explanatory calculations to accompany the example are shown below.

N.B. The names given to variables are those which are used in a computer program used to perform the delivery pipe analysis.

$$\underline{t = 3.0 \text{ secs.}}$$

It is assumed that air cannot rise through a water plug (i.e. water cannot be lost from that plug) until the trailing end of that plug has entered the delivery pipe. This occurs between times $t = 2.0 \text{ secs.}$ and $t = 3.0 \text{ secs.}$

Time taken for bottom plug to reach base of delivery pipe

$$\text{TIME} = \frac{\text{WATLEN} - \text{TOP}(J)}{\text{VDEL}}$$

$$\text{WATLEN} \equiv w_o$$

$$\text{VDEL} \equiv v_D$$

$$= (0.75 - 0.6)/0.3$$

$$= 0.5 \text{ secs.}$$

∴ From $t = 2.5 \text{ secs}$ to $t = 3.0 \text{ secs}$, water is being lost from the bottom of the water plug. The amount lost (DROP) is given by:-

$$\begin{aligned}
 \text{DROP} &= (\text{DT} - \text{TIME}) \times \text{VA} \\
 &= (1 - 0.5) \times 0.1 \\
 &= 0.05
 \end{aligned}$$

$$\begin{aligned}
 \text{VA} &\equiv v_A \\
 \text{DT} &\equiv \Delta t
 \end{aligned}$$

$$\begin{aligned}
 \therefore \text{Length of plug PLUG}_{(J)} &= \text{WATLEN} - \text{DROP} \\
 &= 0.7 \text{ m}
 \end{aligned}$$

$$\begin{aligned}
 \text{Length of plug } N + 1 \quad \text{PLUG}_{(N+1)} &= \text{WATLEN} + \text{DROP} \\
 &= 0.8 \text{ m}
 \end{aligned}$$

Plug (N+1) now gains all the water which is lost from the bottom of plug (N) which at this stage is also plug (J), N being the number of complete water plugs in the delivery pipe.

$$t = 5.0 \text{ secs}$$

$$\text{Length of air plug AIRLEN} = \text{BIRLEN} \times \left(\frac{\text{PA}}{\text{PA} + \text{Head}} \right)$$

$$= 0.75 \times \left(\frac{10.329}{10.329 + 0.5} \right)$$

$$\text{BIRLEN} \equiv a_0$$

$$\text{PA} \equiv H_{\text{Atm}}$$

$$= 0.715 \text{ m}$$

HEAD = Head of water acting on an air plug.

Water plug (N+1) is still gaining water from the plug above it, the position of the top of plug (N+1) is calculated by subtracting the length of air plug (J) from the height of the bottom of water plug (N)

$$\begin{aligned}
 \text{TOP (N+1)} &= \text{BOT(N)} - \text{AIRLEN (N)} \\
 &= 1.0 - 0.715 \\
 &= 0.285 \text{ m}
 \end{aligned}$$

$$t = 8.0 \text{ secs}$$

Between times $t = 7.0 \text{ secs}$ and $t = 8.0 \text{ secs}$ the bottom of plug (N+1) passes the base of the delivery pipe and plug (N+1) effectively becomes plug (N).

The calculations to determine when this happens and the amount of water gained or lost by the plugs involved are similar to those employed at time $t = 3.0$ secs.

$$\begin{aligned}\text{TIME} &= \frac{\text{PLUG (N+1)} - \text{TOP (N+1)}}{V_{\text{DEL}}} \\ &= \frac{1.2 - 1.072}{0.3} \\ &= 0.427 \text{ secs}\end{aligned}$$

$$\begin{aligned}\text{DROP} &= (\text{DT} - \text{TIME}) \times \text{VA} \\ &= (1 - 0.427) \times 0.1 \\ &= 0.057 \text{ m}\end{aligned}$$

$$\begin{aligned}\therefore \text{ At } t = 8.0 \text{ secs } \text{PLUG (N)}_8 &= \text{PLUG (N+1)}_7 + (\text{VA} \times \text{DT}) - \text{DROP} \\ &= 1.2 + 0.1 - 0.057 \\ &= 1.243 \text{ m}\end{aligned}$$

$$\begin{aligned}\text{Also } \text{PLUG (N+1)}_8 &= \text{WATLE N} + \text{DROP} \\ &= 0.75 + 0.057 \\ &= 0.807 \text{ m}\end{aligned}$$

Plug (N) will now remain the same length because the gain from the plug above and the loss to the plug below result in no net change. This condition continues until this plug becomes the uppermost plug or reaches the top of the delivery pipe.

$$\underline{t = 10.0 \text{ secs}}$$

Since plug (J) fully entered the delivery pipe it has been losing 0.1 m of water every second so it is plain to see that it will 'disappear' when $t = 10.0$ secs. However this exact case is not common and for the general case the precise time that plug (J) ceases to exist must be calculated in order to determine when plug (J+1) becomes (J) and only loses water without gaining any from the plug above.

$$\underline{t = 13.0 \text{ secs}}$$

This is a repeat of the case at time $t = 8.0$ secs when plug (N+1) fully enters the delivery pipe and becomes plug (N). The calculations are therefore the same.

$$\underline{t = 20.0 \text{ secs}}$$

Between times $t = 19.0$ secs and $t = 20.0$ secs the uppermost plug has passed out of the delivery pipe (into the header tank) and plug (J+1) has become plug (J). In order to determine the amount of water leaving the delivery pipe (to give a value for the delivery rate Q_p) and the length of the uppermost plug at time = 20.0 secs, it is necessary to calculate what proportion of the time interval had passed when the plug of water left the delivery pipe.

$$\begin{aligned} \text{TIM} &= \frac{\text{TANK} - \text{BOT}(J)}{\text{VDEL} + \text{VA}} \\ &= \frac{5.0 - 4.607}{0.3 + 0.1} \\ &= 0.983 \text{ secs} \end{aligned}$$

TANK = Height of header tank (H_T)

$$\begin{aligned} \text{DROP} &= \text{TIM} \times \text{VA} \\ &= 0.983 \times 0.1 \\ &= 0.0983 \text{ m} \end{aligned}$$

$$\begin{aligned} \text{Quantity delivered } Q &= (\text{PLUG}(J) - \text{DROP}) \times \pi \times \text{DIAD}^2/4 \\ &= (0.343 - 0.0983) \times \pi \times 0.025^2/4 \\ &= 1.201 \times 10^{-4} \text{ m}^3 \\ &= 0.120 \text{ litres} \end{aligned}$$

From this point onwards, these values of Q are summed, then divided by the time over which water has been delivered to give a value of Q_p in litres/min.

The top of the uppermost air plug reached the top of the delivery pipe at the same time as the water plug left it, i.e. 0.983 secs through the 1 sec time interval. Therefore, for 0.017 secs uncompressed air has been leaving the pipe.

$$\begin{aligned}\text{AIRLEN}(J) &= \text{BIRLEN} - ((DT - \text{TIM}) \times \text{VA}) \\ &= 0.75 - ((1 - 0.983) \times 0.3) \\ &= 0.745 \text{ m}\end{aligned}$$

$$\begin{aligned}\text{PLUG}(J)_{20} &= \text{PLUG}(J+1)_{19} + \text{DROP} - (\text{VA} \times \text{DT}) \\ &= 1.234 + 0.0983 - 0.1 \\ &= 1.232 \text{ m}\end{aligned}$$

The pump has now reached a steady state and the delivery pipe ratio will fluctuate between maximum and minimum values. All values are recorded and the max., min. and mean values are calculated at the end of the process.

$$\begin{aligned}R_D &= \frac{\text{TANK}}{\sum_{I=J}^N \text{PLUG}(I)} \\ &= \frac{5}{(1.232 + 1.22 + 0.526)} \\ &= 1.679\end{aligned}$$

$$\underline{t = 23.0 \text{ secs}}$$

$$\begin{aligned}Q &= (\text{TOP}(J) + (\text{VDEL} \times \text{DT}) - \text{TANK}) \times \pi \times \text{DIAD}^2/4 \\ &= (4.855 + 0.3 - 5.0) \times \pi \times 0.025^2/4 \\ &= 0.076 \text{ litres}\end{aligned}$$

$$\underline{t = 25.0 \text{ secs}}$$

Q calculations as for $t = 20.0$ secs.

This series of calculations is continued usually until $t = 200.0$ secs, in increments of 0.1 secs. A flow chart for the program used plus a listing and run of the program are included in Appendix II.

Modifications to This Analysis

Once the program had been found to give satisfactory results for the standard cases, i.e. pipe size constant, small drum diameter etc., methods of improving the correlation over the whole range of tests were studied.

By increasing the relative air velocity in increments of 0.05 m/s in the computer program and carrying out a linear regression analysis to compare theoretical and experimental results, the following values were obtained:-

Relative air velocity m/s	0.15	0.2	0.25	0.3	0.35
Regression coefficient (r^2)	0.798	0.823	0.838	0.819	0.744

i.e. a value of 0.25 m/s gave the best results over the whole range of tests. However, this value did not provide consistent results over the entire range of pump configurations used.

Examination of the results showed that this relative air velocity was related to the velocity in the delivery pipe. By increasing the value of V_A proportionally to V_D , the agreement of theoretical and experimental results was improved, but the closest correlation was obtained (by trial and error) when the following relationship was incorporated into the program:-

$$V_A = \left(\frac{V_D}{0.306} \right)^{0.2} \times V_A'$$

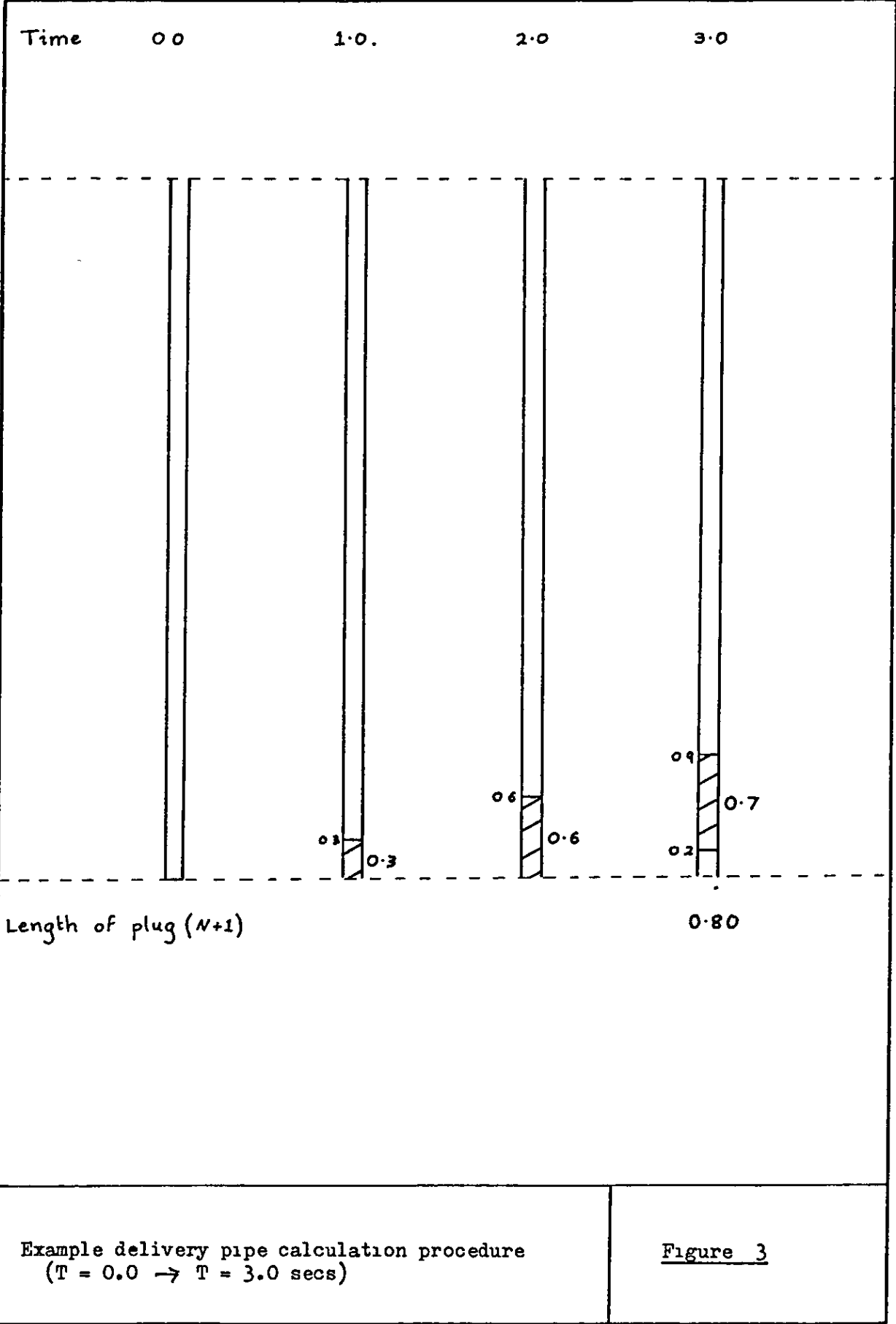
where V_A' is the relative air velocity used for the standard configuration, i.e. 0.25 m/s. This relationship was used in the analysis, although there is scope for further work to verify the relationship. N.B. 0.306 m/s is the velocity of delivery for the standard case (i.e. that configuration corresponding to the V_A' value of 0.25 m/s).

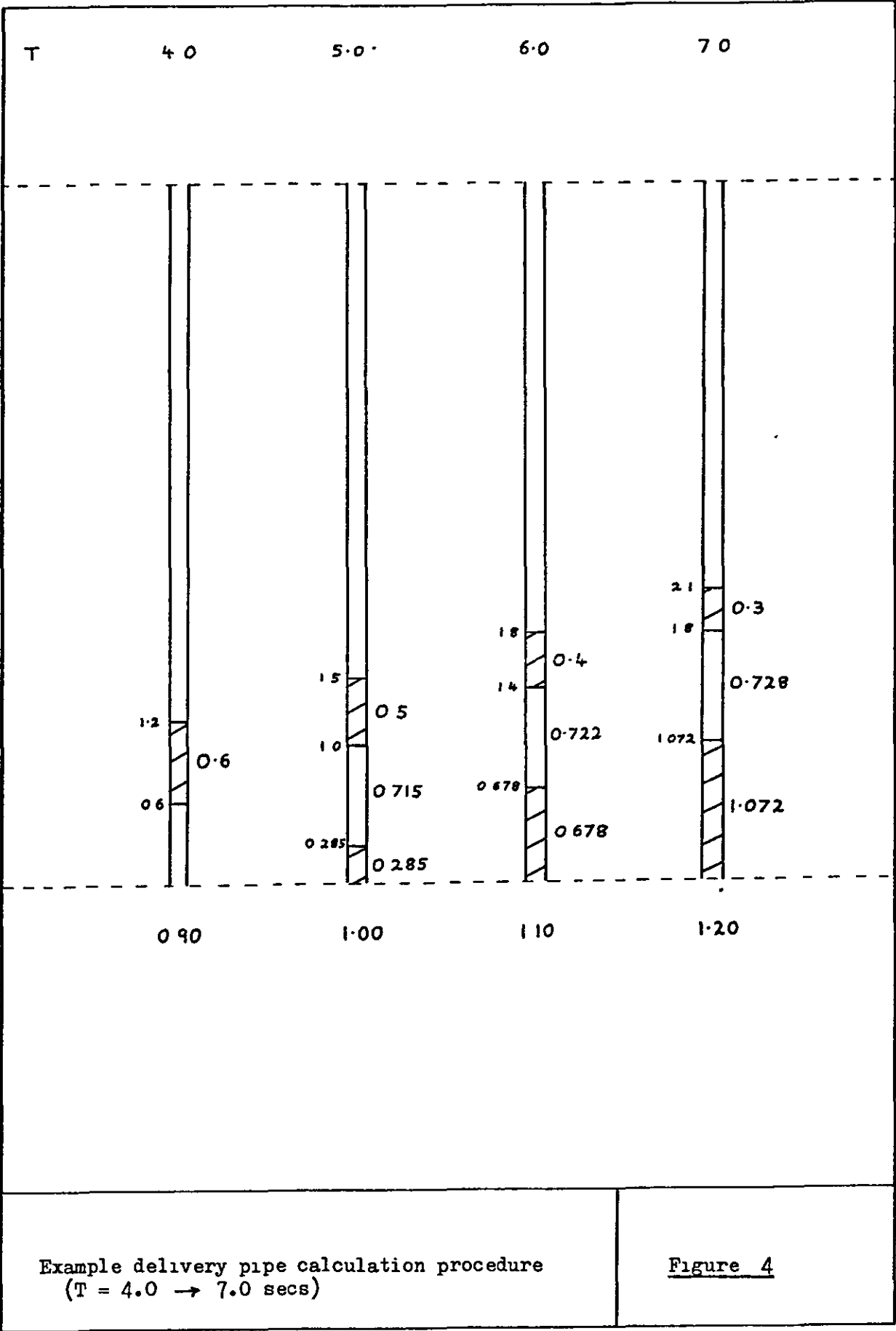
Other values of the exponent in the above equation were experimented with, but 0.2 proved to be the most effective over the whole range of tests. This modification improved the correlation coefficient (r^2) to 0.839.

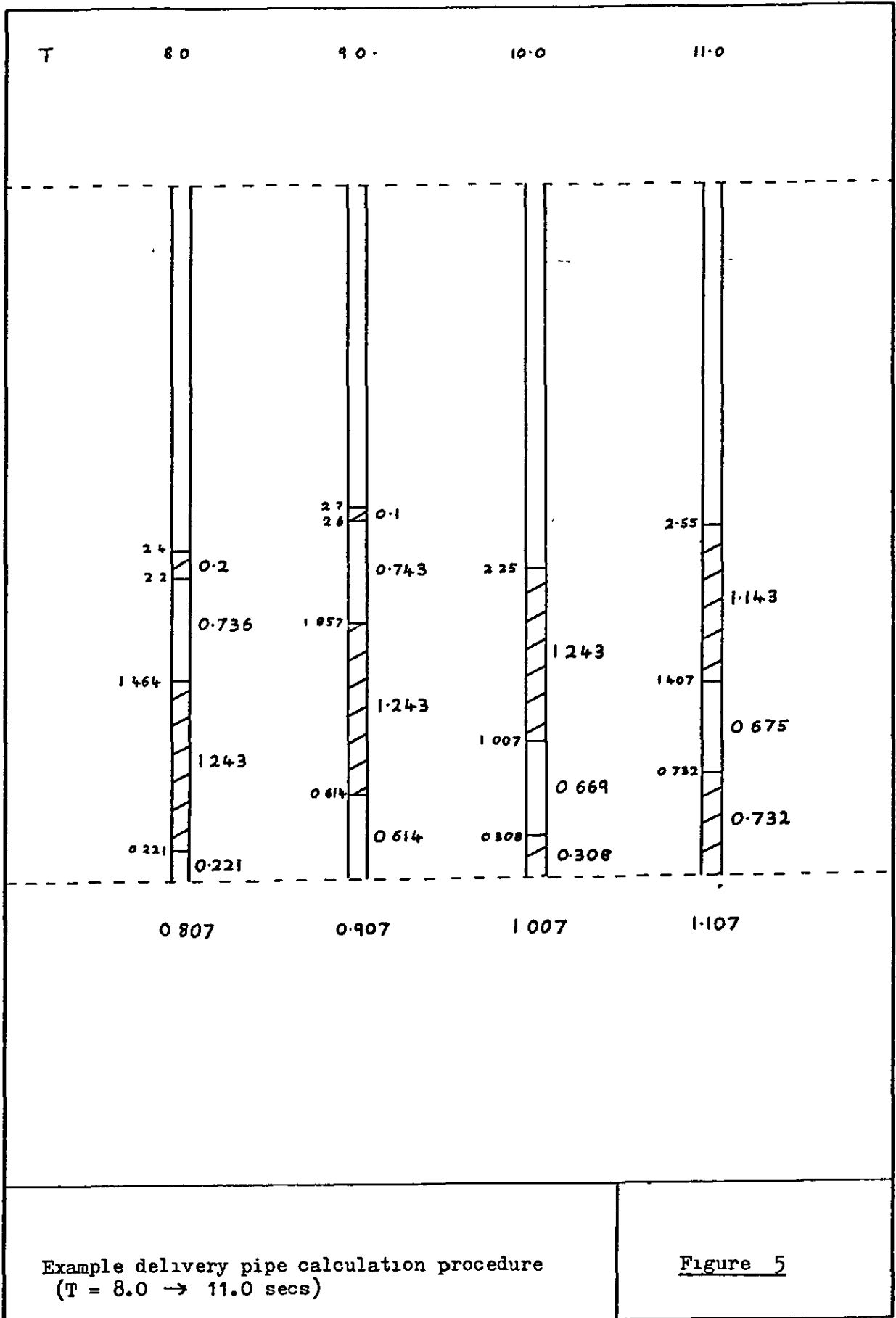
Since the base of the delivery pipe is assumed to be connected directly to the outlet of the pump, then plug (N+1) must also be assumed to be in the last coil of the pump and will therefore be losing water by the mechanism known as spillback which is explained fully in the next section. Account of this has been taken in the program and calculations inserted to determine the amount of water lost by spillback from plug (N+1).

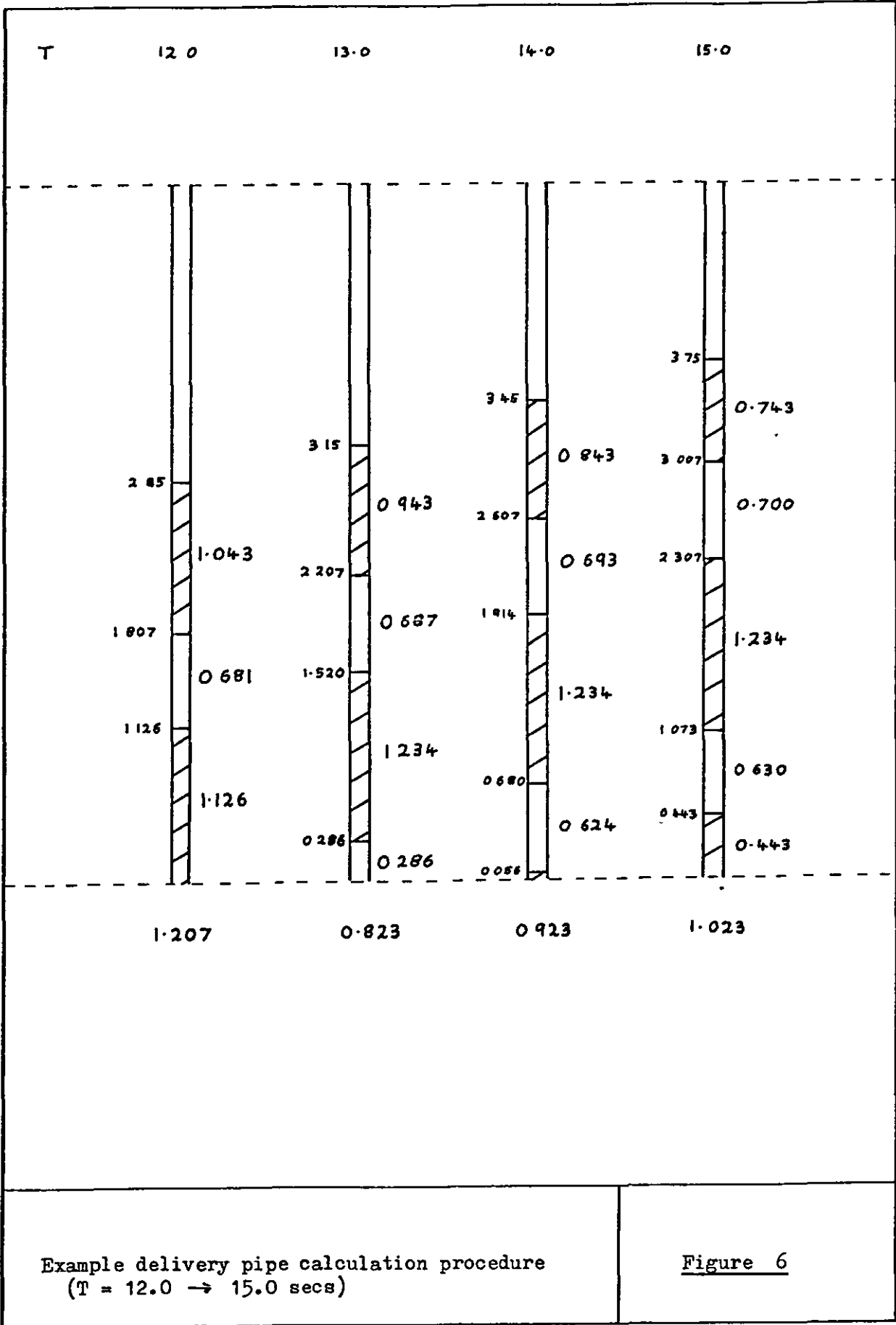
Correlation of theoretical and experimental results is discussed in section 3.7.

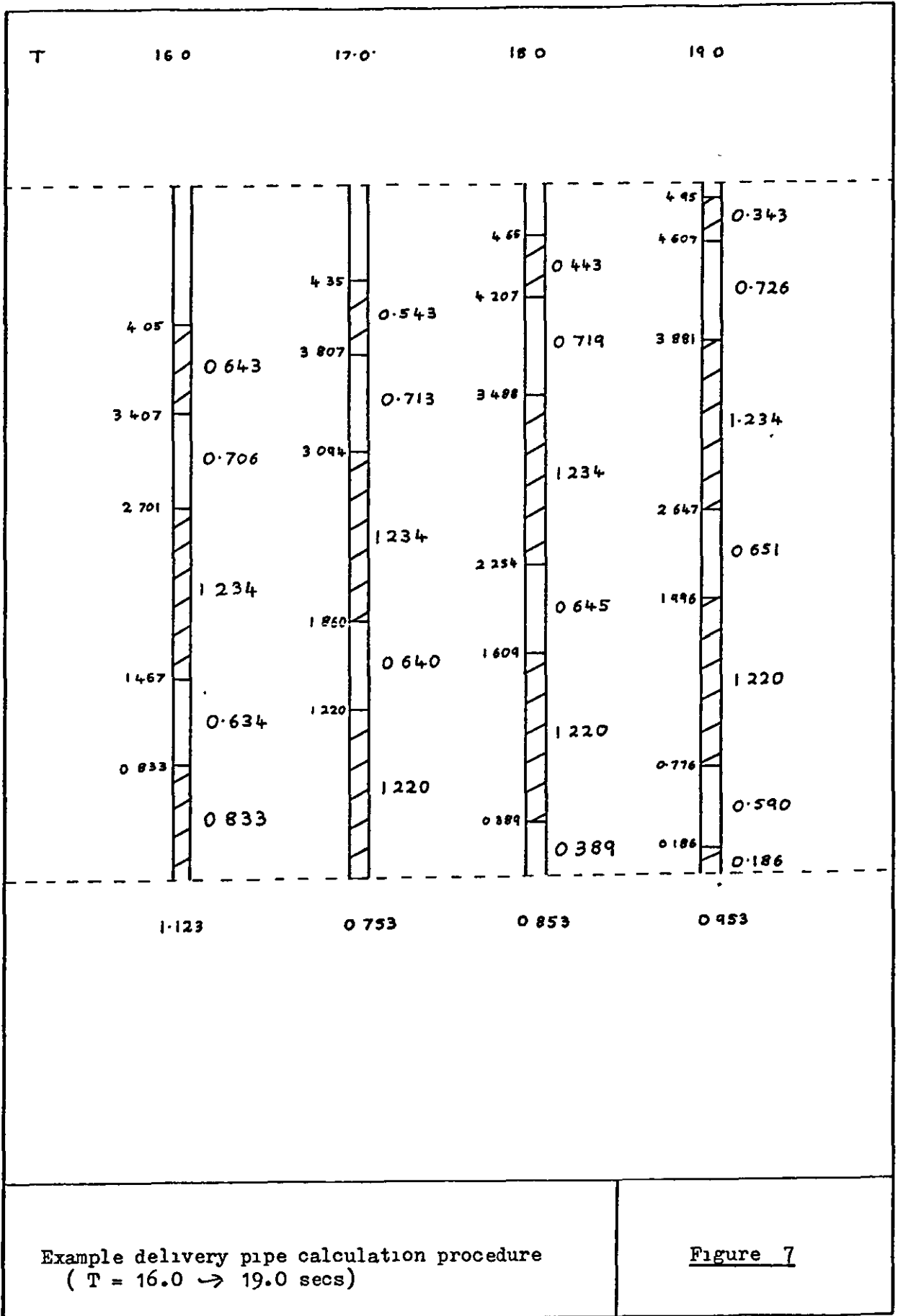
Figures 3 to 9 follow.

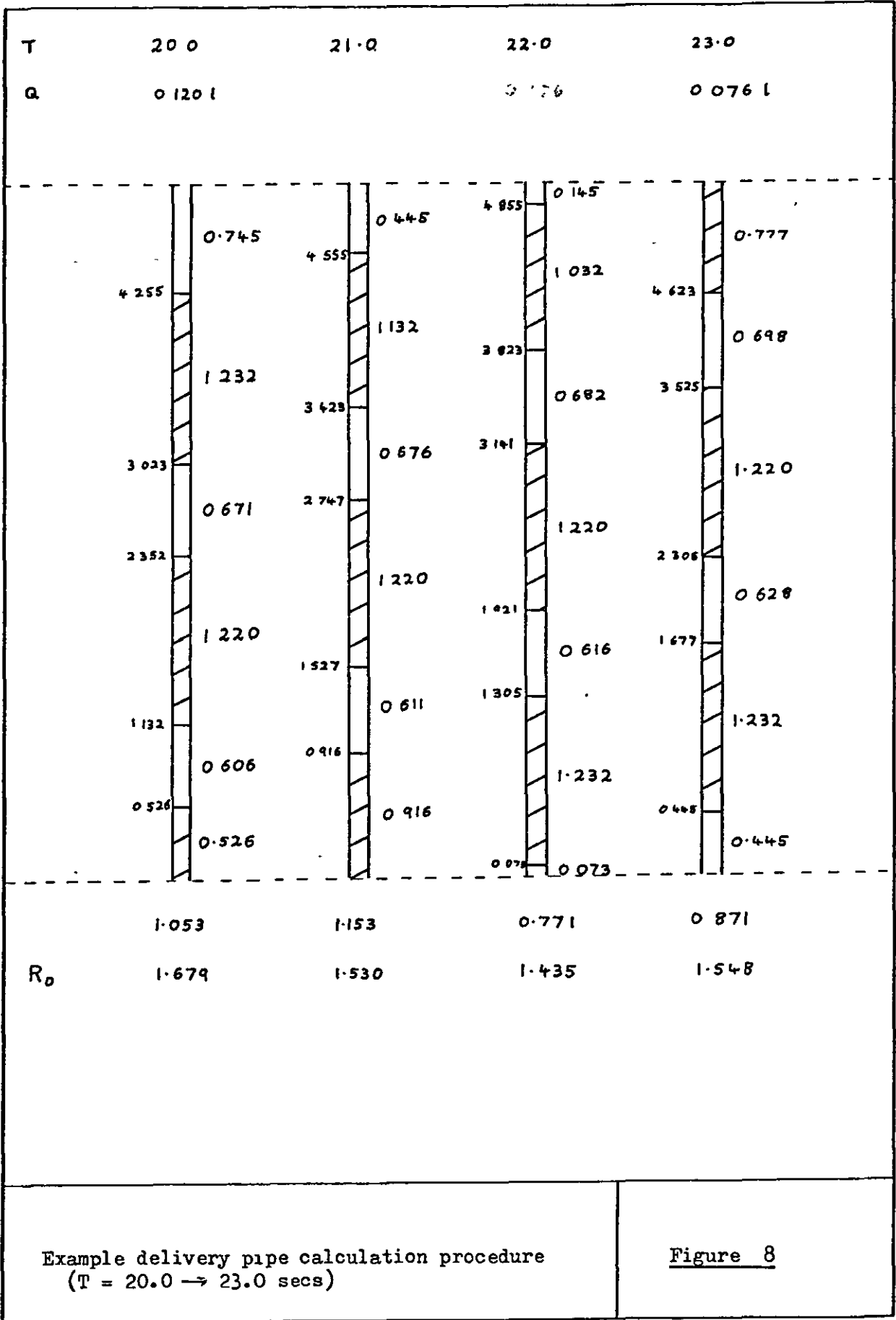






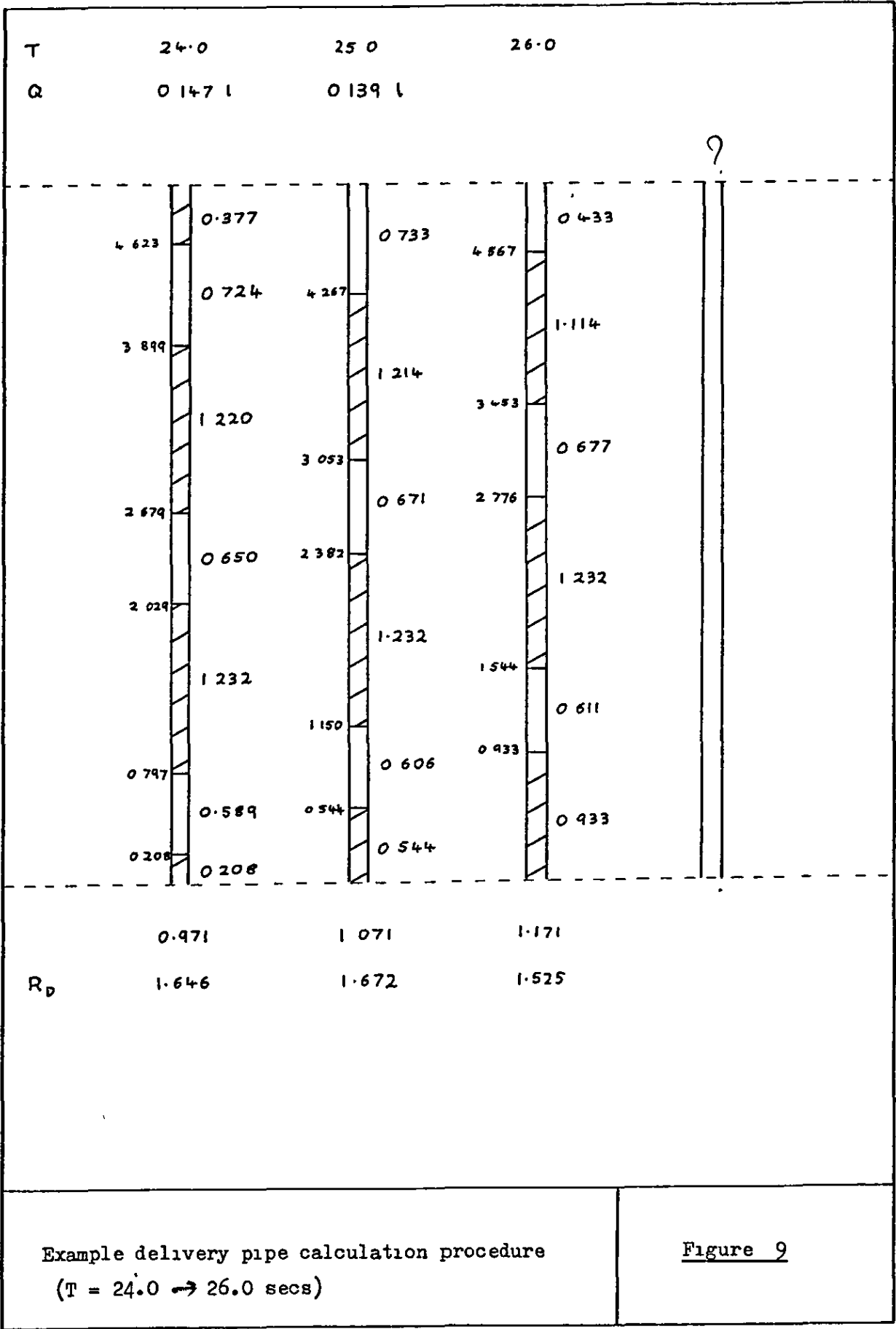






Example delivery pipe calculation procedure
(T = 20.0 → 23.0 secs)

Figure 8



3.4 Analysis of Water Levels in the Coils

The pump consists of n coils, and when in operation head differences are set up across these coils to oppose the head of water in the delivery pipe. These levels are retained when the pump is stopped and can therefore be measured and plotted. Figure 10 below shows two typical head difference patterns for a pump of n coils.

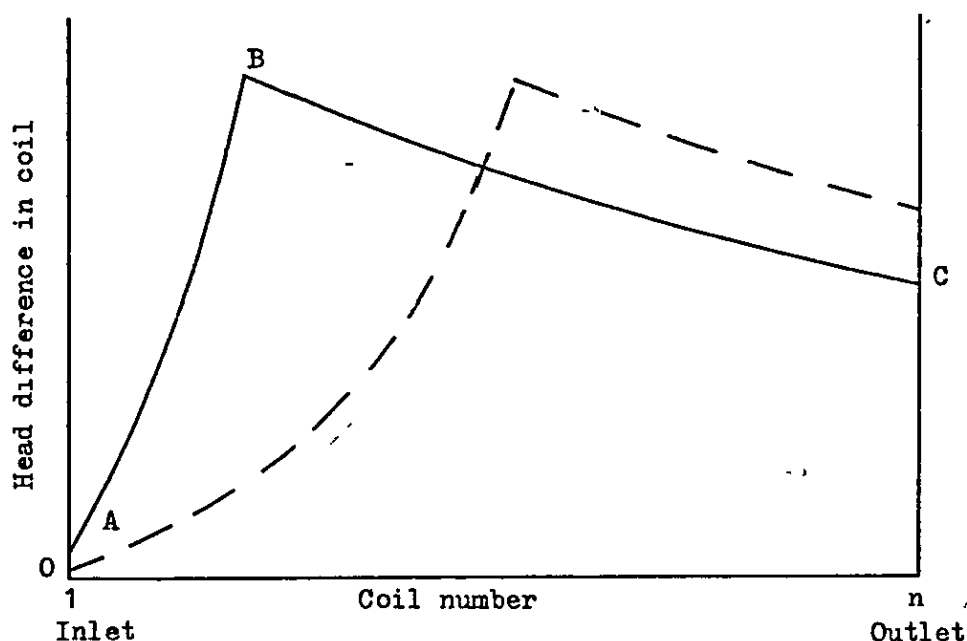


Figure 10

The full line is obtained when the pump is lifting near its maximum limiting height and the broken line is the pattern obtained when the same pump is lifting to a lower height.

Since the head in the delivery pipe is either known or can be worked out using the theory described in the previous section, the

analysis of the pump starts from the last coil and works towards the inlet coil.

The portion B - C of the plot is known as the spillback line. It is so termed because in these coils the upper water levels (i.e. those on the rising side of the pump) are at the crowns of the coils and water can be seen running over the top of the coils and spilling into the lower water levels on the falling side of the pump. By this mechanism water moves from one plug of water back into the previous plug of water, i.e. towards the inlet of the pump.

The assumptions made in the analysis of the water levels in the coils are as follows:-

- i) With no head applied to the delivery of the pump, the velocity of the air and water within the pump's coils is equal and opposite to the peripheral velocity of the mean radius of the coils.
- ii) In order to preserve mass continuity, the velocity of a water/air unit (relative to the pipe wall) is proportional to the length of that unit, i.e. as air is compressed, the length of the unit, and its velocity are reduced.
- iii) Each coil is in a vertical plane, i.e. the horizontal displacement of the coils along the axis of the drum is not considered.
- iv) Spillback does not occur until the water level reaches the crown of the inner wall of a coil, i.e. there is assumed to be no adherence of water to the pipe wall. Such effects would be difficult to quantify and must be ignored at this stage.
- v) The quantity of water spilling from one water plug to another is solely due to the difference in velocities between the water plug and the coil.
- vi) Dynamic losses are assumed to be insignificant.

Consider a pump rotating but with no applied head. The plugs of air and water will retain their inlet lengths and positions as they move along the pump towards the outlet, their position in space (viewed along the axis of the pump) will remain constant and their rotational velocity will be equal and opposite to that of the coils.

As the head at the outlet is increased the air plugs will become compressed according to the equation:-

$$a_c = \left(\frac{H_{Atm}}{H_{Atm} + H} \right) \times a_o$$

For the last coil (c_n) the value of H will be the head being applied from the delivery pipe, for successive coils towards the inlet the value of H will be the applied head less the sum of the head differences in the coils between the considered coil and the outlet, so for any coil (c_m) the head is given by

$$H_m = H - \sum_{i=m+1}^n h(i) \quad \text{where } h = \text{head differences across a coil}$$

Therefore an air plug becomes more compressed as it approaches the outlet and this produces a slowing effect in order to satisfy continuity laws, so for a unit of length $w_o + a_c$, the velocity relative to the coils is given by

$$v_c = v_o \times \left(\frac{w_o + a_c}{w_o + a_o} \right) = v_p \left(\frac{w_o + a_c}{w_o + a_o} \right)$$

$$\therefore v_c = \frac{v_p}{2 \pi R_c} \left[w_o + a_o \left(\frac{H_{Atm}}{H_{Atm} + H} \right) \right]$$

Therefore the plugs slow down as they approach the outlet and so move backward relative to the coil thus setting up static head differences across the coils.

As the applied head is further increased, the trailing end of the plug of water (i.e. the upper water level) will reach the crown of the coil and water will begin to spill over the top and into the previous plug of water. In practice, spillback begins before this upper water level reaches the crown of the coil, due to water adhering to the inside of the pipe and being carried over. The amount of water involved increases as the distance from the water surface to the crown of the coil decreases until full spillback occurs as the water level moves above the highest level of the inner wall of the pipe.

Once the upper water level is at the crown of the coil, the amount of spillback is governed by the relative velocity deficiency between the plug of water and the piping of the coils.

As before:-

$$v_c = \frac{v_p}{2 \pi R_c} \cdot \left[w_o + a_o \left(\frac{H_{Atm}}{H_{Atm} + H} \right) \right]$$

Relative velocity deficiency $v_r = v_p - v_c$

$$\begin{aligned} \therefore v_r &= \frac{2 \pi N R_c}{60} - \frac{v_p}{2 \pi R_c} \cdot \left[w_o + a_o \left(\frac{H_{Atm}}{H_{Atm} + H} \right) \right] \\ &= \frac{2 \pi N R_c}{60} - \frac{N}{60} \cdot \left[w_o + a_o \left(\frac{H_{Atm}}{H_{Atm} + H} \right) \right] \\ &= \frac{N}{60} \cdot \left[2 \pi R_c - w_o - a_o \left(\frac{H_{Atm}}{H_{Atm} + H} \right) \right] \end{aligned}$$

This is the rate at which water is being lost from the trailing end of the water plug.

Therefore in one revolution, the length lost due to spillback is given by:-

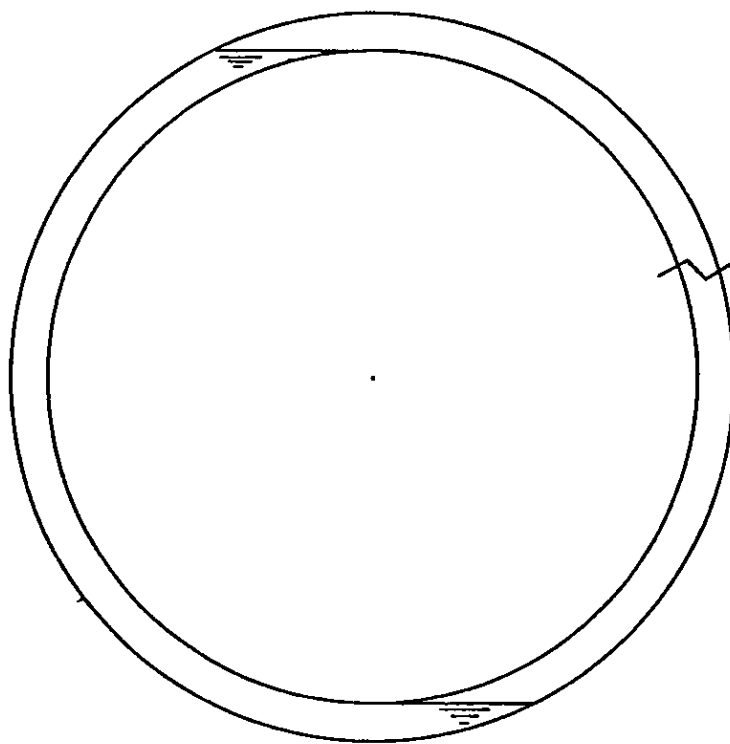
$$L = T \times v_r$$

$$\text{Since } T = \frac{60}{N}$$

$$\text{Then } L = 2\pi R_c - w_o - a_o \left(\frac{H_{Atm}}{H_{Atm} + H} \right)$$

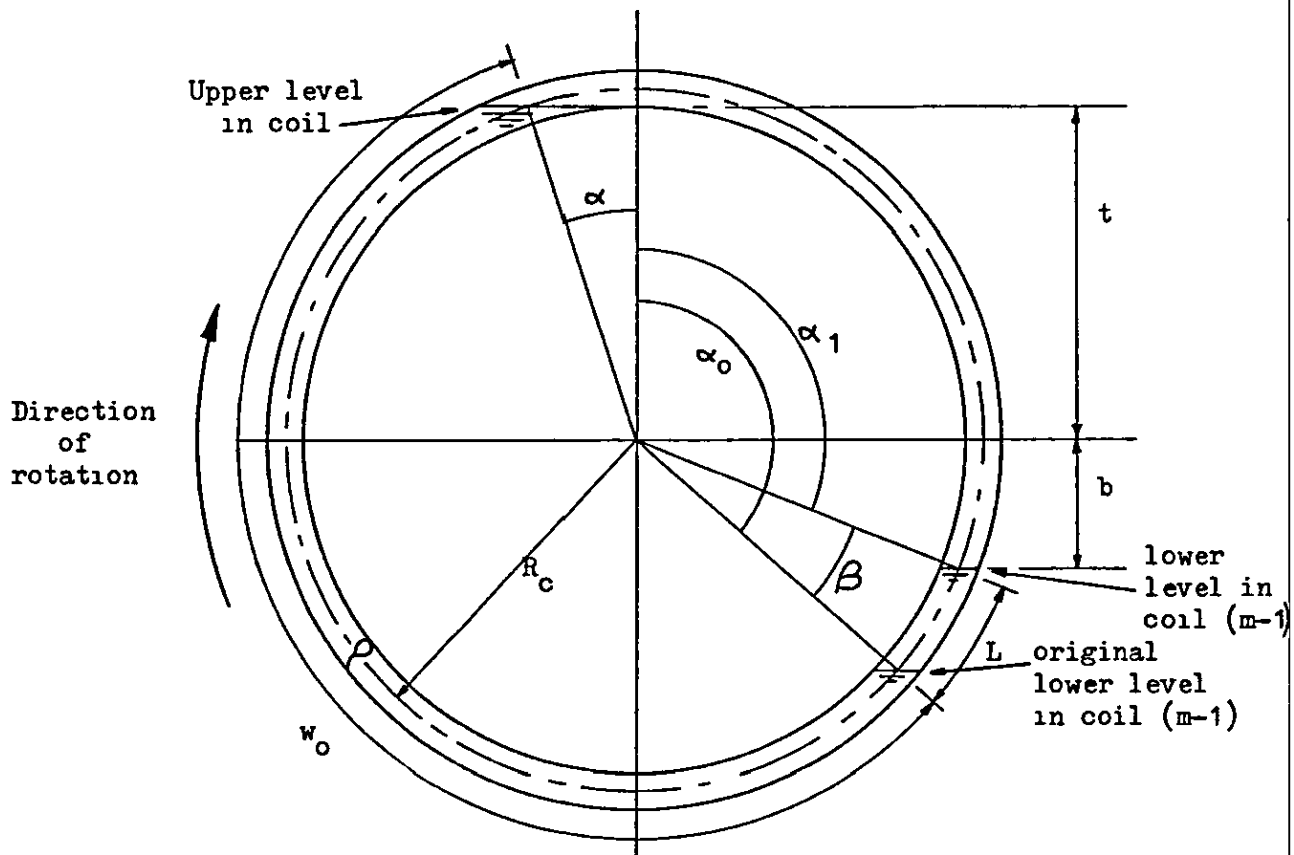
This is the amount of water lost in one revolution and is independent of the speed of rotation of the pump. The water has been lost from the plug nearest the outlet and gained by the previous plug, but because the time interval concerns one revolution of the pump, this plug now occupies coil n.

