On the effects of production and maintenance variations on machinery performance

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Abstract *The variations introduced during the production and maintenance of rotating machinery components are correlated with the vibration and noise emanating from the final system during its operational lifetime. Vibration and noise are especially unacceptable elements in high-risk systems such as helicopters and aircraft engines, resulting in premature component degradation and a potentially unsafe flying environment. In such applications, individual components often are subject to 100 per cent inspection following production and during operation through rigorous maintenance, resulting in increased product development cycles and high production and operation costs. In this work, the aim is to provide engineers with a technique to evaluate vibration modes and levels for each component or subsystem prior to putting them into operation. This paper presents a preliminary investigation of the correlation of manufacturing and assembly variations with vibrations, using an experimental test rig. A factorial design is used to study the effects of various factors. Challenges in developing a process monitoring and inspection methodology to predict performance quality are identified, followed by a discussion of future work.*

Practical implications

In this work, the aim is to provide engineers with a technique to evaluate vibration modes and levels for each component or subsystem in aircraft or rotorcraft, prior to putting them into operation, hence avoiding premature and costly failures. Such techniques will enable testing and quality control of components in an accurate way, by studying the effects of manufacturing and assembly errors during production and maintenance using information technology tools. Specifically, a systematic technique to test ball bearing variations is proposed and explored using an experimental test rig. Challenges in developing a process monitoring and inspection methodology to predict performance quality are identified, which are intended to aid in the follow-up development of an effective technique to be used in practice.

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Variations in production and maintenance

The intended function of a component can be compromised if there are variations introduced during production and maintenance, which result in undesired side effects. Designers and manufacturing engineers use a combination of tools (six-sigma, inspection, statistical process control, Taguchi's robust design method, error budgeting, etc.) to assess and eliminate variation, with the goal of producing higher quality parts with less scrap or rework, hence reducing the time and cost of product development (Bothe, 1997; Bralla, 1999; Carter, 1997; Chen and Thornton, 1999; Frey and Otto, 1997; Tata and Thornton, 1999). However, there is still a continuing need to develop products with better performance in faster and less costly ways. The idea of predicting and controlling variation has resulted in several promising methods (Frey and Otto, 1997; Chen and Thornton, 1999; Kazmer and Barkan, 1996; Suri and Otto, 1999). Design tools that contain variation information have been shown to facilitate the design process by reducing the number of iterations involved, estimating critical parameters, and identifying possible problems in terms of quality and cost (Bralla, 1999).

In rotating machinery, functionality and performance can be hindered by excessive vibrations resulting from variations and defects in the individual components. Two of the significant factors that cause undesired vibrations are manufacturing and assembly errors. In this work, knowledge of the correlation between these variations and vibrations is seen as a crucial piece of information. To this end, the relationship between manufacturing and assembly variations, and vibration patterns in rotating machinery is explored (Tumer and Huff, 2000). A typical production cycle for rotating machinery components involves regular quality control steps to assure that the specifications are met satisfactorily. In particular, rotating components in high-risk applications such as rotorcraft transmissions are subject to an intense inspection process during manufacturing, before and after assembly, as well as during operation (maintenance). In such cases, 100 per cent inspection of the parts' manufacturing and assembly tolerances are typically required, increasing the development time and the cost of producing such parts. A prediction of potential deviations from the intended functional requirements will not only reduce safety risks by avoiding premature failure, but also shorten the product cycle by avoiding scrap, rework, and inspection, as well as decrease costs associated with unplanned maintenance, hence reducing the overall cost of producing and operating such products (Tumer *et al.*, 1998, 2000).

This work presents a preliminary look at the issues in developing a systematic process monitoring and inspection methodology for predicting potential performance problems in rotating machinery components. In this light, the paper first explores an empirical correlation between variations introduced during production and maintenance, and, variations detected in the vibrational signatures from rotating machinery. First, the details of the empirical study and experimental design are presented for the case of ball bearings on a desktop test rig. The analysis of the vibration data collected from the experimental setup is presented next, exploring the frequencies and the statistical changes in the total power (variance) of the vibration data. Conclusions are then presented to discuss the challenges in developing a methodology to assess and predict the quality and performance of such components when installed in their operating environment.

