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Experimental design for characterization of force transmissibility through roller bearings in electric machines and transmissions

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Experimental Design for Characterization of Force Transmissibility through Bearings in Electric Machines and Transmissions

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Abstract

With the increasing stringent emissions legislation on ICEs, alongside requirements for enhanced fuel efficiency as key driving factors for many OEMs, there are many research activities supported by the automotive industry that focus on the development of hybrid and pure EVs. This change in direction from engine downsizing to the use of electric motors presents many new challenges concerning NVH performance, durability and component life. This paper presents the development of experimental methodology into the measurement of NVH characteristics in these new powertrains, thus characterizing the structure borne noise transmissibility through the shaft and the bearing to the housing. A feasibility study and design of a new system level test rig have been conducted to allow for sinusoidal radial loading of the shaft, which is synchronized with the shaft's rotary frequency under high-speed transient conditions in order to evaluate the phenomena in the system. The present work introduces a new component level test rig that can predict the response of new EV and hybrid systems using different types of rolling element bearings such as deep groove ball bearings, angular contact roller bearings, tapered roller bearings and the cylindrical roller bearings. Moreover, it is possible to investigate the influence of factors such as bearing clearance and the amount of axial bearing preload which will be used to further explore the capability of bearing CAE tools. The test rig has multiple novel elements compared to those previously developed. In particular, the rotational speed of the shaft, which significantly exceeds that of previously reported rigs, and the excitation frequency ramp up at the same rate as the frequency of the shaft enabling the phenomena found in Hybrid and EVs. The sinusoidal radial load is supplied using a loading device featuring a single load point to minimize undesired excitation effects. With respect to structure borne noise the system response is captured through the vibrational displacement of the shaft and bearing housing.

Introduction

With the absence of ICE low frequency noise masking, certain NVH issues have become more prominent such as wind (0.1-0.3+ kHz) and road noise (0.5-1.3 kHz), particularly in EVs [1]. In the mid frequency band, gear whine (0.2-2.3 kHz) [2] is prevalent during accelerations and decelerations; the frequency increases proportionally with the motor speed. With these new powertrain systems e-machine and inverter noise (~8 kHz) [3] are also introduced, where the combination of these noises is considered irritant to the human ear. Continued development of hybrid vehicles has resulted in increased complexity of the powertrain systems,

introducing new and additional transmission paths for subsequent NVH issues [4].

Gear whine is produced as a result of gear pair meshing under loaded conditions [5]. The whining can either develop as an air or structure borne noise. The most significant element is the structure borne noise that propagates through the shaft into the supporting bearing.

A gear pair test rig has previously been modelled, predicting gear whine from transmission error by Neusser et al. [6]. The test rig was simplified from a six speed manual transmission to a gear mesh model in order to isolate the constituent parts that were responsible for producing whine. The rig and bearing housing was mounted on a heavy base. Furthermore, the FRF spectrum simulation results revealed that the ideal scenario displayed the meshing frequencies as most prominent; whereas in the transmission error case, it had a much more broadband response up to 3 kHz.

The vibrational characteristics of shaft-bearing structures have been investigated both experimentally and mathematically [7,8]. Rajab [7] originally investigated how the radial loading through a static shaft affected the transverse motion of the bearing housing supported roller bearing. He produced a test rig, which utilized an electro-mechanical shaker to apply a load. The load was to imitate the power transfer through a gear pair. This incorporated the steady state force, as well as the time variant component that induces the transverse vibration in the plate structure, representing the transmission housing. The time variant force is produced via a multitude of effects including the elastic properties of gear meshing and transmission error. It was shown that the length of the shaft affected the displacement of the plate; and thus, by increasing the length of the shaft it induced a bending moment at the bearing with the result being an increase in angular deflection of the shaft. Lin replicated Rajab's study, varying the bearing offset. It was found that the bearing offset had a limited effect on the force transmissibility through the bearing [9]. Lin also showed how the force transmissibility varied with the direction of the excitation force, finding it was due the various mode shapes of the plate. Lim and Singh produced a formulation for the modeling of bearing stiffness within a similar shaft-bearing-plate system [8]. Validation of the model was carried out experimentally, proving its capabilities to predict the flexural motion of the exemplary transmission housing [10].