In order to calculate the head difference in this coil we need to know the position of both the upper and lower water levels. The upper is at the crown of the coil, the position of the lower level is determined by considering its initial position and the effect of the water gained by spillback.



The above diagram shows the theoretical initial positions of both the upper and lower water levels in the coil for a depth of immersion of 0.5. However, as the plug of water moves into the present coil it gains the length of water L from the previous plug and this raises the lower water level. As this lower level is raised the plug of water has to move further round the coil to preserve its original head difference and this may start spilling before it occupies the last coil. This process of spillback will continue until a steady state is reached, the head differences forming the characteristic shape shown in Figure 10.

The following diagram illustrates how the amount of spillback determines the head difference across a coil.



$$t = R_c - \frac{d_p}{2}$$

$$\alpha = \cos^{-1} \left(-\frac{t}{R_c} \right)$$

$$\alpha_0 = 2\pi - \alpha - \frac{w_0}{R_c}$$

Since $\beta = \frac{L}{R_c}$

then $\alpha_1 = \alpha_0 - \beta$

$$b = R_c \cdot \sin \left(\alpha_1 - \frac{\pi}{2} \right)$$

$$h_{m-1} = t + b$$

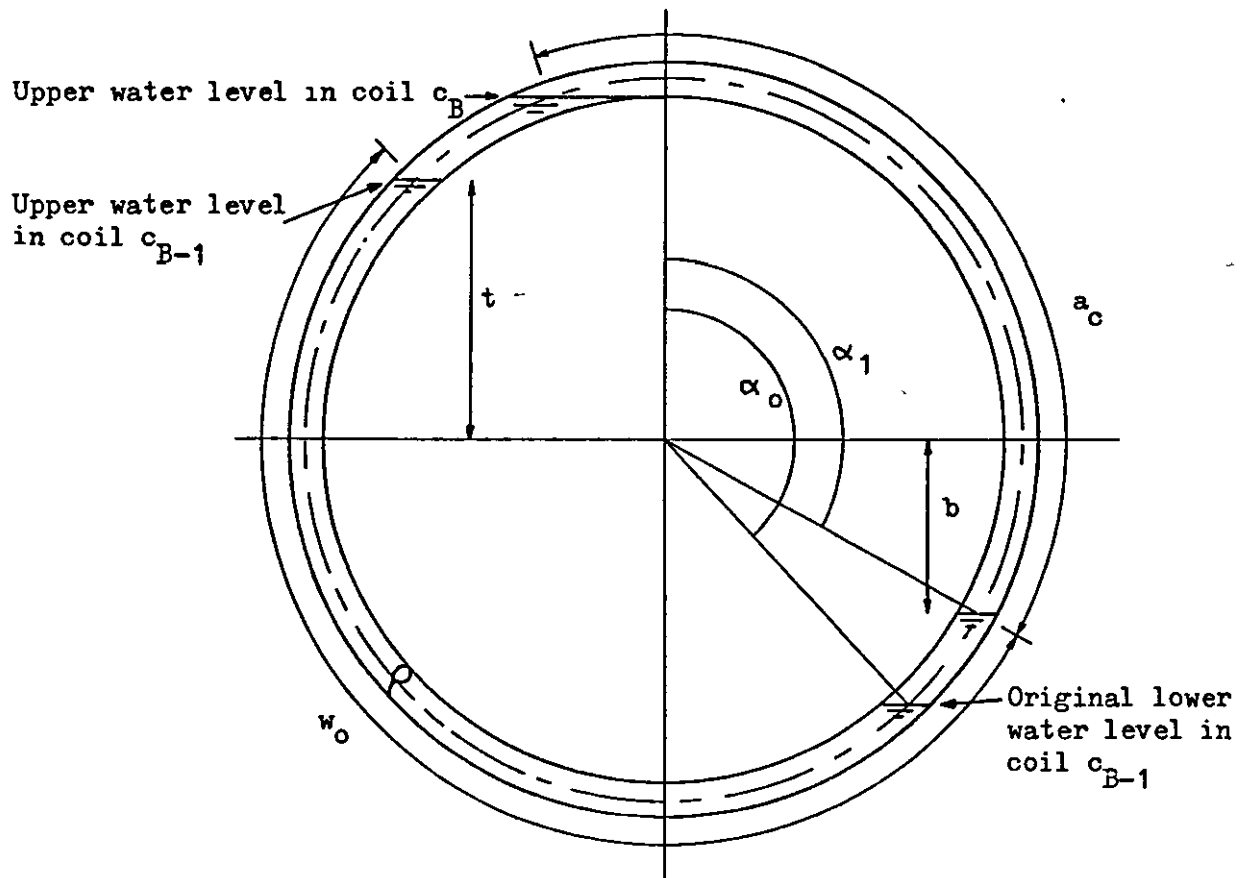
This gives the head difference in coil $m-1$ which is now taken as coil m because the pump has moved on one revolution.

Similar calculations may be carried out for each of the n coils, but for each coil:-

$$H_m = H - \sum_{i=m+1}^n h(i)$$

These calculations deal with the spillback line, portion B - C of the plot in Figure 10. From B to A the levels in the coils decay until the pressure in the pump drops to atmospheric at the inlet. No spillback takes place in these coils as the upper water levels are no longer at the crown of the coils, therefore the lengths of the plugs of water are as they were at the inlet although the air plugs are still subject to compression from the pressure in the pump. Since no spillback is involved, these coils are simpler to analyse, the position of the levels being dependant upon the compressed lengths of the air plugs. Consider the coil at B (c_B) which contains the maximum head difference in the pump, water is spilling into this coil from coil c_{B+1} but, although the upper water level in coil c_B is near or at the crown, it is assumed that no spillback takes place from this coil, so this is the last coil in which the head difference is calculated using the spillback mechanism. In the coils from c_B to the inlet, the head difference calculation involves determining the position of the lower water level and combining this with the known length of the water plug to locate the position of the upper water level.

(Diagram overleaf)



Again the original lower water level is used, the value of α_0 being the same as was used in the coils involved in the spillback line.

$$a_c = a_o \left(\frac{H_{Atm}}{H_{Atm} + H} \right)$$

$$\alpha_1 = \alpha_o + \frac{w_o}{R_c} + \frac{a_c}{R_c} - 2\pi$$

$$b = R_c \cdot \sin \left(\alpha_1 - \frac{\pi}{2} \right)$$

$$t = R_c \cdot \sin \left(\alpha_1 + \frac{w_o}{R_c} - \frac{3\pi}{2} \right)$$

$$h_{(B-1)} = t + b$$

For the next coil the starting value of α (α_0 in this case) is the value of α_1 in this coil.

These calculations give a decay line for the portion of the plot B - A and the severity of the decay is related to the pressure in the pump at point B. In the program written to perform these calculations (see Appendix III) the decay line is determined for each point along the spillback line. When certain conditions are met (see below) the program will stop and give the number of coils required on a pump of given dimensions to develop the head which is fed into the program.

The conditions are that the head difference across the inlet coil should be less than $0.15 R_c$. This value has been found in practical tests to be the maximum that the pump can contain before it begins to stall, i.e. a head difference of over this value in the first coil indicates that the pump is developing on or above its maximum head and is in danger of being overloaded. The other condition is that the residual head at the inlet should be near to atmospheric pressure. The discrepancy allowed depends on the size of the pump and the head being developed but should be of the order of $0.05 H$.

3.5 Presentation of Results

A total of sixty-four tests were done on the laboratory model of the pump, covering the range of configurations outlined in section 2.0.

Graphs 1 to 3 show the experimental and theoretical values of the delivery pipe ratio R_D for all the tests. The results are grouped and labelled according to the particular parameter being varied, and successive points within a group represent the increased height of delivery for that configuration of pump. Tests 60-64 are supplementary

tests carried out on the large drum to confirm agreement of experimental and theoretical results.

The results for the head difference patterns need a separate graph for each group of tests and also for each comparison of experimental and theoretical values. Therefore only one or two typical comparisons from each group are shown and these serve to demonstrate the general case.

Test numbers

Parameter varied	Test numbers
Depth of immersion	1 - 25
Speed of rotation	26 - 36
Bore of delivery pipe	37 - 41
Bore of coils	42 - 52
Diameter of coils	53 - 59
Depth of immersion for large drum	60 - 64

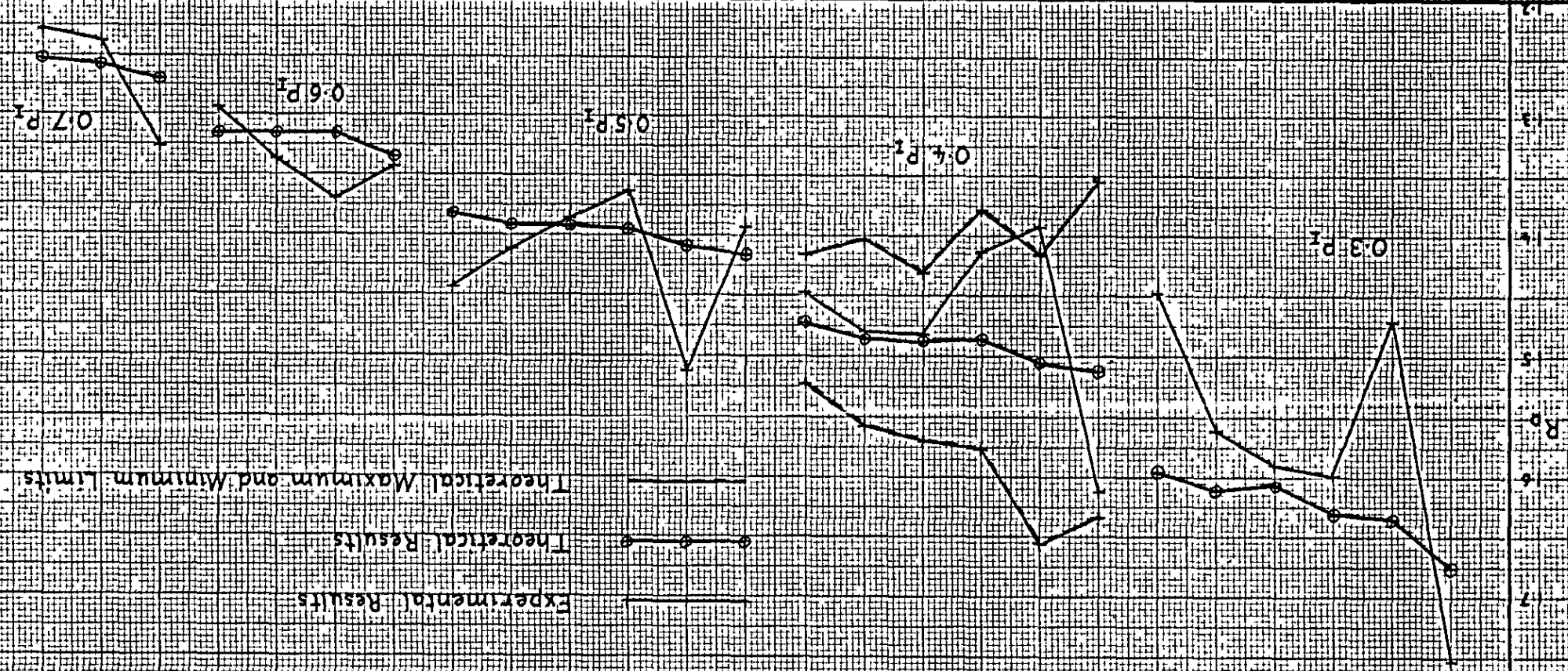
3.6 Experimental and Theoretical Results

Graphs 1 to 40 follow overleaf.

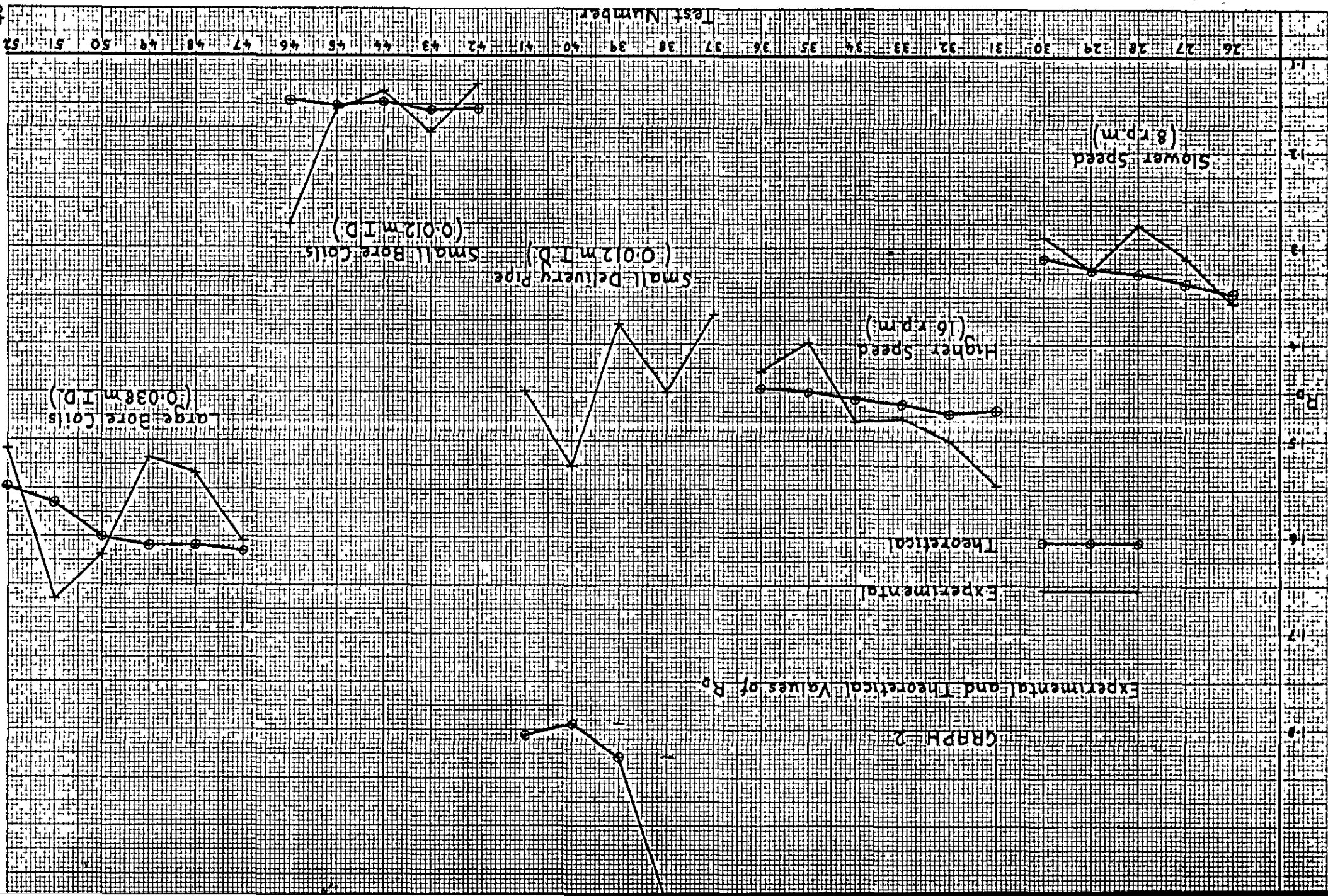
Experimental and Theoretical Values of R_D

GRAPH 1

Experimental Results
Theoretical Results
Theoretical Maximum and Minimum Limits

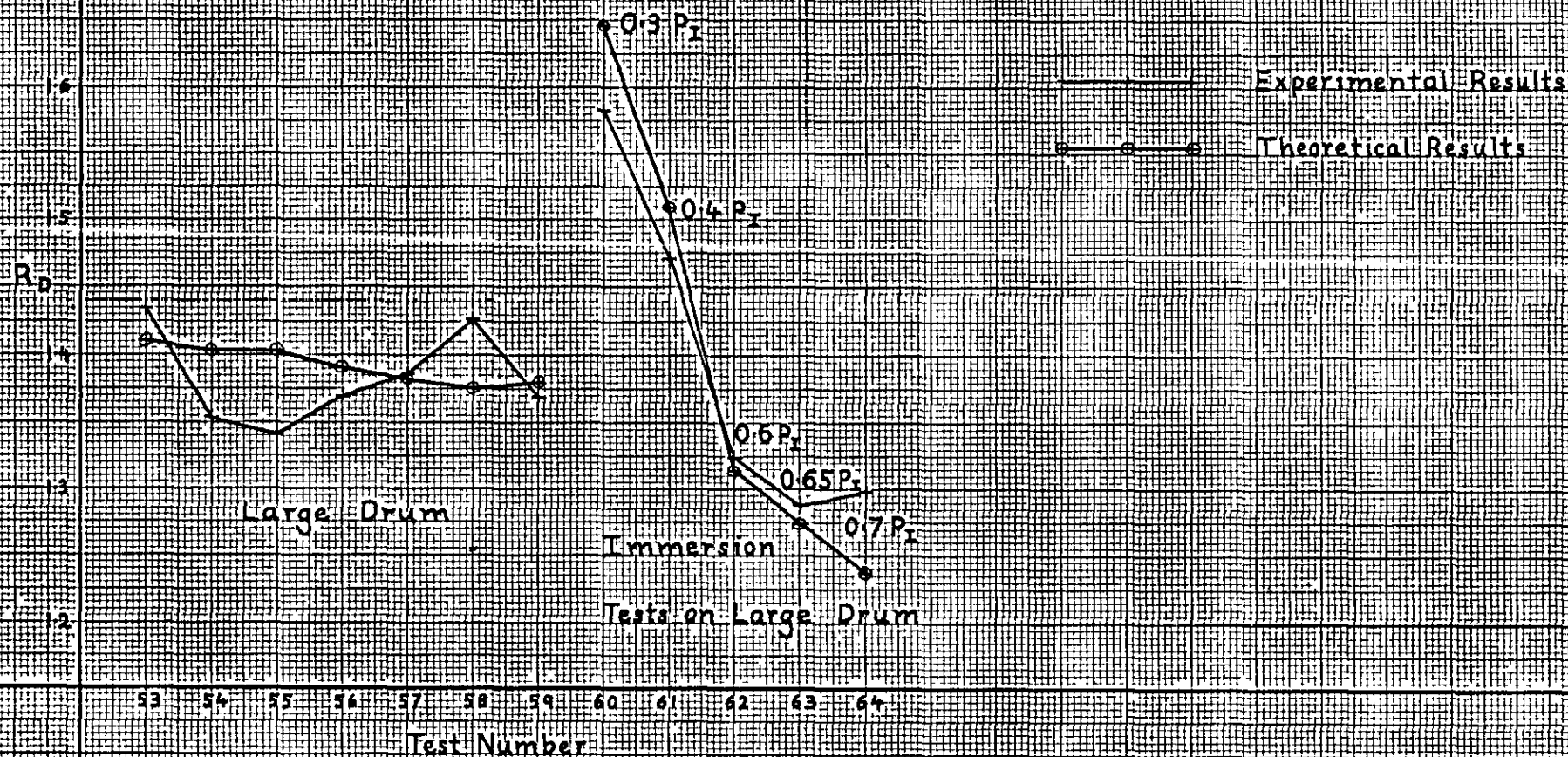


Test Number



GRAPH 3

Experimental and Theoretical Values of R_D

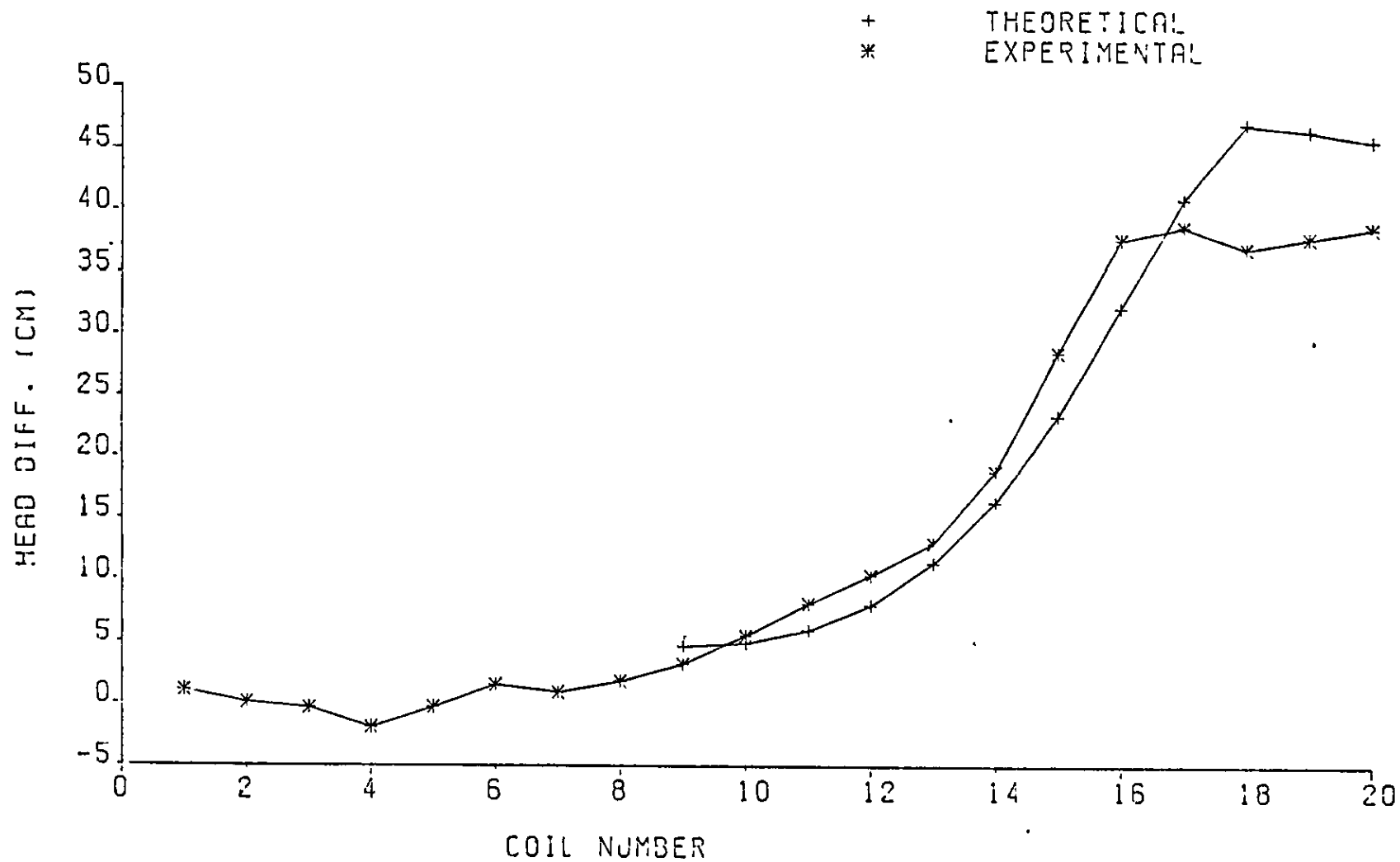


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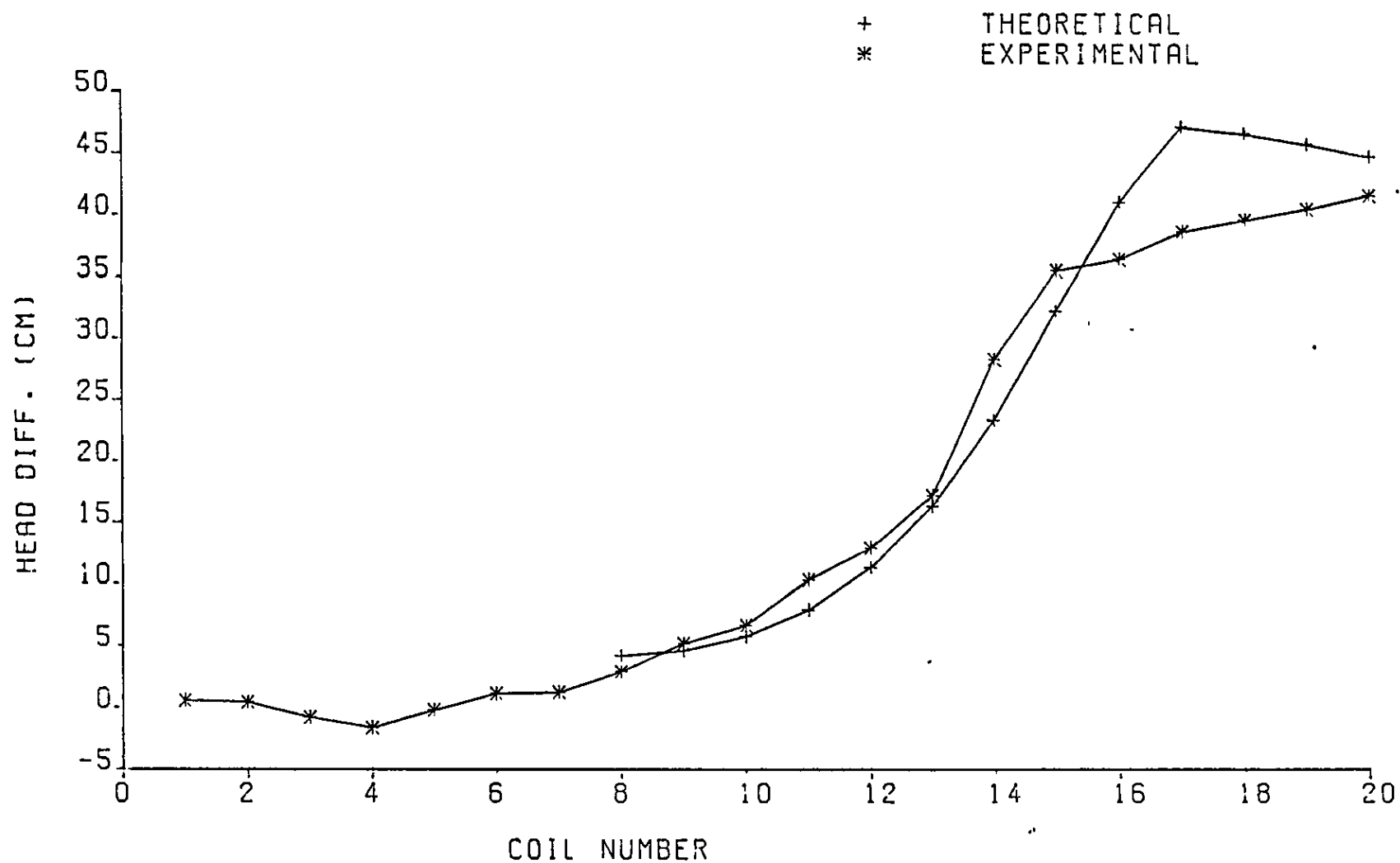
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0.3 IMMERSION , 4M. LIFT

GRAPH 4

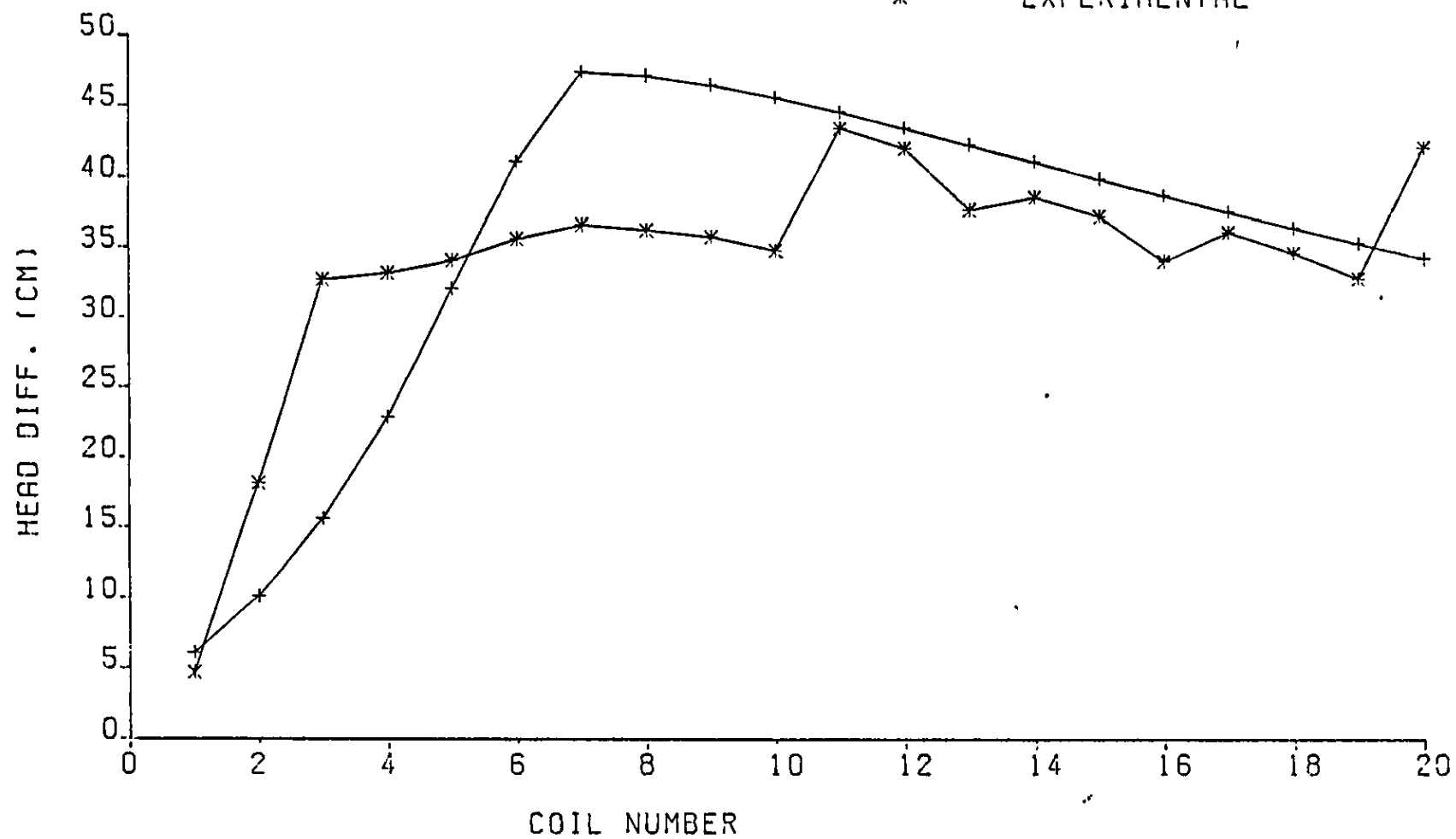


0.3 IMMERSION , 5M. LIFT GRAPH 5

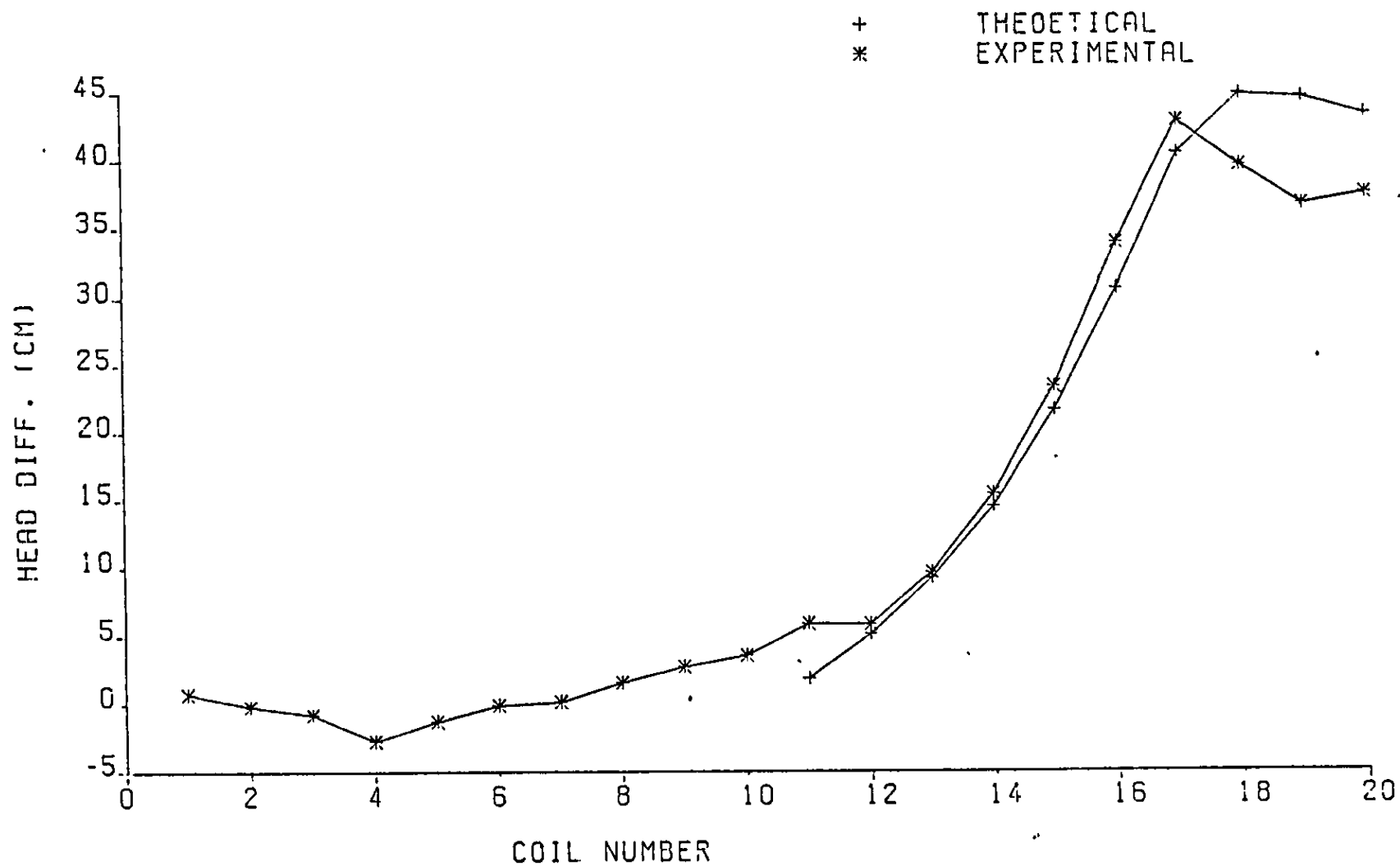


0.3 IMMERSION , 10M. LIFT GRAPH 6

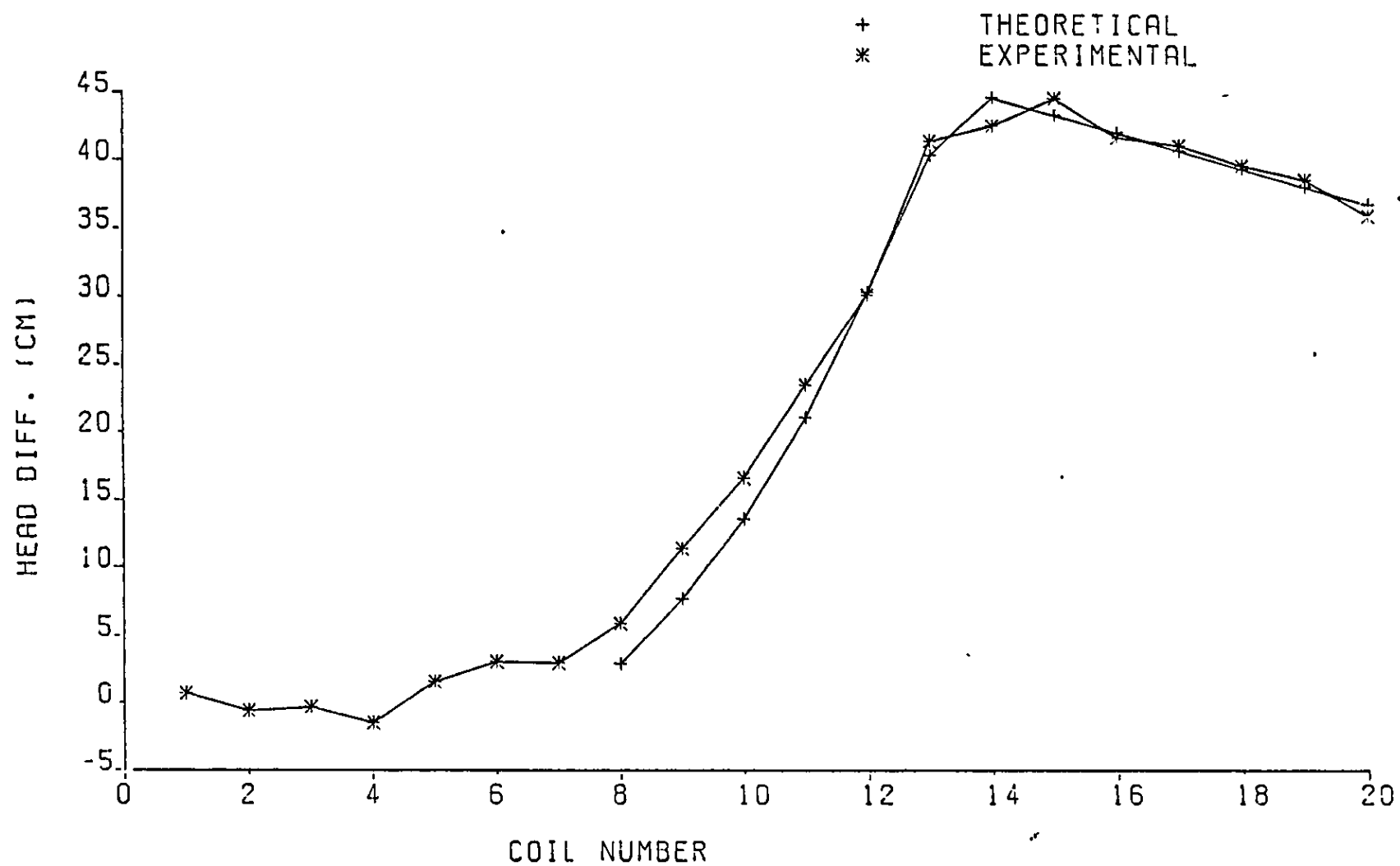
+ THEOETICAL
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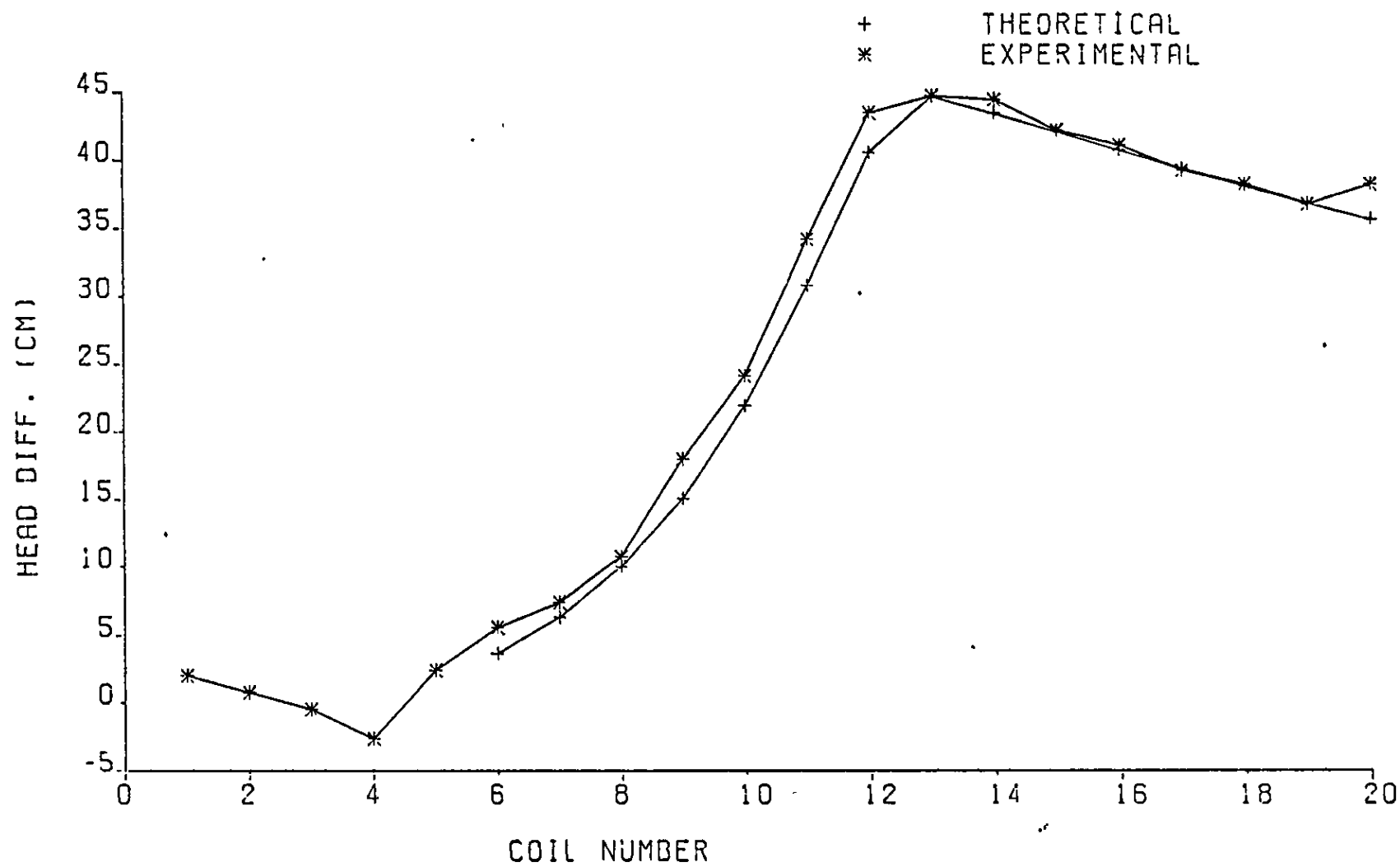
0.4 IMMERSION , 4M. LIFT GRAPH 7