Empirical study of variations

Ball bearings are selected as the initial focus of this study because of their tendency to contain waviness and roughness errors on their inner and outer raceway surfaces, as well as their tendency to move within the bearing housing due to assembly errors (Harris, 1991). An experimental test rig is used to test the bearings. The machinery fault simulator (MFS), shown in Figure 1, is a desktop test rig which houses a rotating shaft with weights to introduce balance faults, bearings attached at several locations to introduce bearing defects, and a gearbox attached to the main shaft by means of a belt to introduce gear defects. This first set of experiments involves the motor main shaft, and the two ball bearing assemblies supporting the main shaft. Several types of bearings are provided for testing by the manufacturer, including a set of healthy and faulty ball bearings. In this study, three "good" bearings were used for three production runs, making a total of nine different bearing/assembly combinations in the experiment.

Ball bearing variations

A defect-free ball bearing has perfectly circular inner and outer races, constant thrust loading, and no radial loads (Meyer, 1980; Shigley and Mischke, 1989). Any deviation from these intended functions is a defect. The most typical failures include point defects on the outer race, inner race, the rolling elements, and the cage: empirical formulas exist to compute the frequency at which these defects would appear in a vibration spectrum, usually a function of the shaft speed, ball diameter, number of balls, pitch diameter, and the contact angle (Mitchell, 1993; Taylor, 1980; Wowk, 1991). The relative amplitudes of these vibrational frequencies indicate the change in the bearing condition.

As opposed to point defects, problems due to manufacturing and assembly errors are difficult to isolate as single frequencies. Assembly errors are

Figure 1. Mechanical fault simulator (MFS) manufactured by SpectraQuest

typically due to the bearing being mishandled after manufacturing, resulting in an improper mount or fit in the bearing housing for a system, or an improper mountor loading within the bearing cage. Such errors can result in looseness, unbalance, or misalignment, which are phenomena that typically result in high first, second, and third harmonics (Brandlein and Eschmann, 1999; Wowk, 1991). Manufacturing errors, defined here as surface irregularities on bearing surfaces (typically under a micrometer in magnitude, not including form errors), are present in the form of surface roughness and surface waviness (Braun and Datner, 1979; Harris, 1991; Ono and Okada, 1998; Su, 1993; Wensing and Nijen, 1996; Whitehouse, 1994). Surface roughness patterns are high-frequency (noise-like) components, resulting in high-frequency components in the vibration spectrum. Waviness patterns are low-frequency components with a periodic pattern, resulting in low-frequency components in the vibration spectrum. Vibration testing and waviness testing are typically performed on components as part of quality control, but typically, overall levels and thresholds are used for determining the quality of such components (Baldanzini and Beraldo, 1999; Chong and Yi, 1999; Harris, 1991).

Experimental design

Since the primary focus of the experiment was to examine the combined effects of manufacturing variations, assembly inaccuracies, and operating speed, a three-way factorial design was used which allows the assessment of interactions between the factors as well as their main effects using analysis-of-variance (ANOVA) techniques (Montgomery, 1991). In the present design, Factor A represents three fixed levels of shaftspeed (30Hz, 45Hz, and 80Hz). Factor B represents three assemblies of the bearings and is considered a random factor since the primary interest is in generalizing to the population of assembly operations. With regard to Factor B, the three re-assemblies are conceptualized assimulating typical part replacement procedures, where the slightest misalignment, misfit, or looseness of the bearings in their housing can cause undesired vibrations that can be hazardous to the overall system's health. A rigorous re-assembly protocol was imposed to emulate the strict procedures followed when replacing helicopter transmission gear units (Huff *et al.*, 2000). Factor C represents the different bearing stocks from which three individual bearings were selected. Bearings from different stocks are assumed to contain manufacturing variations on their surfaces. This factor is also treated as a random factor since the population from which the bearings were drawn is of interest, not the particular three bearings themselves. The three factors with three levels each result in 27 experimental test conditions. Each test condition is replicated 11 times to provide within cell variance for proper statistical analysis (Montgomery, 1991).