This paper presents a new design for a test rig with the capability to characterize the force transmissibility using a motor-shaft-bearing system, mounted within a transmission. Using open literature, the test rig has been designed and developed such that minimal noise is produced from a close to ideal excitation source, to replicate gear

meshing and from the motor and support structures. Since it has previously been shown transmission error would reduce the signal to noise ratio of measurement necessary to generate a portrayal of the response of the bearing and housing [6], a novel method is utilized to excite the shaft without inducing additional frequencies.

Test Rig Design

A motor-shaft-bearing system has been designed and manufactured to investigate the phenomenon of structure borne noise through bearings and their respective housing. To develop a test rig that maximizes the signal/noise ratio from the bearing housing, individual components that could be a potential source of noise generation were reviewed. The test rig is a modification of an existing rig used by Karthikeyan [11], comprising of a CNC spindle mounted to two isolation beds as shown in Figure 1.

A vertical orientation was first investigated to remove the effect of gravity. This proved to be negligible since further complications in achieving a refined system and isolating components effectively proved to be more dominant.

Figure 1: CAD Design of the Initial Test Rig

1. Spindle
2. Upper Isolation Bed
3. Lower Isolation Bed

Baseline Testing – Vibration transmission through Isolation Beds

Tests were carried out to benchmark acceleration results around the spindle and isolation beds to ascertain the levels of damping that were being achieved within the system. It is essential that background vibrations and resonant frequencies are kept to a minimum within the operating speed range. The spindle and control unit were run using a MATLAB control system to acquire data.

Baseline Test 1 – A tri-axial accelerometer and a single beam laser vibrometer were used to measure acceleration on the isolation bed. Results from both measurement methods correlate well as the spindle

speed ramps up with time (Figure 2), confirming that the acceleration is part of the dynamic response of the system.

Figure 2: Rotational Speed Ramp on the Spindle

Baseline Test 2 - Three accelerometers were used with appropriate amplification on the spindle, upper and lower bed. These results can be seen in Figures 3-5 respectively. The purpose of this test was to assess the vibration transmission from the spindle through both isolation beds. This was used to assess the transmission to the test brackets and whether this noise was permissible.

Figure 3: Acceleration on Spindle

Figure 4: Acceleration on Upper Isolation Bed

Figure 5: Acceleration on Lower Isolation Bed

Acceleration is damped from 390 ms^{-2} (resonance) at the spindle to 27 ms^{-2} at the lower bed where the test bracket would be located. This configuration did not remove enough noise at the test location.

3. Shaft
4. Test Bracket Holder
5. Shaker

Mitigation Testing

Due to resonance at certain frequencies during the ramping of the spindle, the range of speeds that can be tested is reduced. Noise mitigation strategies were investigated to reduce peak acceleration for the operating speed range. Sorbothane [12] isolating sheets were implemented into the rig to dampen mechanical vibration between the upper and lower bed, and the lower bed and ground. These refinements produced promising results presented in Figure 6. The acceleration recorded on the lower bed was reduced across the operating range of speed and over resonant frequencies, with the peak reduced from 27 ms^{-2} to 13 ms^{-2} .

Figure 6: Lower Isolation Bed Acceleration with Sorbothane Sheet

Whilst Sorbothane sheets provided further damping, additional mitigation techniques were required to refine the dynamics of the bracket further, in order to observe the least possible effect of background vibration. Isolating components of the system from the peak accelerations at resonance is critical to achieve a higher range of speeds. This configuration, however, introduced difficulties in alignment as mechanical datums could no longer be used.

Based on the results generated in the horizontal isolated configuration, a design matrix was developed to critically assess the individual components that would make up the test rig to ensure requirements were being met. The final design can be seen in Figures 7 and 8.