0.4 IMMERSION , 6M. LIFT GRAPH 8

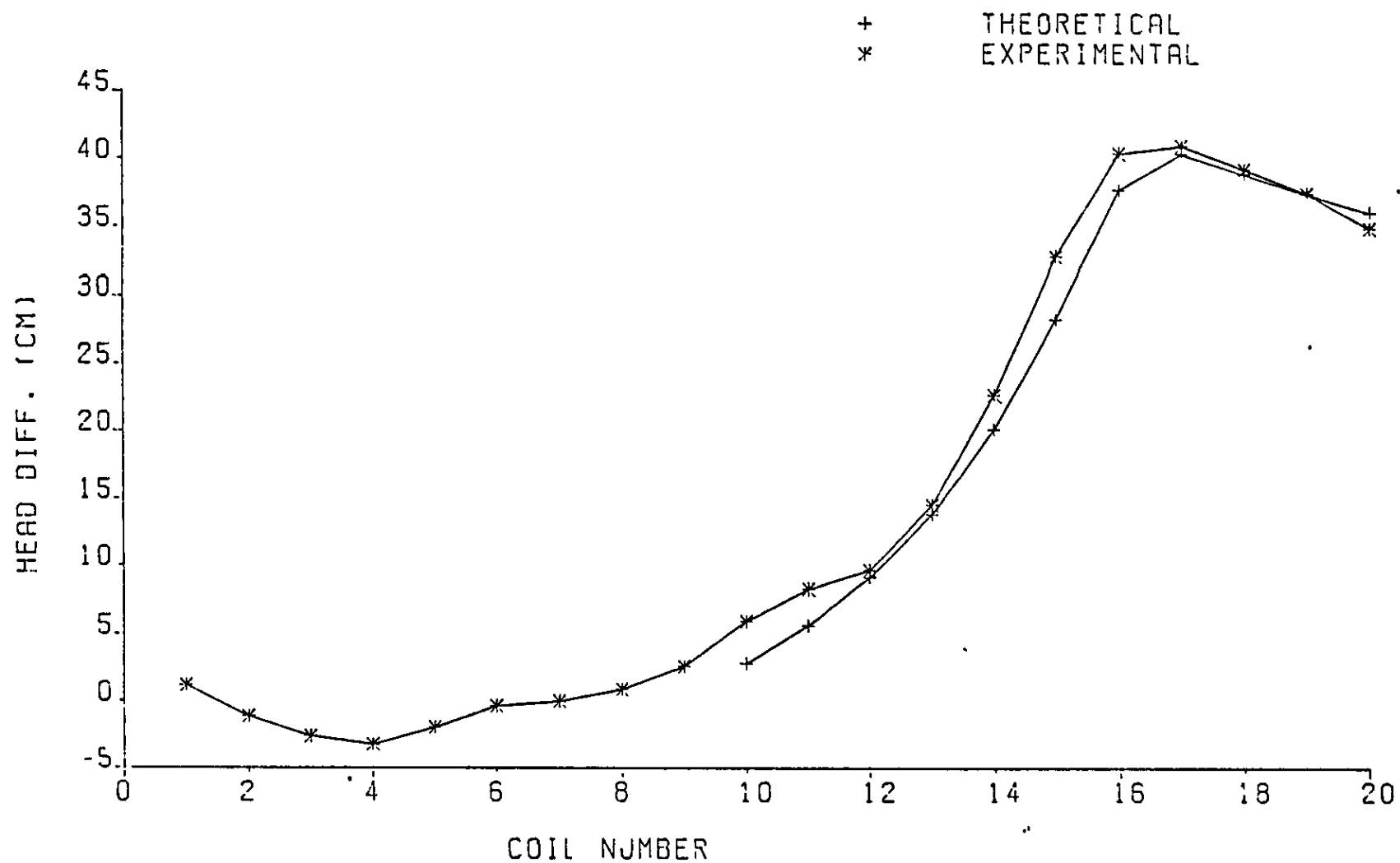


0.4 IMMERSION , 7M. LIFT GRAPH 9

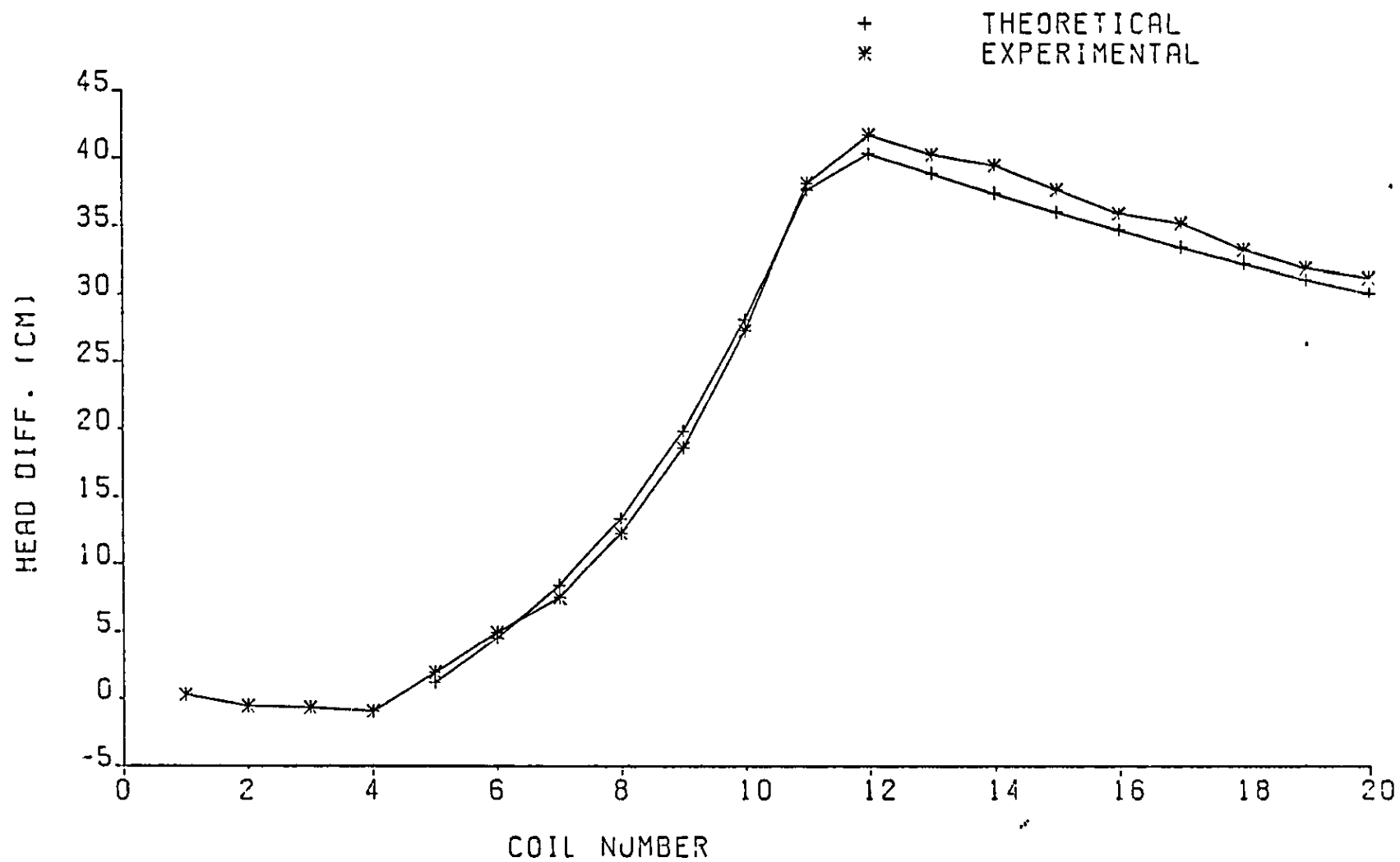


0.5 IMMERSION , 4M. LIFT

GRAPH 10

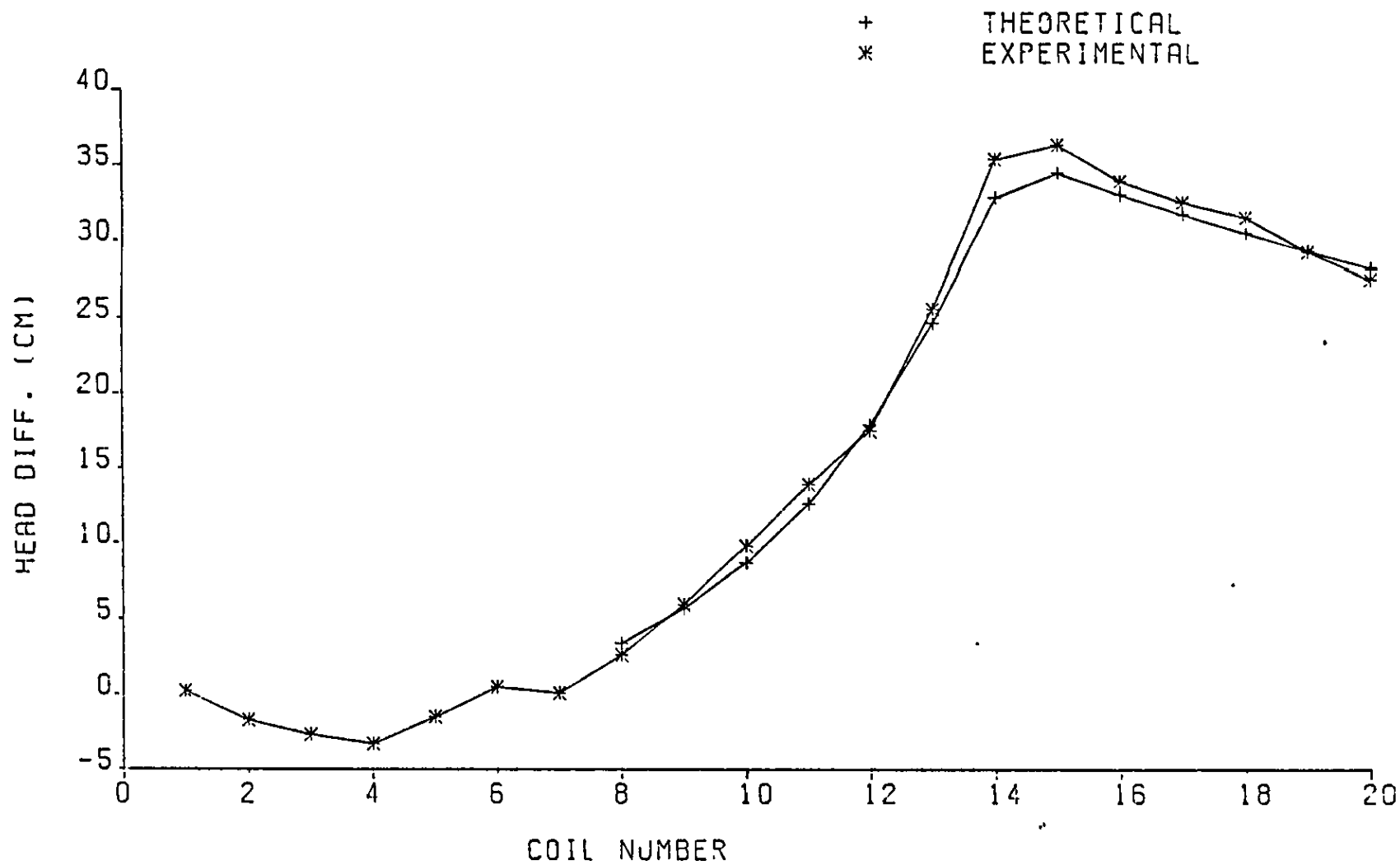


0.5 IMMERSION , 6M. LIFT GRAPH 11

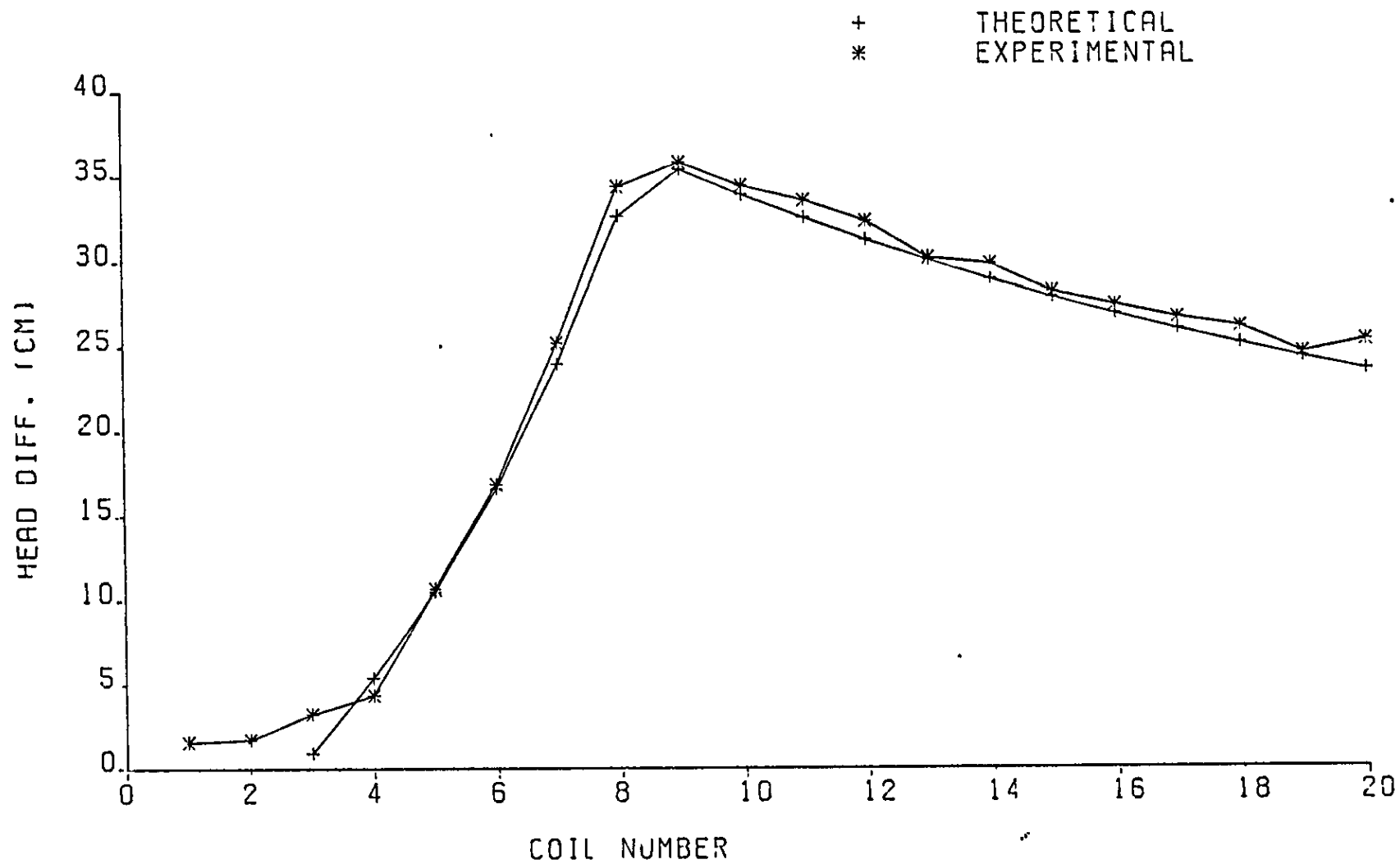


0.6 IMMERSION , 4M. LIFT

GRAPH 12

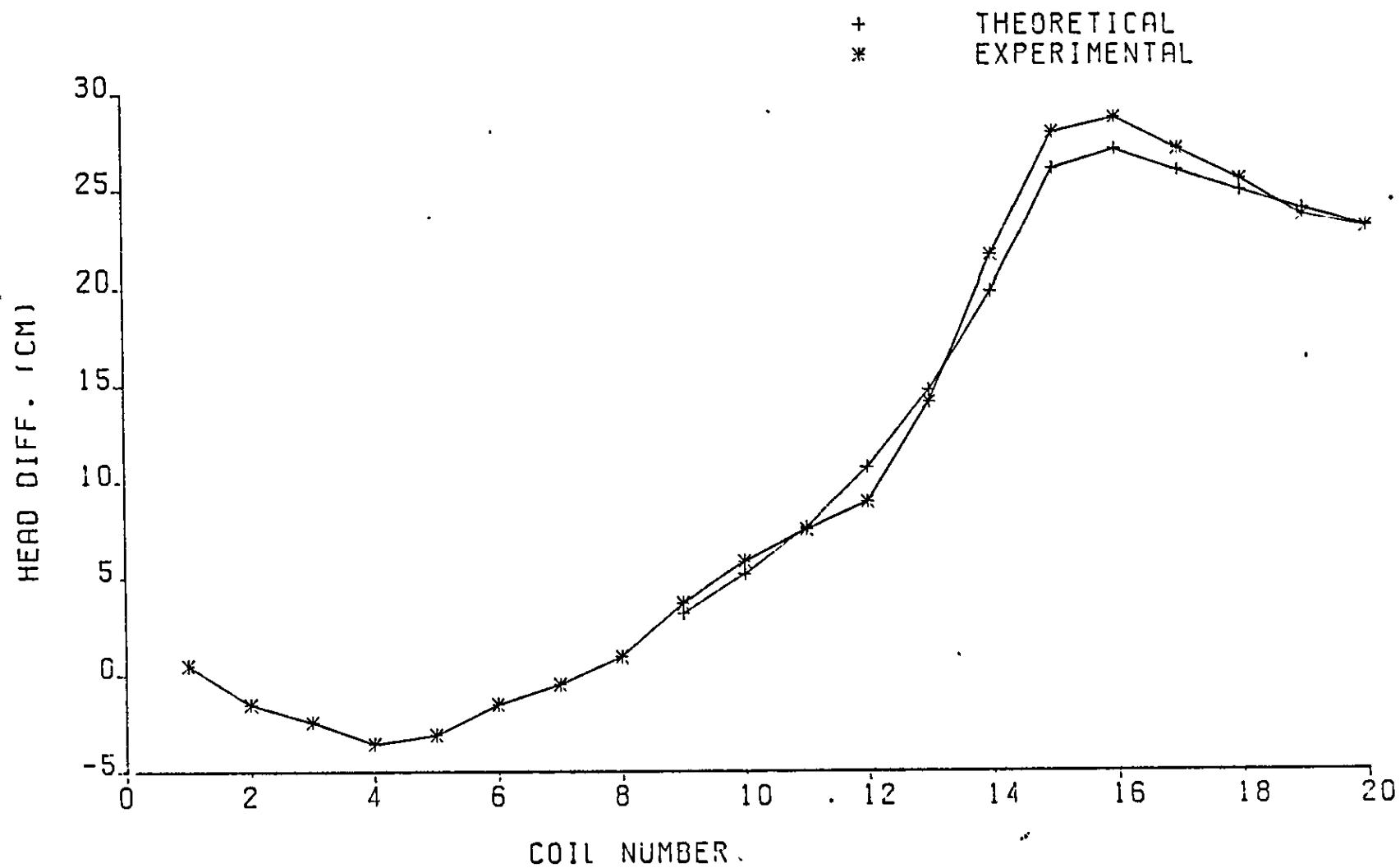


0.6 IMMERSION , 6M. LIFT GRAPH 13



0.7 IMMERSION , 3M. LIFT

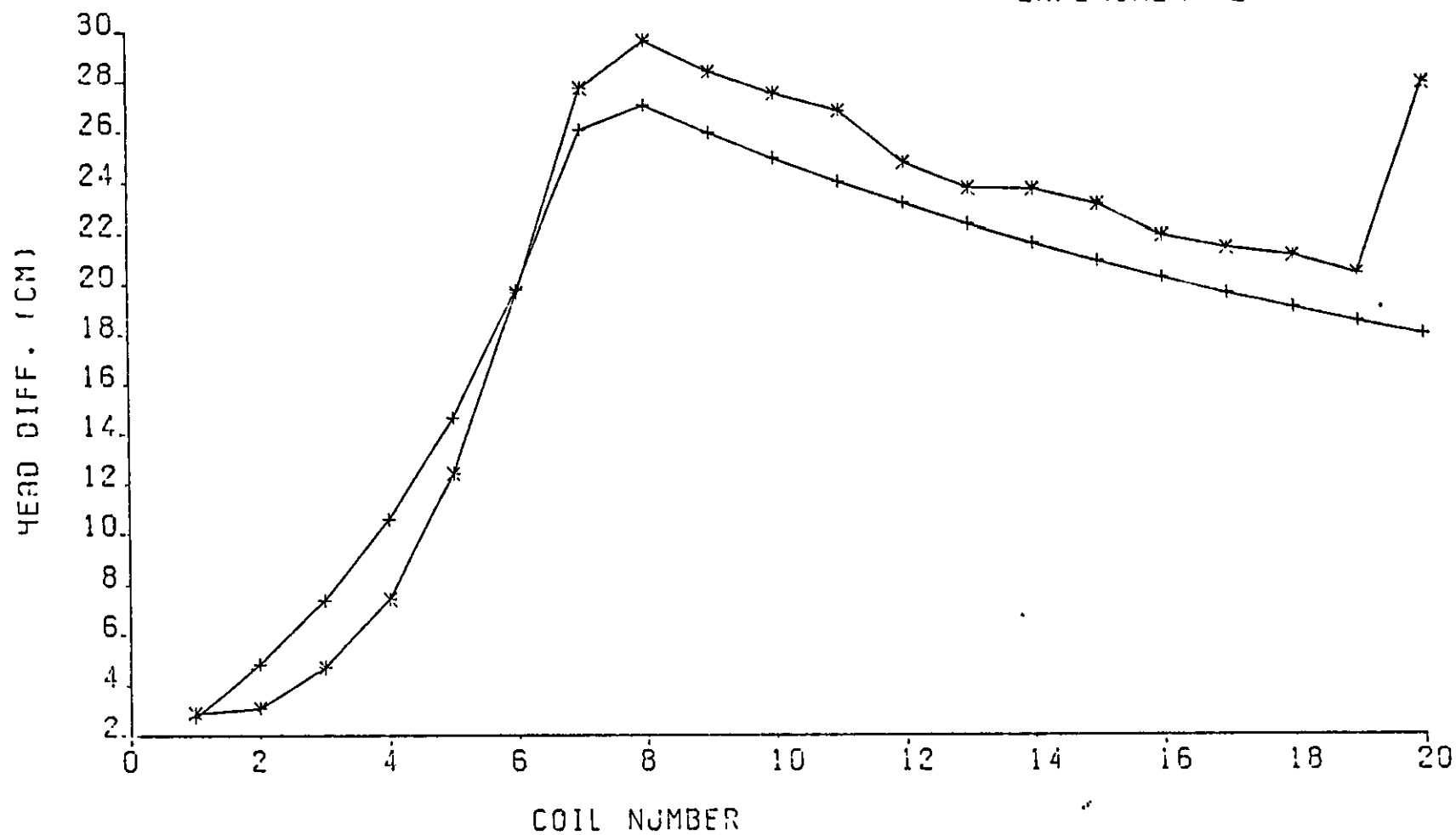
GRAPH 14



0.7 IMMERSION , 5M. LIFT

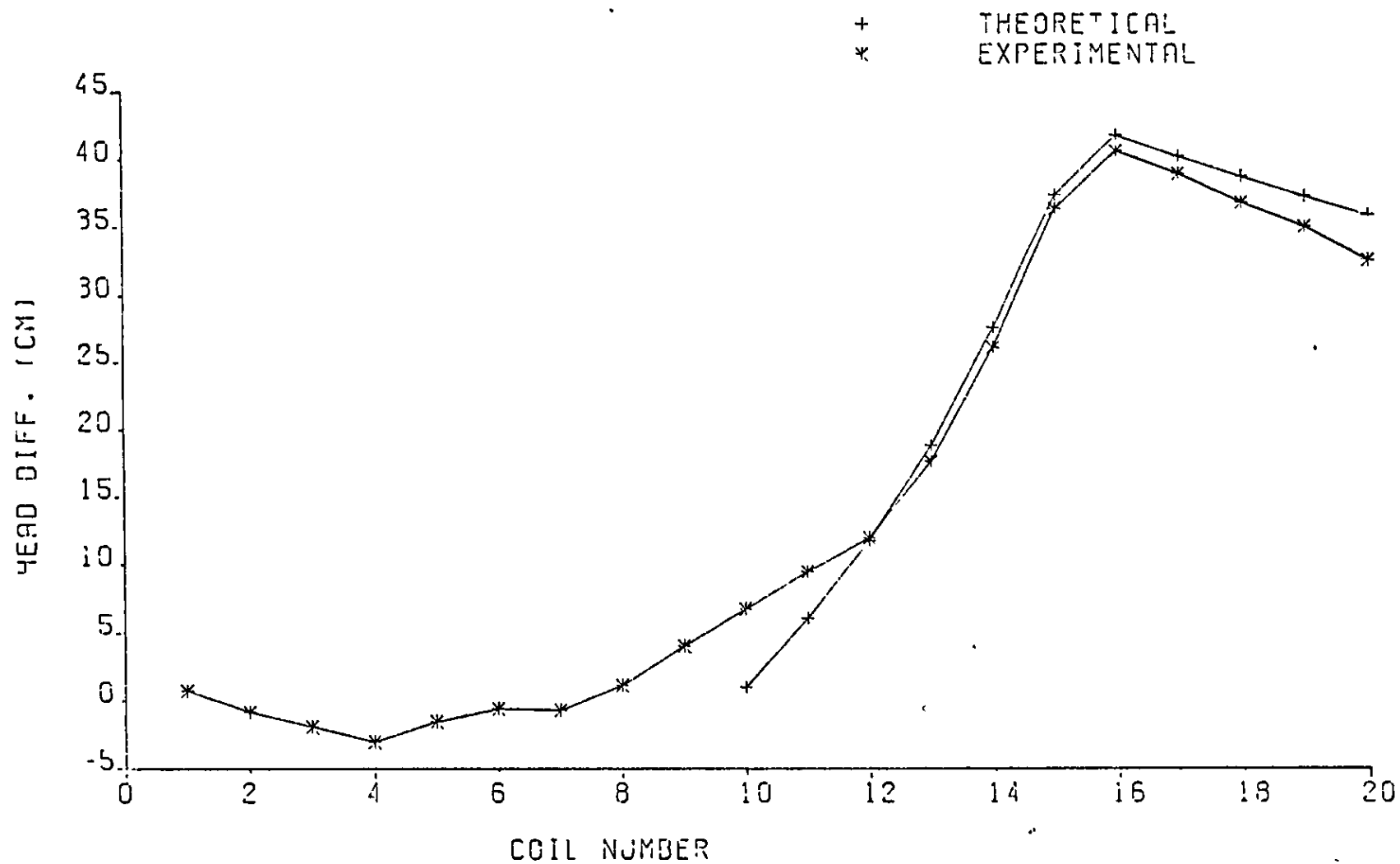
GRAPH 15

+ THEORETICAL
* EXPERIMENTAL

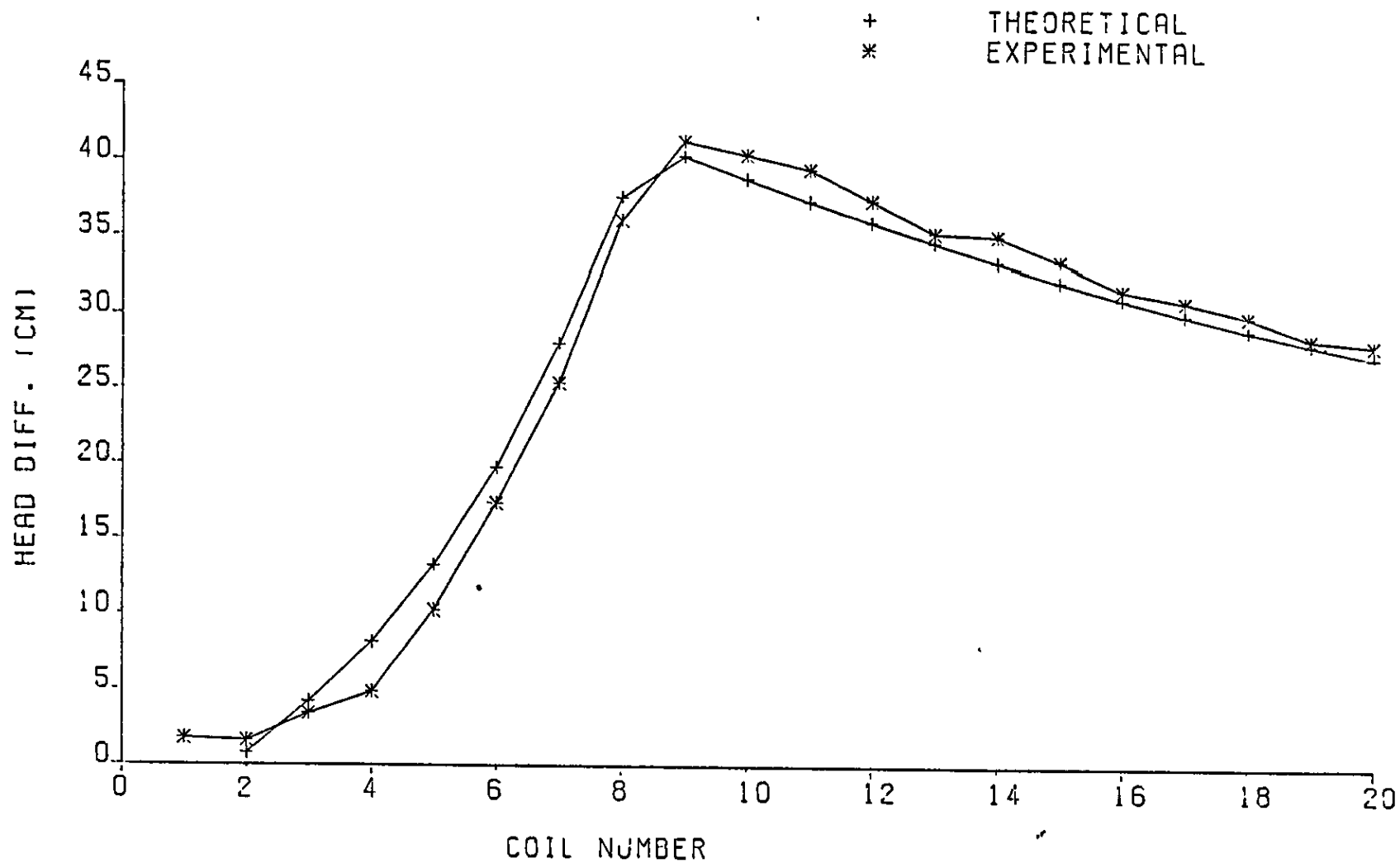


SLOWER SPEED , 4M. LIFT

GRAPH 16

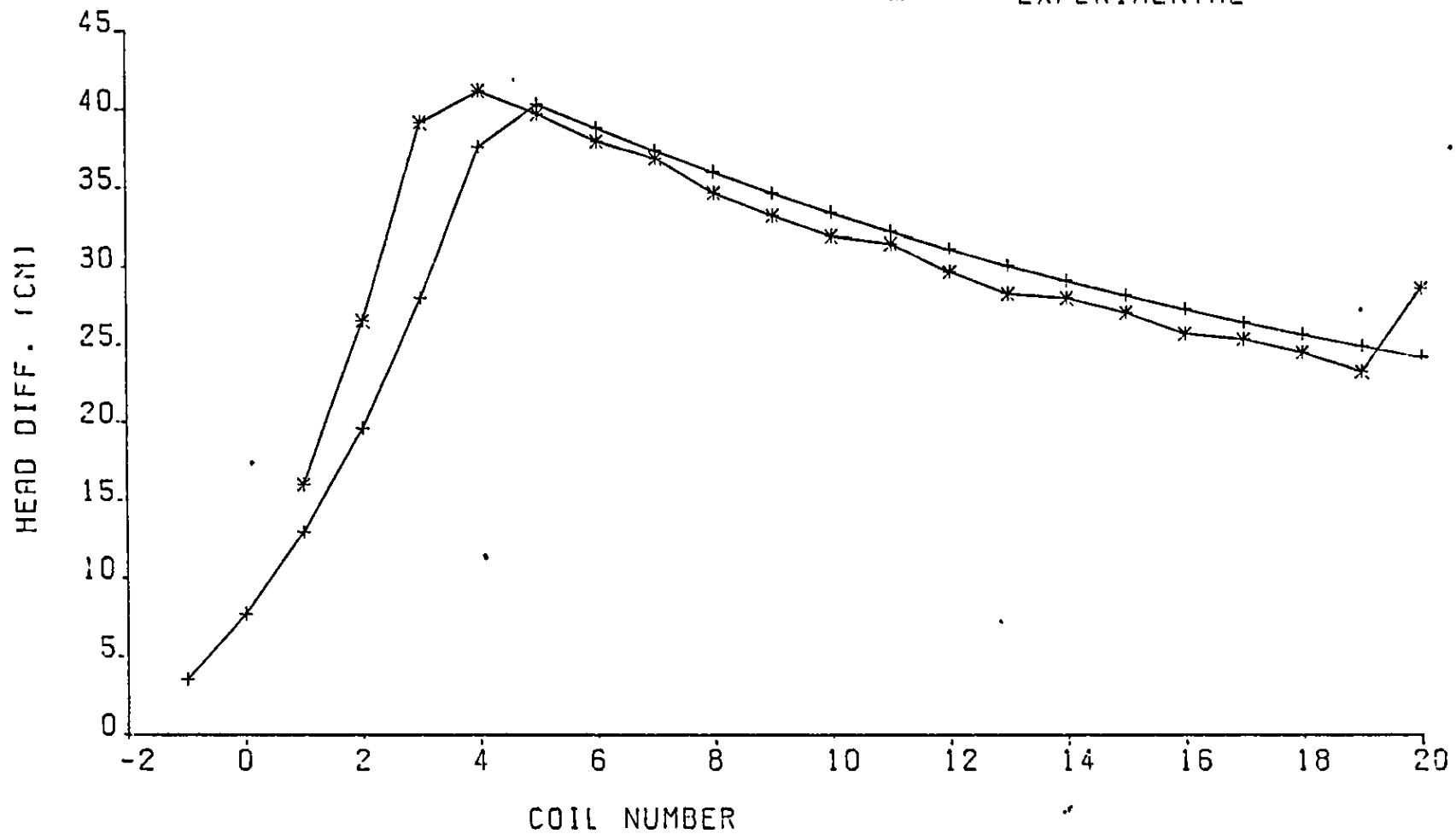


SLOWER SPEED , 7M. LIFT GRAPH 17

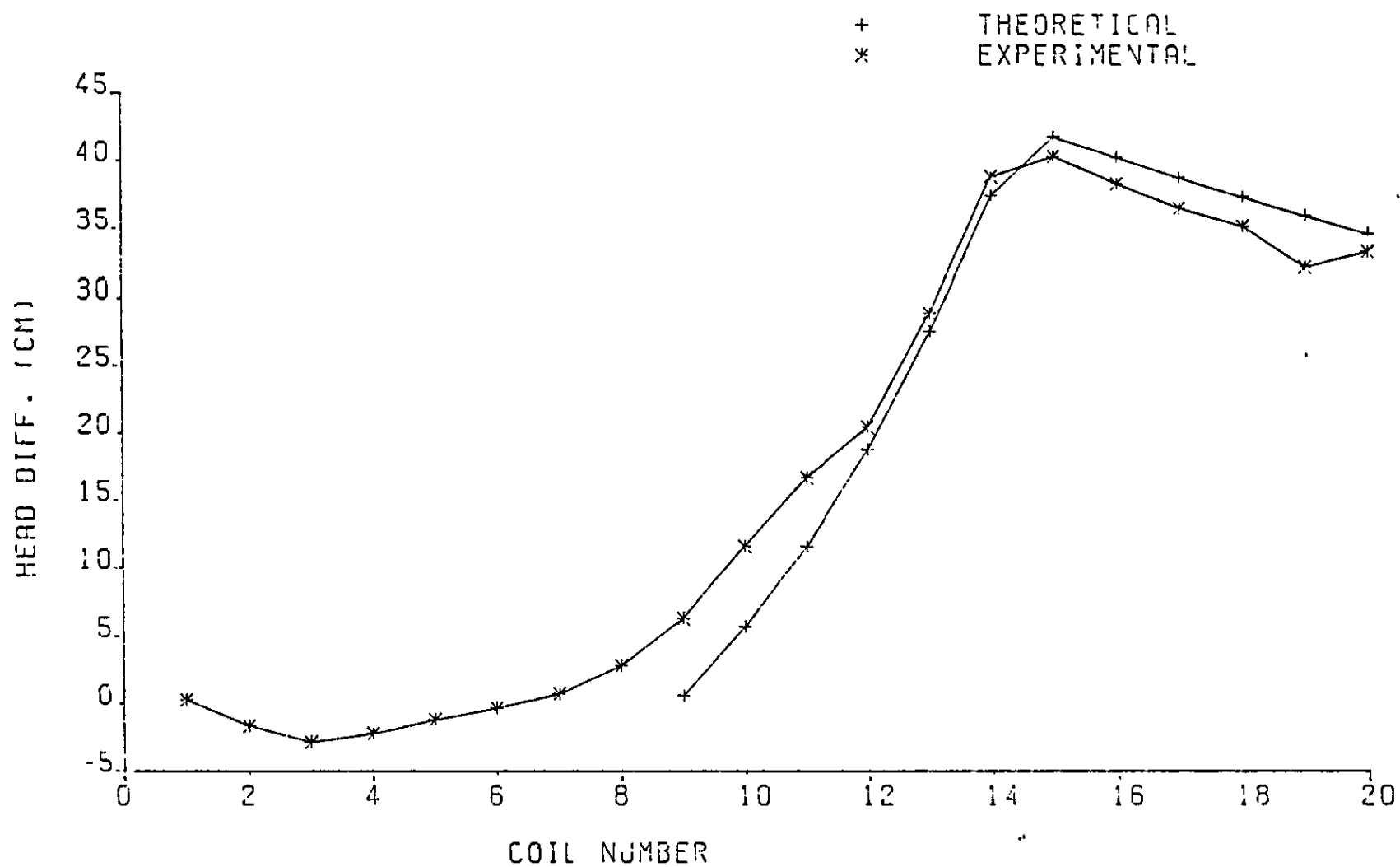


SLOWER SPEED , 8M. LIFT GRAPH 18

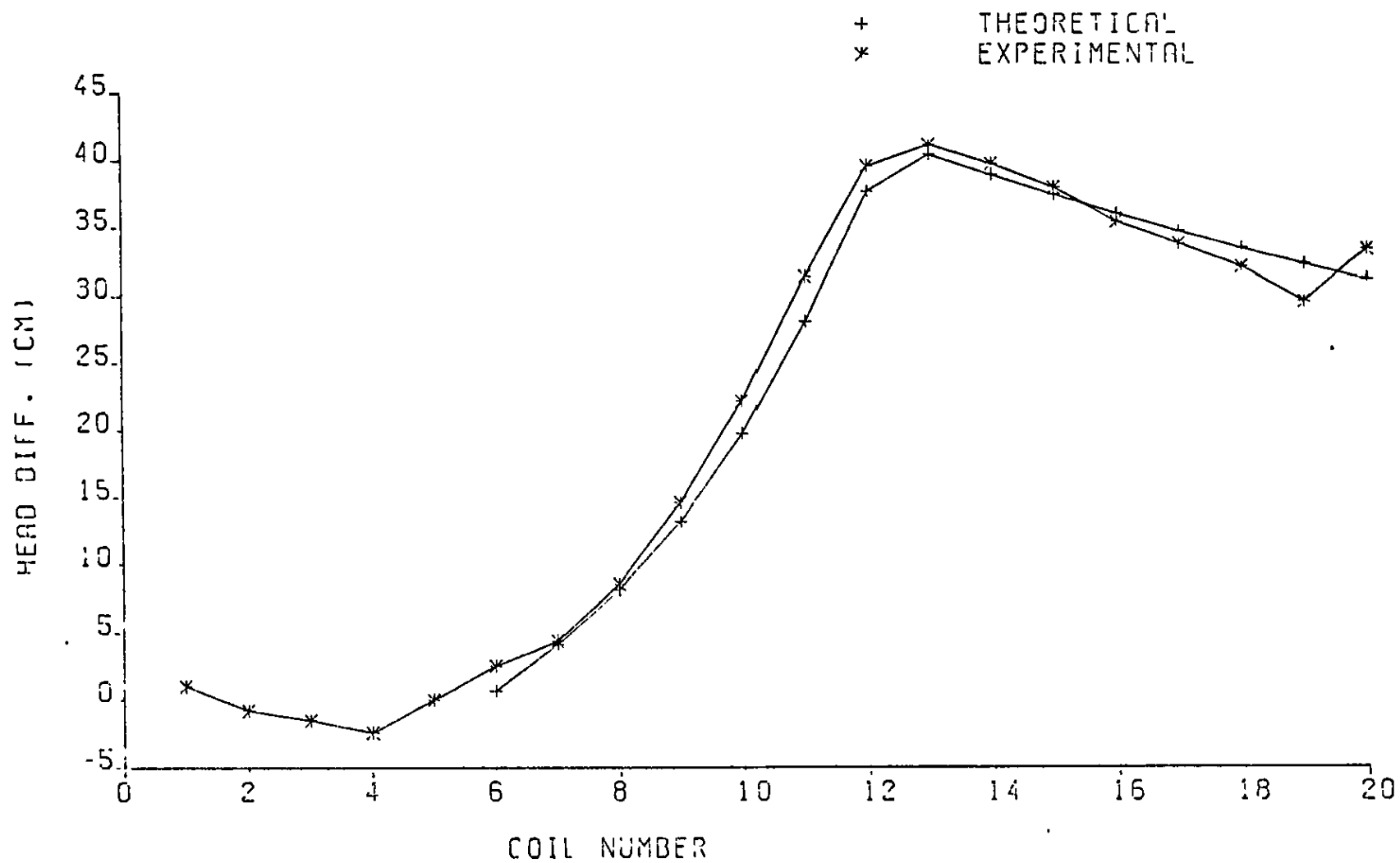
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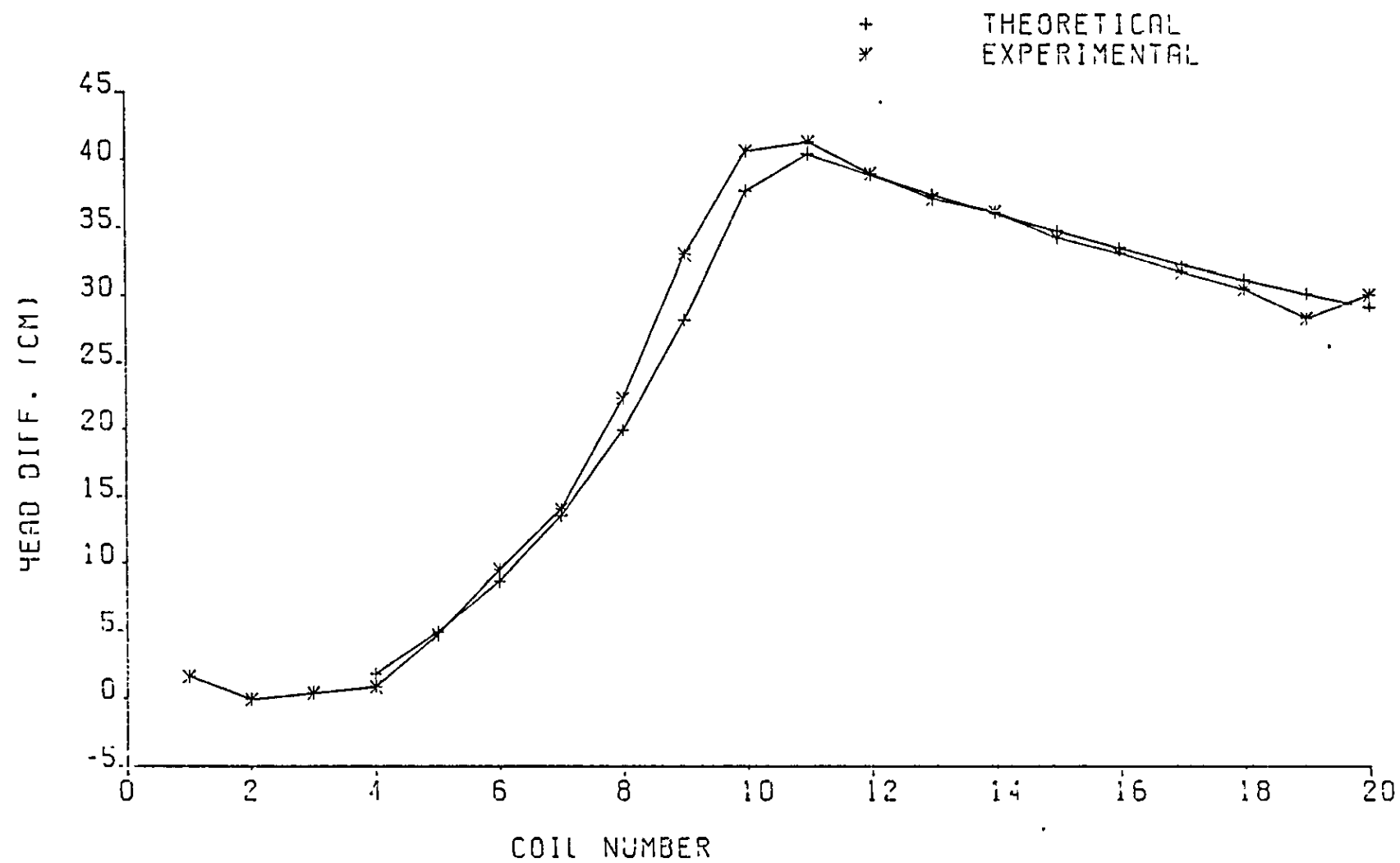
FASTER SPEED . 5M. LIFT GRAPH 19