Computation of expected frequencies

The bearings used for this study are SKF bearings with $n = 8$ balls, with a pitch diameter of $D_m = 1.1228$ in (0.0285m), a ball diameter of $d = 0.2813$ in (0.0072m), an inner ring thickness of $t = 0.1083$ in (0.0210m), an inner race circumference of $IR = 2.6437$ in (0.0671m), and an outer race circumference of $OR = 5.0916$ in (0.1293m). The defect frequencies can be computed as a function of the shaft rotational speed, which rotates at a frequency of 30Hz, 45Hz, or 80Hz, depending on the run condition, and a function of the ball diameter, pitch diameter, and number of balls. A simple geometry of a ball bearing is shown in Figure 2 for reference.

A point defect on the outer race causes an impact each time a ball crosses it, resulting in N impacts for every revolution of the cage. Since the impacts are of short duration, the spectra will exhibit many harmonics. The computed cage frequency f_{cage}, and the corresponding outer race fault frequency f_{outer} are presented in Table I for each shaft speed condition. A flaw on a single ball will alternately strike the inner and outer races, resulting in periodic forces at twice the ball spin frequency. Each impact is brief, and hence results in many harmonics. In addition, the contact points rotate at the cage precession frequency, which shows up as sidebands at $\pm f_{cage}$. The computed ball spin frequency $f_{ballspin}$ and the corresponding ball fault frequency $f_{ballfault}$, are also presented in Table I. Finally, a flaw on the inner race will be impacted by each of the N balls in sequence, at an inner race contact frequency $f_{IRcontact}$, each will be brief and lead to harmonics. In addition, the fault location rotates with the shaft, resulting inside bands at $\pm f_{shaft}$. The resulting inner race fault frequency and the inner race contact frequency are presented in Table I.

The rotation of a ball about its own axis is f_R, and the rotation of the inner race is f_{IR}. The vibration produced by waviness on the surface of a ball is computed as f_R times the number of waves per ball circumference. The

Table I.
Computed defect frequencies (all frequencies in Hz; harmonics at $K = 1, 2, 3, \ldots$)

f_{shaft}	f_{cage}	f_{outer}	$f_{ballspin}$	$f_{ballfault}$	$f_{IRcontact}$	f_{inner}
30	11.24	$K \times 89.93$	56.11	$K \times 112.2 \pm 11.24$	18.76	$K \times 150.0 \pm 30$
45	16.86	$K \times 134.9$	84.17	$K \times 168.3 \pm 16.86$	28.14	$K \times 225.1 \pm 45$
80	29.97	$K \times 239.8$	149.63	$K \times 299.3 \pm 29.98$	50.02	$K \times 400.2 \pm 80$

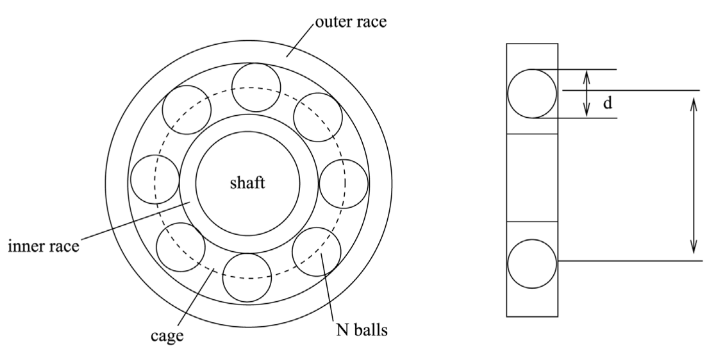


Figure 2.
Ball bearing geometry

vibration produced by inner race waviness is computed as f_{IR} times the number of waves per inner race circumference. Similarly, the vibration produced by outer race waviness is computed as the cage frequency f_{cage} times the number of waves per outer race circumference. The assumption is that any ball rolls over all the waves in the outer raceway in one cage revolution, resulting in a ball passage frequency over an individual wave cycle on the outer race. Unfortunately, there is no way of knowing the total number of waves on the outer race, inner race, or the ball surfaces, unless detailed measurements of the surface profiles are collected from the dismantled bearings. As a result, until such a measurement is made, there are no reported values for these expected frequencies. The measurement of the bearing surfaces will be performed at a future date, once the types of experiments to be conducted using these bearings have been exhausted.