Figure 7: CAD Design of the Test Rig

1. Hinge
2. Test Bracket

Figure 8: Photo of the Test Rig Prior to Testing

Test Brackets

Modular aluminum 7075 end brackets were designed to accommodate the five different bearing types summarized in Table 1:

Table 1: Summary of Bearing Types

Bearing Type	Designation	C _{dyn} /C _{stat} [kN]	Reference Speed [rpm]
Deep Groove Ball Bearing	6205	14.8 / 7.8	28000
Angular Contact Ball Bearing	7205 BECD	30.8 / 51	17000
Cylindrical Roller Bearing	NU205	28.6 / 27	14000
Needle Roller Bearing	NA49/32	30.8 / 51	10000
Tapered Roller Bearing	30205 J2/Q	34.1 / 32.5	13000

With the exception of the needle roller bearing, all bearings have been selected to have a 52 mm outside diameter and 25 mm bore; common to the test bracket and shaft. The application of a tight fitted or clamped adaptor ring was considered for the smaller outer diameter of the needle roller bearing; however this may influence structure borne noise transmissibility significantly. A separate bracket with a reduced outer diameter would provide the most consistent test method.

For the test bracket, runout and positional tolerances have been referenced to datums that will be common to dependent components. Referencing the mounting holes on polar coordinates around the central axis reduces radial misalignment which would result in additional stress on the bearings. Total radial and axial runout was specified at ISO Tolerance Grades IT4/2 and IT4 respectively for high speed and load applications [13].

Test Bracket Holders - Flame-cut cast iron blocks were manufactured to mount the brackets. The large, rigid structure provides damping to increase S/N, with an adjustable axial distance enabling varying shaft lengths. The test bracket is constrained to four

small contact areas on the holder, allowing freedom of vibration for the bracket at measurement locations.

Shaft and Coupling – A long shaft was chosen to increase the bending moment and angular deflection at the test bearing. This should therefore increase the S/N, however, the test rig has been designed to be modular such that the shaft length can be varied aptly if required. The shaft is made from SAE620; with a polished section at the point contact to prevent excessive wear. DLC Coating was investigated due to its low-friction, high-hardness properties, however manufacturing constraints prohibited this. To reduce the torsional vibration induced by the spindle a polymeric jaw coupling was selected, rated to 19,000 rpm allowing providing suitability for an ultra-high-speed test rig.

Shaker and Hinge – An electromagnetic shaker was selected to provide the force input to the mechanical point loading system as it can produce a ramping sinusoidal force. The stinger from the shaker is attached to one side of a moment arm, and the other is the single point load acting upon the shaft. The moment arm was developed to apply radial load to the shaft. Force is transmitted to the hinge through a ball joint with adjustable internal clearance. To ensure a hard contact, a steel rather than PTFE lining was chosen. The ball joint is attached to the shaker via a hex turnbuckle nut and threaded bar to allow for adjustability of the point load using the moment about the mounting pin. This enables a 2 kN load with permissible fluctuations about a mean of 5-10 %.

Table 2: Test Rig Criteria

Quantities to be varied (overview)	
Speed	For all configurations, a speed sweep shall be carried out (run-up duration 30 or 60 s)
Axial Preload	0/100/1000 N
Clearance	Tight (C1) / Normal (CN) / Loose (C4)
Radial Load (Mean)	0/100/2000 N
Radial Load Fluctuation	Only a single variant (~5 to 10 % of mean value)
Radial Load Frequency	In ideal case ramped up according to shaft rotary speed

Axial Preload

Methodologies for applying bearing preloads and axial adjustments were evaluated by Morris [14]. One method is to apply axial preload to the shaft using a high-speed shaft locking nut, and ground washer to eliminate runout. A strain gauge is used to measure the tensile strain of the shaft to attain the preload. A torque vs. preload curve is then to be developed and its repeatability analyzed to assess whether the nut can be removed, or the strain gauge unplugged, whilst still providing results within the specified tolerance.

Radial Load Excitation Methodology

In order to simulate conditions, present in hybrid EVs, excitation will be applied to the shaft in the form of a sinusoidal radial load (Figure 9). This excitation will ramp up proportionally to the shaft speed representative of an excitation order, simulating phenomena like gear engagement. The load will vary through 0 N / 100 N / 2000 N.

Auxiliary Bearing– A method developed for exciting a rotating shaft is to use an auxiliary bearing, with excitation being provided by a shaker-pushrod configuration. Plain spherical bearings were selected due to having the advantage of allowing angular misalignment, some axial loading and high radial loads. As there are no rolling elements, no frequency shall arise from their rotation. However, due to the necessity of a stiff connection between bearing and shaft for optimum transmission of forces, a dry sliding contact would need to be a necessity. The sliding velocity requirements of the rig, combined with the required load proved to be too large to make the steel/steel sliding contact bearings a viable option. It has been found that complications arise with attenuated vibration transmission through the auxiliary bearing [15].