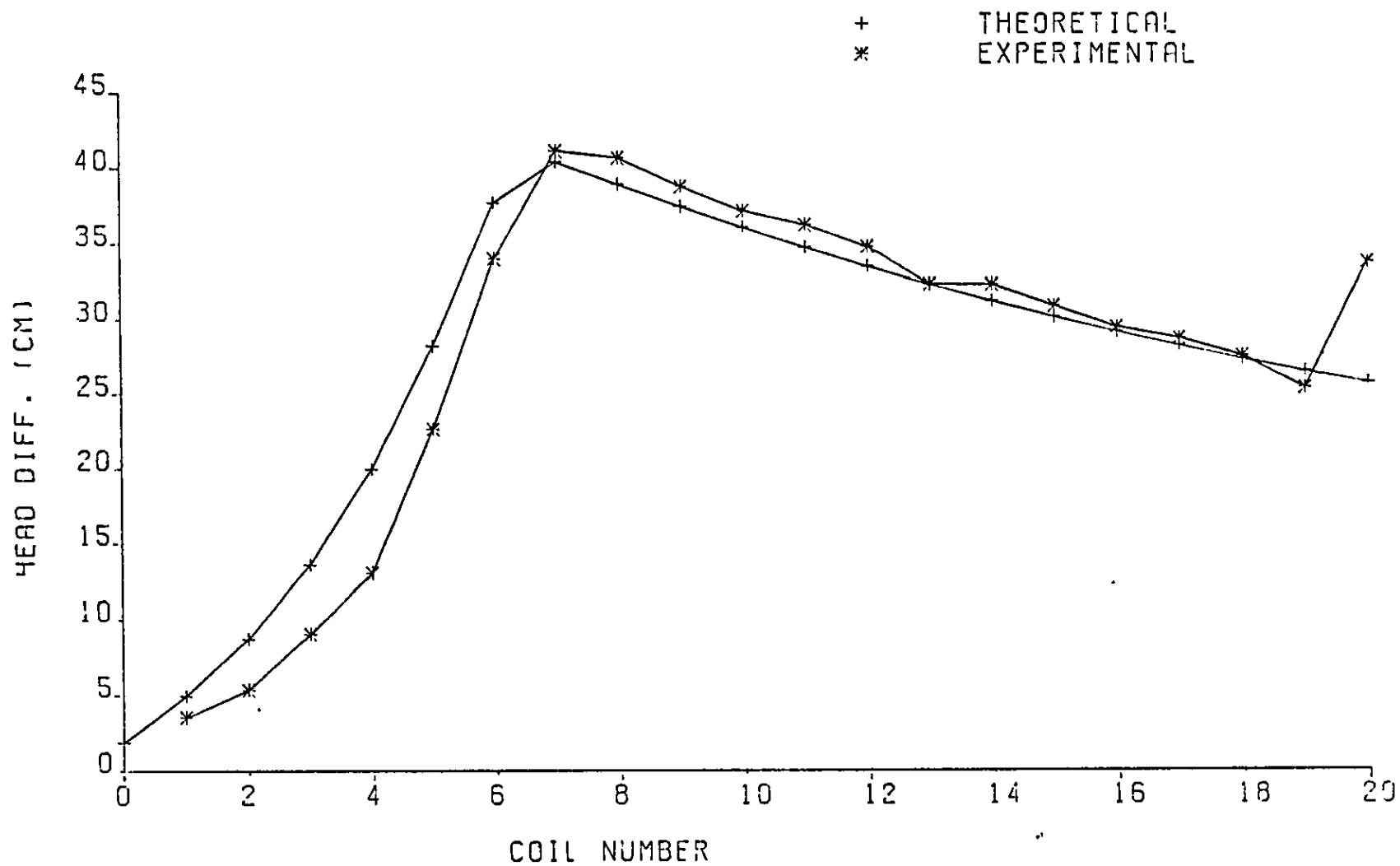
FASTER SPEED , 6M. LIFT GRAPH 20



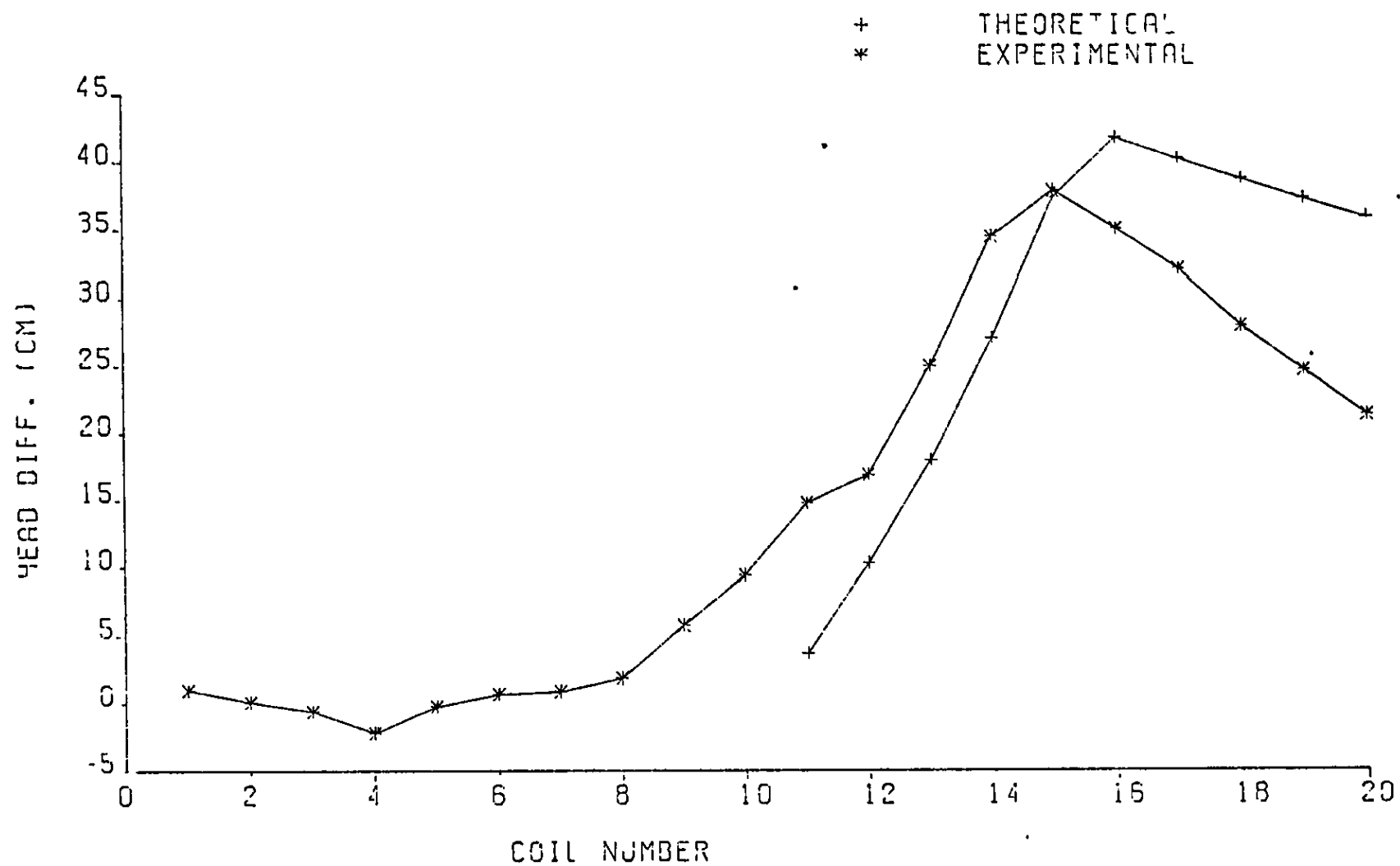
FASTER SPEED , 7M. LIFT GRAPH 21



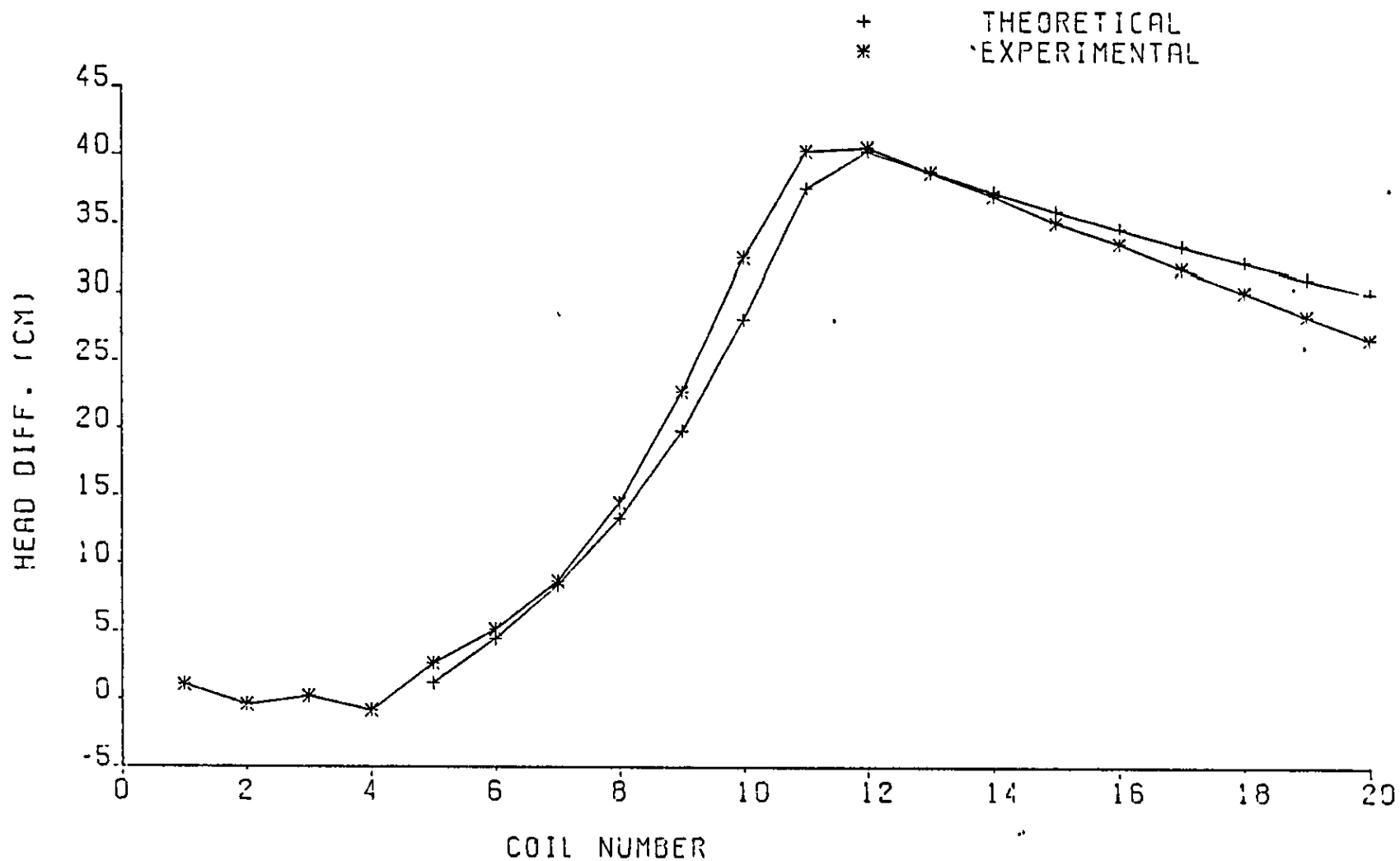
FASTER SPEED , 8M. LIFT GRAPH 22



SMALL DELIVERY PIPE , 4M. LIFT GRAPH 23

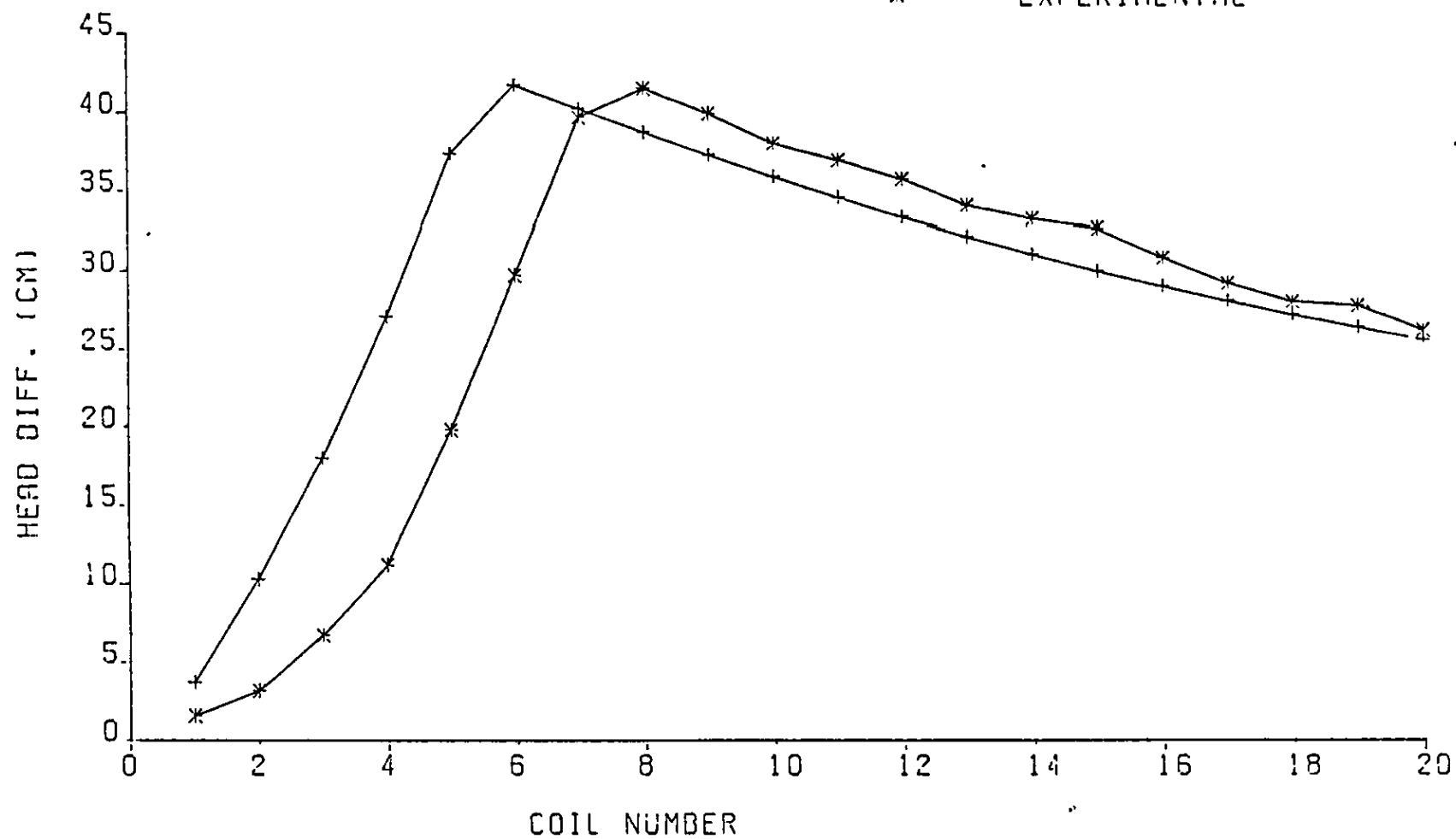


SMALL DELIVERY PIPE , 6M. LIFT GRAPH 24



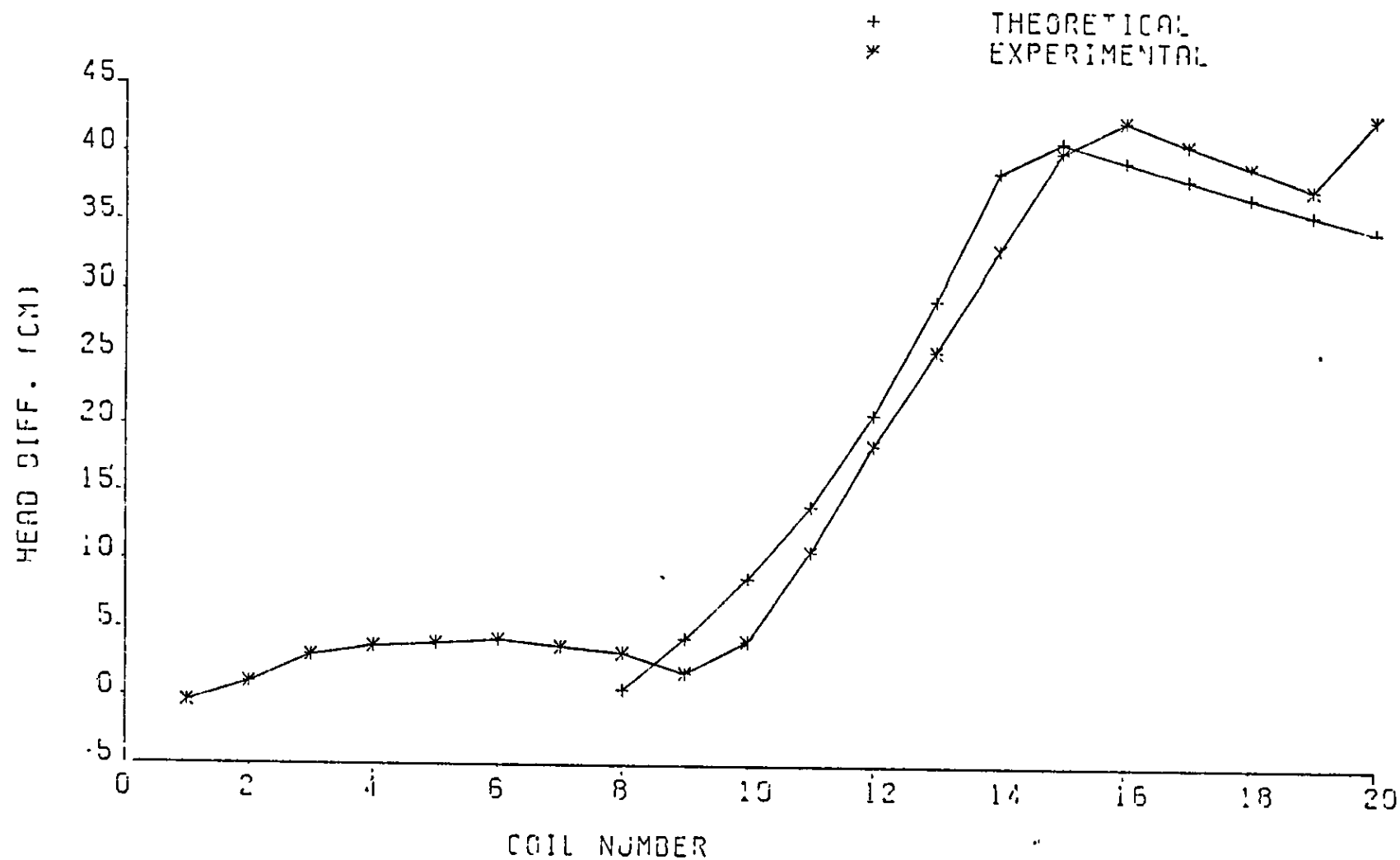
SMALL DELIVERY PIPE , 8M. LIFT GRAPH 25

+ THEORETICAL
* EXPERIMENTAL

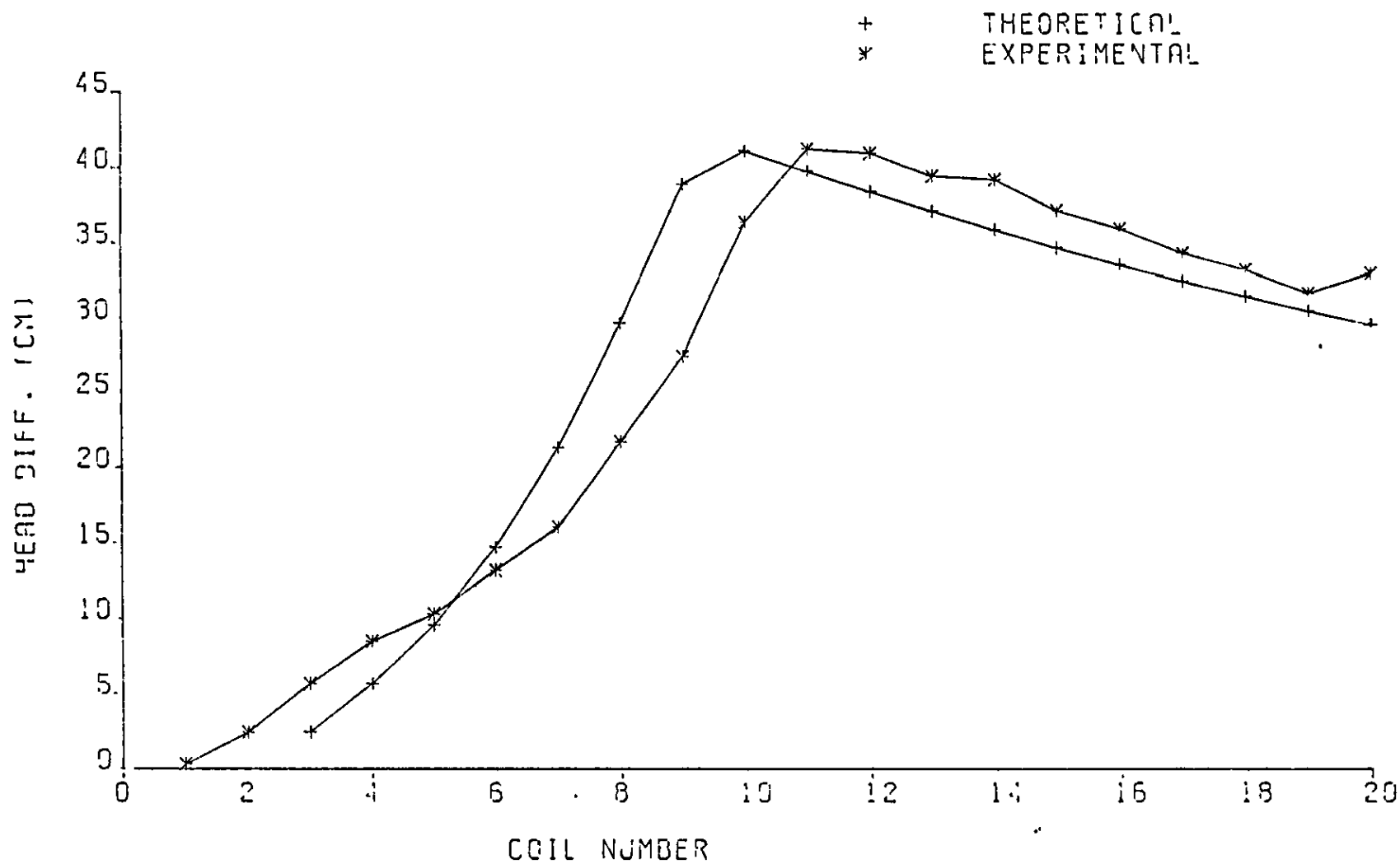


SMALL BORE COILS , 4M. LIFT

GRAPH 26

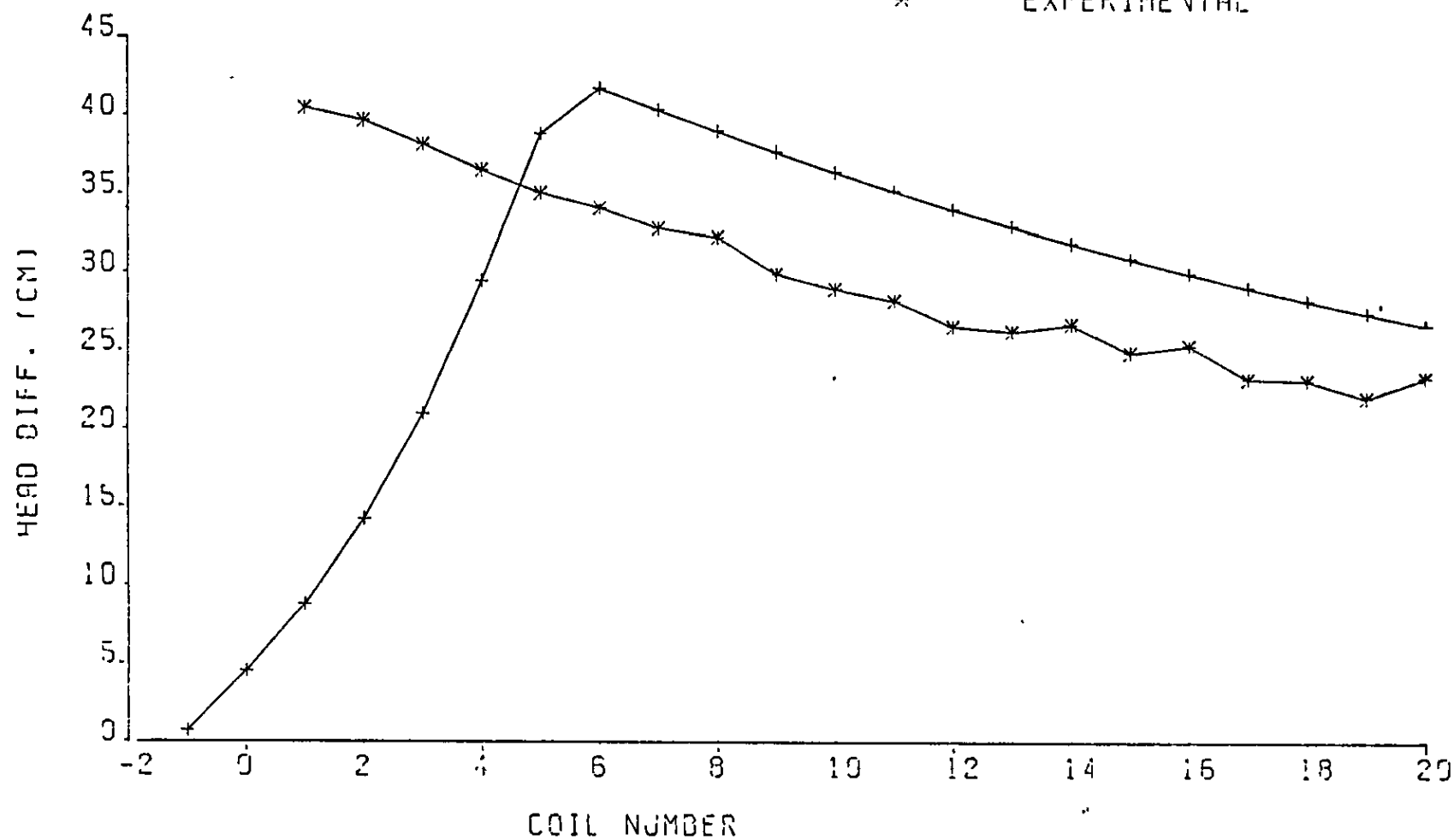


SMALL BORE COILS , 6M. LIFT GRAPH 27



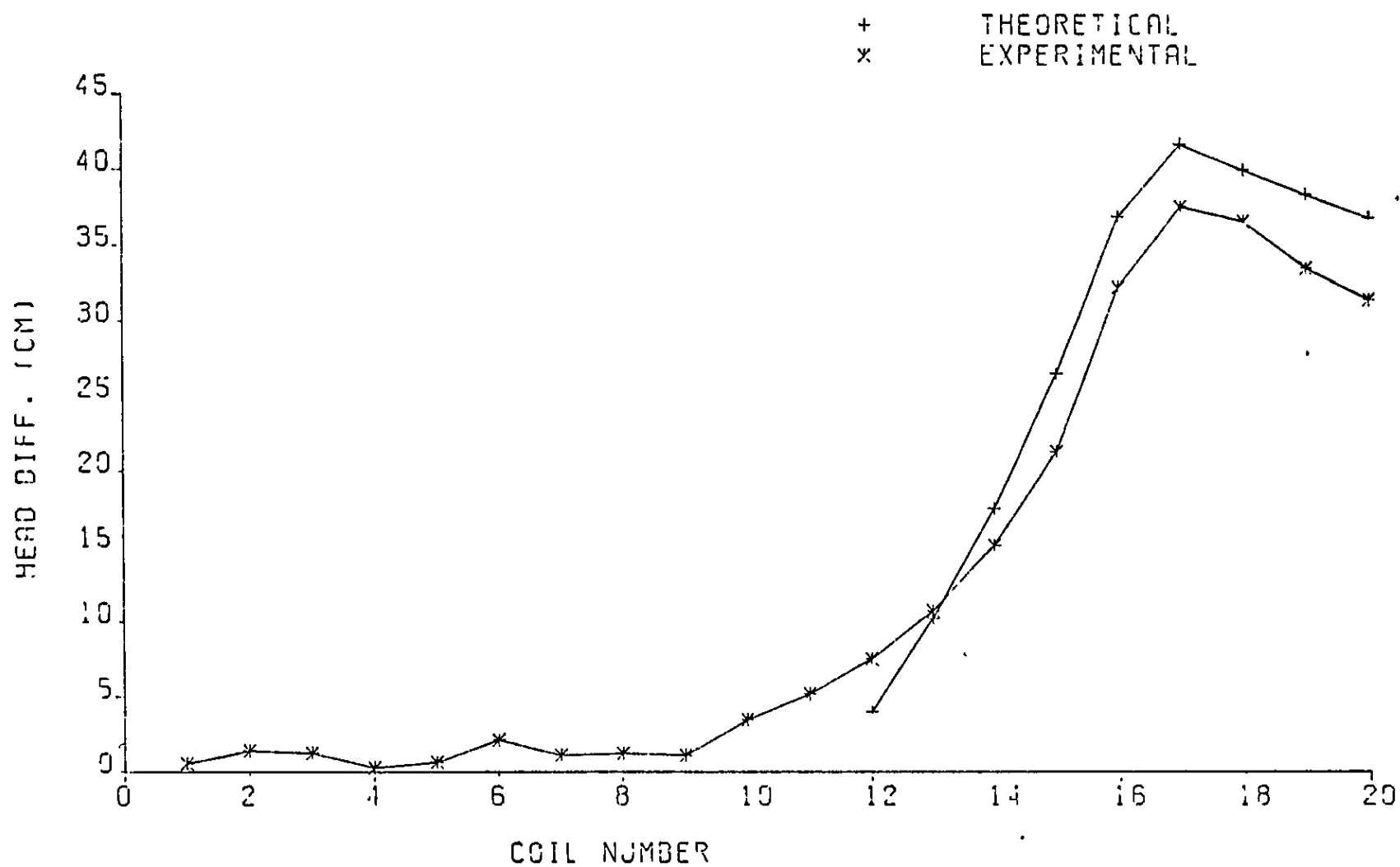
SMALL BORE COILS , 8M. LIFT GRAPH 28

+ THEORETICAL
* EXPERIMENTAL



LARGE BORE COILS , 4M. LIFT

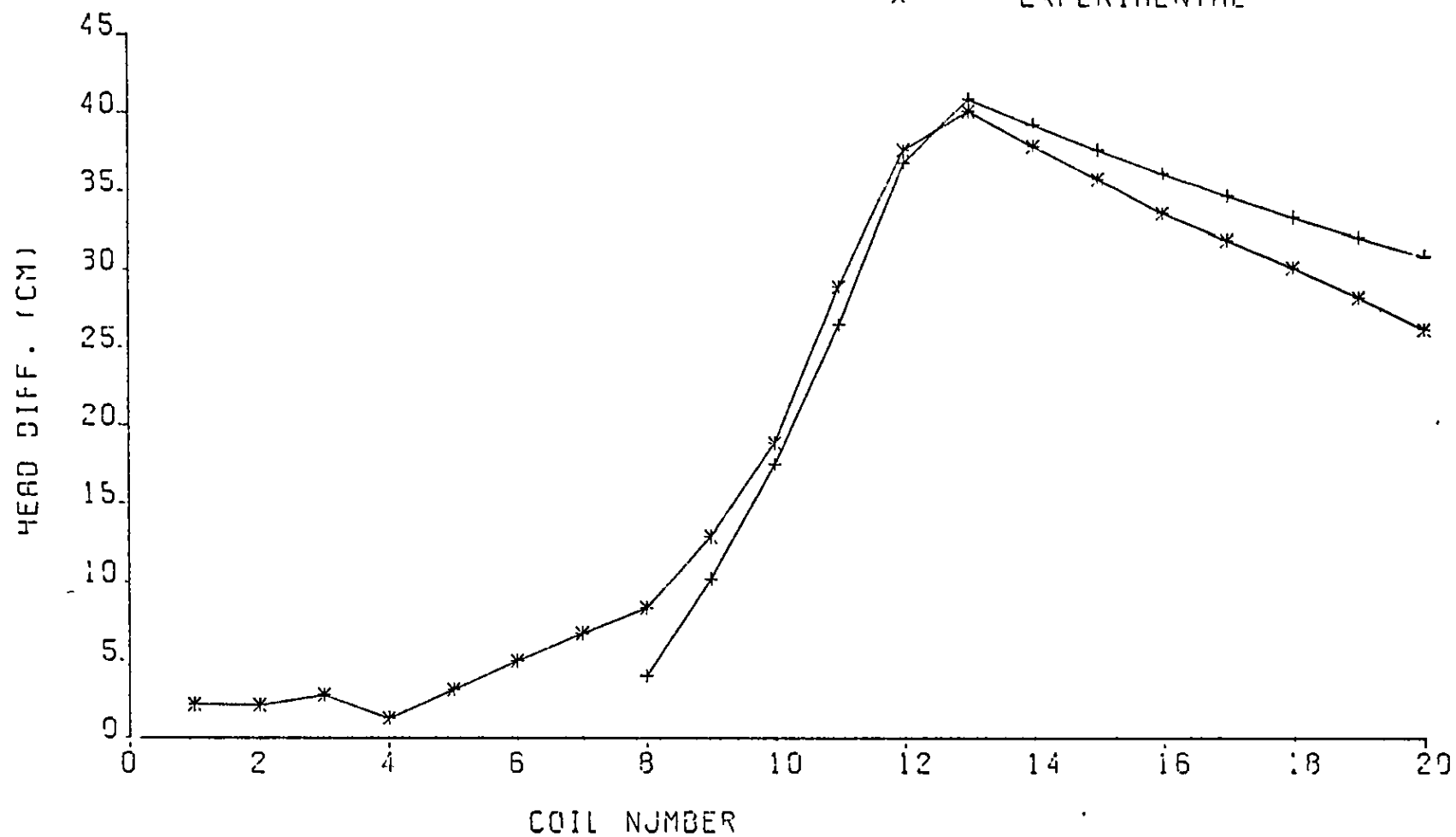
GRAPH 29



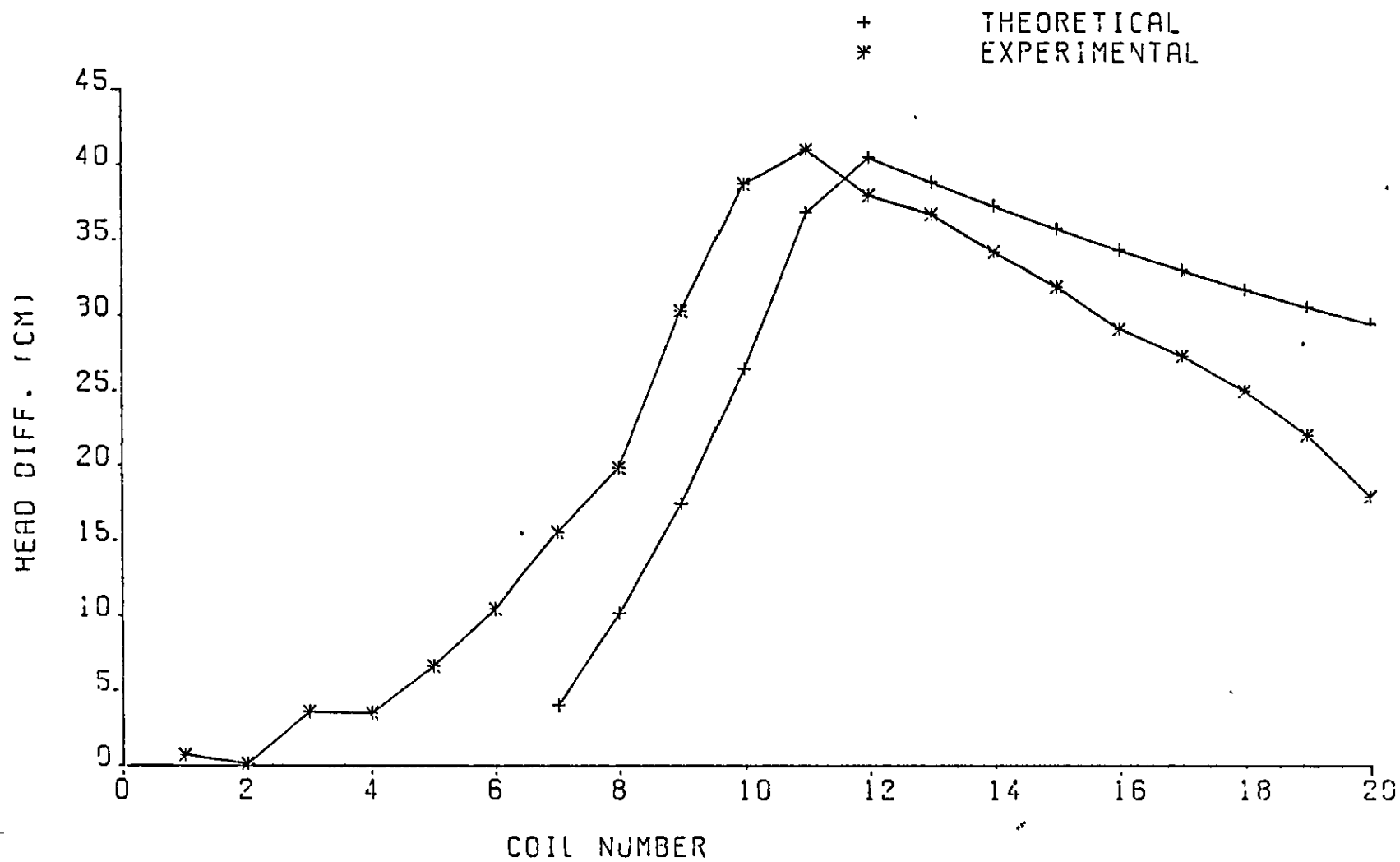
LARGE BORE COILS , 6M. LIFT

GRAPH 30

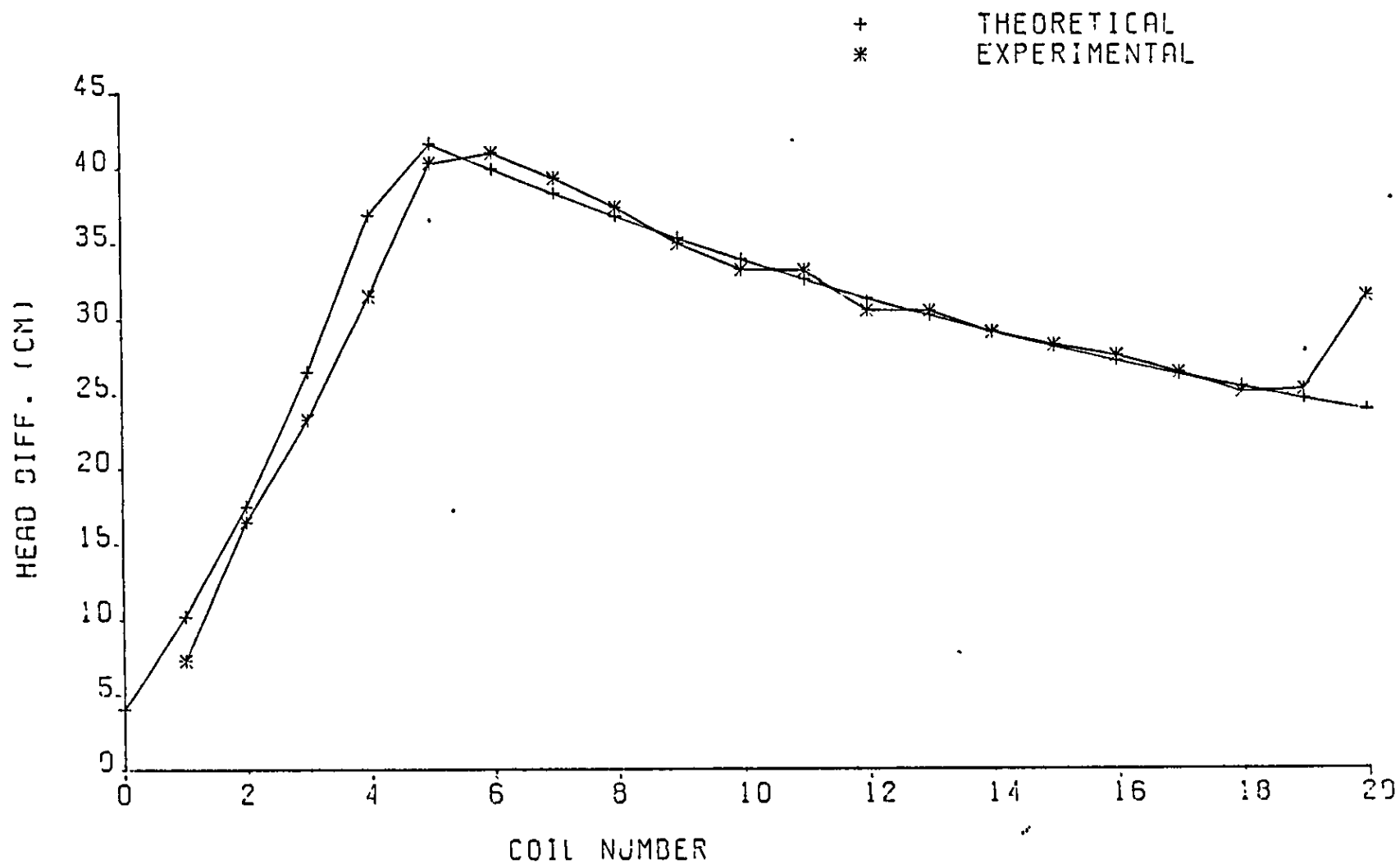
+ THEORETICAL
* EXPERIMENTAL



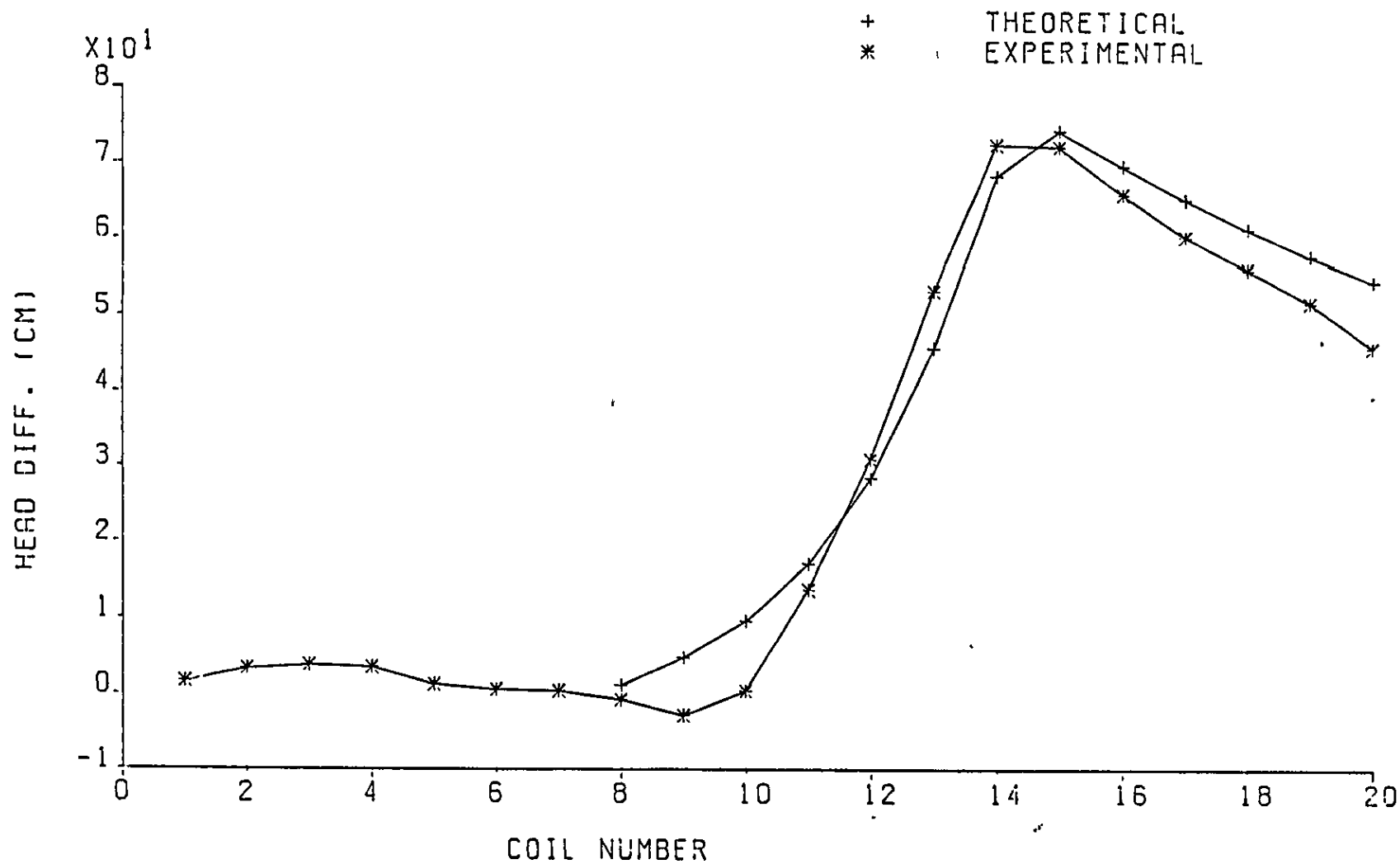
LARGE BORE COILS , 7M. LIFT GRAPH 31



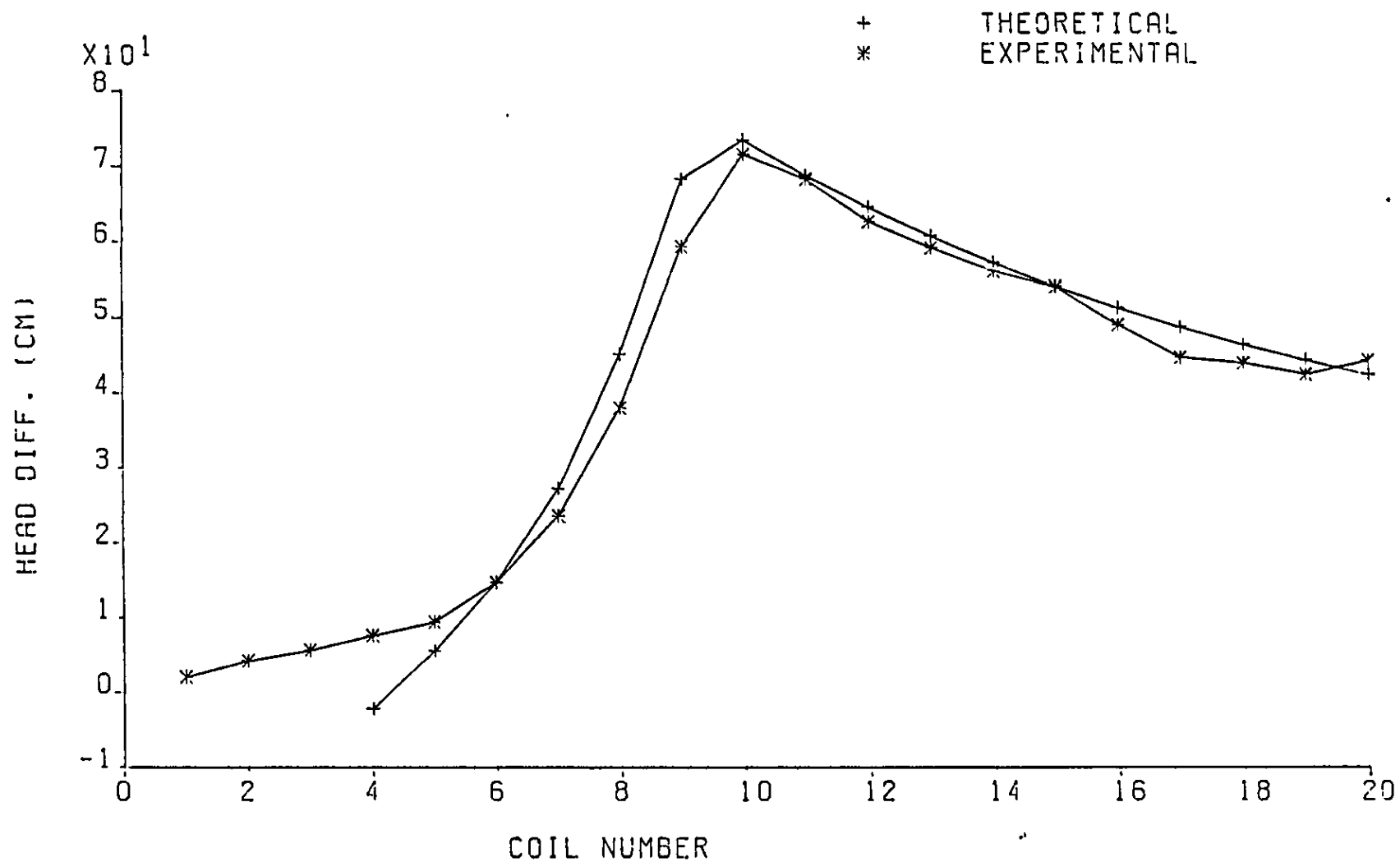
LARGE BORE COILS , 9M. LIFT GRAPH 32



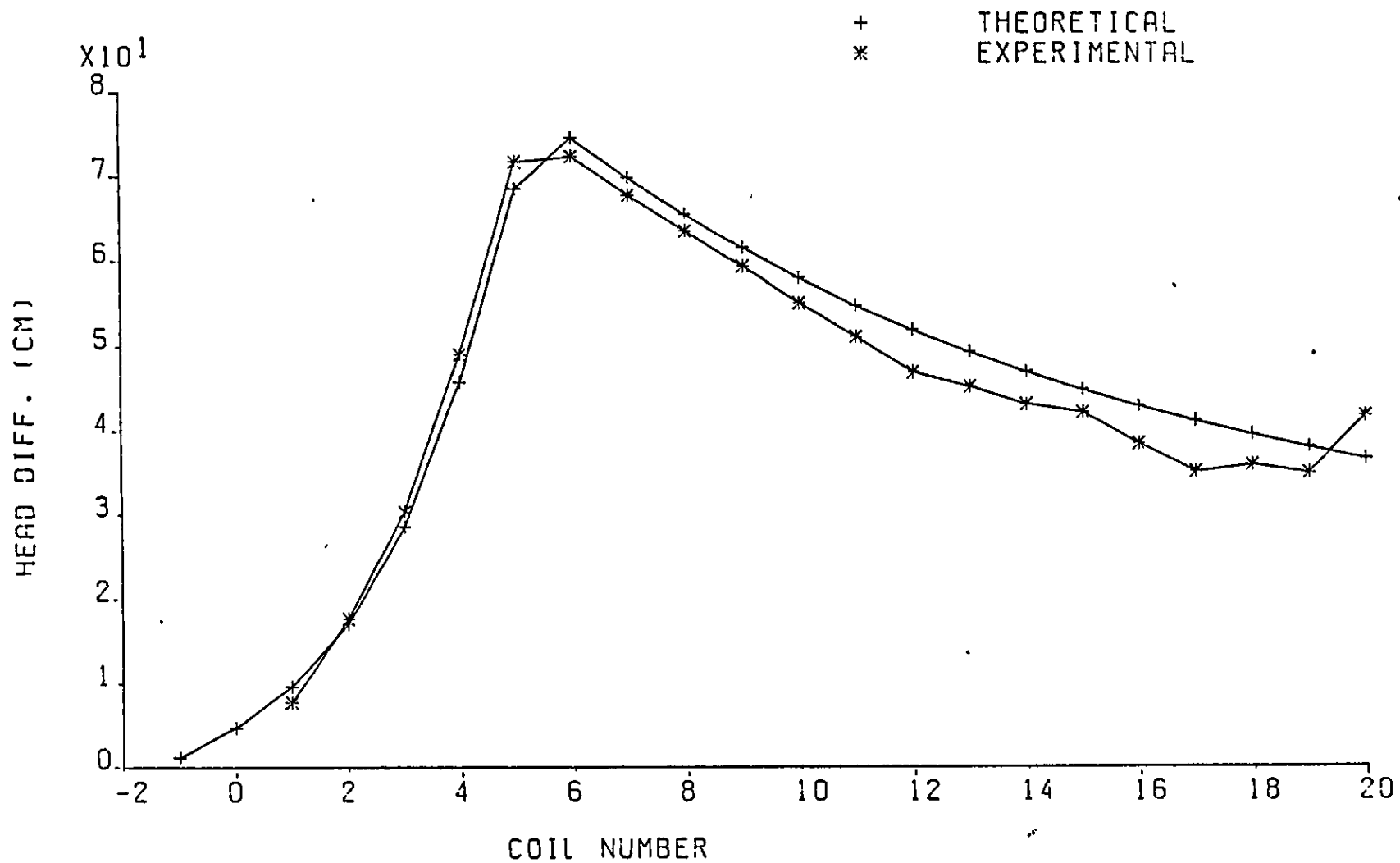
LARGE DIAMETER COILS , 8M. LIFT GRAPH 33