Analysis of empirical data

In order to maintain consistency with vibration data collected on NASA test rigs and research aircraft, two single axis Endevco accelerometers were mounted near the bearing housings. During each experimental run, accelerometer and temperature sensors were sampled at 10kHz for 2.5sec, using a data acquisition system containing many of the same components used aboard the NASA test-rigs and research aircraft. The anti-aliasing filter was set at 5kHz. The data collection process was controlled by a LabVIEW software package (ALBERT) specifically developed by NASA for test-rig operations (Huff *et al.*, 2000, 2002).

Power spectral analysis of vibration data

To investigate the frequency characteristics of the vibration data and isolate specific defect frequencies, the standard power spectrum is used for this preliminary study. Despite many problems faced by power spectral analysis, especially in the presence of time-varying trends, to be compatible with industry standards, this paper uses the standard technique to get an initial global picture (Tumer *et al.*, 2000). More advanced methods and metrics will be explored in a future study.

Figure 3 provides an overall picture of the frequency content for components 1, 2, and 3, assemblies 1, 2, and 3, for the low speed runs (30Hz). The top three spectra correspond to the first bearing component, assembled into the same housing three times. The next three plots correspond to the second bearing component, and the last three correspond to the third bearing component. Similar plots are studied for each speed, and each replicate.

An analysis of the power spectra indicates that the overall magnitudes of the frequency components are higher for component 2 than for component 1. This effect is repeated over all the replicates, and speeds. In addition, the spectra for component 2 indicate the presence of higher frequency components, whereas the spectra for component 1 are less significant in the higher frequencies. This pattern points to a possible defect in component 2,

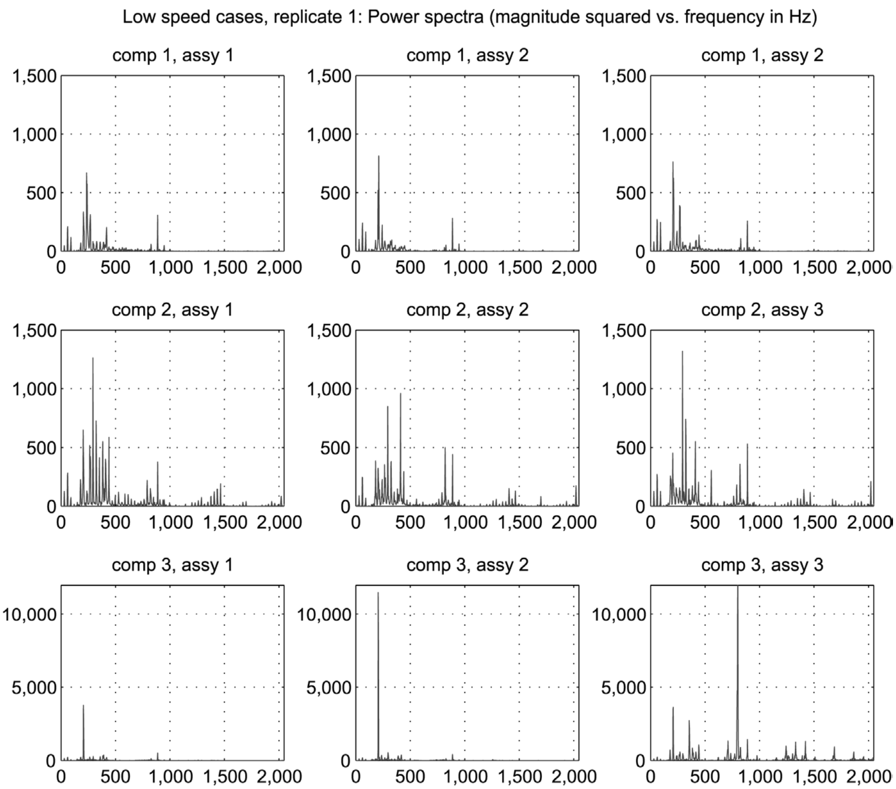


Figure 3.
Power spectral analysis, low-speed case, replicate 1

potentially in the form of surface waviness errors either on the outer race or the inner race. This conclusion will not be finalized until the inner and outer race surface profiles are measured and compared for the two bearings. Component 3, on the other hand, implies a different set of conclusions. The magnitudes of a couple of the harmonics are much higher in comparison to the spectra for the first two components. In addition, a lot more variability is present between the different replicates, possibly indicating an unbalance error within the bearing, since the frequency content is not repetitive or consistent.