Non-contact magnetic bearing - Using an active magnetic bearing would provide support for the shaft, as well as introducing a sinusoidal excitation force that could be ramped up with the angular acceleration of the shaft. Moreover, no noise due to additional contacting components would be introduced. However, when being investigated, the maximum excitation forces used in previous studies were up to 490 N [16]. For this application, where excitation forces of 2 kN are required, this method is not suitable.

Shaft Imbalance – Another methodology could be to add an eccentric mass to the shaft. However, this would cause difficulty when extracting the FRF data as the method would not enable for a swept range of shaft speeds as a shaft rotational frequency close to its natural frequency would be required for the necessary excitement [17].

Single Point Mechanical Load - To negate issues surrounding noise generation from the excitation device, for example ball pass frequencies in auxiliary bearings, a method was devised in which a mechanical single point load could be applied to the shaft. Therefore, the full specified 2 kN of sinusoidal load could be achieved.

Radial Load Measurement - A permanently preloaded force transducer has been selected to measure the radial load applied to the shaft. The transducer has a force range of 1000 N tensile and 5000 N compressive. It serves as a secondary purpose to ensure that the point contact remains with the shaft at all times through the full amplitude range of the shaker.

Support Bearing – To support the shaft adequately a second bearing is required. This will however reduce the S/N because of potential bearing defects, which produce ball pass, fundamental train and ball spin frequencies. A 6205 deep groove ball bearing was selected as the support bearing for the initial baseline testing to match the bearing within the test housing.

Figure 9: Experimental set up

Instrumentation

Accelerometers – Tri-axial over single axis accelerometers offer the benefit of only a single mounting point on the bearing housing is required. In addition to this, less mass and modifications would be added to the structure and to the surface. The methodology of mounting to the bearing housing must also be considered as this will affect the measurement. Stud mounted accelerometers must be mounted exactly normal to the surface to ensure minimal loss of stiffness [16]. To choose the location of the accelerometer a non-permanent fixing method was used to ensure that it did coincide with any nodes of the modes; otherwise the magnitude of the vibration would not be observed.

Laser Vibrometer – An alternative to accelerometers for measuring the acceleration of the bearing housing is to use single point laser vibrometers. This offers the advantage of nondestructive and non-interfering (stiffness/mass) measurement. A scanning laser vibrometer with reflective tape on the shaft is to be used to measure the angular velocity.

Alignment

Laser Alignment – Due to the high-speed nature of the rig, parallel (offset) and angular misalignment across the coupling and between the test brackets is critical. Permissible coupling and shaft angular misalignment are specified at 0.01 mm / 100 mm and offset specified at 0.01 mm for 18,000 rpm running speeds.

Perpendicularity between the faces of the bearing bores with respect to each other, as well as to the surface of the optical breadboards (flatness ± 0.01 mm over 1 m²) is a source of both types of misalignment. Tight tolerances during design, as well as mechanical alignment using shims during assembly corrected tolerance stack-up that arose through manufacturing.

Summary/Conclusions

A rig has been designed and created for characterization of force transmissibility through roller bearings in electric machines and transmissions. The ultra-high-speed rig has been developed with the capability of running at speeds up to 18,000 rpm, whilst maximizing

the S/N from the bearing housing. Introduction of Sorbothane sheets have attenuated the peak acceleration response of the system, thus providing additional damping to the rig.

The rig will be used to further evaluate NVH characteristics from current and future EV powertrain systems, whilst being used to build upon the already well established existing CAE tools available in the commercial software for future trends.

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Definitions/Abbreviations

CAE	Computer Aided Engineering
DLC Coating	Dimond-like Carbon Coating
EV	Electric Vehicle
FRF	Frequency Response Function
ICE	Internal Combustion Engine
NVH	Noise, Vibration and Harshness
OEM	Original Equipment Manufacturer
PTFE	Polytetrafluoroethylene
S/N	Signal-to-Noise Ratio