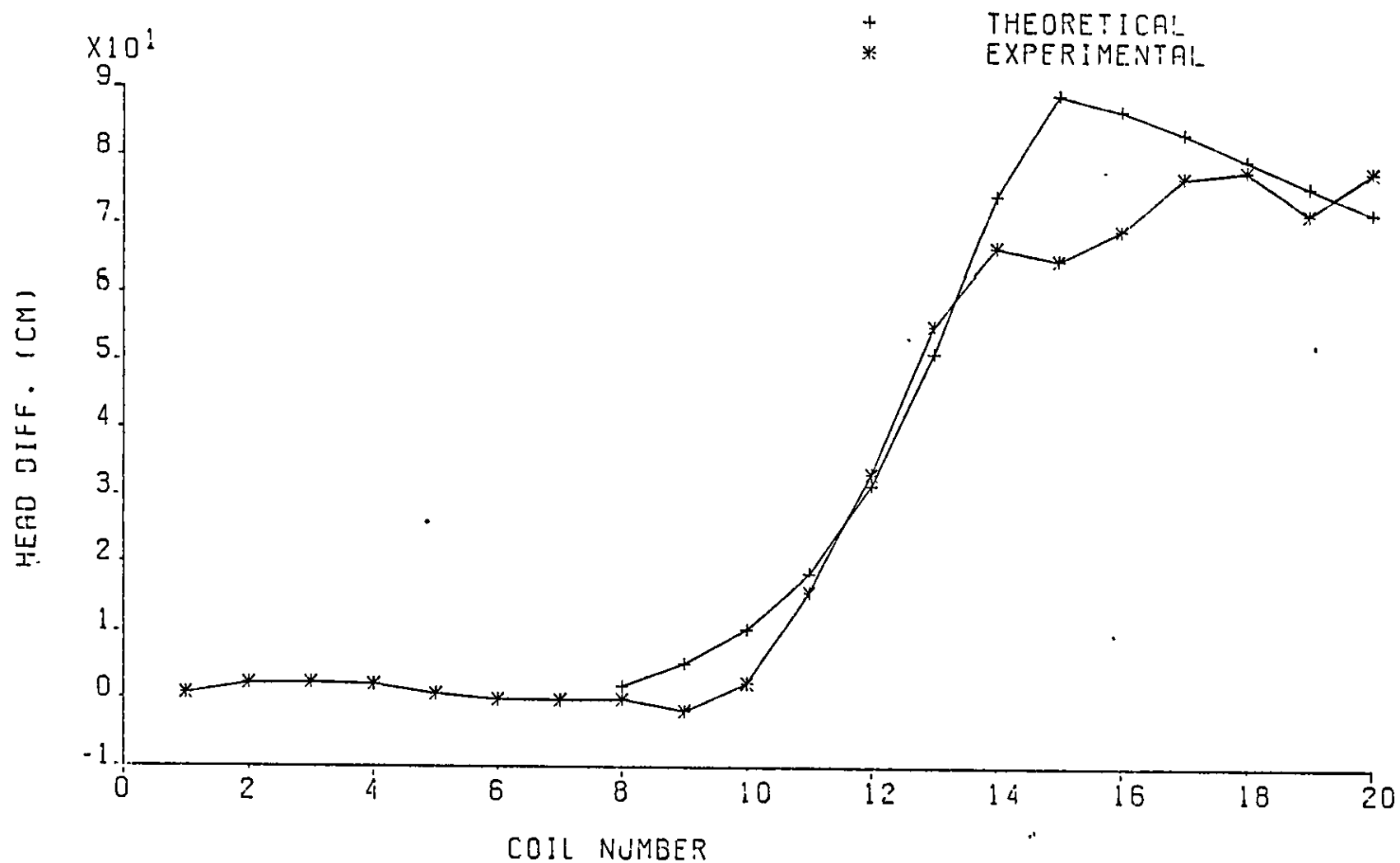
LARGE DIAMETER COILS . 11M. LIFT GRAPH 34



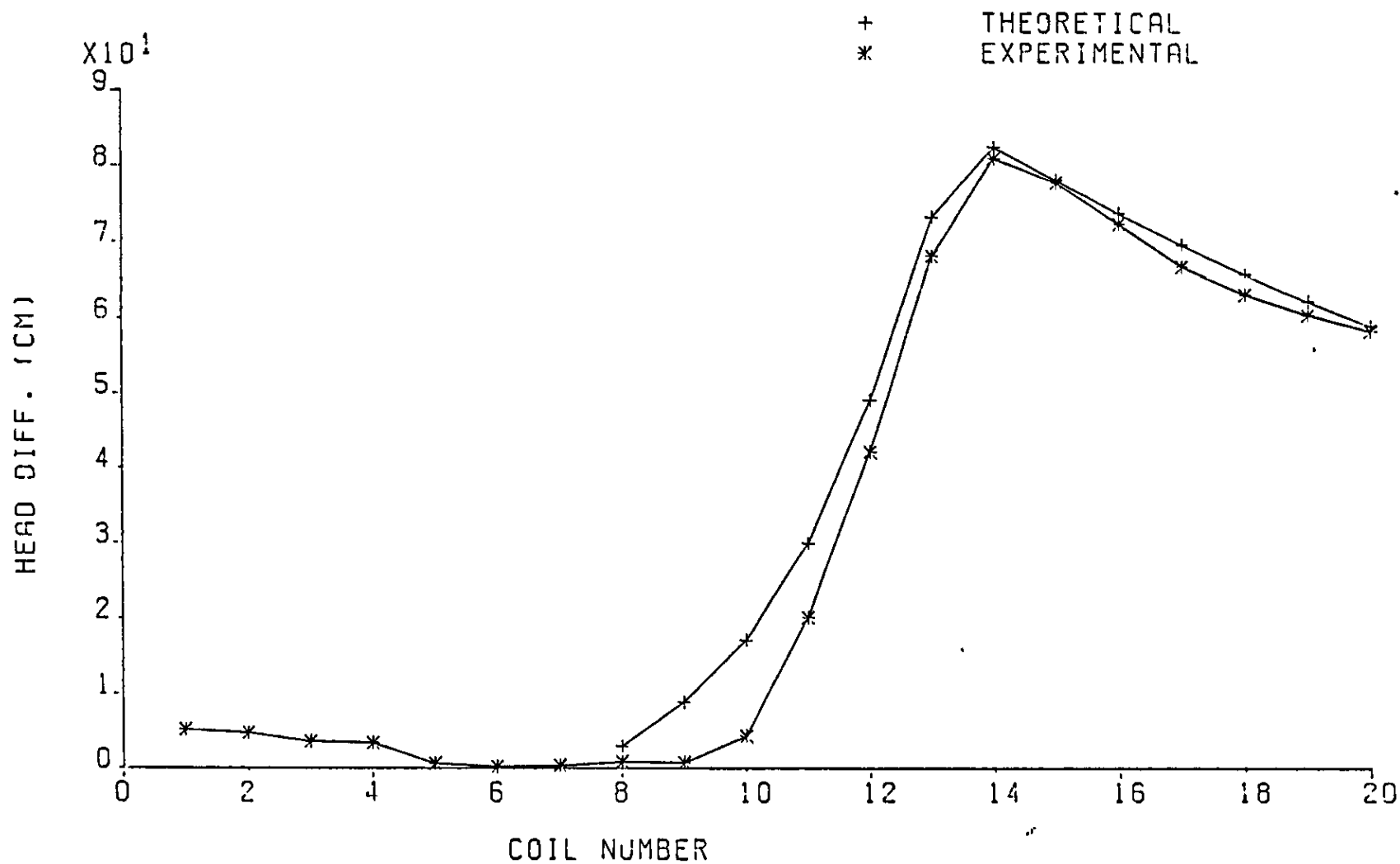
LARGE DIAMETER COILS , 13M. LIFT GRAPH 35



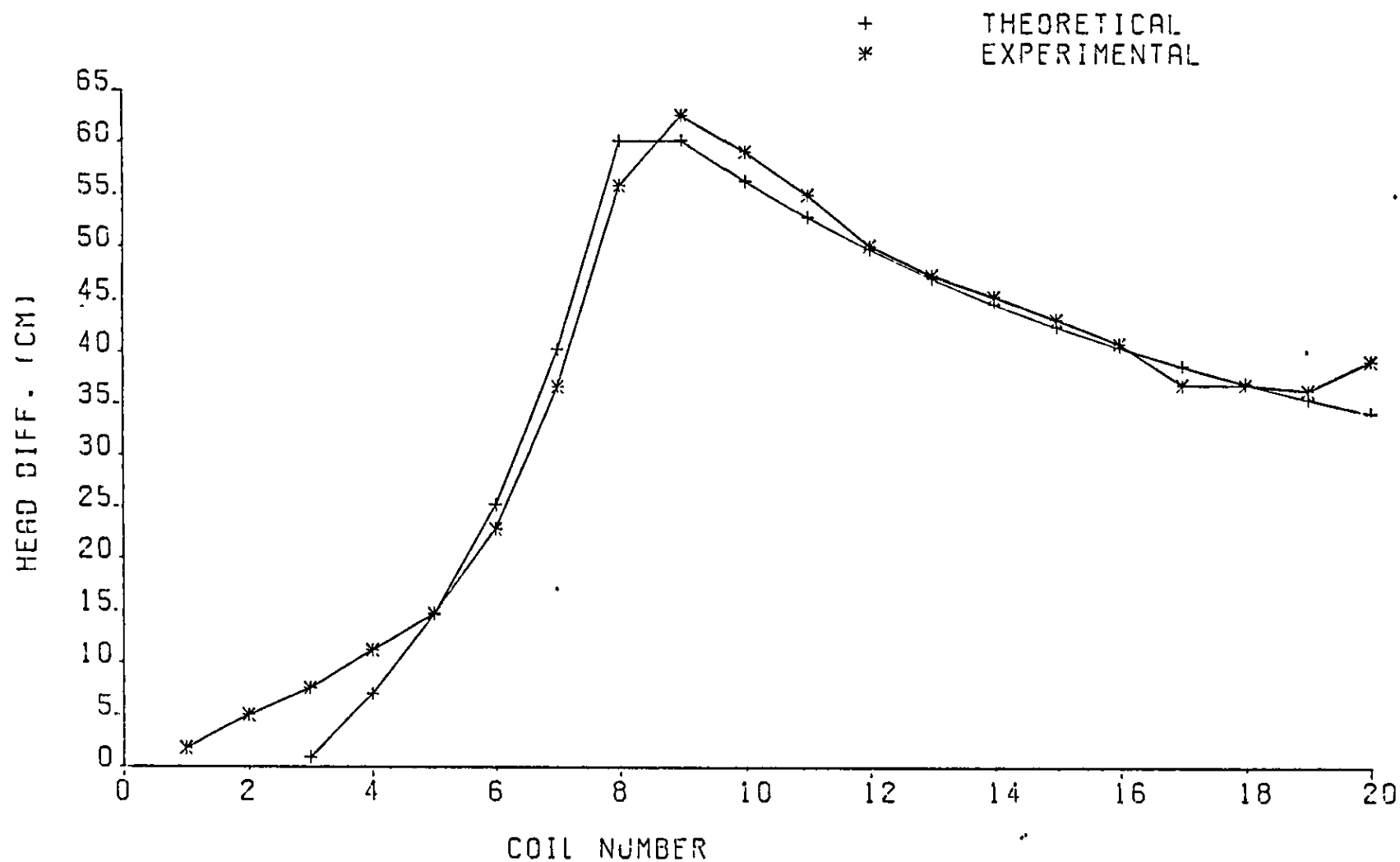
0.3 PI , LARGE DIA. COILS ., 10M. LIFT GRAPH 36



0.4 PI , LARGE DIA. COILS , 10M. LIFT GRAPH 37

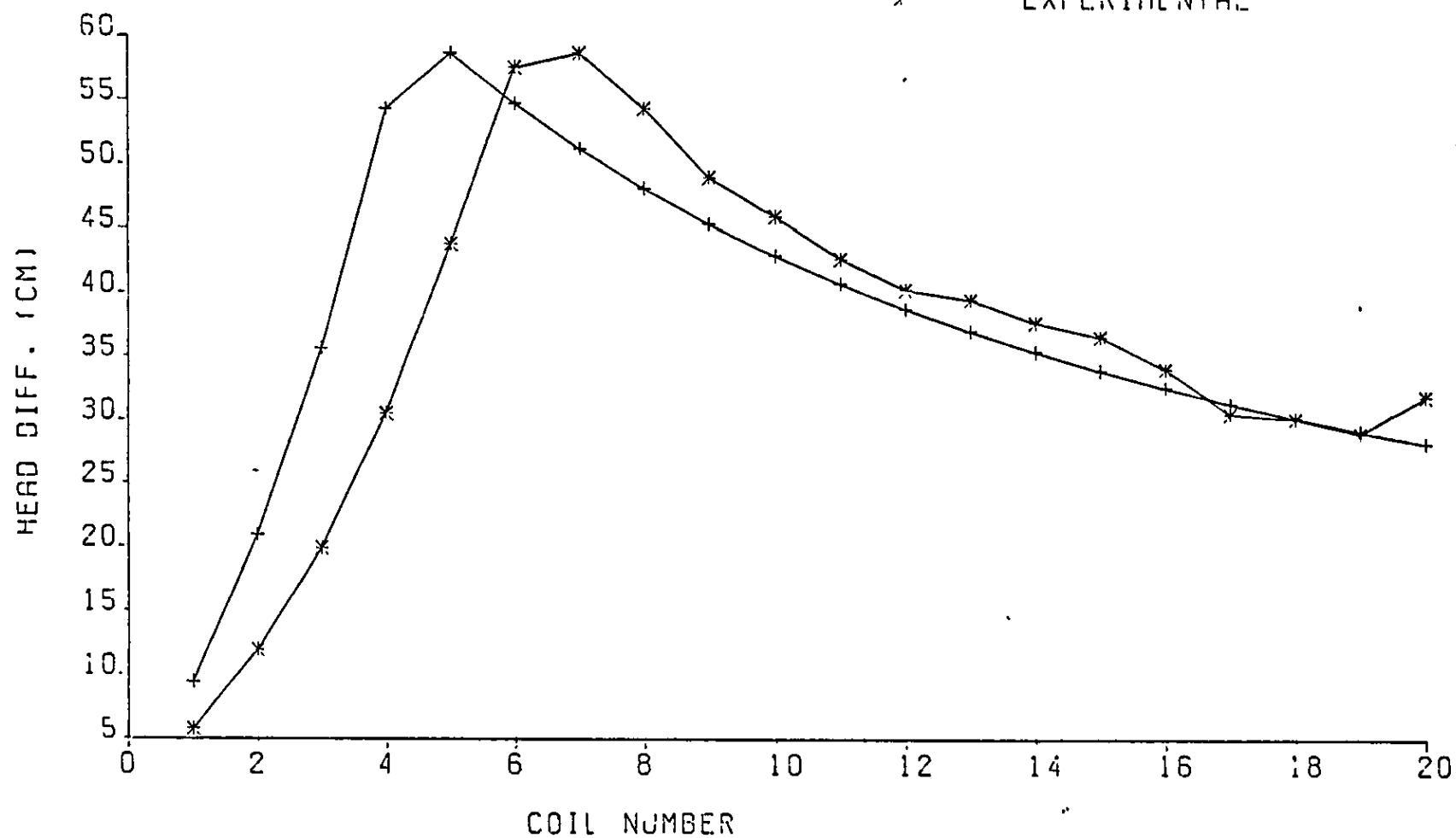


0.6 PJ , LARGE DIA. COILS , 10M. LIFT GRAPH 38



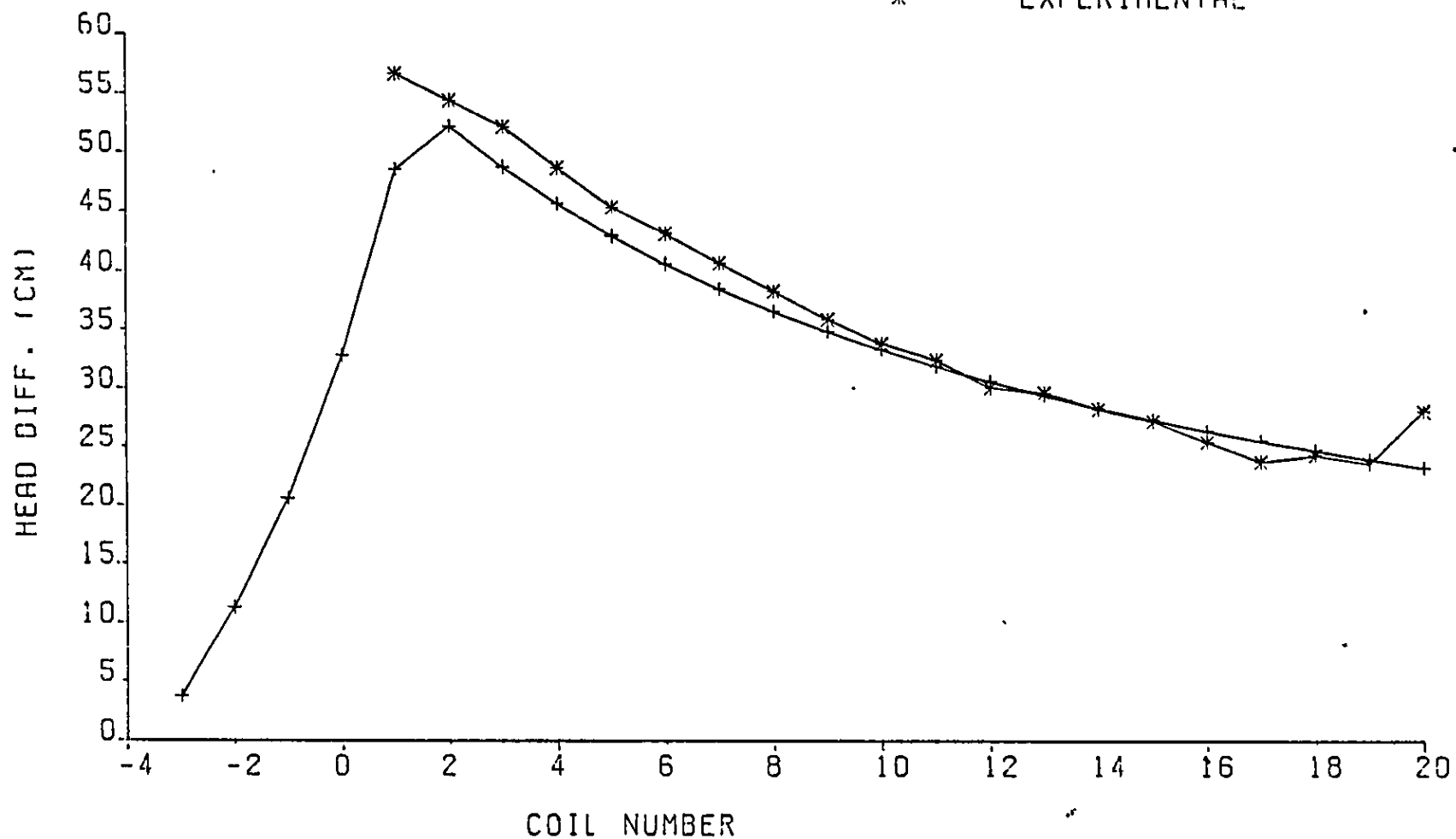
0.65 PI , LARGE DIA. COILS , 10M. LIFT GRAPH 39

+ THEORETICAL
* EXPERIMENTAL



0.7 PI , LARGE DIA. COILS , 10M. LIFT GRAPH 40

+ THEORETICAL
* EXPERIMENTAL



3.7 Discussion of Results

3.7 (a) Delivery Pipe Ratios

In all, sixty-four tests were carried out on twelve different arrangements of the laboratory pump. In each of these tests the developed head was measured and the experimental value of the delivery pipe ratio R_D was calculated. The ratios are plotted against the number of the relevant test on Graphs 1 to 3, the grouping showing the ratios for one particular arrangement as labelled. The relevant input values (section 2.2) were then fed into the program, and the mean value of R_D plotted against test number on the same graphs. For the 0.4 immersion tests, the maximum and minimum values of R_D have also been plotted to show the range within which R_D may vary.

The results for the depth of immersion tests are good, especially in the practical range of the pump 0.4 - 0.7 immersion; at 0.3 the pump does not conform to the general theory as the lower water levels in the coils tend to be at the bottom of the coil resulting in air bubbles rising up through the plugs of water in the coils of the pump. This disrupts the usual steady state of the plugs of air and water in the pump and hence this case is not covered by the usual analysis. In practice, the minimum value of R_D would be used as this corresponds to the maximum head that the pump will need to develop. This process will act as a safety factor, ensuring that the pump will be working well within its range for most of the time.

Graph 2 shows the results for further tests done on speed of rotation and bore of piping used. The three speeds used on the medium sized pump (0.48 m dia.) were 8, 12 and 16 r.p.m. and the two groups of tests on this graph are for 8 r.p.m. (tests 26 - 30)

and 16 r.p.m. (tests 31 - 36). The tests for 12 r.p.m. are those using the standard configuration, medium drum size, 0.5 immersion, 0.025 m bore piping on the pump and for the delivery pipe, and 12 r.p.m. speed of revolution, i.e. tests 13 - 18. The correlation on these tests is again good and the results show that an increase in speed of the drum leads to a beneficial increase in the ratio R_D . This is because the plugs of air and water spend less time in the delivery pipe, reducing the time for which the air plugs can rise up through the water plugs. The reverse effect is also true, with a very slow speed of revolution, the air plugs can rise to the top of the delivery pipe leaving only a continuous plug of water filling the pipe and leading to a high head of water acting on the pump. This effect is well illustrated by these tests.

Tests 37 - 41 require special consideration as they proved very difficult to measure and give very poor correlation with the theoretical values. In these tests a delivery pipe was used with a bore of 0.012 m, the piping on the drum being the standard 0.025 m bore. This gives a reduction in cross-sectional area from $6.25 \times 10^{-4} \text{ m}^2$ to $1.44 \times 10^{-4} \text{ m}^2$, a factor of 4.34. This has two main effects on the plugs in the delivery pipe; firstly that the reduction in the area results in a proportional increase in the speed of the plugs. The effect of increasing the speed of delivery has already been explained for tests 31 - 36, where the notable increase in R_D was due to an increase in speed of 1.33 times. So for an increase in speed of 4.34 times it can be expected that the theory will predict a massive increase in the value of R_D . Secondly, the lengths of the plugs are also increased by this factor, causing a water plug with a length of 0.9 m in the coils of the drum to have a length of 3.91 m in the delivery pipe. This reduces the number of plugs in the delivery pipe and

gives a great variability of R_D , especially at the lower heads when the delivery pipe may be almost filled by one plug of water (low R_D) or one plug of air (high R_D). The maximum and minimum values for test 37 (lifting to 4 m) were 5.879 and 1.098 respectively.

Where the theory and the practice differ is in the connection between the last coil of the pump and the bottom of the delivery pipe which in the theory is assumed to be of negligible length. In practice there is a pipe leading from the coil to the axle of the drum and from there a 1 m length of horizontal piping leading to the connection at the bottom of the delivery pipe. When the pump was stopped, water fell back down the delivery pipe into this rigid horizontal pipe (which was 0.025 m bore) and so disrupted the levels in the coils. An attempt was made to solve this problem by trying to record the position of the levels as the pump was rotating but this method was quite inaccurate. The second attempt involved recording the levels when the pump had stopped, calculating the lengths of the plugs of water, and recalculating the positions in the coils where the upper levels had been seen to be at the crown of the coils. This method proved to be the more accurate although the assumptions are not strictly valid as the reduction of head in the delivery pipe lessens the compression of the air plugs which would be occurring when the pump is working. The pumping head was then calculated by summing the repositioned head differences in all of the coils.

Although these laboratory results are inaccurate, it can still be shown that the program works correctly in this situation by examining tests 47 - 52. In these tests a 0.038 m bore pipe was used on the pump, leading into the standard 0.025 m bore delivery pipe, which gives rise to a similar reduction in cross sectional area as

in tests 37 - 41 although in this case the factor is only 2.31. Good agreement is achieved here in a case where the horizontal connecting pipe is of the same size as the delivery pipe preventing water draining out of the delivery pipe to occupy a larger space. Also in tests 42 - 46 we have a comparable situation but this time there is an increase in area by a factor of 4.34 and the two main effects of this are the reverse of those previously described for the opposite case, leading to a reduction in R_D and a far smaller variability, the maximum and minimum values for test 42 (again lifting to 4 m) being 1.172 and 1.123 respectively.

Graph 3 shows the results of test carried out on a larger pump of 0.9 m diameter. The immersion tests were for 0.3, 0.4, 0.6, 0.65, and 0.7 immersion, all lifting to a height of 10 m. The good correlation between theoretical and practical results shows that computer predictions are not limited to one size of pump and the following example shows the program working for a configuration of pump not used in the laboratory tests.

Example:- A pump of 0.75 m mean coil diameter and immersed to 0.55 of its diameter is required to lift water to 10 m. It is expected that the pump will have an operating speed of 10 r.p.m., and to deliver the quantity of water needed a pipe of bore 0.05 m is to be used for the pump. The delivery pipe is of 0.04 m bore. Calculate the maximum head the pump will need to develop, and the increase in this head should the pump slow to 8 r.p.m.

INPUT HEIGHT OF DELIVERY TANK
 ROTATIONAL SPEED OF PUMP
 MEAN COIL DIAMETER
 PROPORTIONAL DEPTH OF IMMERSION
 I.D. OF PIPE ON PUMP
 I.D. OF DELIVERY PIPE
 10,10,0.75,0.55,0.05,0.04

QUANTITY DELIVERED = 22.196 L/MIN
 MAX. RATIO = 1.421
 MIN. RATIO = 1.317
 MEAN RATIO = 1.365
 OK,

As can be seen from the printout, the minimum ratio to be used in the design is 1.317, this gives a developed head of 7.593 m. Slowing the pump to 8 r.p.m. lowers this minimum ratio to 1.242 necessitating a head of 8.052 m. The value given for quantity delivered is just a check for the program and the true value would be calculated by multiplying the volume of water in the inlet coil by the speed of revolution.

$$Q = 1.253 \times \frac{0.05^2}{4} \times \pi \times 10 = 24.60 \text{ l/min.}$$

To calculate the number of coils required to develop this head, we must use the theory outlined in section 3.4. Comparisons of the theoretical and experimental levels in the coils are made in the next section.

3.7 (b) Head Difference Patterns

Comparisons of theoretical and experimental head difference patterns are shown in Graphs 4 to 40. These will be dealt with in groups according to which characteristic of the pump is being varied in the tests.

Depth of Immersion (Graphs 4 to 15)

As can be seen from the graphs, the theoretical prediction of difference patterns conforms well with the experimental results in the range 0.4 - 0.7 immersion. It can be expected that the theory would not be applicable to the tests for 0.3 immersion since, as has previously been explained, this is an unstable case where other mechanisms of operation exist within the pump.

For the range 0.4 - 0.7 the theoretical values have a very strong correlation with the recorded results, and graphs 4 to 15 predict the number of coils needed on a particular pump to develop a given head. (N.B. In order to test this second part of the complete program, the head measured in the particular test, rather than the head calculated by the delivery pipe program, was inputted, thereby eliminating errors transmitted from this first calculation).

Speed of Rotation (Graphs 16 to 22)

Again the correlation is good and the theoretical values are accurate predictions of the test results. This can be expected because in theory the only effect of varying the speed of the pump is on the delivery pipe ratio, and no alteration is made to the way the pump behaves, and this has been shown to be true over the range of speeds tested. It is interesting to note that the program has occasionally predicted the need for a greater number of coils than the twenty which were on the test model. This situation occurs in other tests when

the pump is lifting to its maximum tested value, and perhaps indicates a likelihood that the pump may eventually stall or blowback when attempting this lift with only twenty coils.

Various Pipe Sizes (Graphs 23 to 32)

Graphs 23 to 25 refer to the tests done using a small delivery pipe. Although great difficulty was experienced when recording the water levels in the pump, the correlation is better than expected. Only in graph 23 is there a serious discrepancy and it was in this test that the water levels were recorded as the pump was turning.

Graphs 26 to 28 and graphs 29 to 32 are the patterns obtained when different size piping was used on the drum. The predictions of the spillback lines in these cases show some inaccuracy. The first assumption might be that the bore of the pipe has an effect on the amount of water carried over the crown of a coil and that the greater drag force created by the smaller pipe might increase the spillback. This is not the case, however, as it can be seen that for the tests using a small bore pipe (graphs 26 to 28) the predicted values fall below the actual results. This means that the predicted spillback is greater than the actual spillback and the reverse situation occurs for the larger bore pipe - the predicted values are high due to an under-calculated value of the amount of spillback. The explanation of these errors lies at the outlet of the pump. In the former case there is an enlargement in pipe size between the pump and the delivery pipe which facilitates movement of water out of the pump, thus reducing the amount of spillback. In the case of the larger pipe the constriction at the outlet reduces the rate of exit of the water, causing more water to spillback into the previous coil. This effect can clearly be seen in graph 31 where the uncharacteristic dive in

the spillback line occurs towards the outlet. This dive may only occur as water is passing through the constriction; in the next graph the predictions are exact and it could be that the pump was stopped as air was passing out of the pump, releasing the effect of the constriction. This would indicate that the levels in the spillback line change markedly throughout one revolution of the pump although this was not noted during the testing.

Large Drum (Graphs 33 to 35)

Values for the head differences in the large pump are not simply scaled up from those for the medium sized pump, although the method of calculation is exactly the same. It is pleasing to see that the theory still applies to this larger size of pump and it is hoped that it will be applicable for even larger pumps up to 3 m diameter, a few of which are working in Third World countries.

Immersion Tests on Large Drum (Graphs 36 to 40)

As with the standard sized pump, the only case in which the theory and practice do not agree is with 0.3 immersion. In all the other cases the theory is a good prediction of the obtained results and in the case of 0.7 immersion the theory predicts the need for 24 coils, a fact borne out by the pattern of head differences which shows that when stopped, the pump was in the advanced stages of a stall.

3.8 Examples of the Design Process

In this section two examples of pump design are given, one concerning the standard model of pump as used in the field test, and the other involving a configuration of pump not previously tested.

The design process is carried out by employing the computer programs developed during the analysis of the pump but the examples chosen are those for which design charts have been plotted in section 6.0, and a comparison is made with the values obtained from the design charts.

Example 1

A standard sized oil drum coil pump is placed in a stream flow of 0.75 m/s and floats at half immersion. Calculate the number of coils required to lift water to 8 m if (a) 0.025 m bore pipe is used for the coils and the delivery pipe, and (b) 0.038 m bore pipe is used for the coils in conjunction with a 0.025 m bore delivery pipe.

- (a) Feeding the relevant values into the delivery pipe program gives the following ratios:-

$$\text{Max. ratio} = 1.449$$

$$\text{Min. ratio} = 1.330$$

$$\text{Mean ratio} = 1.376$$

$$\therefore \text{The maximum design head} = 8/1.330 \\ = 6.015 \text{ m}$$

This value is fed into the second program along with the dimensions of the pump, and the resulting number of coils required is 20.

Using the design chart for this standard pump, (Section 6.0) it can be seen that the number of coils required is 20.5 which would be

rounded up to 21. This is greater than the previous value because the design curve is a smoothed approximation to the actual plot which is a wavy line. The smooth curve is drawn on the side of caution, thus resulting occasionally in a positive error.

(b) When the larger pipe is used for the coils, the delivery pipe ratios became:-

$$\text{Max. ratio} = 1.729$$

$$\text{Min. ratio} = 1.372$$

$$\text{Mean ratio} = 1.552$$

$$\begin{aligned} \therefore \text{Maximum design head} &= 8/1.372 \\ &= 5.831 \text{ m} \end{aligned}$$

This gives a computer result of 20 required coils, no design chart has been drawn for this particular configuration of standard sized pump.

A comparison of the two above calculations reveals a basic flaw in the design procedure. Although the R_D values obtained for the larger pipe are obviously the higher of the two (mean value of 1.552 compared to 1.376) the minimum values do not show the same difference (1.372 compared with 1.330) and it is these values that are used to determine the maximum head that the pump must develop. This method of design ensures that the pump is totally capable of achieving the desired lift, but it may also be under-estimating the potential of the pump.

An improved design procedure would result from examining the distribution of the ratios produced by the computer model, and as an alternative to using the absolute minimum value, employ a value a given distance from the mean, e.g. a value above which 90% of the ratios lie.

Tests could be conducted on a laboratory model of the pump to ascertain the validity of any chosen minimum ratio.

Example 2

A large pump of 1.3 m diameter is to be immersed to 0.4 depth of immersion in a stream flow which, with the wheel arrangement used, will produce a rotational speed of 3.67 r.p.m. The pipe used on the pump has a bore of 0.038 m and the delivery pipe a bore of 0.025 m. Calculate the number of coils required to produce a lift of 14.3 m.

From delivery pipe program:-

$$\text{Max. ratio} = 2.246$$

$$\text{Min. ratio} = 1.348$$

$$\text{Mean ratio} = 1.694$$

$$\begin{aligned} \therefore \text{Max. head} &= 14.3/1.348 \\ &= 10.608 \text{ m} \end{aligned}$$

The number of coils needed to develop this head is given by the program as 17.

The number given by the design charts in Section 6.0 is also 17.

4.0 SELECTION OF POWER SOURCE AND TESTS DONE

4.1 Description of Laboratory Work

Initial information on finding a power source for the rotating coil pump was provided by a final year project done in the Department of Civil Engineering during the academic year 1979-1980. Ref. 6

In this project, the student tested and compared two small scale models of waterwheels. The first design was a drum with flat paddles running along the length of the drum parallel to the axis of the drum; the second design had the paddles forming chevrons around the drum, similar to the tread on a tractor tyre. The angle formed at the joining of the two arms of a blade was approximately 90° and both wheels had the same blade area. The tests carried out involved measurement of power, speed and range of depths of immersion over which the two wheels performed, and it was found that the chevron-bladed wheel developed twice as much power, operated over a greater range of depths and was much less susceptible to vibration than the straight-bladed wheel.

For these reasons, the chevron-bladed type of wheel was adopted for use in conjunction with the coil pump. A model was made of a combined waterwheel and coil pump arrangement, this being basically a drum with the blades fixed on to the outside and piping wound around the inside. This model was tested in a metre wide flume in the hydraulics laboratory to determine the effect of varying the depth of immersion, clearance from the bottom of the flume and the speed of flow of the stream. It was realised that this model would bear no real resemblance to the proposed field model in terms of dimensions and number of blades, so the results obtained were treated as general

relationships for the design of the wheel, but specific characteristics only for that one model.

4.2 Results of Laboratory Work

From Graph 41 of speed rotation against depth of immersion (with both speed of flow and clearance kept constant) it can be seen that the maximum speed was obtained at 0.61 immersion. A greater depth of immersion causes water to strike the front face of the blades as they enter the water. While this is happening to some part of the blade at depths of immersion greater than about 0.35, it is at this optimum value of 0.61 that the resultant force taken over the whole blade throughout one revolution is at a maximum. The trailing edges of blades leaving the water on the downstream side of the wheel are also moving against the flow of the water and this too has a slowing effect. It should be noted that the pump was made to lift water to a constant 2.5m during this test, but the effect of increasing the depth of immersion meant that the pump could not develop the head required when immersed to more than 0.62 of the wheel's diameter. The six points in the top righthand region of the graph are, therefore, relating to a situation where power is not being used to lift water, which explains the slight increase in rotational speed at these depths of immersion.

The graph of speed of rotation against clearance from the bottom of the flume (Graph 42) shows the expected exponential decrease in speed as the wheel is given greater clearance (again speed of flow and depth of immersion kept constant, i.e. depth of flow is increased with clearance). Decreasing the clearance reduces the amount of space around the wheel and the water has to move the blades to get past the wheel. The upstream level of water increases and it is this 'backing up' of the water combined with the increased momentum of the flow which

increases the speed of the wheel. The clearance can better be expressed as a dimensionless blockage ratio which takes into account the amount of area at the sides of the wheel through which the water can flow. As this free area of flow is reduced, the velocity increases resulting in increased rotational speed of the wheel. Graph 43 shows the graph of speed against blockage ratio.

The plots of speed of wheel against flow velocity (Graphs 44 to 46) suffer from a gap in the range of flows available in the flume, and also from an unnecessary scatter which was due to lack of control in the depth of flow. However, the two plots do demonstrate an almost direct relationship between speed of rotation and flow velocity, both when pumping and when running free. The comparison of the two plots shows very little variation in the lines depicting the free-running and pumping conditions. This is because most of the power abstracted from the flow is used to overcome friction in the sealed rotary joint which is necessary for the pump's efficient operation, hence the lifting of a small amount of water to 2.5 m requires only a relatively low proportion of the power available, especially towards higher speeds, where the friction losses are greater.

It is realised that these results relate only to the particular model used and the flume in which it was tested. It cannot be said that all waterwheels of a similar design will have the same characteristics. An in-depth study into these waterwheels is needed, but because of the number of variables present and the difficulty of making a wheel on which these variables could be quickly adjusted, the study would need to take the form of a full research project. Among the variables which may affect the characteristics of a wheel are:-

Number and spacing of blades

Depth of blades

Angle at apex of blades

'Presentation' of blades (because each arm of a blade is planar, the face of the blade is only normal to the drum surface at one point along the length of the arm. Altering the position of this point would change the aspect of the blade presented to the water).

In a final year project done in the academic year 1980-1981 (report not yet published) ^{Ref. 8} it was discovered that when the model described above had its number of blades reduced from 12 to 6, the peak power produced was markedly increased. This is thought to be due to the fact that with 12 blades interference was taking place and the upstream blades were 'shadowing' the downstream blades and effectively blocking the flow on to these blades as they passed below the axis. This gives an indication towards an optimum number of blades which would be a worthwhile topic to investigate.

Graphs 41 to 46 follow.

Graph 41

Speed v. Depth of Immersion

Speed
of
Rotation
(r.p.m.)

0.3

0.4

0.5

0.6

0.7

Proportion of Wheel's Diameter Immersed in Flow

11

10

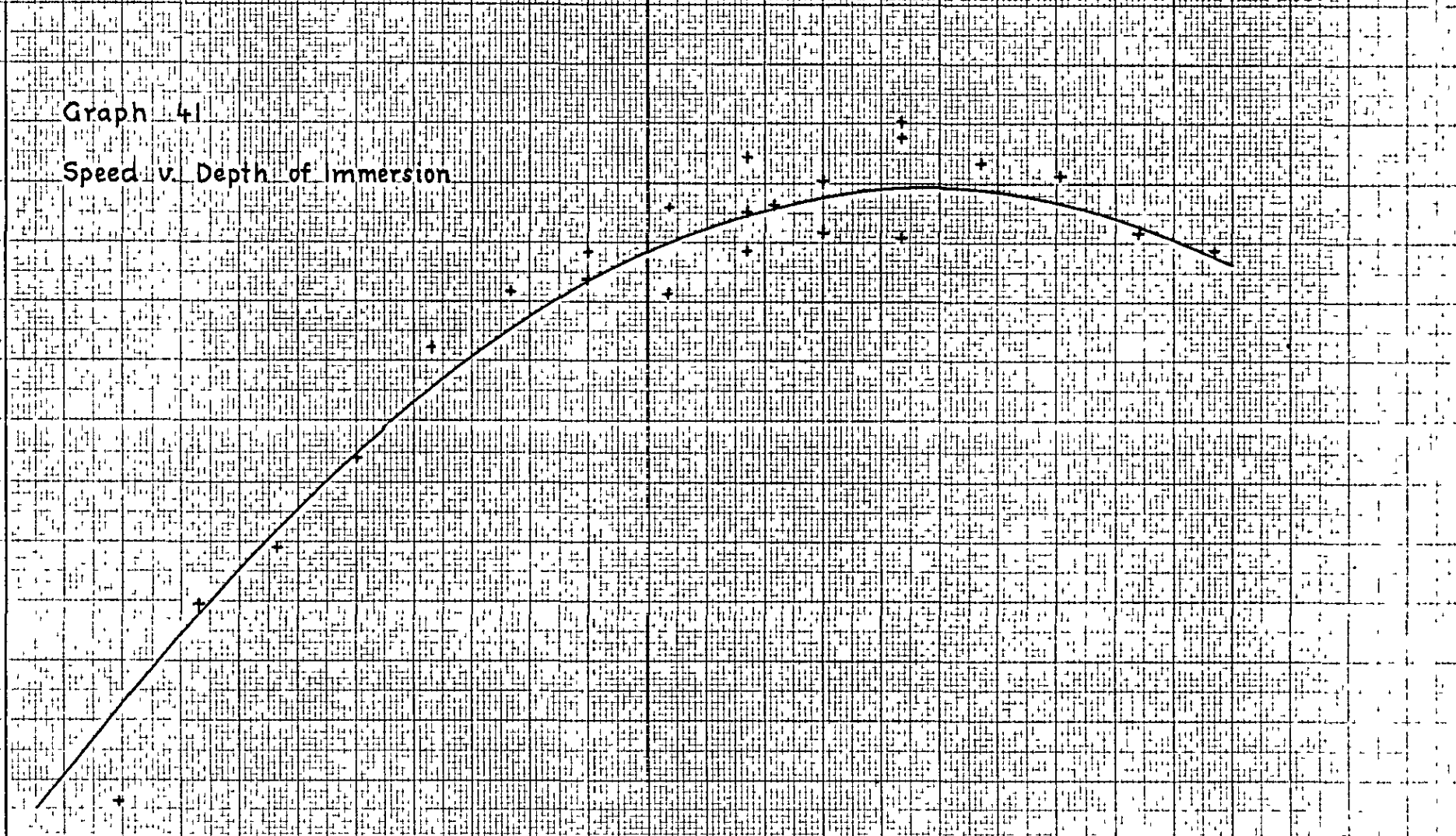
9

8

7

6

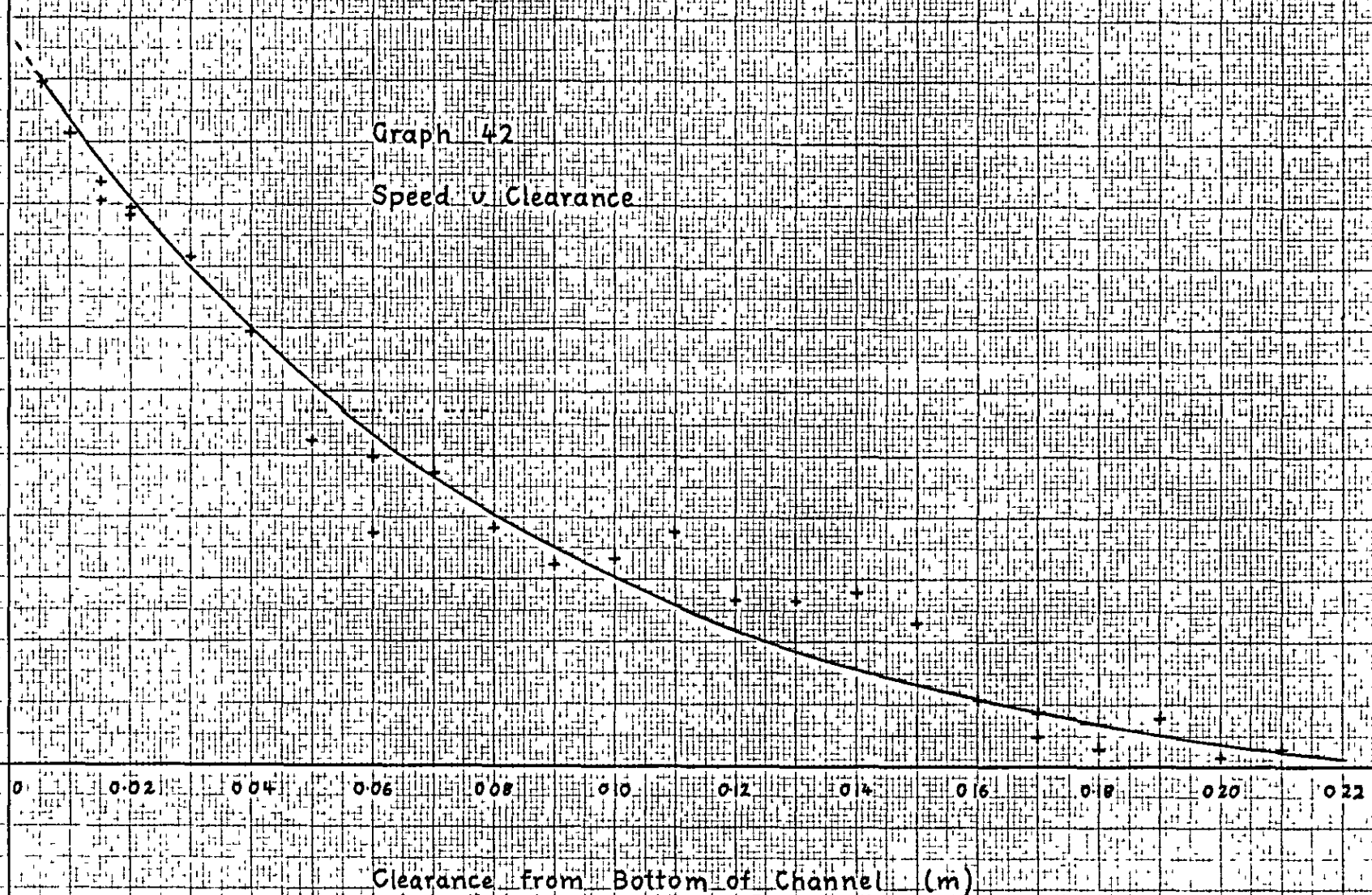
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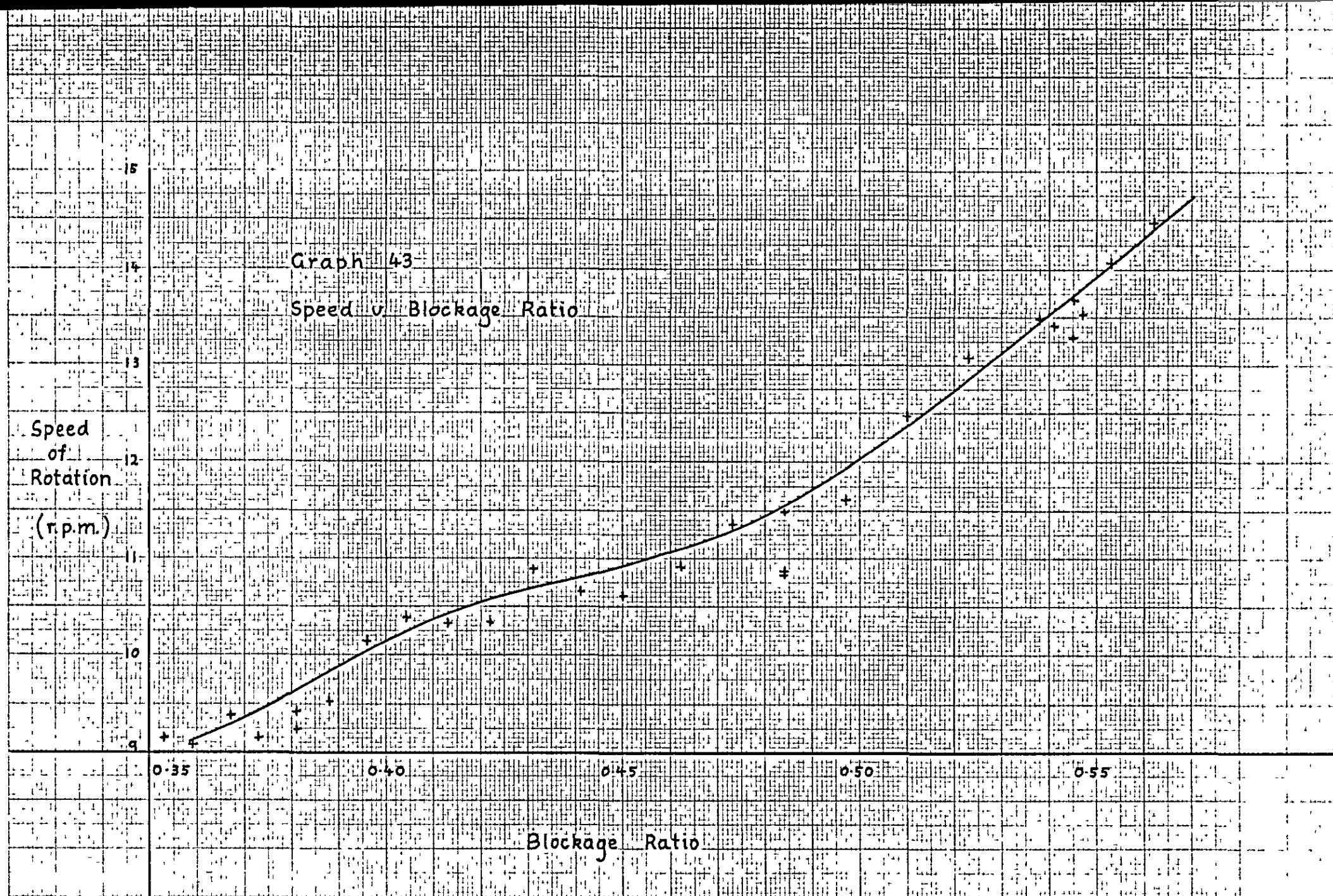


Graph 42

Speed v Clearance

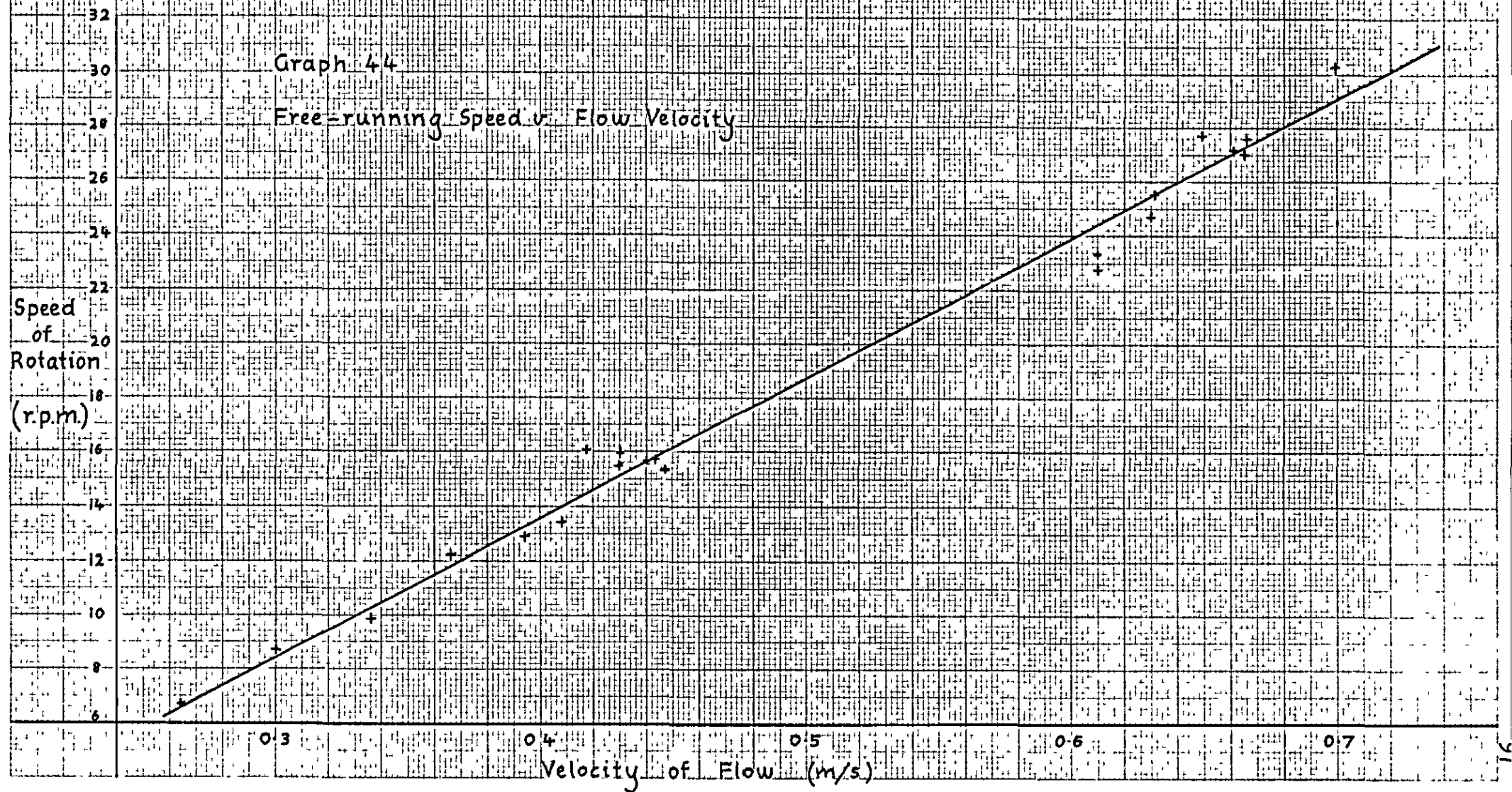
Speed
of
Rotation
(r.p.m.)