Overall, the spectra from all the cases display the typical frequency content for rotating machinery, composed of the shaft rotational frequency and its many harmonics. The presence of harmonics can also be indicative of potential misalignment or mechanical looseness problems, which can both be introduced due to assembly errors. In general, the magnitudes of these shaft harmonics dominate the spectra, possibly masking any of the smaller effects that would be indicative of defects. However, a detailed analysis of the spectra for low and medium speeds shows a couple of frequencies that are not multiples of the shaft frequency, corresponding to the fault frequencies computed in Table I (Tumer and Huff, 2000).

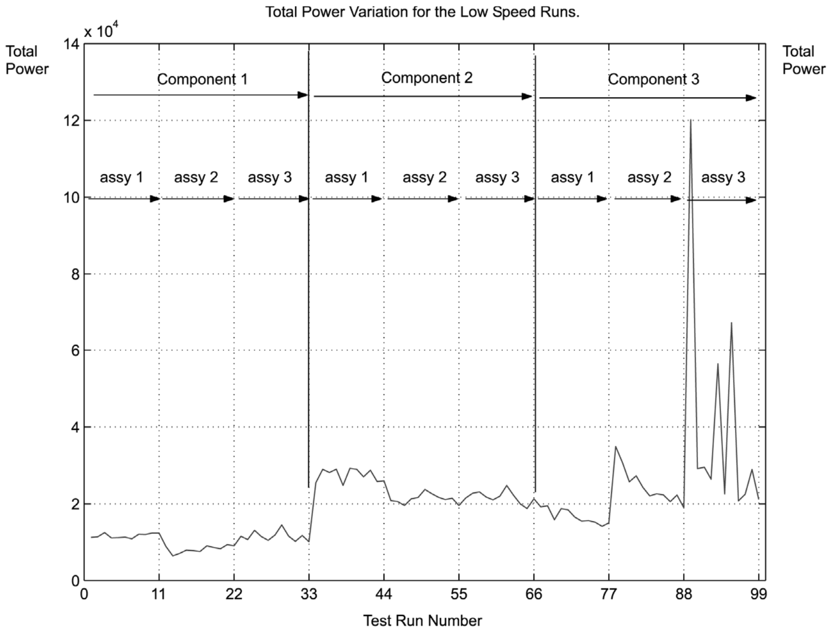
Analysis of trends in total power

A general observation from the analysis above is that the total power in the power spectra varies for each of the bearing components, for each three speeds, in a manner that is suggestive of some variation in the components. To investigate this observation further, the total power in the frequency domain (or variance in the time domain) is analyzed for each test condition. The change in the total power for each component and assembly instances for the low, medium, and high speed cases is shown in Figure 4. The x-axis in these plots corresponds to each test run, for each component and assembly conditions, with 11 replicates for each run. For the case of the low-speed runs, component 1 shows the lowest total power levels, with a consistent increase for component 2, which validates earlier observations from the power spectral analysis. Component 2 has some characteristic that results in higher total power, possibly due to manufacturing errors. The total power levels are initially lower for component 3 than for component 2, but increase slightly for the rest of the runs. In particular, component 3 manifests a large amount of variability in the case of the third assembly. It is possible that an assembly error was introduced during the final disassembly and reassembly phase, hence resulting in the spectral energy to vary as the magnitude and direction of the impact forces on the bearings and shaft vary due to assembly errors. For the remaining components, it is difficult to observe any changes due to the different assembly conditions.

For the medium-speed runs, more variability is introduced amongst the replicates as the shaft speed in the experiments is increased. The total power for component 2 is slightly higher than for components 1 and 3, just as was observed for the low speed runs. It is difficult to observe any differences due to the three assembly conditions. Similarly, for the high-speed runs, the variability in the total power is so significant that it is difficult to isolate any specific patterns for the different bearing components.

Analysis of variance in vibration characteristics

The analyses of the frequency content and total power for the test conditions show some preliminary trends that indicate that manufacturing defects (represented in the form of three different bearing components) have an effect on the vibrational energy. Since speed is a major factor in the amount of vibrational energy emanating from the bearing fault simulator setup, it is reasonable to expect a second and third order interaction between speed and manufacturing errors, and possibly assembly errors. An analysis of variance approach is used to study these interaction effects, as well as the individual effects and their significance for the vibrational energy. The response metric is total power of the power spectra (which equals the variance of the vibration series in the time domain.) The main assumption for an ANOVA approach is that the data are sampled from a normal distribution (Montgomery, 1991). The normality assumption is met for the originating data. However, as observed from the analysis presented above, the total power varies greatly as a function



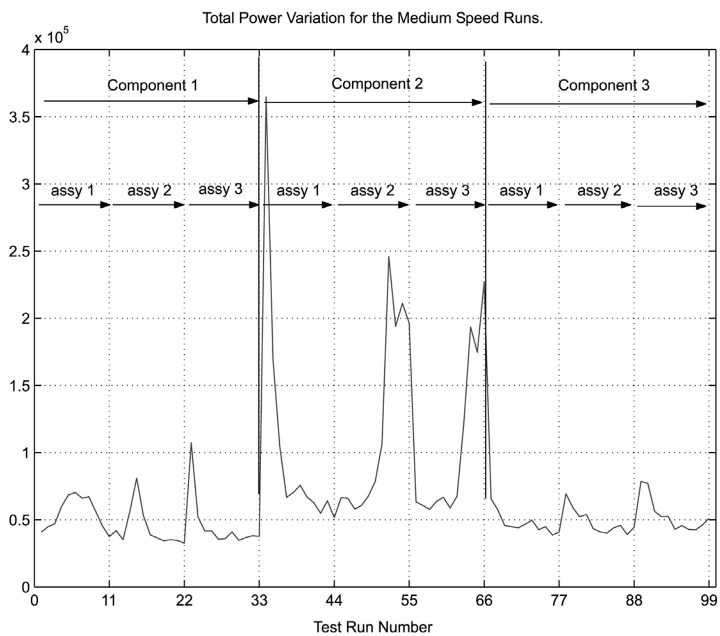


Figure 4.
Total power variation for low, medium and high speed runs

(*continued*)

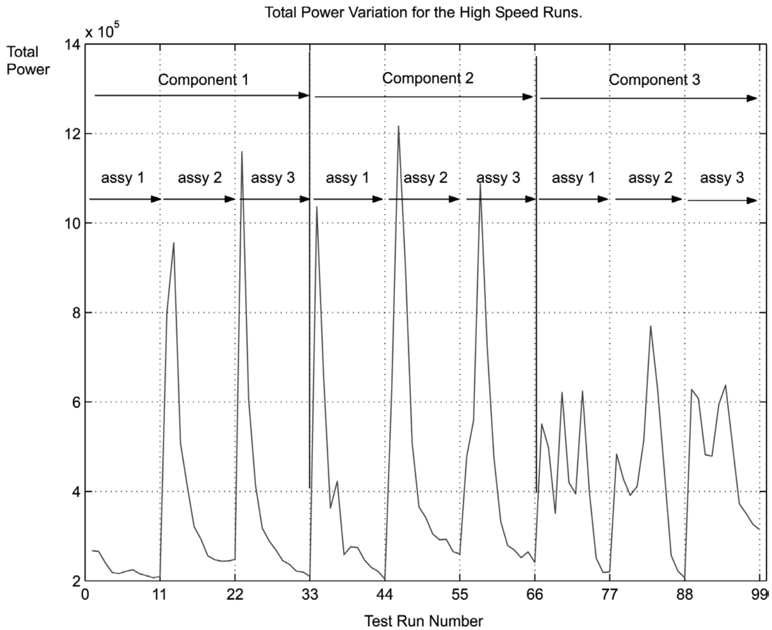


Figure 4.

of speed, hence casting some doubt to the validity of the results. A typical solution to such a problem is to transform the response metric to make it behave as a normal distribution, which was done in this paper.