Graph 44

Free-running Speed v. Flow Velocity



Graph 45

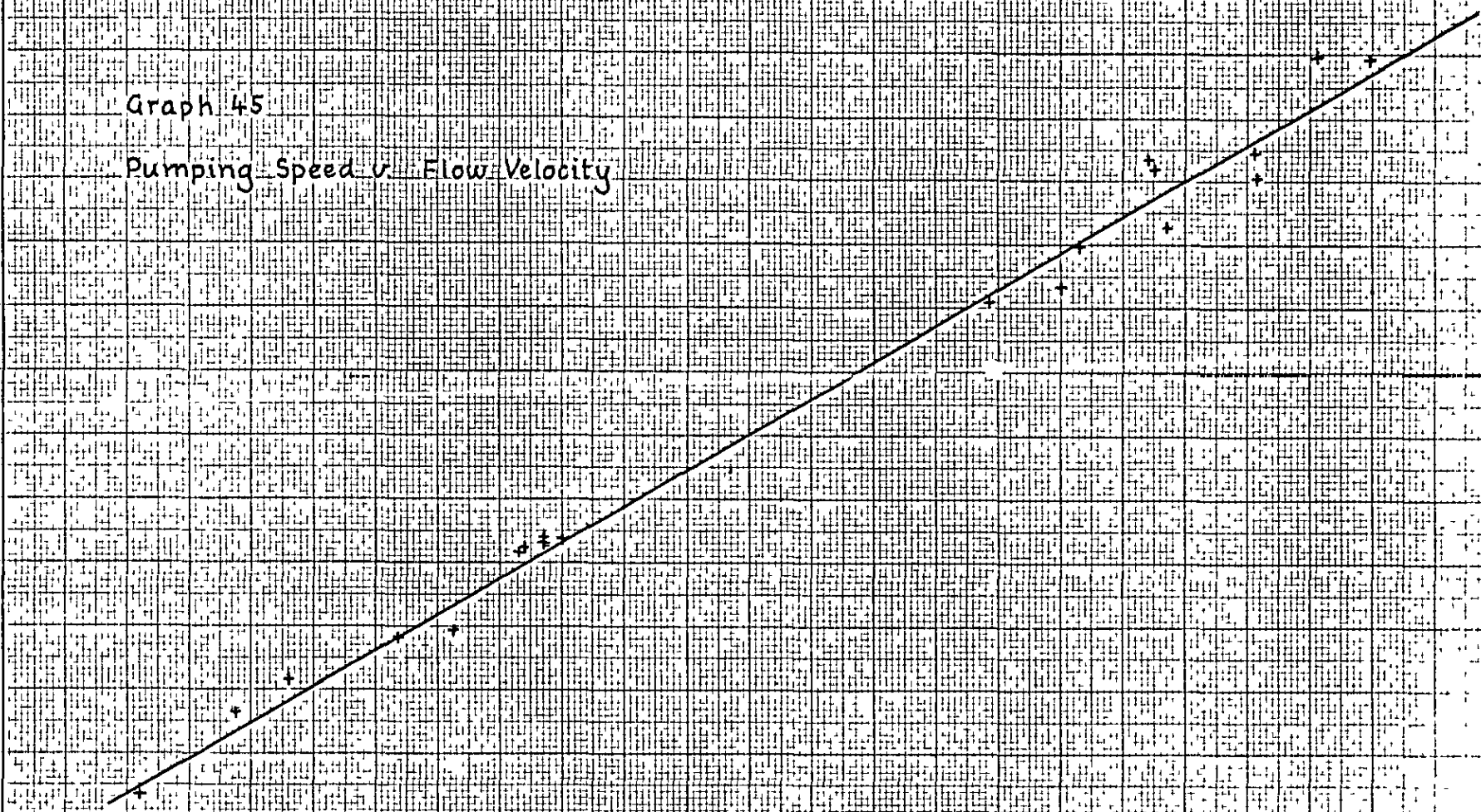
Pumping Speed v Flow Velocity

Speed
of
Rotation
(r.p.m.)

Velocity of Flow (m/s)

30
28
26
24
22
20
18
16
14
12
10
8
6

0.3 0.4 0.5 0.6 0.7



Graph 46

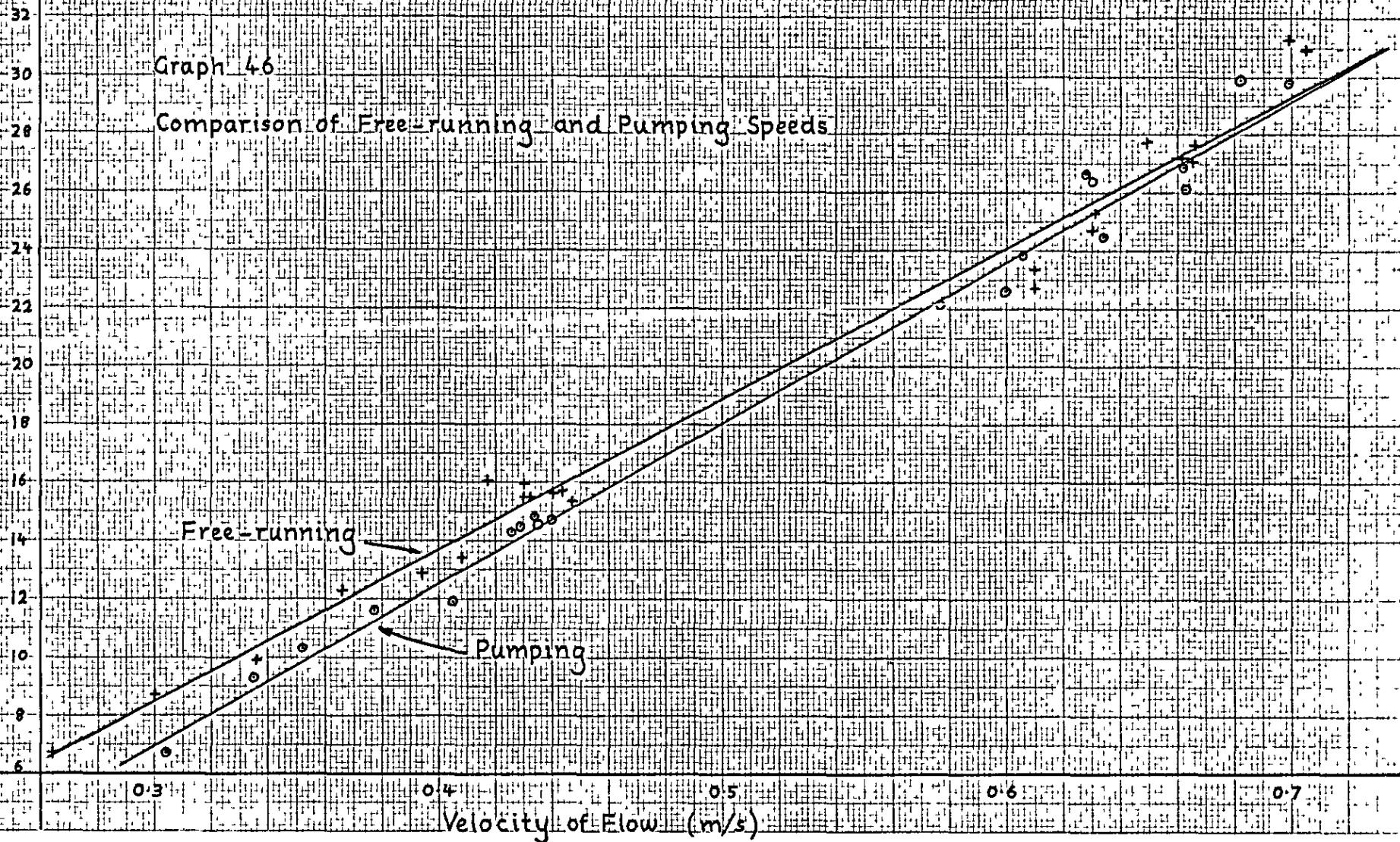
Comparison of Free-running and Pumping Speeds

Speed
of
Rotation
(r.p.m.)

Free-running

Pumping

Velocity of Flow (m/s)



4.3 Recommendations for Further Tests

The tests done on the model in the laboratory flume yielded results and characteristics applicable to that particular model only, although the general form of the plots may hold true for all waterwheels of this type. In order to formulate general rules governing all the different possible arrangements of the chevron-bladed waterwheel, a complete hydraulic study needs to be carried out involving a thorough mathematical analysis of the mechanisms affecting the performance of the wheel and the way in which this performance is affected by altering the many variables concerned.

Investigation of the force on the blades as they pass through the water could be particularly helpful and this could be done by fixing strain gauges to the faces of the blades and recording their output throughout one revolution. Thus the effects of varying the blade size, angle and number and also the interaction of the blades on each other can be studied.

From the results of such a study it would be possible to predict the performance of any given size and shape of waterwheel and also to recommend a size of waterwheel in a situation where the standard oil drum was neither available nor suitable. Possible improvements to the wheel include shrouding and ventilation of the blades and it would be useful to know over what ranges these methods increase the performance and efficiency of the wheel.

5.0 TESTS ON FIELD MODEL

5.1 Construction of the Working Pump

Throughout the design and construction of the pump to be used in the field tests, it was kept in mind that if the pump were to be suitable for Third World use, then the technology and materials used would need to be readily available in developing countries. To this end, the model was constructed largely of scrap materials and standard fittings, and the skills used in the making of the pump were kept to a minimum. The main stumbling block was the sealed rotary joint which needed accurate lathe machining and was made of plastic which is not readily available. However, it was felt that this component of the pump could be manufactured in bulk by more industrialised communities and transported to the intended location of the pumps.

For the actual drum itself an old 45 gallon oil drum was used. Initially it was planned to cut the vanes out of the shell of the drum itself, bending them up into position and leaving sections of the outer surface in position to provide rigidity and strength. This method was impractical, however, for two reasons: (a) when the curved surface of the drum was cut, the individual sections (those to be bent up to form blades) were unstable and moved freely; and (b) it was impossible to bend one part of a section normal to the former surface of the drum whilst leaving the other part in position.

Therefore this method of construction of the waterwheel part of the pump was abandoned and instead the vanes were cut out of flat sheets of steel and spot welded on to the surface of the drum. This was much simpler and although new sheet steel was used for the vanes, steel cut from the body panels of scrap cars or lorries would work quite adequately.

The outlet arrangement (including the rotary sealed valve) were constructed, as in laboratory models, from Darvic plastic, and this was attached to a circular steel plate that replaced the end of the drum which was removed in order to place the coiled pipe and bouyancy inside the drum. This circular plate was fixed to the drum by means of self-tapping screws around its circumference.

At the inlet end of the drum, the bearing was constructed from Darvic plastic and was bolted to the drum, whilst at the outlet end the bearing was formed by placing a ring around a flat section of the outlet joints.

The material originally used to provide buoyancy for the pump was small chips of polystyrene, packed in plastic bags. By adjusting the position and volume of these bags the uplift produced by the bouyancy was altered until the pump floated level and with the axis a few centimetres above the water level. When working, the weight of water inside the coils would bring the axis down to water level giving a depth of immersion of 0.5.

To hold the pump in position in the stream, galvanised steel poles were driven into the bed of the stream and various methods of fixing the pump to the poles were tried throughout the course of the field work. The first restraining apparatus used were two 12.5 mm diameter rods, roughly 900 mm long, which were threaded at both ends. The poles (galvanised steel) were situated 900 mm upstream from the intended position of the pump and the bars pushed through holes drilled in the poles at water level. Bolts either side of the poles held the bars firm. The bars were screwed into cored holes in the bearing casings.

This process of threading the bars and coring the holes requires a lathe and suitable dies. An alternative way of fixing the bars to

the poles would be to drill small holes into the bars into which split pins would be placed. The bearing cases could be attached to the bar by bending a metal strip around the casing and bolting this to the bar, through a hole drilled in the bar.

As the total length of piping in the pump exceeds 40 m, it is likely that two or more pieces will need to be joined together. It is essential that the joints are air-tight, as any leaks will result in a loss of pressure. The method used was to push both ends of pipe in to a tubular metal sleeve which had been smeared with glue. The ends were firmly held on to the sleeve by tightening hose-clips around the pipe. This type of connection was also employed to attach the flexible piping on to the rigid outlet column.

During the field tests, the original model was improved upon, and these modifications are described in the next section.

5.2 Modifications Made to Pump During Field Tests

As well as modifications to the pump, this section also describes modifications made to the channel in which the pump was tested. The section of the stream chosen for the tests consisted of a narrow channel with a depth of 0.4 m and a flow of 0.4 m/s, and a broad, shallow region. Preparation of the stream involved excavation of the main channel and the building of a gabion barrier across the shallow region. After completion of the excavation, a channel was formed, 1.5 m wide and 0.55 m deep with a flow velocity of about 0.7 m/s. In addition to these rearrangements, boards were also placed in the stream, slightly upstream of the pump, to vary the flow rate on to the pump.

The delivery pipe was supported from a nearby tree and connected to a larger diameter down pipe so that the quantity of water being

pumped could be measured. It was ensured that the delivery pipe was made open to the atmosphere at the top of the lift so that no siphoning mechanism was operating.

The pump was installed supported by rigid bars and was allowed to rotate freely before the delivery pipe was connected, thus giving an idea of how much power was required when pumping commenced. The free-running speed was in the region of 6 r.p.m. and when the delivery pipe was connected the pump slowed down and eventually stopped before any water was pumped to the required height (9.5 m). The two main reasons for this failure were, (i) the polystyrene pieces used as bouyancy were presenting too much friction as they passed through the water inside the drum; (ii) as the flow was hitting the blades it was deflecting sideways and escaping around the sides of the drum instead of travelling under the drum and thus pushing the paddles along.

So the first modifications involved a change of buoyancy material and shrouding of the water wheel. Instead of the polystyrene chippings, old car inner tubes were used. These fitted well around the inside of the drum leaving a space down the middle which was filled with two more inner tubes pushed lengthwise into the space. The shrouds were made the same depth as the blades and spot-welded on to the ends of the drum and also on to the trailing edges of the vanes (see Figure 11).

These modifications resulted in a large improvement of the water wheel, the free-running speed increasing to 9 - 10 r.p.m. and pumping speed being maintained at 8 r.p.m., delivering 3.0 l/min to 9.5 m.

The next modification was to drill ventilation holes in the shroud. This was to prevent water trapped between the trailing edge of the vanes and the shroud from being lifted above the water surface and thus consuming power. With this modification, water was still

lifted but all drained out of the holes before it was carried over the top of the wheel.

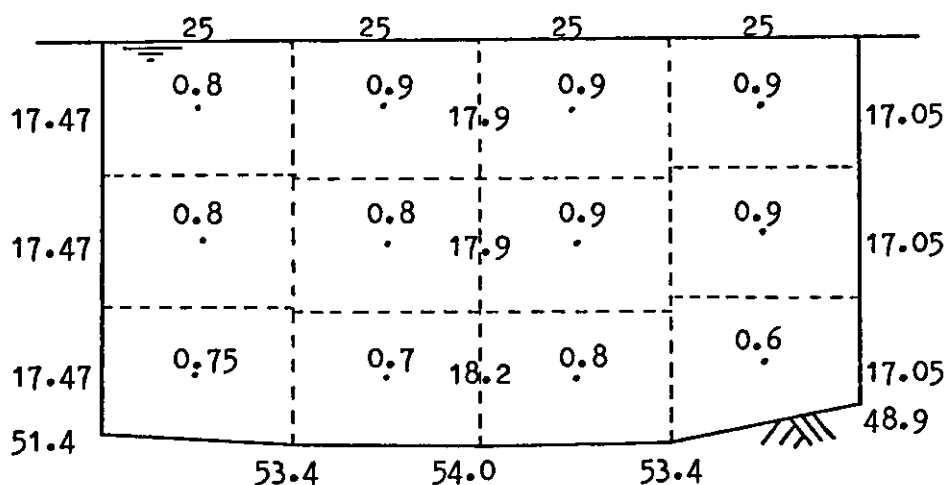
It was this final design that was used for field tests.

5.3 Performance of the Field Test Pump

When the pump was deemed to be operating satisfactorily, tests were carried out to simulate the situation which would occur when the pump was installed in a stream with varying flow rate.

Firstly, the pump was removed and the depth of the channel was measured across the width of the pump, indicating the bed profile shown in Figure 12. The depth at one point was measured throughout the tests, to show any rise or fall in the stream level, and in all the tests the flow was measured by taking velocity readings with a flowmeter in twelve points over the channel as shown in the example below.

The first test was done with a complete dam across the shallow part of the stream and boards deflecting all of the flow on to the pump. In successive tests the boards were removed and the dam gradually breached to reduce the flow to the pump. The velocities recorded for the first test are as shown below.



All measurements in cms;
velocities in m/s.

$$\begin{aligned} \text{Total flow of stream} = & 0.8 (0.0433 \times 2) + 0.75 \times 0.0433 + \\ & 0.9 (0.0444 \times 3) + 0.7 \times 0.0444 + 0.9 (0.0422 \times 2) + \\ & 0.6 \times 0.0422 \end{aligned}$$

$$Q_{ST} = 0.425 \text{ m}^3/\text{s}$$

$$\text{Mean velocity (weighted average)} v_{ST} = 0.813 \text{ m/s}$$

With the pump installed, the depth remained the same and the respective velocities were:-

$$\begin{array}{cccc} 0.8 & 0.8 & 0.9 & 1.0 \\ 0.8 & 0.8 & 0.85 & 1.1 \\ 0.7 & 0.6 & 0.6 & 0.55 \end{array}$$

$$Q_{ST} = 0.413 \text{ m}^3/\text{s}$$

$$v_{ST} = 0.790 \text{ m/s}$$

Pump data:-

$$\text{Speed} = 10 \text{ r.p.m.}$$

$$Q_p = 3.8 \text{ l/min}$$

The results of the five tests carried out are summarised in the table below:-

Test	Q_{ST} m^3/s	v_{ST} m/s	Pump speed r.p.m.	Q_p l/min	Q_p l/rev
1	0.413	0.790	10.0	3.8	0.380
2	0.360	0.641	7.33	2.7	0.368
3	0.293	0.522	5.75	2.2	0.383
4	0.262	0.467	4.0	1.7	0.425
5	0.262	0.467	3.25	1.1	0.328

See Graph 47

Cut-away diagram of field test model

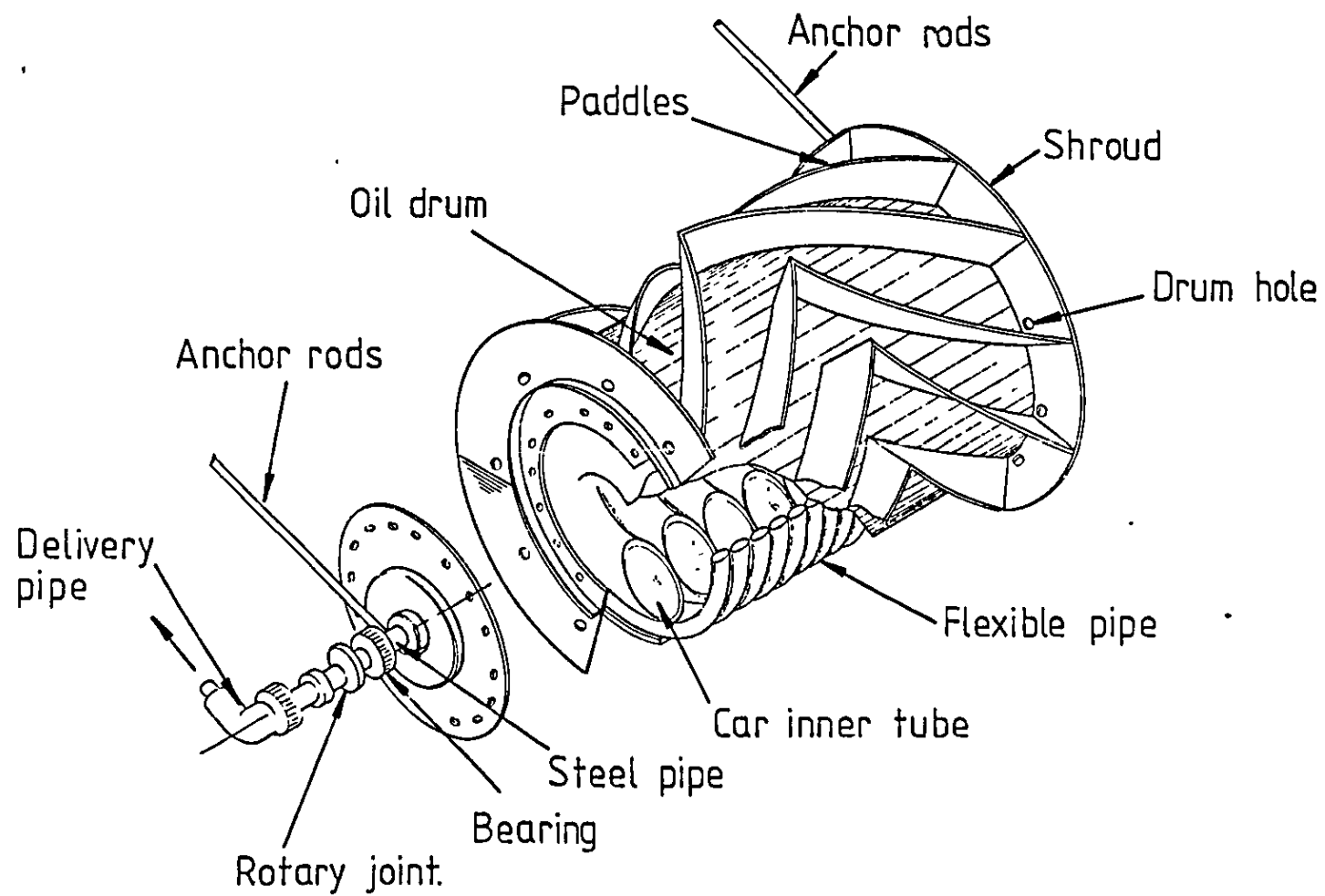
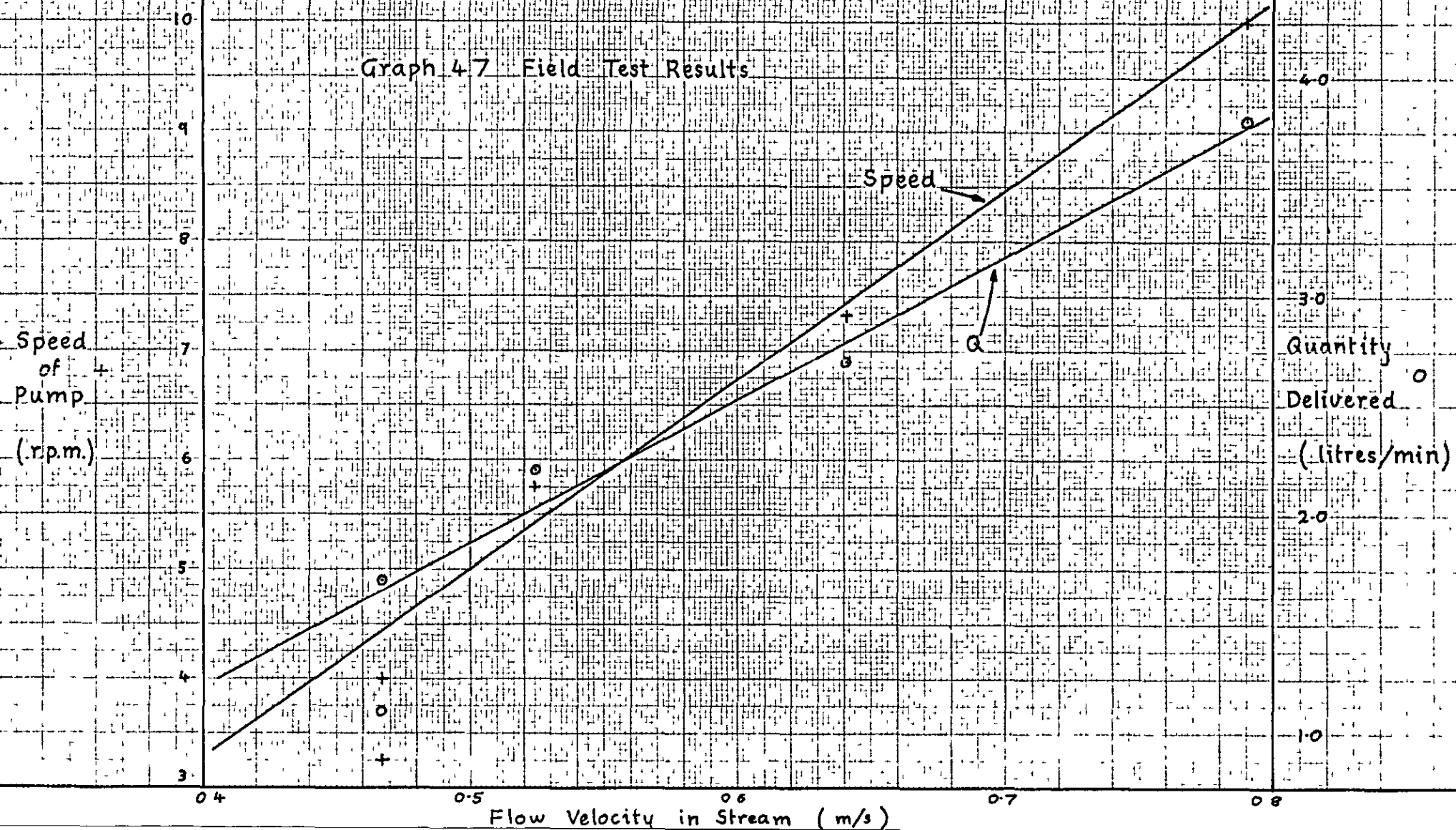


Figure 11

Graph 4.7 Field Test Results



5.4 Discussion of Performance of the Field Test Pump

It should firstly be noted that although the whole of the shallow section of the stream was made open between tests 4 and 5, the results show no reduction of the flow. It is unlikely that this was the case and the flow recorded in test 5 may be assumed to be erroneous, a fact confirmed by the points on Graph 47 which correspond to this test. The speed of rotation recorded would suggest a flow velocity of around 0.4 m/s but the pumping rate for this test is also low as is the quantity of water pumped per revolution which suggests that the inlet of the pump had risen out of the water, an effect of buoyancy shift inside the drum which would unlevel the drum and result in a speed reduction. In Test 4, the value of Q_p/rev is high, indicating a buoyancy shift the other way, again resulting in a slowing effect, although the error is not so pronounced in this case.

Taking these errors into account, it can be seen that the relationship between speed of flow and speed of revolution is as expected from the laboratory tests, i.e. a straight line. Very little change in depth of flow took place throughout the tests so the clearance off the bottom of the stream was fairly constant, thus giving a true test of speed comparisons with little scatter.

The pumping height of 9.5 m was the maximum to which the delivery pipe could be raised up a nearby tree. The horizontal distance from the pump to the base of the tree was almost 10 m, so the delivery pipe was inclined at roughly 45° . Recent work done on sloping delivery pipes indicates that they behave very similarly to vertical deliveries, and that the value of delivery pipe ratio R_D is constant for a particular pumping height regardless of the angle of inclination. ^{Ref. 7.} Therefore, the same program can be used to calculate R_D and give a value for the

head developed by the pump to achieve this lift.

INPUT HEIGHT OF DELIVERY TANK
 ROTATIONAL SPEED OF PUMP
 MEAN COIL DIAMETER
 PROPORTIONAL DEPTH OF IMMERSION
 I.D. OF PIPE ON PUMP
 I.D. OF DELIVERY PIPE
 9.5, 19, 0.544, 0.5, 0.025, 0.025

QUANTITY DELIVERED = 3.593 L/MIN
 MAX. RATIO = 1.410
 MIN. RATIO = 1.322
 MEAN RATIO = 1.369
 OK,

The minimum ratio given is 1.322, i.e. a maximum head of $9.5/1.322 = 7.186$ m will need to be developed to ensure that this lift can be coped with. Feeding this value into the coil pattern program results in the values shown on Graph 48 indicating that, as only 24 of the 26 coils are being used, the pump is not lifting to its maximum possible height. By increasing the value of the head fed into the program we eventually obtain a pattern for 26 coils as shown in Graph 49. The head developed ($\sum h$) in this case is 7.65 m and by using a trial and error method with the delivery pipe program, this gives a maximum lift for the pump of 10.18 m.

The pump with the 16 large bore coils lifted to 7.5 m, the delivery pipe ratios for which are shown below:-

$$R_D \text{ max} = 1.819$$

$$R_D \text{ min} = 1.316$$

$$R_D \text{ mean} = 1.567$$

when $R_D = 1.316$, head developed = 5.70 m.

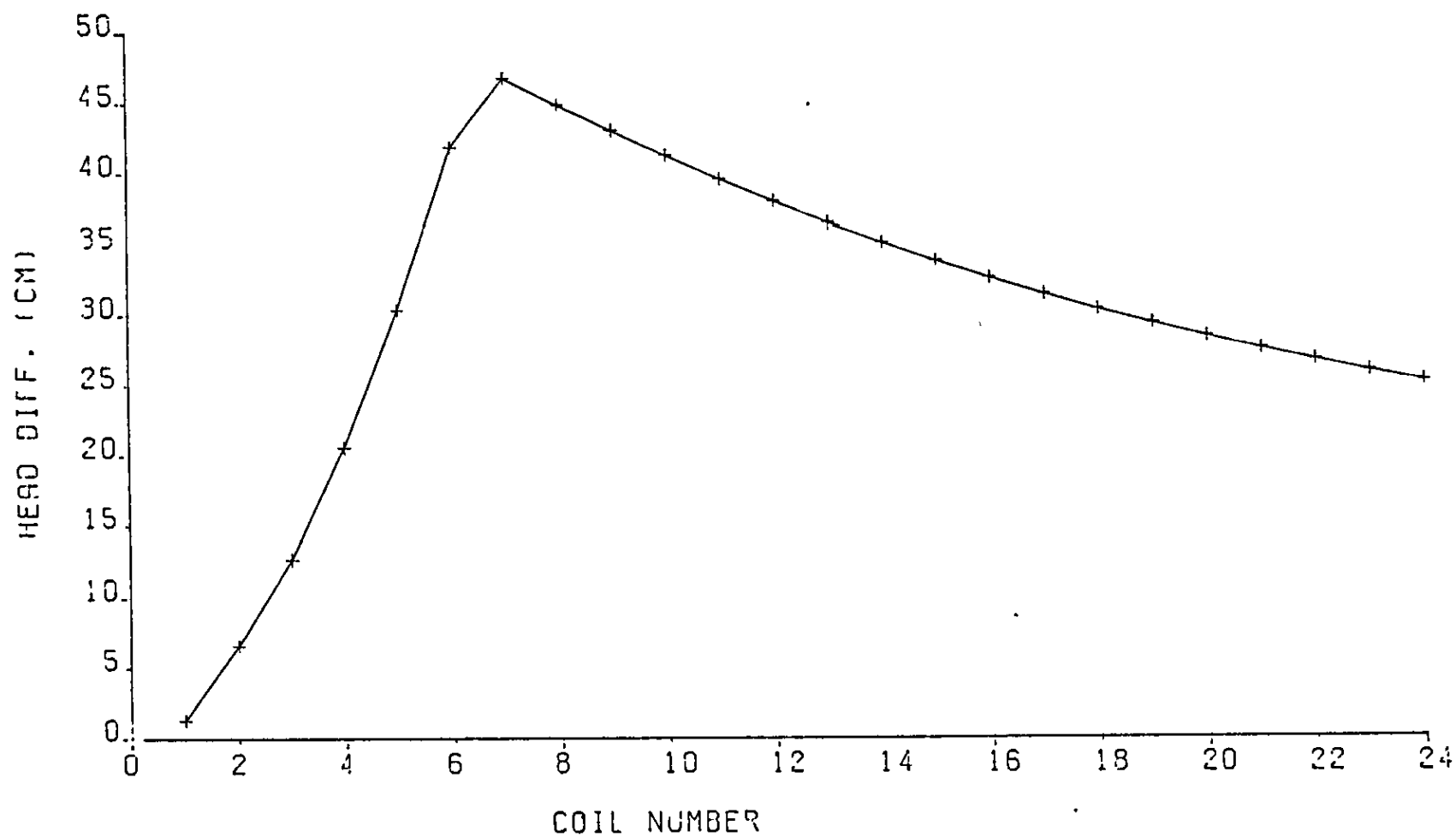
When developing this head, the pump requires 19 coils (see Graph 50). However, if the mean ratio is used, the head developed is 4.79 m which

requires only 15 coils (see Graph 51). It can be assumed therefore that while developing this head the pump can occasionally produce higher pressures in excess of those predicted by the program, but the minimum ratio should be used in any design of a pump to ensure that the lift can definitely be achieved 100% of the time.

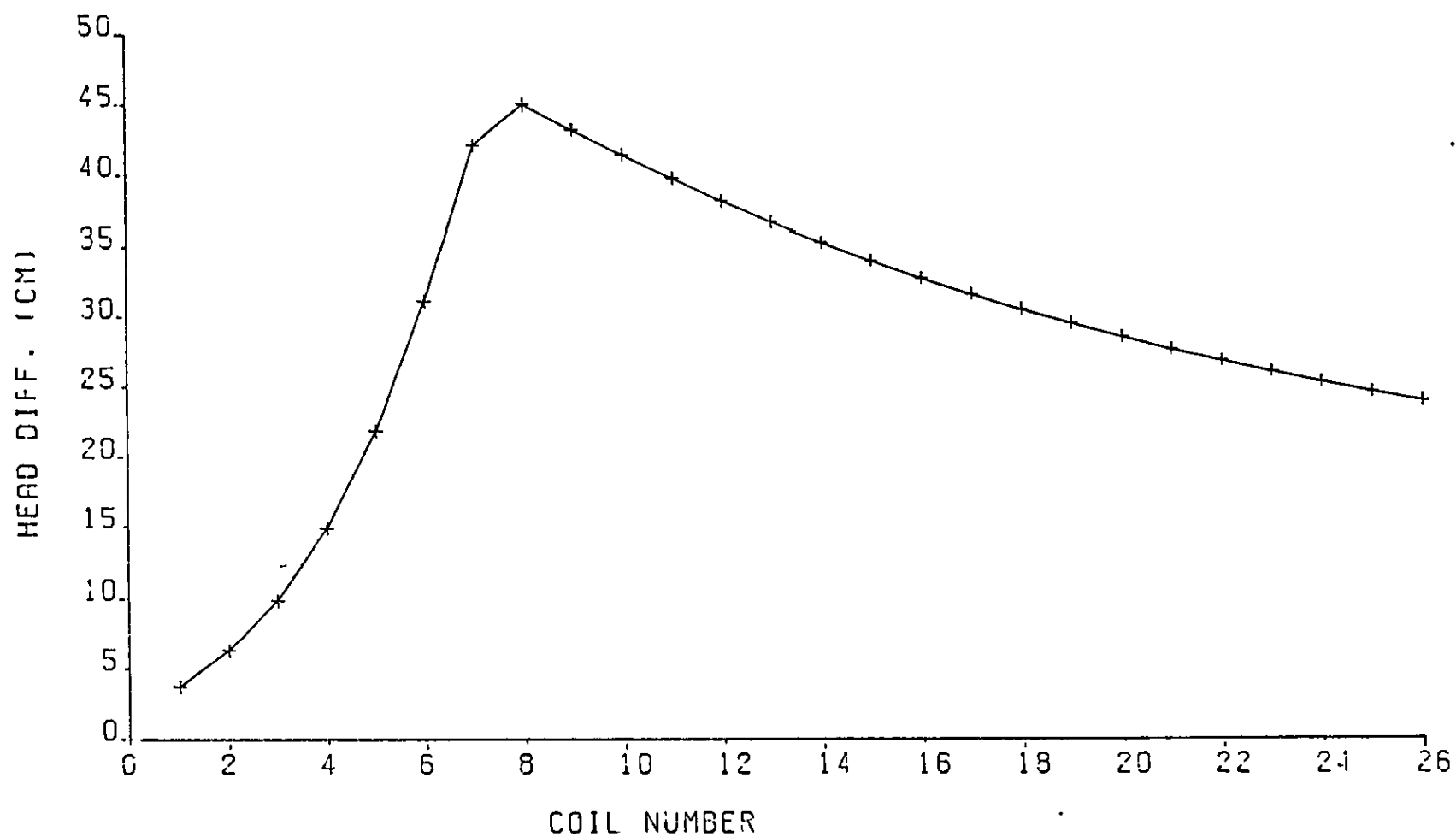
Although no continuous stream flow measurements were taken during the field tests to obtain any values of the efficiency of the pump, observations made during the laboratory testing of the waterwheel suggest that overall efficiencies of between 3% and 9% can be expected of this arrangement. Since a great proportion of the power absorbed is used to overcome friction in the bearings, the value of the efficiency increases with the lift. Redesign of a more suitable bearing for the shaft of the drum should result in a better performance, with overall efficiencies possibly reaching 20%. From an economic point of view, the efficiency of the pump is not important as the power source is free. However, any gain in performance will improve the chances of the coil pump being accepted as a viable alternative to more costly and complicated pumps.

FIELD PUMP LIFTING TO 9.5M.

GRAPH 48

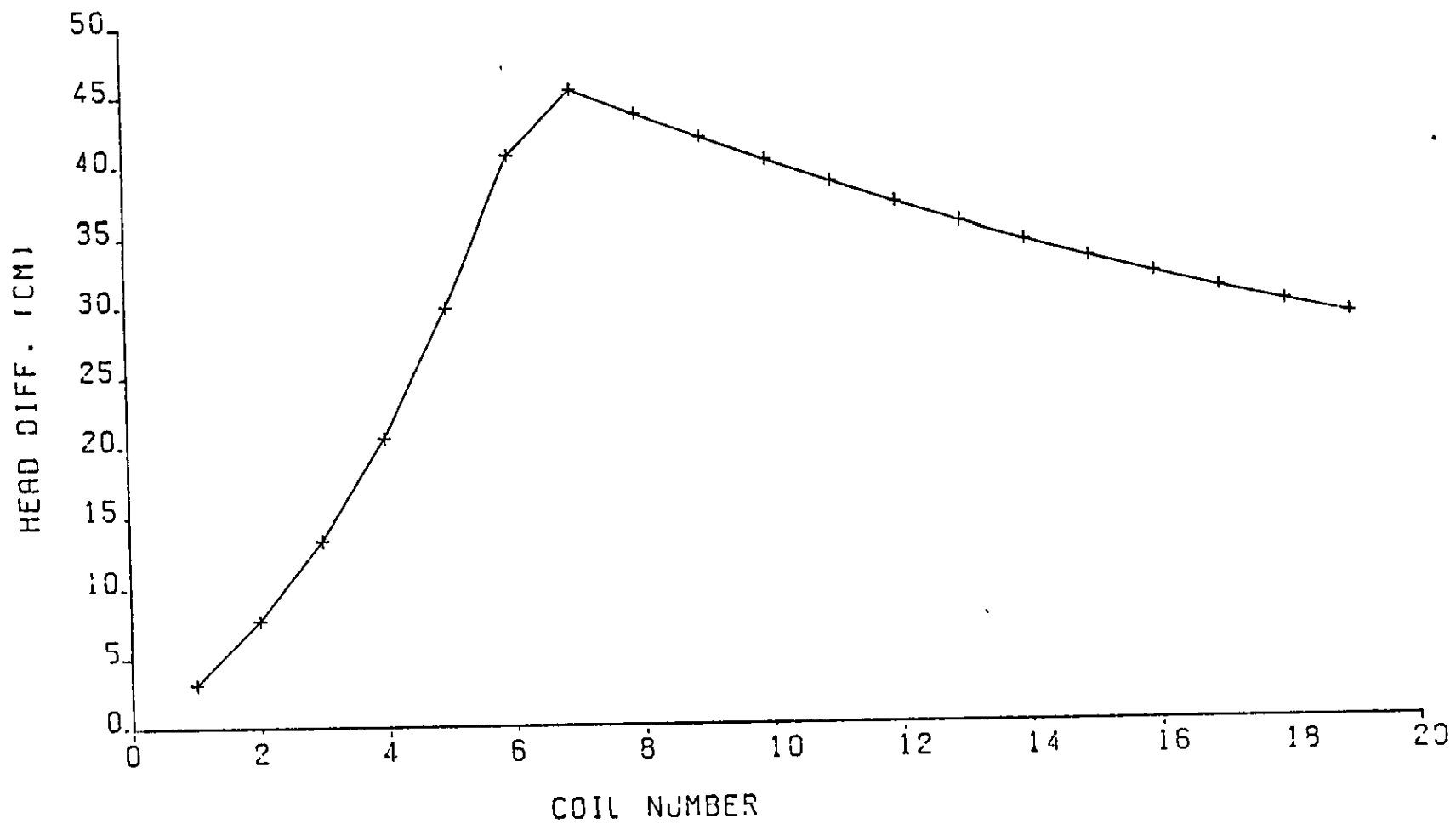


FIELD PUMP DEVELOPING 7.65M. HEAD GRAPH 49

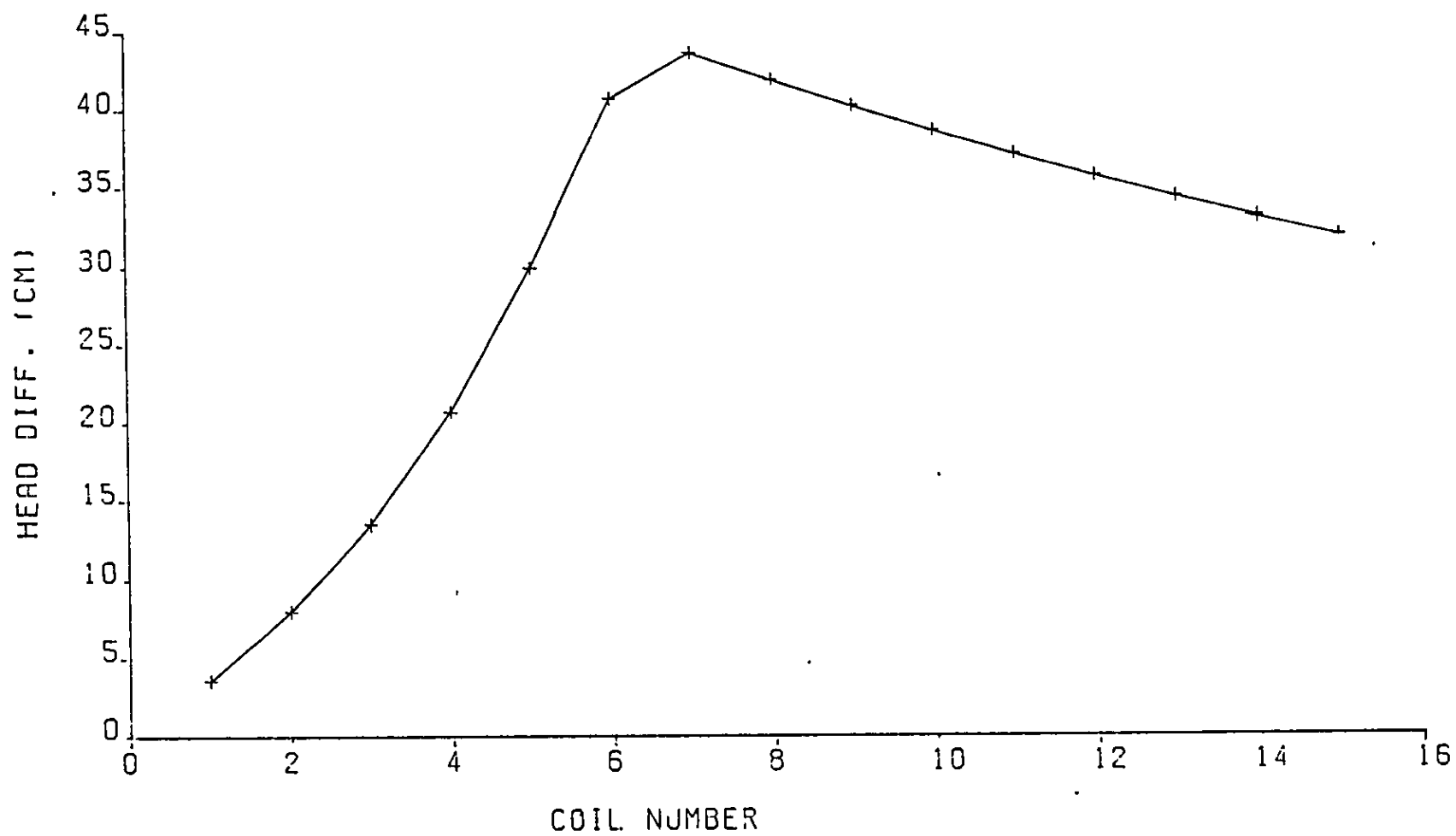


FIELD PUMP LIFTING TO 7.5M. (LARGE PIPE)

GRAPH 50



7.5M. LIFT , MEAN RATIO (HEAD =4.79M.) GRAPH 51



6.0 DESIGN CHARTS

One of the main aims of the research and development done on the coil pump is to produce a comprehensive design and construction manual for use by engineers and technicians in the Third World. Since there are no simple formulae relating the number of coils to the lift produced, then charts must be used which give a number of design solutions for various sizes of pump.

The principle problem in producing these design charts concerns attempting to group the variables governing the pump's performance (radius of coils, bore of pipes, speed of rotation, depth of immersion) into dimensionless quantities in order to produce a small number of charts which will be applicable to a great many variations of pump specification. All these variables have an effect on both the delivery pipe ratio and the number of coils needed to develop a particular head and the only possible grouping is in terms of radius of coils, i.e. a chart can be drawn for one peripheral velocity, coil bore and delivery pipe bore, but covering a range of coil radii.

To relate the lift of a particular pump to the number of coils required, two charts need to be drawn. The first relates the lift of the pump (relative to its diameter) to the head developed by the pump (relative to its diameter). This is done by computing the delivery pipe ratios for a number of cases with common values of peripheral velocity and pipe bore and with diameters within the specified range, and plotting these on the chart. A number of different plots are drawn representing different depths of immersion. This chart is termed the LH chart (Lift \rightarrow Head).

The second chart gives the number of coils required to develop this relative head and can cover a greater range of specifications

than the previous chart due to it being totally independent of the delivery pipe bore and only slightly varied by the peripheral velocity of the pump. This chart is termed the HN chart (Head \rightarrow Number of coils) and when combining the two charts to give plots of the relationship between lift and number of coils, one HN chart may be used in conjunction with many LH charts.

Two examples of design charts are shown, one for a small pump ($0.3 \text{ m} \leq D \leq 0.5 \text{ m}$) with the following specifications:-

Coil bore = 0.025 m

Del. pipe bore = 0.025 m

Periph. vel. = 0.3 m/s

and the second for a larger pump ($1.0 \text{ m} \leq D \leq 1.5 \text{ m}$) with values of:-

Coil bore = 0.038 m

Del. pipe bore = 0.025 m

Periph. vel. = 0.25 m/s

For both pumps both the LH, HN and combined charts are drawn showing the plots for 0.4, 0.5, 0.6 and 0.7 proportional depth of immersion.

Also included is a design chart for the standard oil drum pump, when fitted with 0.025 m bore piping throughout and rotating at 10 r.p.m. Since only one diameter is being considered the true values of lift and head can be used and the combined chart gives a direct relationship between lift and number of coils needed.

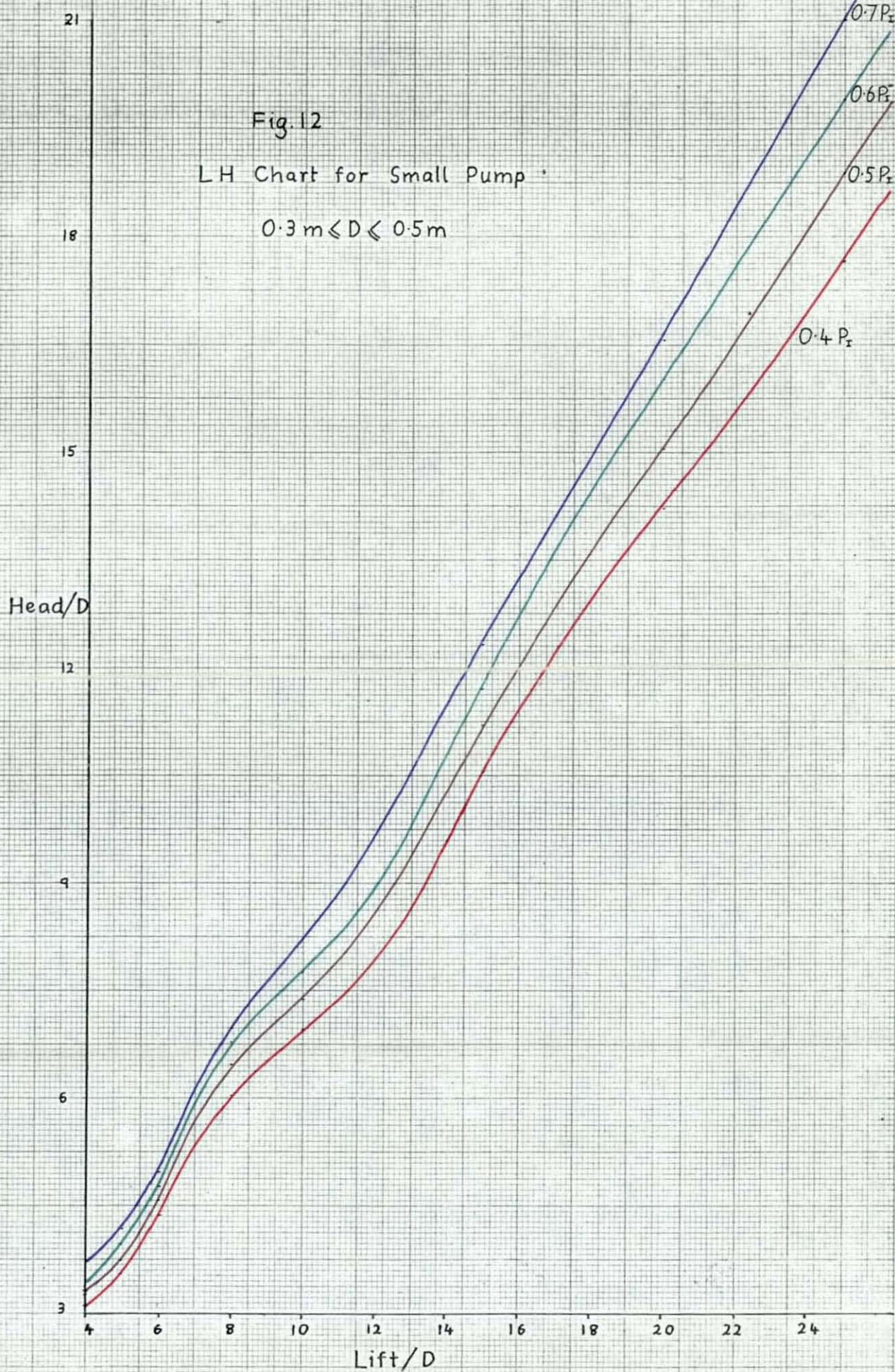


Fig 13
HN Chart for Small Pump

Head/D

21

18

15

12

9

6

3

 $0.4 P_r$ $0.5 P_r$ $0.6 P_r$ $0.7 P_r$

No. of coils

6 8 10 12 14 16 18 20 22 24 26 28 30 32 34 36

Fig. 14

Combined Chart for Small Pump

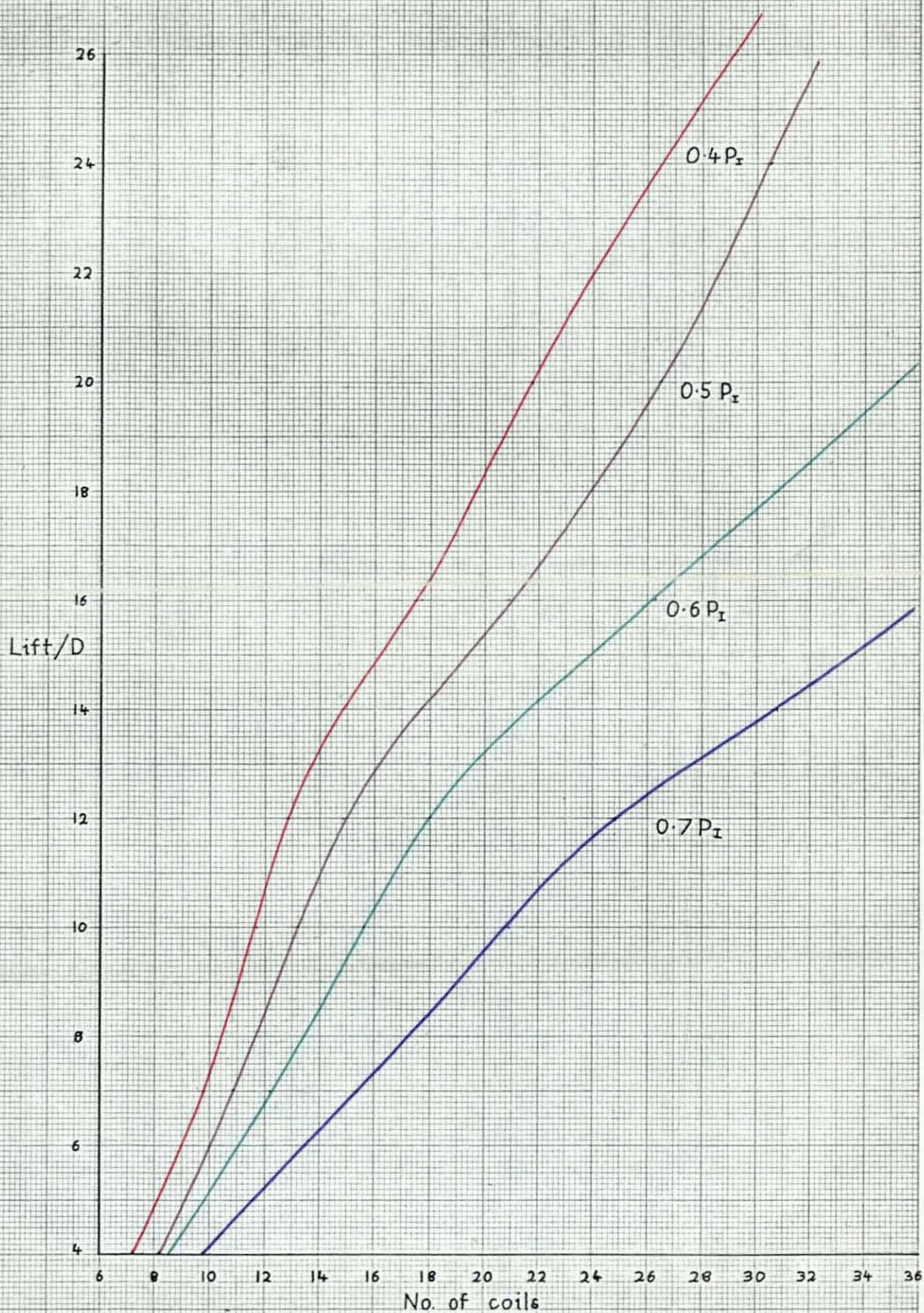


Fig.15

LH Chart for Large Pump.

 $1.0\text{m} \leq D \leq 1.5\text{m}$

Head/D

12

11

10

9

8

7

6

5

4

Lift/D

4

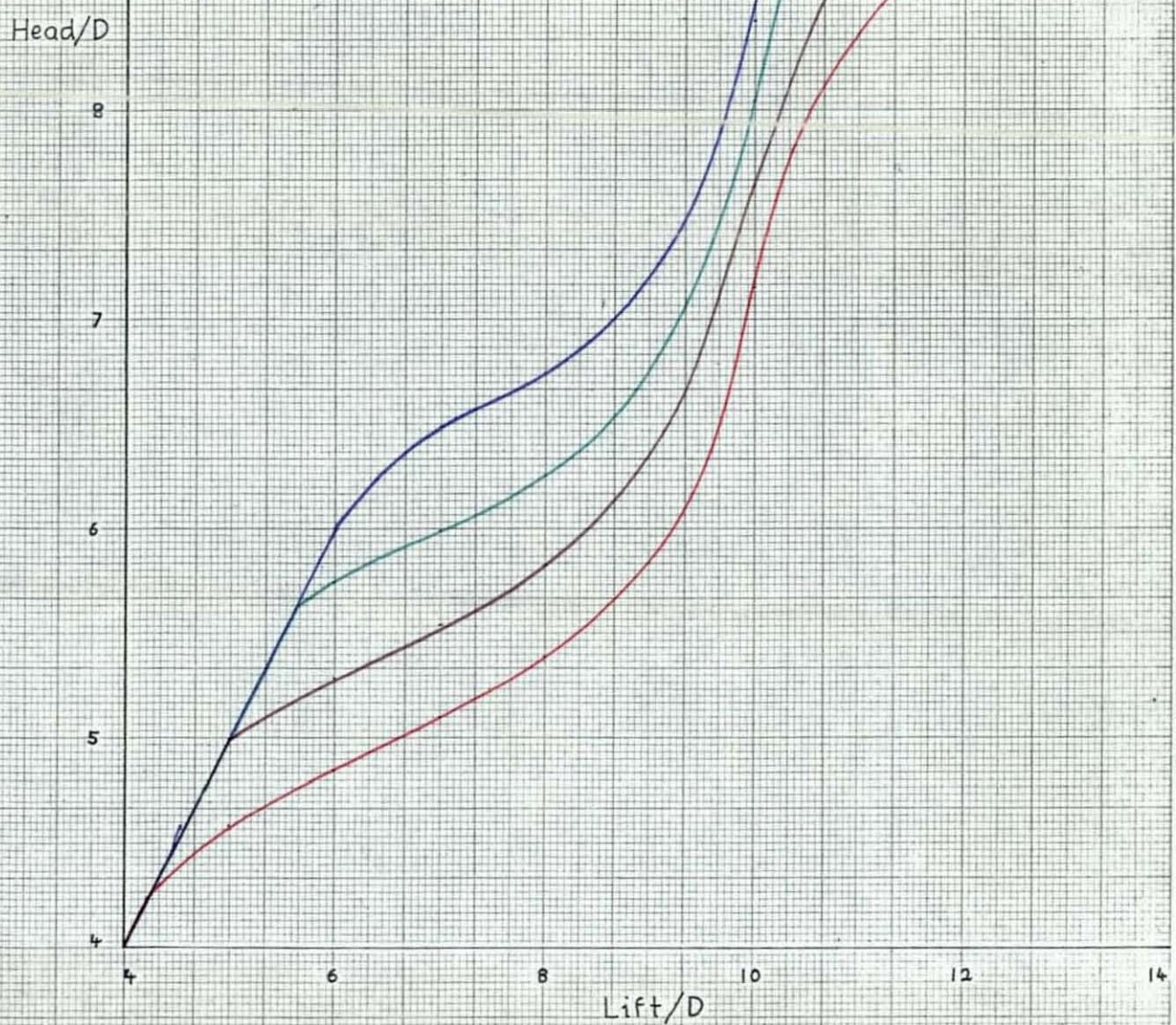
6

8

10

12

14

 $0.7P_r$ $0.6P_r$ $0.5P_r$ $0.4P_r$ 

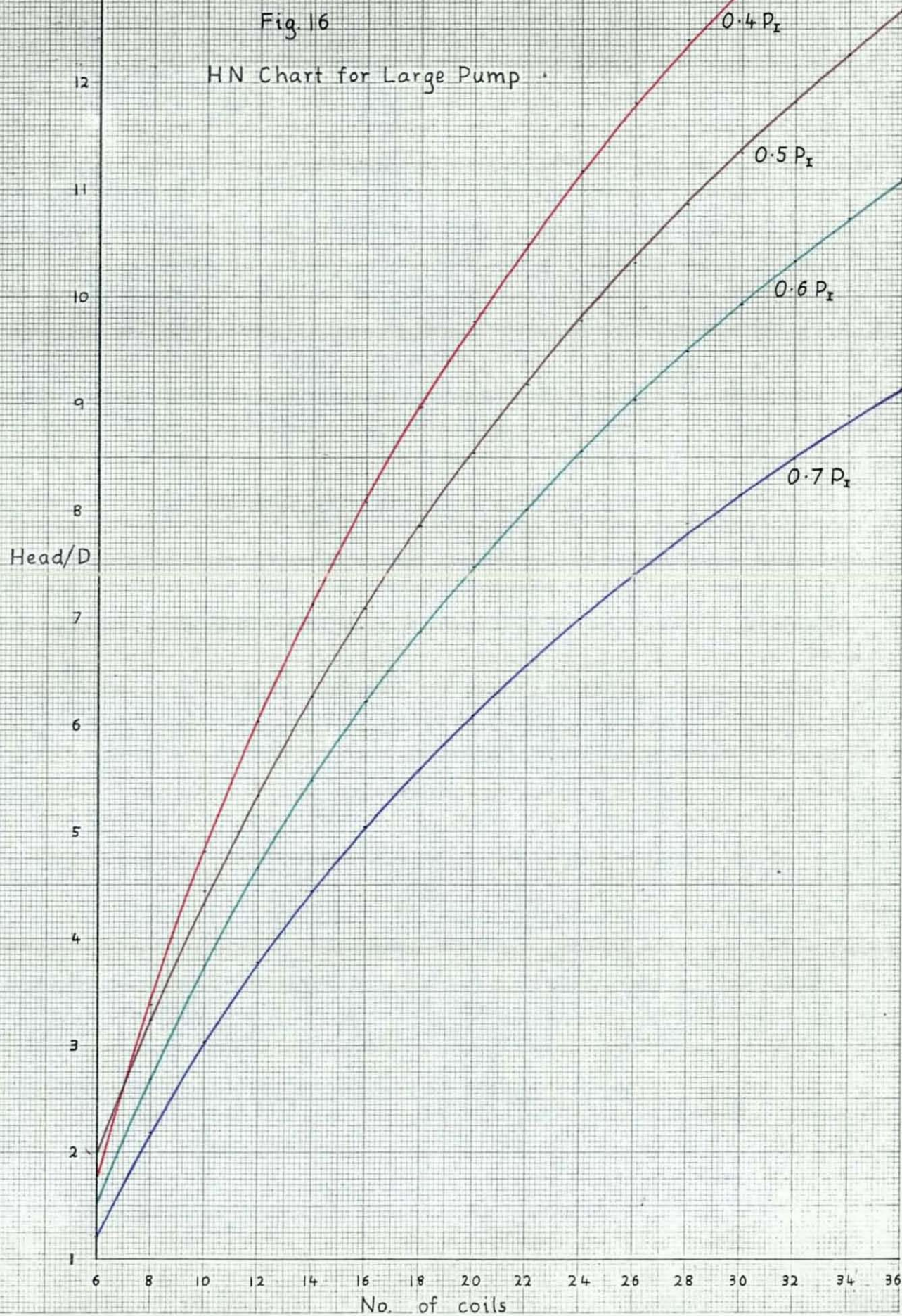
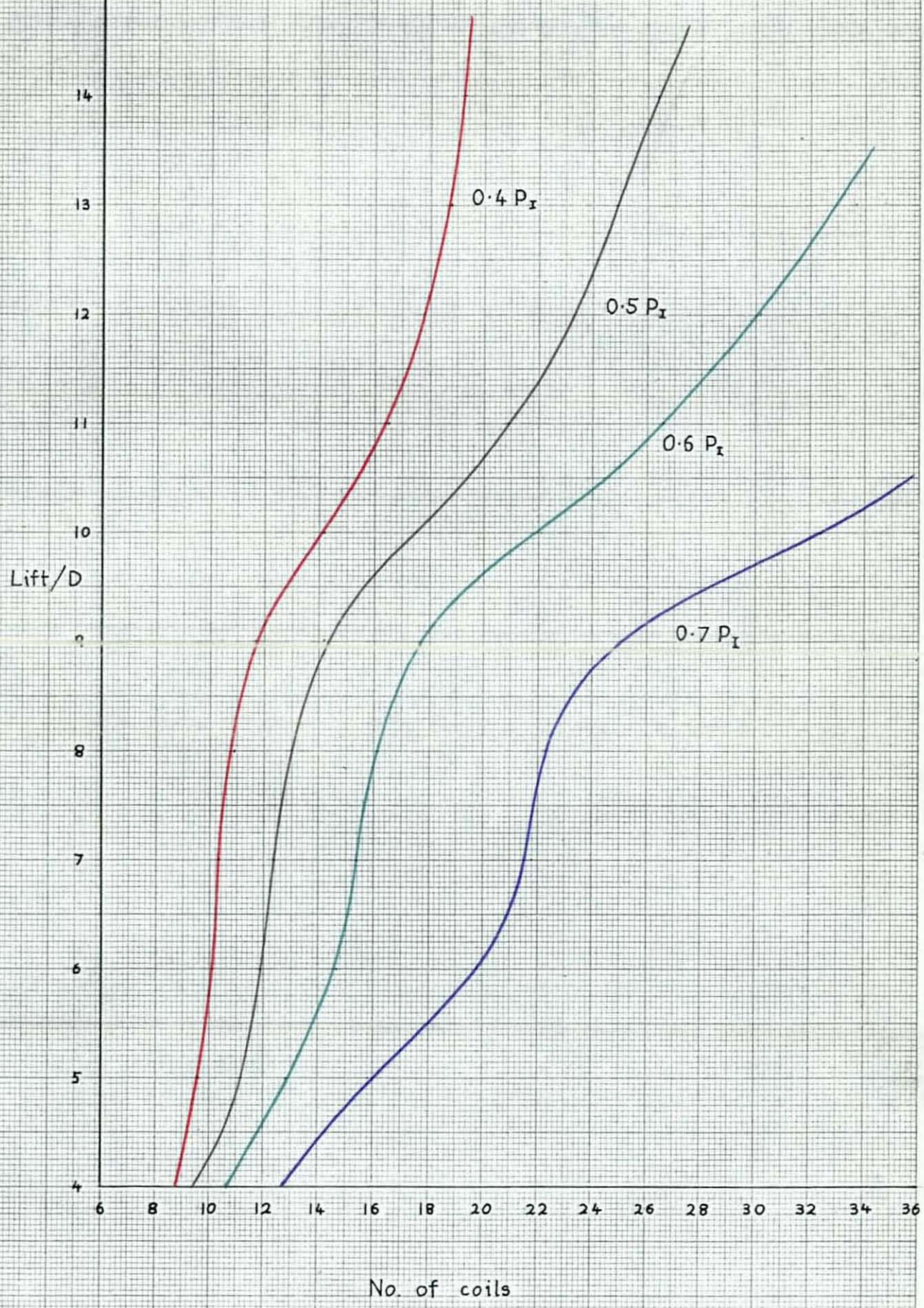


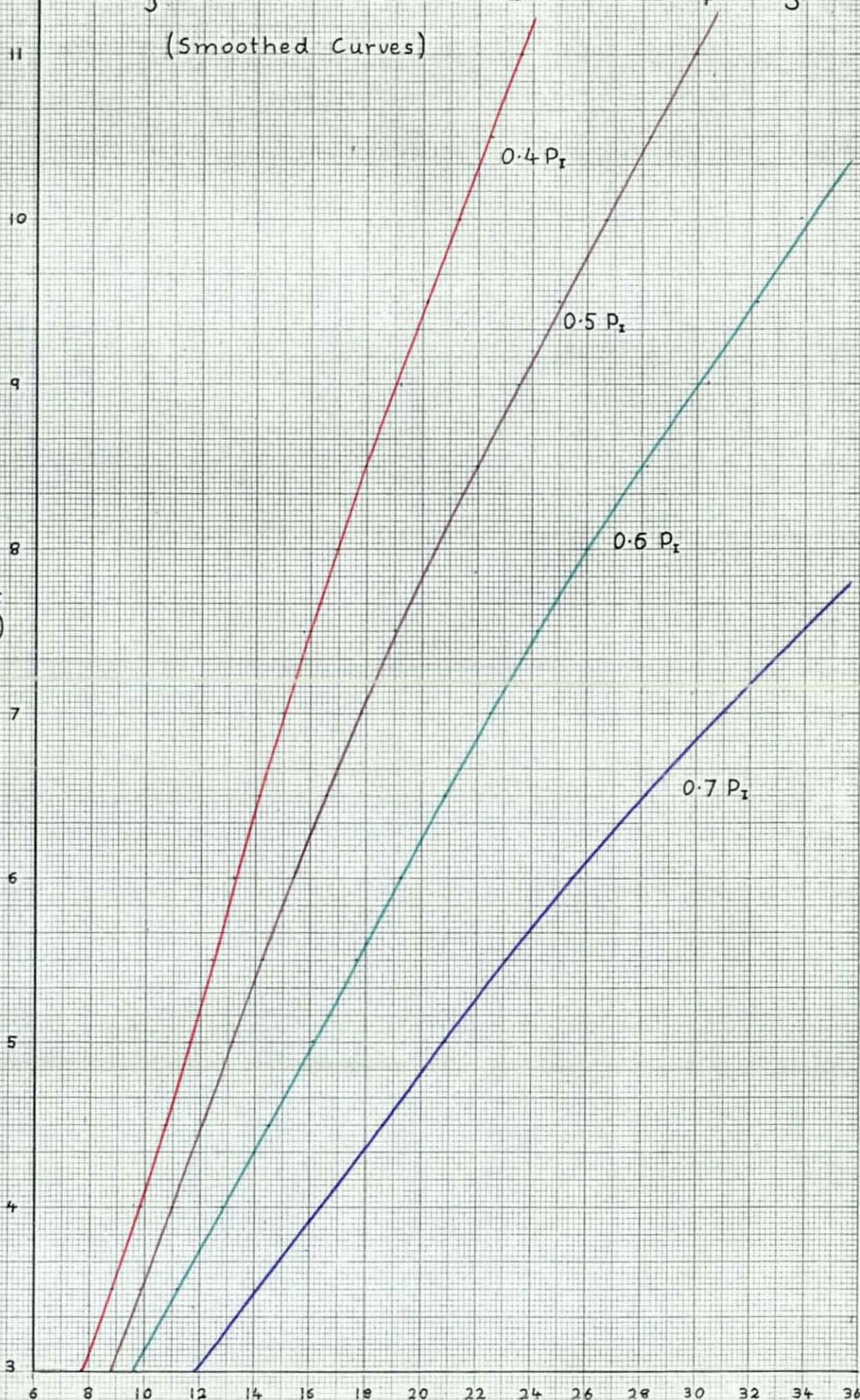
Fig.17

Combined Chart for Large Pump



Design Chart for Standard Sized Field Pump Fig. 18
(Smoothed Curves)

Lift
(m)



Example Design Chart - Table of Values Used

$0.3 \text{ m} \leq D < 0.5 \text{ m}$ $v_p = 0.3 \text{ m/s}$ I.D. of both pipes = 0.025 m

LH Chart

Lift (m)	D (m)	L/D	0.4 P_I		0.5 P_I		0.6 P_I		0.7 P_I	
			Ratio	H/D	Ratio	H/D	Ratio	H/D	Ratio	H/D
1.2	0.3	4	1.280		1.201		1.131		1.066	
1.6	0.4	4	1.289	3.112	1.208	3.316	1.139	3.251	1.070	3.742
2.0	0.5	4	1.287		1.210		1.138		1.071	
1.5	0.3	5	1.402		1.326		1.256		1.190	
2.0	0.4	5	1.412	3.538	1.335	3.746	1.264	3.953	1.197	4.176
2.5	0.5	5	1.426		1.343		1.275		1.205	
1.8	0.3	6	1.382		1.319		1.264		1.212	
2.4	0.4	6	1.372	4.371	1.310	4.578	1.256	4.777	1.205	4.978
3.0	0.5	6	1.364		1.303		1.248		1.199	
2.4	0.3	8	1.311		1.226		1.186		1.151	
3.2	0.4	8	1.325	6.037	1.240	6.462	1.182	6.772	1.147	6.978
4.0	0.5	8	1.340		1.248		1.176		1.142	
3.0	0.3	10	1.433		1.350		1.282		1.205	
4.0	0.4	10	1.448	6.956	1.366	7.368	1.292	7.761	1.217	8.220
5.0	0.5	10	1.432		1.356		1.288		1.228	
4.5	0.3	15	1.441		1.364		1.293		1.221	
6.0	0.4	15	1.421	10.552	1.346	11.194	1.283	11.707	1.223	12.309
7.5	0.5	15	1.403		1.311		1.268		1.212	
6.0	0.3	20	1.422		1.348		1.284		1.226	
8.0	0.4	20	1.399	14.229	1.328	15.051	1.267	15.778	1.211	16.507
10.0	0.5	20	1.396		1.311		1.252		1.198	
7.5	0.3	25	1.405		1.334		1.272		1.202	
10.0	0.4	25	1.414	17.648	1.313	18.878	1.254	19.917	1.189	18.485
12.5	0.5	25	1.431		1.326		1.240		1.179	

Values for H N Chart

$0.3 \text{ m} \leq D \leq 0.5 \text{ m}$ $v_p = 0.3 \text{ m/s}$ I.D. of both pipes = 0.025 m

No. of Coils	Head Developed/D			
	$0.4 P_I$	$0.5 P_I$	$0.6 P_I$	$0.7 P_I$
6	2.029	1.894	1.877	1.681
8	3.802	3.501	3.252	2.752
10	5.555	5.061	4.554	3.752
12	7.219	6.524	5.771	4.692
14	8.792	7.899	6.914	5.577
16	10.281	9.194	7.991	6.414
18	11.690	10.418	9.009	7.209
20	13.026	11.576	9.974	7.964
22	14.294	12.676	10.890	8.685
24	15.501	23.721	11.761	9.374
26	16.651	14.758	12.595	10.033
28	17.750	15.710	13.392	10.665
30	18.802	16.622	14.156	11.273
32	19.808	17.495	14.890	11.858
34	20.774	18.334	15.595	12.423
36	21.703	19.141	16.274	12.967

Design Chart Values for Larger Pump

$1.0 \leq D \leq 1.5$ m; $v_p = 0.25$ m/s; coil bore = 0.038 m; pipe bore = 0.025 m;
delivery pipe bore 0.025 m

Lift (m)	D (m)	L/D	0.4 P_I		0.5 P_I		0.6 P_I		0.7 P_I	
			Ratio	\bar{H}/D	Ratio	\bar{H}/D	Ratio	\bar{H}/D	Ratio	\bar{H}/D
4	1.0	4	1.000		1.000		1.000		1.000	
5	1.25	4	1.000	4	1.000	4	1.000	4	1.000	4
6	1.5	4	1.000		1.000		1.000		1.000	
5	1.0	5	1.084		1.000		1.000		1.0000	
6.25	1.25	5	1.092	4.582	1.000	4.998	1.000	5	1.000	5
7.5	1.5	5	1.098		1.001		1.000		1.000	
6	1.0	6	1.229		1.126		1.035		1.000	
7.5	1.25	6	1.240	4.839	1.136	5.283	1.045	5.745	1.000	6
9	1.5	6	1.251		1.145		1.053		1.000	
7	1.0	7	1.356		1.249		1.156		1.069	
8.75	1.25	7	1.374	5.100	1.265	5.539	1.169	5.993	1.080	5.489
10.5	1.5	7	1.388		1.277		1.179		1.087	
8	1.0	8	1.476		1.367		1.269		1.177	
10	1.25	8	1.494	5.391	1.383	5.820	1.282	6.250	1.187	6.742
12	1.5	8	1.482		1.374		1.289		1.196	
10	1.0	10	1.429		1.334		1.264		1.204	
12.5	1.25	10	1.396	7.153	1.308	7.634	1.242	8.041	1.185	8.429
15	1.5	10	1.369		1.288		1.225		1.170	
12	1.0	12	1.339		1.271		1.212		1.165	
15	1.25	12	1.334	8.929	1.252	9.577	1.196	10.028	1.151	10.417
18	1.5	12	1.359		1.236		1.182		1.140	
14	1.0	14	1.452		1.322		1.211		1.139	
17.5	1.25	14	1.473	9.504	1.339	10.466	1.225	11.441	1.127	12.375
21	1.5	14	1.494		1.352		1.235		1.128	

H N Values for Larger Pump

$1.0 \leq D \leq 1.5 \text{ m}$; $v_p = 0.25 \text{ m/s}$; coil bore = 0.038 m;

delivery pipe bore = 0.025 m.

No. of Coils				
	0.4 P_I	0.5 P_I	0.6 P_I	0.7 P_I
6	1.714	1.806	1.435	1.201
8	3.378	3.232	2.683	2.186
10	4.802	4.447	3.742	3.037
12	6.035	5.329	4.661	3.783
14	7.120	6.261	5.473	4.446
16	8.087	7.096	6.202	5.045
18	8.961	7.854	6.864	5.592
20	9.758	8.547	7.470	6.095
22	10.491	9.186	8.030	6.561
24	11.170	9.780	8.551	6.996
26	11.804	10.335	9.038	7.404
28	12.397	10.856	9.495	7.788
30	12.864	11.347	9.927	8.151
32	13.392	11.812	10.335	8.495
34	13.892	12.253	10.723	8.822
36	14.368	12.673	11.093	9.133

Values for Standard Pump Design Chart

$D = 0.544 \text{ m}$ Speed = 10 r.p.m.

I.D. of both pipes = 0.025 m

Lift (m)	L/D	0.4 P_I		0.5 P_I		0.6 P_I		0.7 P_I	
		Ratio	Head	Ratio	Head	Ratio	Head	Ratio	Head
3	5.51	1.404	2.140	1.333	2.251	1.273	2.357	1.217	2.465
3.5	6.43	1.327	2.638	1.271	2.754	1.225	2.857	1.180	2.966
4	7.35	1.281	3.123	1.232	3.247	1.191	3.359	1.154	3.466
4.5	8.27	1.345	3.346	1.258	3.577	1.178	3.820	1.135	3.965
5	9.19	1.408	3.551	1.319	3.791	1.242	4.016	1.169	4.277
5.5	10.11	1.417	3.881	1.342	4.098	1.278	4.304	1.220	4.508
6	11.03	1.370	4.380	1.305	4.598	1.250	4.800	1.198	5.008
6.5	11.95	1.350	4.815	1.276	5.094	1.227	5.297	1.180	5.508
7	12.87	1.387	5.047	1.299	5.389	1.213	5.771	1.166	6.003
7.5	13.79	1.427	5.256	1.335	5.618	1.254	5.981	1.177	6.372
8	14.71	1.404	5.698	1.330	6.015	1.268	6.309	1.210	6.612
8.5	15.62	1.372	6.195	1.306	6.508	1.249	6.805	1.196	7.107
9	16.54	1.387	6.489	1.288	6.988	1.232	7.305	1.183	7.608
9.5	17.46	1.417	6.704	1.320	7.197	1.235	7.692	1.172	8.106
10	18.38	1.414	7.072	1.336	7.485	1.265	7.905	1.187	8.425
10.5	19.30	1.387	7.570	1.315	7.985	1.255	8.367	1.200	8.750
11	20.22	1.387	7.931	1.298	8.475	1.241	8.864	1.189	9.251

Values for Standard Pump Design Chart

$D = 0.544 \text{ m}$ Speed = 10 r.p.m.

I.D. of both pipes = 0.025 m

No. of Coils	Head Developed (m)			
	0.4 P_I	0.5 P_I	0.6 P_I	0.7 P_I
6	1.259	1.177	1.047	0.875
8	2.245	2.080	1.798	1.468
10	3.165	2.873	2.486	2.012
12	4.020	3.623	3.119	2.515
14	4.814	4.316	3.704	2.983
16	5.554	4.961	4.249	3.420
18	6.244	5.562	4.759	3.830
20	6.891	6.126	5.237	4.216
22	7.500	6.656	5.688	4.581
24	8.073	7.157	6.114	4.927
26	8.616	7.631	6.519	5.257
28	9.131	8.081	6.904	5.571
30	9.621	8.509	7.271	5.872
32	10.088	8.919	7.621	6.159
34	10.535	9.310	7.957	6.435
36	10.963	9.685	8.280	6.701

CONCLUSIONS

The conclusions to this investigation are stated below:-

- i) The essential element in determining the potential lift of a coil pump is the calculation of the water levels in the coils. Sufficiently good agreement was achieved between theoretical and experimental levels for a range of differing pump configurations.
- ii) The assumption that dynamic effects (i.e. viscous drag etc.,) within the coils of the pump had a negligible effect on the pump's performance was found to be justified.
- iii) Below a proportional depth of immersion of 0.4, the movement of water within the pump was unstable and the pump's internal mechanisms did not correspond with those assumed in the derivation of the proposed theory.
- iv) The analysis was shown to give equally good results over a range of drum diameters and pipe bores.
- v) It was shown that the derived theory could be used for design purposes to predict the number of coils required on a particular pump to develop any given head.
- vi) The program used to calculate the ratio of lift:head (the delivery pipe ratio) gave an agreement with experimental values which is sufficient for most design applications. However, further work needs to be done to verify these calculations over a range of delivery pipe bores.
- vii) The chevron-bladed waterwheel used in conjunction with the coil pump proved to be very well suited for the application.
- viii) Although only general relationships governing the response of

the waterwheel to varying flow regimes were obtained, there is scope for a great deal of research to be done on this wheel.

- ix) The low overall efficiency of the stream powered pump is largely due to the high proportion of power used to overcome friction in the bearings.
- x) Successful field tests showed that construction of a working pump using low technology methods and materials is a feasible proposition.
- xi) The design charts produced give a useful guideline to the design of a pump, thereby conveying the computer calculated results into a practical situation. However, for a fully accurate pump design, a complete analysis as outlined in the thesis would be necessary.
- xii) These design charts tend to err on the side of safety, i.e. by predicting too many rather than too few coils for a given situation.

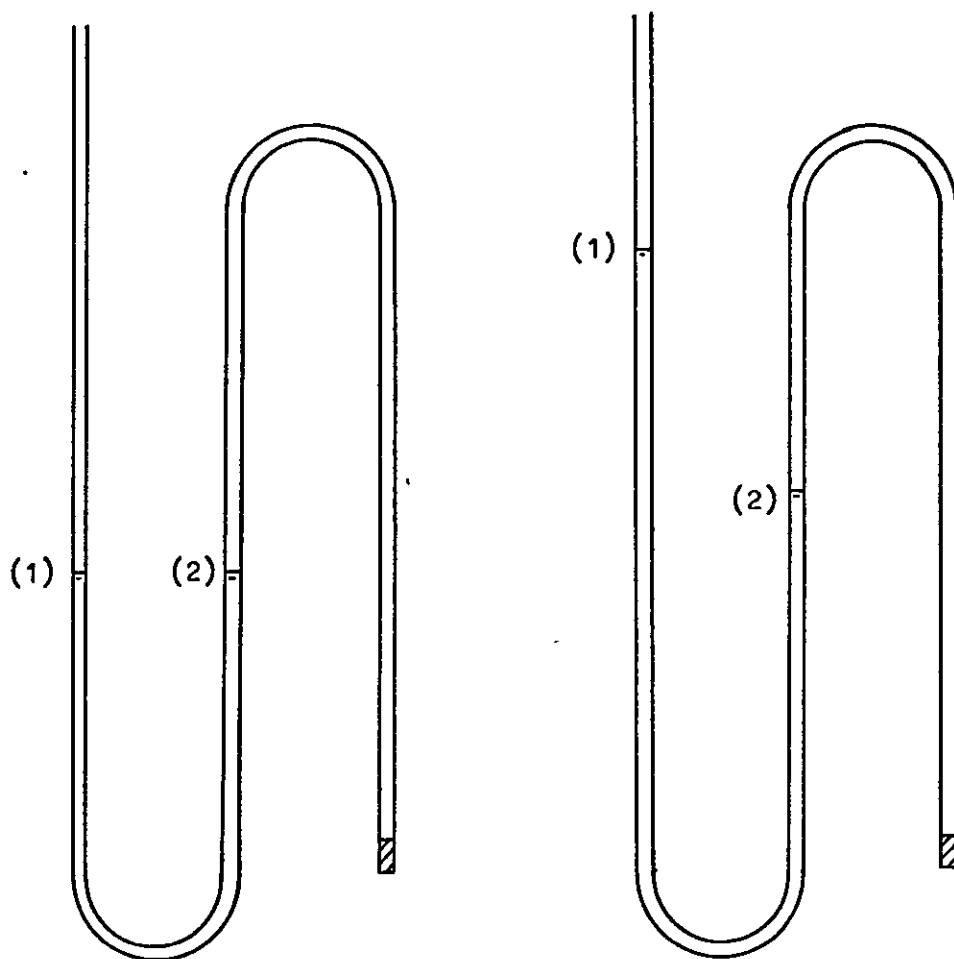
APPENDIX ITEST ON THE COMPRESSIBILITY OF AN AIR PLUG

This test was carried out to verify that the relationship $P_1 V_1 = P_0 V_0$ is applicable to the analysis of the coil pump, and that the expansion of the pipe is negligible, thereby showing that the volume of a plug of air is proportional to the length of pipe occupied.

Method

A length of clear flexible tubing was arranged as shown in the diagrams below. An amount of water was poured into the pipe and the bung inserted into the end of the pipe. Levels (1) and (2) were recorded. The position of level (2) was marked on the pipe in order to evaluate the original length of air.

More water was then poured into the top of the pipe causing compression of the air plug and a difference in the water levels (1) and (2) as shown in the right hand diagram. This process was repeated, measuring levels (1) and (2) thus giving several values of pressure and volume of air.