These initial results validate the general observations from the power spectral analysis. Table II shows the ANOVA results for the log-transformed total power metric, providing more reliable conclusions about the significance of each individual and interaction effect. *SS* is the sum-of-squares value, *DF* is

Source	*SS*	*DF*	*MS*	*F*	Sig.
Intercept (hyp)	37,221.742	1	37,221.742	4,047.495	0.000
Error	18.687	2.032	9.196		
Speed (hyp)	454.178	2	227.089	101.344	0.000
Error	8.314	3.710	2.241		
Assembly (hyp)	0.807	2	0.404	1.439	0.510
Error	0.277	0.987	0.281		
Manufacturing (hyp)	18.224	2	9.112	4.154	0.117
Error	7.843	3.576	2.193		
Speed × assembly (hyp)	1.467	4	0.367	0.904	0.505
Error	3.245	8	0.406		
Speed × manufacturing (hyp)	9.118	4	2.280	5.620	0.019
Error	3.245	8	0.406		
Assembly × manufacturing (hyp)	1.277	4	0.319	0.787	0.565
Error	3.245	8	0.406		
Speed × assembly × manufacturing	3.245	8	0.406	3.188	0.002
Error	34.352	270	0.127		

Table II. Analysis of variance results for log transformed total power the degrees of freedom, *MS* is the mean square value, *F* is the statistic value to assess significance, to be coupled with the significance level which indicates the probability that the effect is significant (Montgomery, 1991).

The results from the transformed metric indicate that speed is the main effect with a probability of $\alpha = 0.001$, followed by the third order interaction between speed, manufacturing errors, and assembly errors (probability of $\alpha = 0.002$). The second order interaction of speed and manufacturing is the third significant effect with a probability of $\alpha = 0.019$. Manufacturing errors could be assessed as significant, if the acceptable probability level were set at $\alpha = 0.2$ (significance level of $\alpha = 0.117$). Assembly error does not show to be a significant factor by itself, or in its interactions with the other two factors.

Discussion of results: towards a methodology

The analysis of the vibration data discussed in the above sections point to an initial correlation between production/maintenance variations and vibrational signature changes. Specifically, a random sample of bearings drawn from a population of bearings is assumed to contain manufacturing defects in the form of surface waviness errors. The results above have shown a possible manufacturing defect in the second bearing used for the experiments. The analysis of variance based on the experiments indicates that manufacturing errors and their interactions with the different shaft rotational speeds can have a relatively significant effect on the total power in the vibrational energy, whereas no significant effects were found due to assembly/maintenance variations. This paper showed a preliminary study to determine whether a correlation can be established using empirical data, collected in a controlled test environment, using bearings as the main focus of the study. The approach described is recommended by the authors as a preliminary assessment method for any rotating machinery system under production and through maintenance. The work and results described in this paper demonstrate the necessity and need of addressing the problem of correlating production and maintenance errors with vibrations during operation. A complete methodology which will enable production and maintenance engineers to assess the performance quality of the components prior to putting them in their operating state is currently being explored.

While providing an initial empirical correlation between production variations and vibrational energy, the difficulty in analyzing the vibrational signatures for specific defects is clearly demonstrated throughout this paper. This difficulty presents a serious roadblock to the development of tools to assess the performance characteristics of rotating components during product development and maintenance. Several important challenges were identified that need to be addressed before developing a systematic methodology. First, the high degree of variability in the experimental data indicates that the production variations are buried inside large factors which need to be taken into account. Currently, the methods and metrics used for such analyses cannot provide a reliable correlation between production variations and vibrational signatures, which makes it difficult to provide the correct information into the development cycle for early assessment of errors. Future work includes a measurement of the surface profiles of the bearings used in this study, test runs with pre-fabricated production errors, as well as analytical models of such correlations as a basis for comparison. Second, there is a clear difficulty in quantifying and assessing the effect of assembly/maintenance errors on the vibration signature. Future work includes developing a quantifiable metric necessary to feed such information into the product development cycle. Currently, assembly/maintenance errors are being studied in a separate experiment, using an OH58 helicopter transmission test rig at the NASA Glenn Research Center, with the purpose of investigating reinstallation effects on the main pinion housing vibrations (Huff *et al.*, 2000). In the helicopter work, overall power levels show significant changes before and after reassembly, indicating the potential effect of assembly errors on the vibrational signature. Additional experiments are currently being planned using the MFS and the OH58 test rigs to address the rest of the challenges. Future work will concentrate on addressing the challenges identified above, so that automated tools can be developed to feed this information into the production and maintenance phases in a systematic and reliable way.

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