Theory

For any gas $PV^\gamma = \text{const. (K)}$

$$\therefore \log P + \gamma \log V = K' \quad K' = \log K$$

$$\therefore \log P = -\gamma \log V + K'$$

For a straight line plot $y = mx + c$

So by plotting $\log P$ on the ordinate and $\log V$ on the abscissa we obtain a straight line of gradient $(-\gamma)$

Results

Atmospheric pressure $\equiv 10.329$ m head of water

Levels (m)		Head (m H ₂ O)	P $\times 10^5 \text{ N/m}^2$	V $\times 10^{-3} \text{ m}^3$	log P	log V
(1)	(2)					
0.920	0.904	10.345	1.015	1.577	5.006	-2.802
1.430	1.034	10.725	1.052	1.513	5.022	-2.820
1.926	1.149	11.106	1.089	1.457	5.037	-2.837
2.361	1.245	11.445	1.123	1.410	5.050	-2.851
3.316	1.448	12.197	1.197	1.310	5.078	-2.883
3.668	1.517	12.480	1.224	1.276	5.088	-2.894
4.138	1.599	12.868	1.262	1.236	5.101	-2.908
4.663	1.668	13.324	1.307	1.202	5.116	-2.920

$$\text{From Graph A.1, } -\gamma = \frac{5.006 - 5.108}{-2.8025 + 2.915} = -0.907$$

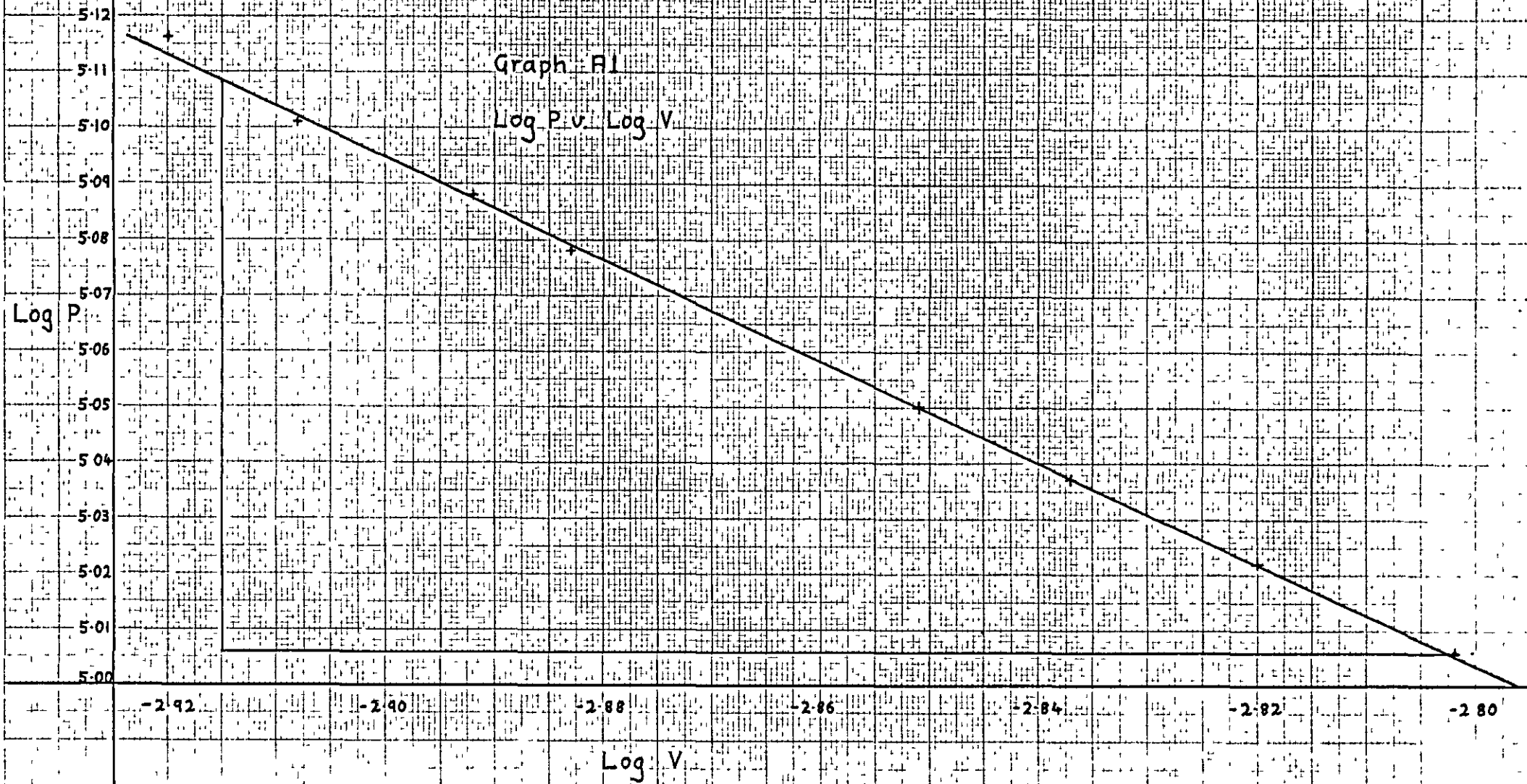
$$\therefore \gamma = 0.907$$

Conclusion

Since the value of γ can only lie between 1.0 and 1.4, the lower value of 1.0 must be taken. However, this gives an error of 10% in the experiment which can only be attributed to incorrect

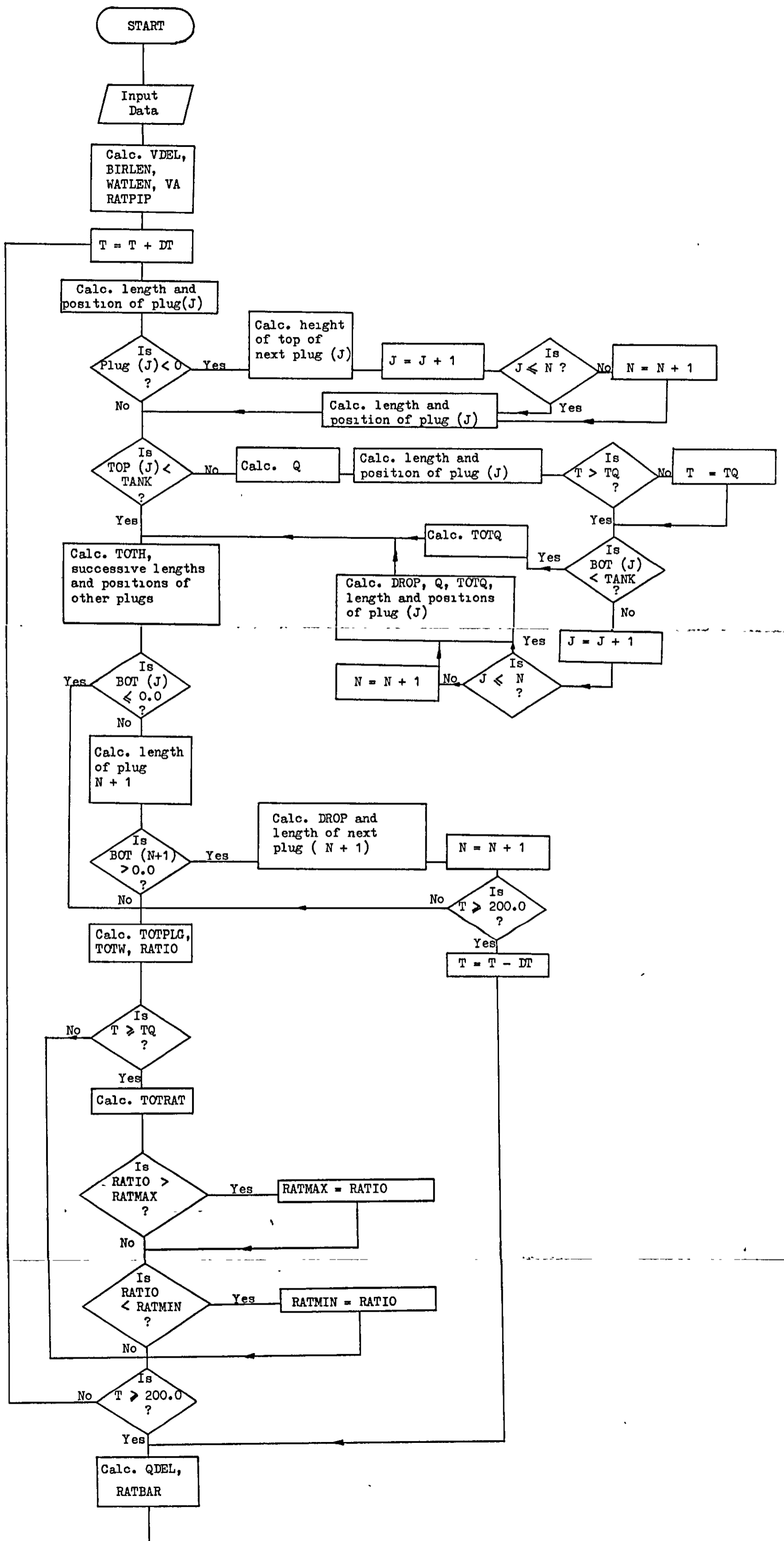
measurement of the atmospheric pressure.

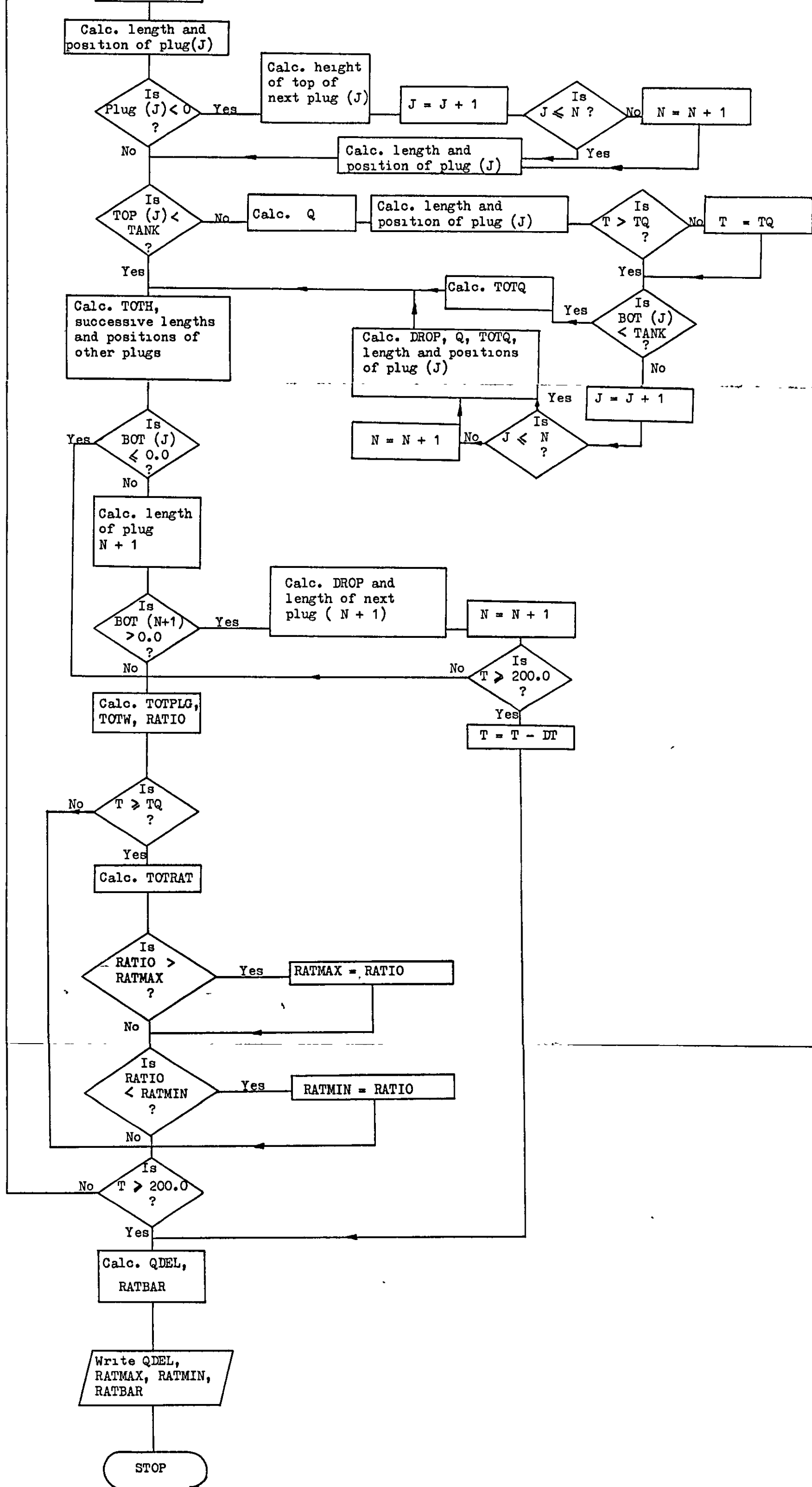
The value of 1.0 for γ corresponds to an isothermal change, i.e. one in which the work done on the gas can be recovered and no energy is lost in heat. This is the case in the coil pump, as the air is compressed very slowly and then recovers its original volume as it travels up the delivery pipe.



APPENDIX II

FLOW CHART, LISTING AND EXAMPLE RUN
OF DELIVERY PIPE PROGRAM





Dictionary of Variable Names Used In
Delivery Pipe Program

AIR	Length of uncompressed air left at the top of the delivery pipe at the end of a time interval during which the bottom end of a water plug has left the delivery pipe.
AIRLEN	Compressed length of an air plug.
BIRLEN	Original length of air plug in the delivery pipe.
BOT	Height of the trailing end of a water plug above the base of the delivery pipe.
BOTAIR	Height of the trailing end of an air plug.
COMB	Combined length of plug (N+1) and compressed air plug while still in the pump.
DEFF	Relative velocity deficiency in the last coil of the pump.
DIAD	I.D. of the delivery pipe.
DIAP	I.D. of the pipe used on the pump.
DROP	Length of water lost from the bottom of a water plug during one time interval.
DT	Time interval.
HEAD	Head of water above an air plug.
HT	Last height of water plug before its length diminishes to 0.
HTN	Height of the top of the next water plug at the moment this happens.
HTNI	Height of the top of this same water plug at the end of the time interval.
J	Counter for the uppermost plug of water.
N	Counter for the lowest complete plug of water.
PA	Atmospheric pressure measured in metres head of water.
PLEN	Length of water plug (N+1) while still in the pump.
PLUG	Length of a water plug.
Q	Quantity of water passing out of the top of the delivery pipe in one time interval.
QDEL	Quantity delivered in litres/min.
RATBAR	Mean value of delivery pipe ratio R_D .

RATMAX	Maximum value of delivery pipe ratio R_D .
RATMIN	Minimum value of delivery pipe ratio R_D .
RATPIP	Ratio of the cross-sectional areas of the two pipes.
RES1-6	Reserve stores.
SPEED	Velocity of plug (N+1) while in the last coil of the pump.
T	Time measured in seconds.
TANK	Height of the delivery tank above the base of the delivery pipe.
TIM	Time taken during one interval for the bottom of the uppermost plug of water to reach the top of the delivery pipe.
TIME	Time taken during one interval for the trailing end of plug (N+1) to enter the delivery pipe.
TIMET	Time taken during one interval for the length of the uppermost water plug to diminish to 0.
TOP	Height of the leading end of a water plug.
TOPAIR	Height of the leading end of an air plug.
TOTPLG	Total length of complete water plugs in the delivery pipe.
TOTQ	Cumulative quantity of water passing out of the top of the delivery pipe.
TOTRAT	Summation of delivery pipe ratios from time TQ to the end of the calculation.
TOTW	Total head in the delivery pipe.
TQ	Time at which water first passes the top of the delivery pipe.
VA	Relative velocity of an air plug through a water plug.
VDEL	Velocity of water plugs up the delivery pipe.
VP	Peripheral velocity of the pump.
WATLEN	Original length of a water plug in the delivery pipe.

Listing of delivery pipe program

```

      DIMENSION TOP(110),BOT(110),PLUG(110),AIRLEN(110),TOTALR(110),
      CBOTAIR(110)
      WRITE(1,1)
1  FORMAT('INPUT HEIGHT OF DELIVERY TANK',/, 'ROTATIONAL SPEED OF PUMP'
      CP',/, 'MEAN COIL DIAMETER
      C',/, 'PROPORTIONAL DEPTH OF IMMERSION',/, 'I.D. OF PIPE ON PUMP'
      C',/, 'I.D. OF DELIVERY PIPE')
      READ(1,*)TANK,REV,D,PI,DIAP,DIAD
      VP=REV*3.14159*D/60.0
      WATLEN=(D/2.0)*(3.14159+2.0*ASIN(2.0*PI-1.0))
      BIRLEN=3.14159*D-WATLEN
      DT=1.0
      VA=0.25
      PA=10.329
      T=0.0
      J=1
      N=1
      T0TQ=0.0
      TQ=9999.0
      RATMAX=0.0
      TOTRAT=0.0
      RATMIN=9999.0
      RATPIP=(DIAP**2.0)/(DIAD**2.0)
      VDEL=VP*RATPIP
      BIRLEN=BIRLEN*RATPIP
      WATLEN=WATLEN*RATPIP
      VA=VA*(DIAD)/(0.025)
      VA=(VDEL/0.326)**0.2+VA
      TOP(J)=WATLEN
      BOT(J)=0.0
      DO 5 I=1,110
      PLUG(I)=WATLEN
5  CONTINUE
9  T=T+DT
      Q=0.0
      TOP(J)=TOP(J)+(VDEL*DT)
      IF(BOT(J).LT.0.0) GO TO 19
      BOT(J)=BOT(J)+(VDEL+VA)*DT
      GO TO 21
19 BOT(J)=BOT(J)+(VDEL*DT)
21 PLUG(J)=TOP(J)-BOT(J)
      IF(PLUG(J).LE.0.0)GO TO 6
      GO TO 7
6  TIMET=RES6/VA
      HT=RES4+(TIMET*VDEL)
      HTN=HT-BIRLEN
      HTNI=HTN+(VDEL*(DT-TIMET))
      J=J+1
      IF(J.LE.N)GO TO 3
      N=N+1
3  PLUG(J)=PLUG(J)+RES6-(VA*DT)
      TOP(J)=HTNI
      BOT(J)=TOP(J)-PLUG(J)
7  RES4=TOP(J)
      RES5=BOT(J)
      RES6=PLUG(J)
      IF(TOP(J).LE.TANK)GOTO 50
      Q=(TOP(J)-TANK)*(DIAD**2.0)+0.7854
      TOP(J)=TANK
      PLUG(J)=TOP(J)-BOT(J)
      RES6=PLUG(J)
      IF(T.GT.T0)GOTO 8

```

```

      8 IF (TOT(J).LE.TANK)GOTO 47
      GOTO 17
49 TOTQ=TOTQ+Q
      GOTO 50
17 J=J+1
      IF (J.LE.N)GO TO 111
      N=N+1
111 TIM=(TANK-RES3)/(VDEL+VA)
      DROP=TIM+VA
      Q=(TANK-RES3-DROP)*(DIAD+2.0)+1.7854
      TOTQ=TOTQ+Q
      AIR=BIRLEN-((DT-TIM)*VDEL)
      TOP(J)=TANK-AIR
      PLUG(J)=PLUG(J)-(DT+VA)+DROP
      BOT(J)=TOP(J)-PLUG(J)
50 RES3=BOT(J)
      HEAD=0.0
      DO 10 I=J,N
      IF (BOT(J).LT.0.0) GO TO 33
      HEAD=HEAD+PLUG(I)
      GO TO 34
33 HEAD=HEAD+TOP(J)
34 AIRLEN(I)=BIRLEN*(PA/(PA+HEAD))
      TOPAIR(I)=BOT(I)
      BOTAIR(I)=TOPAIR(I)-AIRLEN(I)
      TOP(I+1)=BOTAIR(I)
      BOT(I+1)=TOP(I+1)-PLUG(I+1)
      IF (BOT(J).LE.0.0) GO TO 54
10 CONTINUE
      HEAD=0.0
      DO 11 I=J,N
      HEAD=HEAD+PLUG(I)
11 CONTINUE
      IF (TOP(N+1).GT.0.0) GOTO 64
      GOTO 65
64 HEAD=HEAD+TOP(N+1)
65 PLEN = PLUG(N+1)/RATPIF
      COMB=PLEN+BIRLEN*(PA/(PA+HEAD))
      SPEED=VP*COMB/(PLEN+BIRLEN)
      DEFF=VP-SPEED
      PLUG(N+1)=PLUG(N+1)+(VA+DT)-(DEFF+DT)
      TOP(N+1)=BOTAIR(N)
      BOT(N+1)=TOP(N+1)-PLUG(N+1)
      IF (BOT(N+1).GT.0.0)GOTO 20
      RES1=TOP(N+1)
      RES2=PLUG(N+1)
54 TOTPLG =0.0
      DO 45 I=J,N
      IF (BOT(J).LT.0.0) GO TO 55
      TOTPLG=TOTPLG+PLUG(I)
      GOTO 45
55 TOTPLG=TOTPLG+TOP(I)
45 CONTINUE
      IF (TOP(N+1).LT.0.0) GOTO 61
      TOTW=TOTPLG+TOP(N+1)
      GOTO 62
61 TOTW=TOTPLG
62 RATIO=TANK/TOTW
      WRITE(1,101)T,RATIO
101 FORMAT('T =',I3,' SECS   RATIO = ',F6.3)
      IF (T.GE.TQ) GOTO 46
      GOTO 47
46 TOTRAT=TOTRAT+RATIO
      IF (RATIO.GT.RATMAX)GOTO 81
      GOTO 82

```

```

81 RATMAX=RATIO
82 IF(RATIO.L1.RATMIN)GOTO 83
   GOTO 84
83 RATMIN=RATIO
84 CONTINUE
47 IF(T.GE.200.0) GOTO 70
   GOTO 9
20 TIME=(RES2-RES1)/VDEL
   DROP=(DT-TIME)*VA
   PLUG(N+1)=RES2+(VA*DT)-DROP
   BOT(N+1)=TOP(N+1)-PLUG (N+1)
   N=N+1
   PLUG (N+1)=PLUG(N+1)+DROP
   IF(T.GE.200.0)GOTO 705
60 CONTINUE
   GOTO 54
705 T=T-DT
70 QDEL=(TOTQ/(T-TQ))*60000
   RATBAR=TOTRAT*DT/(T-TQ+DT)
   WRITE(1,100)QDEL,RATMAX,RATMIN,RATBAR
100 FORMAT(//,'QUANTITY DELIVERED = 'F6.3 ' L/MIN'
C,/, 'MAX. RATIO = ',F7.3,/, 'MIN. RATIO = ',F7.3,/,
C' MEAN RATIO = ',F7.3)
   CALL EXIT
   END
FUNCTION ASIN(X)
Z=X/SQRT(1-X**2)
ASIN=ATAN(Z)
RETURN
END

```

OK,

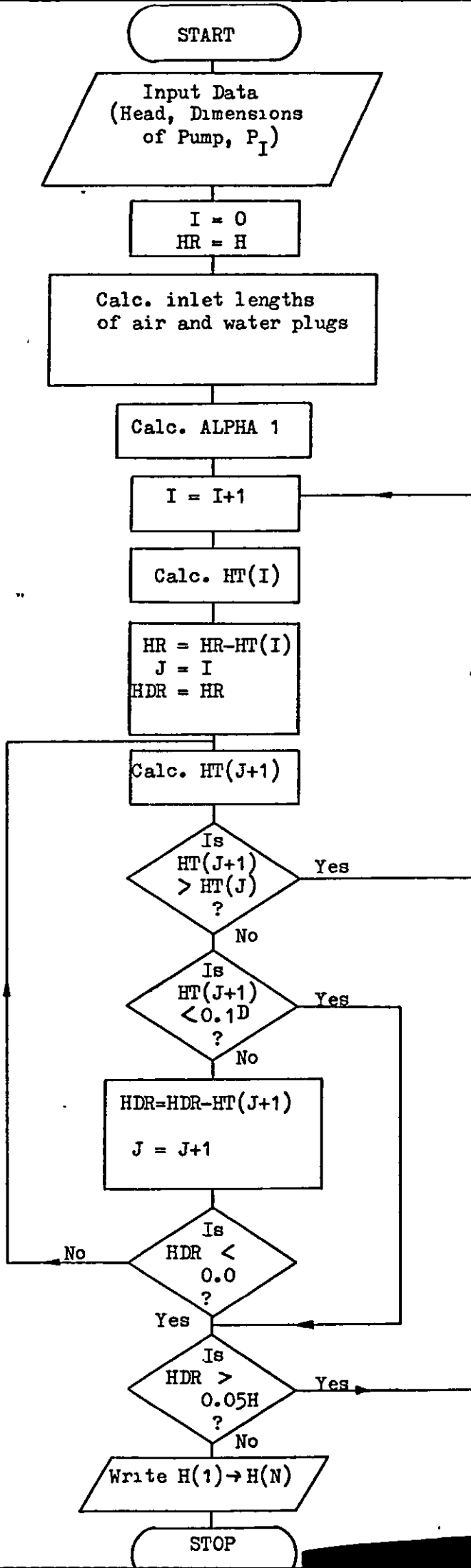
Example run of delivery pipe
program (T = 0 → 54 secs)

INPUT HEIGHT OF DELIVERY TANK
 ROTATIONAL SPEED OF PUMP
 MEAN COIL DIAMETER
 PROPORTIONAL DEPTH OF IMMERSION
 I.D. OF PIPE ON PUMP
 I.D. OF DELIVERY PIPE
 7.5, 16, 0.6, 0.4, 0.035, 0.03

T = 1 SECS	RATIO = 9.791
T = 2 SECS	RATIO = 6.848
T = 3 SECS	RATIO = 4.338
T = 4 SECS	RATIO = 4.175
T = 5 SECS	RATIO = 4.116
T = 6 SECS	RATIO = 3.040
T = 7 SECS	RATIO = 2.411
T = 8 SECS	RATIO = 2.719
T = 9 SECS	RATIO = 2.335
T = 10 SECS	RATIO = 1.944
T = 11 SECS	RATIO = 1.989
T = 12 SECS	RATIO = 1.844
T = 13 SECS	RATIO = 1.602
T = 14 SECS	RATIO = 1.673
T = 15 SECS	RATIO = 1.563
T = 16 SECS	RATIO = 1.454
T = 17 SECS	RATIO = 1.679
T = 18 SECS	RATIO = 1.803
T = 19 SECS	RATIO = 1.569
T = 20 SECS	RATIO = 1.505
T = 21 SECS	RATIO = 1.844
T = 22 SECS	RATIO = 1.655
T = 23 SECS	RATIO = 1.470
T = 24 SECS	RATIO = 1.725
T = 25 SECS	RATIO = 1.454
T = 26 SECS	RATIO = 1.528
T = 27 SECS	RATIO = 1.918
T = 28 SECS	RATIO = 1.683
T = 29 SECS	RATIO = 1.478
T = 30 SECS	RATIO = 1.736
T = 31 SECS	RATIO = 1.468
T = 32 SECS	RATIO = 1.515
T = 33 SECS	RATIO = 1.885
T = 34 SECS	RATIO = 1.671
T = 35 SECS	RATIO = 1.468
T = 36 SECS	RATIO = 1.725
T = 37 SECS	RATIO = 1.459
T = 38 SECS	RATIO = 1.518
T = 39 SECS	RATIO = 1.895
T = 40 SECS	RATIO = 1.675
T = 41 SECS	RATIO = 1.472
T = 42 SECS	RATIO = 1.726
T = 43 SECS	RATIO = 1.787
T = 44 SECS	RATIO = 1.558
T = 45 SECS	RATIO = 1.544
T = 46 SECS	RATIO = 1.819
T = 47 SECS	RATIO = 1.587
T = 48 SECS	RATIO = 1.477
T = 49 SECS	RATIO = 1.773
T = 50 SECS	RATIO = 1.681
T = 51 SECS	RATIO = 1.475
T = 52 SECS	RATIO = 1.600
T = 53 SECS	RATIO = 1.811
T = 54 SECS	RATIO = 1.575

APPENDIX III

FLOW CHART, LISTING AND EXAMPLE RUN
OF HEAD DIFFERENCES PROGRAM



Dictionary of Variable Names Used In
Program for Determination of Head Differences

ALPHA	Angle formed at the centre of the coil between the lower water level and the vertical.
ALPHA1	Original value of ALPHA before spillback is taken into account.
AIR	Length of an uncompressed air plug at inlet.
ATU	Atmospheric head of water.
BETA	Angle subtended at the centre by the length of water which spills back in one revolution.
BORE	Bore of the coils.
COMP	Length of a compressed plug of air.
D	Diameter of the coils.
DEFF	Difference between distance travelled by the coil and that travelled by a unit of water and compressed air in one revolution.
DEPTH	Vertical distance from centre of coil to the lower water level.
H	Head which the pump is required to develop.
HDR	Head remaining at end of spillback line.
HR	Head acting in a coil.
HT	Head difference across a coil.
I	Counter.
J	Counter.
K	Counter.
L	Counter.
N	Number of coils.
PI	Proportional depth of immersion.
PVEL	Peripheral velocity of the coils.
RADM	Mean radius of the coils.
REV	Speed of pump in r.p.m.
SPEED	Velocity of a unit of water and compressed air.
TO	Vertical distance from centre of the coil to the uppermost point in the inner wall of the coil.
UNIT	Length of a unit of water and compressed air.
WAT	Length of water plug at inlet.

Listing of head differences program

```

      DIMENSION COMP(60),UNIT(60),SPEED(60),DEFF(60),IETA(60)
      &,ALPHA(60),DEPTH(60),HT(60),THECD(60),ALPHAT(60),I(60)
      &,XAXIS(60),HTP(60)
      LOGICAL MARK4
      WRITE(1,10)
10  FORMAT('INPUT',/, 'HEAD TO BE DEVELOPED',/, 'MEAN DIAMETER OF COILS'
      C,/, 'PROPORTIONAL DEPTH OF IMMERSION',/, 'ROTATIONAL ',
      C'SPEED OF PUMP',/, 'BORE OF COILS')
      READ(1,*)H,D,PI,REV,BORE
      I=0
      HR=H
      PVEL=REV*D*3.14159/60.0
      ATM=10.329
      RADH=D/2.0
      A=0.05*H
      B=0.15*RADH
      WAT=(D/2.0)*(3.14159+2.0*ASIN(2.0*PI-1.0))
      AIR=3.14159*D-WAT
      TO=RADH-(BORE/2.0)
      ALPHA1=6.2832-(WAT/RADH)-ACOS(TO/RADH)
15  I=I+1
      COMP(I)=AIR*ATM/(ATM+HR)
      UNIT(I)=WAT+COMP(I)
      SPEED(I)=(UNIT(I)/(AIR+WAT))+PVEL
      DEFF(I)=6.2832*RADH-(SPEED(I)*60.0/REV)
      BETA(I)=DEFF(I)/RADH
      ALPHA(I)=ALPHA1-BETA(I)
      DEPTH(I)=RADH*SIN(ALPHA(I)-1.5708)
      HT(I)=DEPTH(I)+TO
      HR=HR-HT(I)
      J=I
      HDR=HR
      ALPHA(J)=ALPHA1
17  COMP(J+1)=AIR*ATM/(ATM+HDR)
      BETA(J+1)=COMP(J+1)/RADH
      ALPHA(J+1)=ALPHA(J)+(WAT/RADH)+BETA(J+1)-6.2832
      HT(J+1)=RADH*(SIN(ALPHA(J+1)+(WAT/RADH)-4.7124)+SIN(ALPHA(J+1)
      C-1.5708))
      L=J+1
      K=J-1
      IF(HT(J+1).GT.HT(J)) GOTO 15
      IF(ABS(HT(J+1)).LT.B)GOTO 30
      HDR=HDR-HT(J+1)
      J=J+1
      IF(HDR.LT.0.0) GOTO 30
      GOTO 17
30  IF(HDR.GT.A)GOTO 15
      DO 20 I=1,L
      WRITE (1,16)I,HT(I)
16  FORMAT('H',I2,' =',F8.4,'M')
20  CONTINUE
      WRITE (1,40) L
40  FORMAT('NO. OF COILS =',I2)
      DO 50 I=1,L
      HTP(I)=HT(I)+100.0
      N=L-I+1
50  XAXIS(I)=FLOAT(N)
      CALL FILOUT(XAXIS,HTP,L,.TRUE.,L,1,MARK4)
      CALL EXIT
      END

```

Example Run of Head Differences Program

This program calculates the head differences in the coils of the pump, beginning with the outlet coil and working back along the spillback line towards the inlet.

For every coil on the spillback line the model assumes that this coil is the last on the spillback line (i.e. the coil containing the maximum head difference) and calculates the levels in the non-spilling coils using the residual head remaining at the end of the spillback time. If all the necessary conditions are satisfied by the resulting head difference pattern (see section 3.4) then the program will terminate and the head differences will be outputted. If, however, the conditions are not satisfied, then the next point on the spillback line will be calculated and the same procedure repeated.

Calculation of the non-spilling coils is very sensitive to the initial head used, i.e. that head remaining at the inlet end of the spillback line and it is likely that the first head fed into the program will not produce the characteristic pattern of head differences shown in Graphs 4 - 40. This is demonstrated by the following program runs and graphs; the first value of head used (7.000 m) produces the values shown in Graph A.2. Increasing the head to 7.100 m gives the pattern shown in Graph A.3.

It is between these two plots that the true solution lies and the head is increased from 7.000 m until the next integral solution is found (Graph A.4, head = 7.045) and it is this integral number of coils that should be used on the pump. If the non-integral solution is required, then the plot for the non-spilling coils (which is similar for all heads developed) may be attached to the spillback line in such a way that the sum of the head differences is equal to the required pumping head.

Example run of head differences program (Head = 7.000 m)

```

INPUT
HEAD TO BE DEVELOPED
MEAN DIAMETER OF COILS
PROPORTIONAL DEPTH OF IMMERSION
ROTATIONAL SPEED OF PUMP
BORE OF COILS
7, 0.6, 0.55, 11.0, 0.03
H 1 = 0.2446M
H 2 = 0.2520M
H 3 = 0.2600M
H 4 = 0.2685M
H 5 = 0.2775M
H 6 = 0.2872M
H 7 = 0.2976M
H 8 = 0.3087M
H 9 = 0.3207M
H10 = 0.3336M
H11 = 0.3475M
H12 = 0.3626M
H13 = 0.3788M
H14 = 0.3964M
H15 = 0.4154M
H16 = 0.4359M
H17 = 0.4578M
H18 = 0.4332M
H19 = 0.3225M
H20 = 0.2376M
H21 = 0.1819M
H22 = 0.1530M
H23 = 0.1485M
NO. OF COILS =23
Input the name of the file
T1
OK,

```

Example run of head differences program (Head = 7.100 m)

```

INPUT
HEAD TO BE DEVELOPED
MEAN DIAMETER OF COILS
PROPORTIONAL DEPTH OF IMMERSION
ROTATIONAL SPEED OF PUMP
BORE OF COILS
7.1, 0.6, 0.55, 11.0, 0.03
H 1 = 0.2416M
H 2 = 0.2489M
H 3 = 0.2566M
H 4 = 0.2648M
H 5 = 0.2737M
H 6 = 0.2831M
H 7 = 0.2931M
H 8 = 0.3039M
H 9 = 0.3156M
H10 = 0.3281M
H11 = 0.3416M
H12 = 0.3561M
H13 = 0.3719M
H14 = 0.3889M
H15 = 0.4073M
H16 = 0.4272M
H17 = 0.4485M
H18 = 0.4711M
H19 = 0.4582M
H20 = 0.3885M
H21 = 0.3584M
NO. OF COILS =21
Input the name of the file
T2
OK,

```

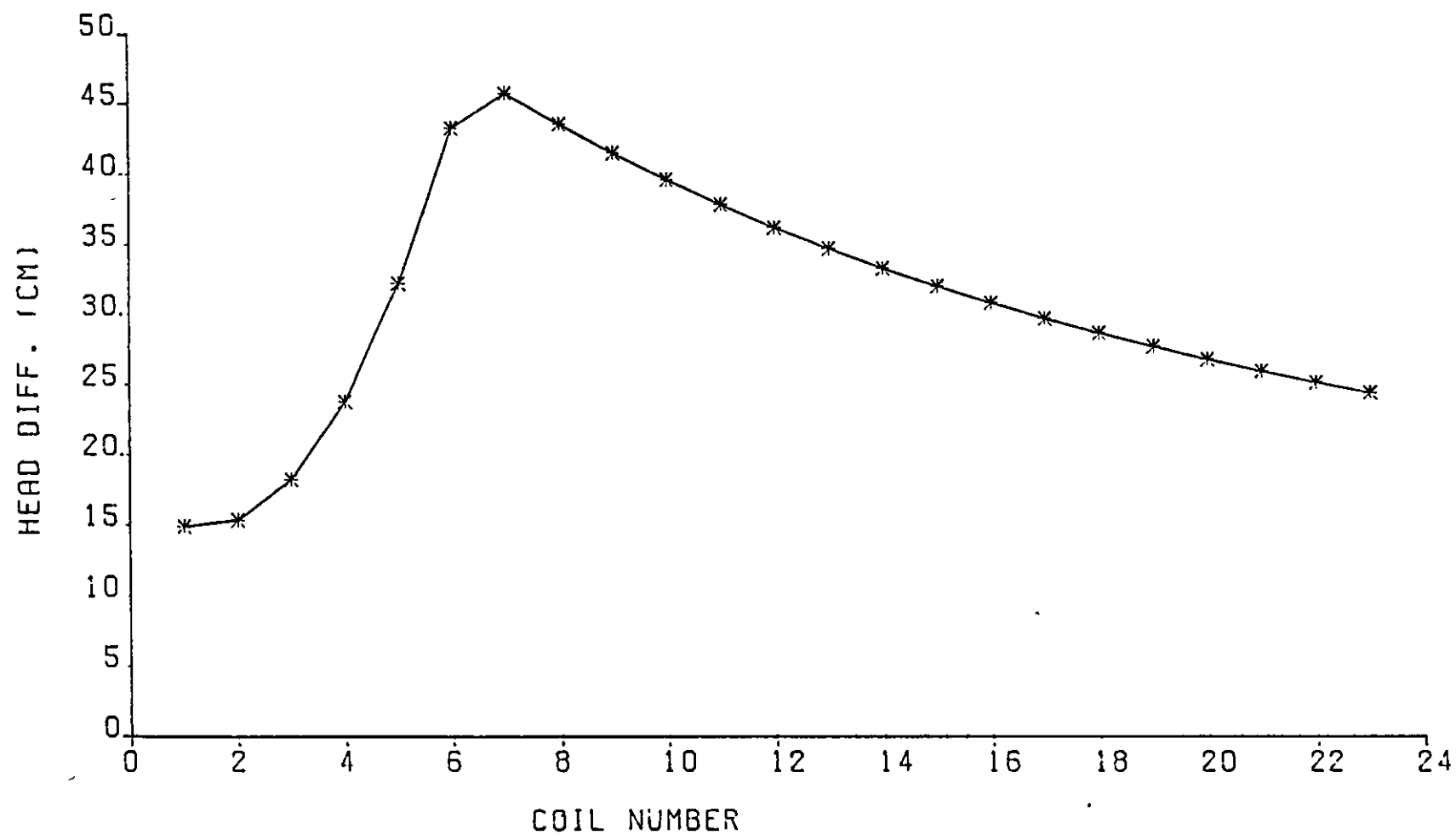
Example run of head differences program (Head = 7.043 m)

```

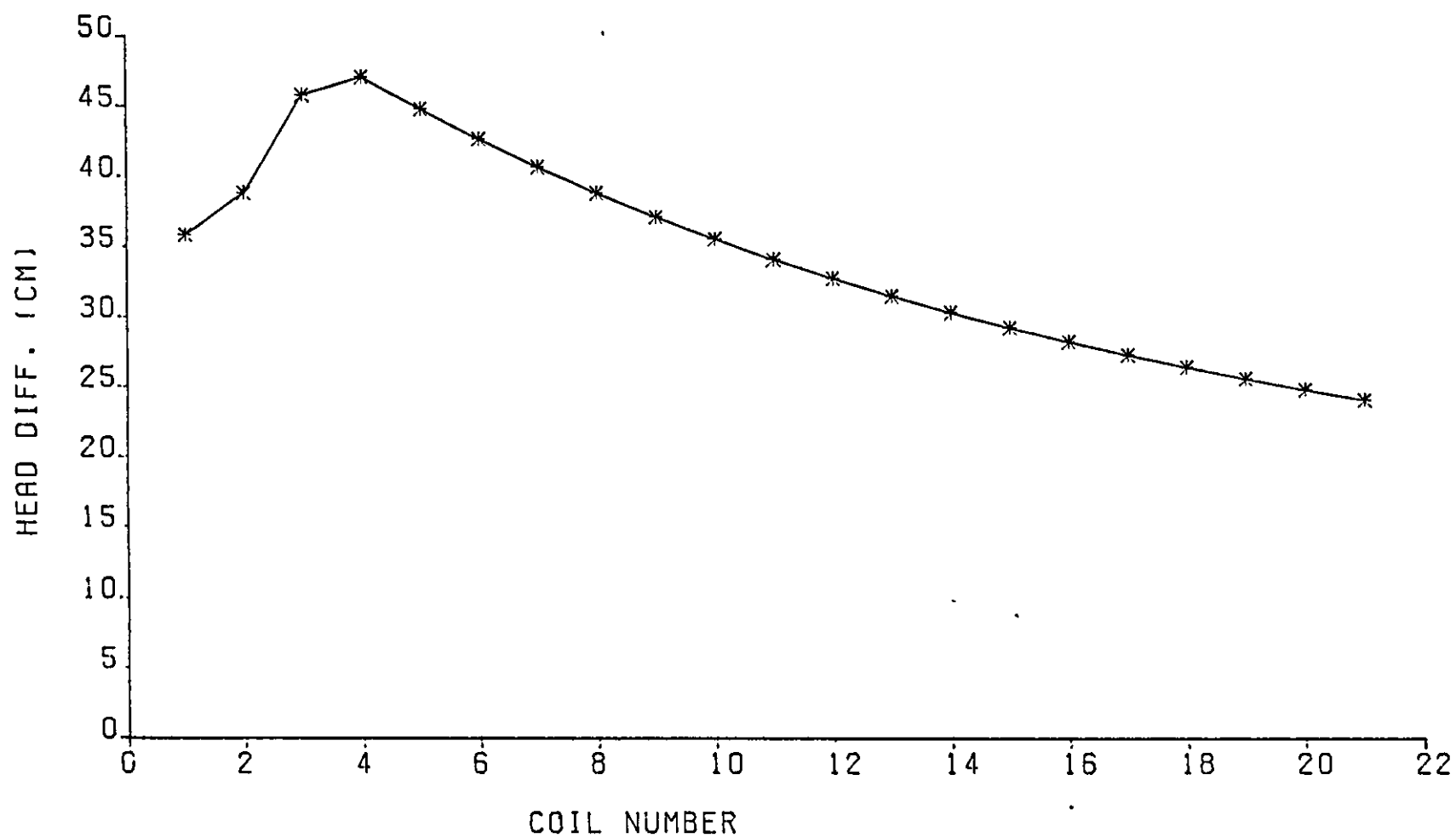
INPUT
HEAD TO BE DEVELOPED
MEAN DIAMETER OF COILS
PROPORTIONAL DEPTH OF IMMERSION
ROTATIONAL SPEED OF PUMP
BORE OF COILS
7.043, 0.6, 0.55, 11.0, 0.03
H 1 = 0.2433M
H 2 = 0.2507M
H 3 = 0.2505M
H 4 = 0.2669M
H 5 = 0.2758M
H 6 = 0.2854M
H 7 = 0.2956M
H 8 = 0.3066M
H 9 = 0.3185M
H10 = 0.3312M
H11 = 0.3449M
H12 = 0.3598M
H13 = 0.3758M
H14 = 0.3932M
H15 = 0.4119M
H16 = 0.4321M
H17 = 0.4537M
H18 = 0.4255M
H19 = 0.3021M
H20 = 0.1997M
H21 = 0.1199M
H22 = 0.0564M
H23 = 0.0012M
NO. OF COILS =23
Input the name of the file
T3
OK,

```

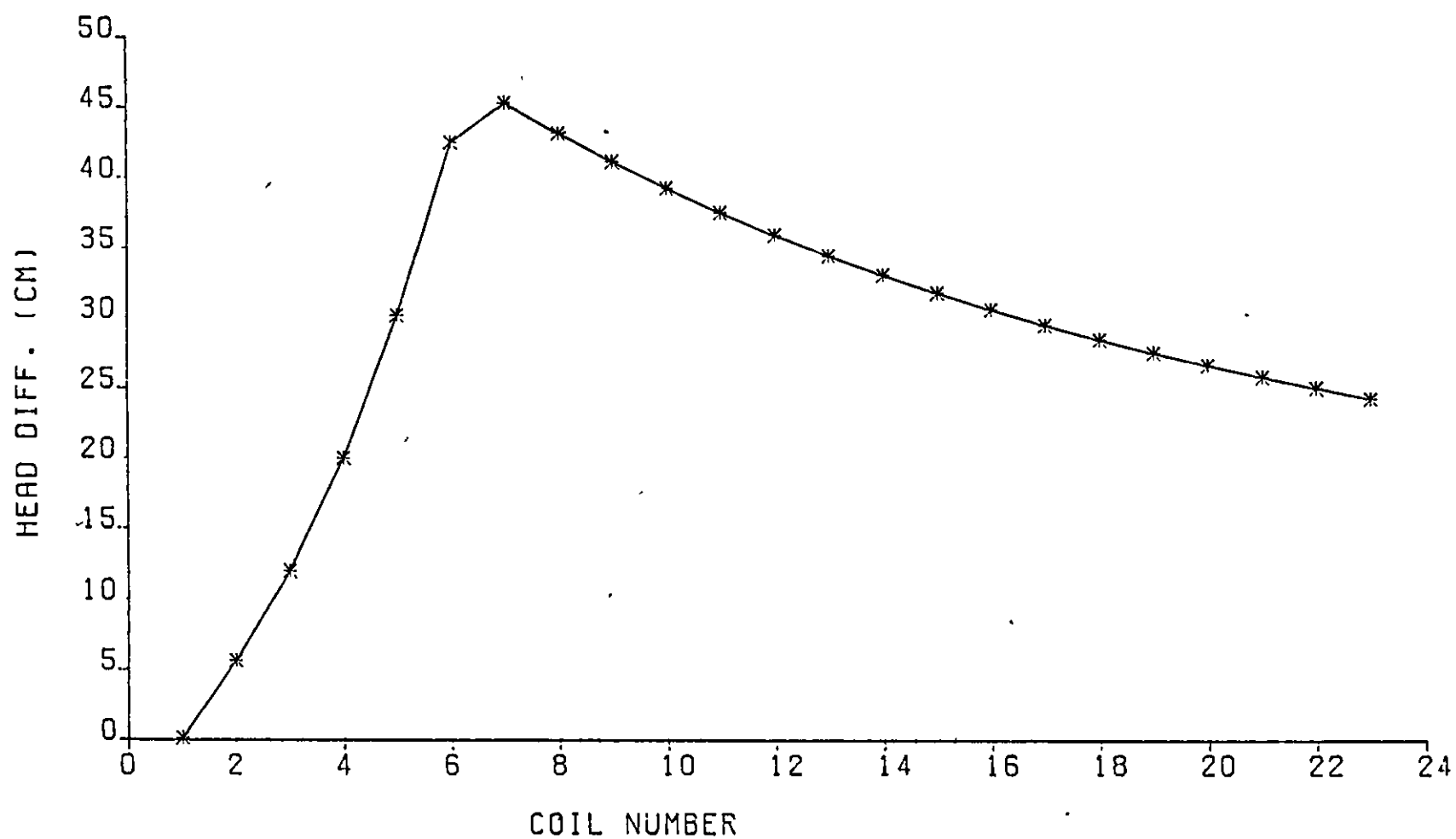
HEAD = 7.000M. GRAPH A.2



HEAD = 7.100M. GRAPH A.3



HEAD = 7.045M. GRAPH A.4



APPENDIX IV

LIST OF ARTICLES WRITTEN BY THE AUTHOR

List of Articles Written by the Author

Articles on the stream-powered rotating coil pump are as follows:-

1. Stream-powered rotating coil pump. Water Services, Dec. 1980, Vol. 84, No. 1018, pp. 706 and 722.
2. Water pump produces a high turnover. New Scientist, 5th February, 1981, Vol. 89, No. 1239, p. 346.
3. Harnessing river flow for pumping. World Water, February 1981, pp. 41-43.
4. Developing pumps for developing countries. Consulting Engineer, April 1981, Vol. 45, No. 4, pp. 50-51.
5. Rotating coil pump. Chartered Mechanical Engineer, May 1981, pp. 43-44.
6. Flow of stream drives oil drum irrigation pump. International Agricultural Development, July/August 1981, pp. 16-17.

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2. WINSTANLEY, C.B.F.: Rotating Coil Pump. Final Year Project, Loughborough University of Technology, April 1977.
3. BAMFORTH, A.: A Low Cost Pump using a Rotating Helical Coil. Final Year Project, Loughborough University of Technology, April 1977.
4. ROBINSON, P.A.: A Rotating Coil Pump. Final Year Project, Loughborough University of Technology, April 1978.
5. ANNABLE, R.J.: Rotating Coil Pump. Final Year Project, Loughborough University of Technology, April 1979.
6. LISTER, G.M.: Waterwheels for Small-Scale Power. Final Year Project, Loughborough University of Technology, March 1980.
7. WILSON, C.: Rotating Coil Pump. Final Year Project, Loughborough University of Technology, April 1981.
8. DALEY, J.R.: Investigation of Waterwheels. Final Year Project, Loughborough University of Technology, April 1981.
9. WEIR, A.: The Stream Driven Hydrostatic Coil Pump for Low Lift Irrigation: Determination of its Economic and Technical Feasibility and Prediction of its Performance Characteristics. University of Dar Es. Salaam, May 1979.
10. STUCKEY, A.T. and WILSON, E.M.: The Stream Powered Manometric Pump. Paper for 'Appropriate Technology in Civil Engineering' Conference, London 1980.

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