Vibration Analysis of a Sensor-Integrated Ball Bearing

This paper presents an analysis of the vibrational behavior of a deep groove ball bearing with a structurally integrated force sensor. The miniaturized force sensor, accommodated within a slot on the bearing's outer ring, provides on-line condition monitoring capability to the bearing. Analytical and finite element models were developed to predict the sensor output due to bearing dynamic load and rotational speed variations. Experimental studies were conducted on a ball bearing to validate the analytical and numerical solutions. Good agreement was found between the model-predicted sensor outputs and the experimental results. The findings validated the approach of integrated-sensing for on-line bearing condition monitoring. [S0739-3717(00)00203-8]

Introduction

Rolling element bearings are widely used as low friction joints between rotating machine components. Since the rotational motion is often a significant function of the overall system, such as wheels on a train, rollers in a paper mill, or a rotor on a helicopter, proper functioning of a bearing over its designed life cycle is of vital importance to ensure product quality, prevent machine damage or even loss of human life [1]. However, because of faulty installation, inappropriate lubrication, overloading, and other unpredictable adverse conditions, premature and sudden failures of individual bearings often occur in real-world applications [2,3]. For improved bearing health and safety monitoring, the research community has been investigating a wide range of topics related to bearing structural design, dynamics, tribology, fault detection, identification, and signal processing techniques [4–7]. Various sensing and measurement technologies have been developed for improved monitoring of bearing operational parameters. Typical representatives include acoustic emission meters, fiber optic monitors, and vibration spectrum analyzers [8–10]. To measure bearing vibrations, eddy current-based displacement sensors have been developed that can be mounted through or to the side of a bearing cap to monitor radial or axial shaft motion with respect to the bearing [11]. Other types of bearing vibration sensors make use of velocity, acceleration, temperature, or spectral energy information [12–16]. While these efforts have considerably advanced the state-of-the-art of bearing condition monitoring, few of them have been widely used for practical, on-line applications. The need for an effective, reliable, and on-line bearing fault detection technique is evidenced by an emergency Airworthiness Directive (AD) recently issued by the Federal Aviation Administration (FAA), requiring “visual inspections” to detect and replace faulty bearings on several Boeing 777 aircraft engines [2]. This situation clearly indicates the need for developing an integrated bearing condition monitoring and self-diagnostic technique for on-line, in-process applications.

A new technique has been developed that involves the direct structural integration of sensing elements (e.g., force, temperature, and acoustic emission sensors) and telemetric data communication electronics into the bearing environment to provide “self-diagnostic” capabilities [17,18]. During bearing operation, the sensing elements continuously measure the real-time fluctuations of the bearing load variations. If predetermined threshold values representing critical bearing operational parameters are exceeded, an alarm signal will be triggered which is then transmitted out of the integrated sensor by a wireless link to the machine control system for corresponding actions. The major advantage of this technique over the previous efforts is the close vicinity of the fault detection mechanism to the fault source. The resulting shorter signal transmission path leads to quicker fault localization. In addition, the sensors will be less vulnerable to background noise and distortion of fault characteristics. The advancement of miniaturization technologies in recent years, including hybrid, surface mount, and MEMS (microelectromechanical systems) technologies, has made it possible to design and package a millimeter-sized sensor module and have it embedded directly into the outer ring of a bearing. Complemented by wireless data communication and advanced signal processing technologies such as wavelet [19,20] and neural-fuzzy logic [21], the miniaturized sensor module can be functionally expanded to a “telemetric probe” that not only enables on-line bearing condition monitoring, but is also applicable for the diagnosis and prognosis of other manufacturing equipment and systems.

To accommodate the sensor module within the bearing structure, the outer ring of the bearing needs to be modified. The stress-strain relationship and structural modifications of such a modification have been studied to provide a design framework [22]. Of equal importance are the dynamic implications associated with the sensor integration. Since a bearing’s vibrational characteristics are affected by its structural modifications, the dynamic response under varying loads and rotational speeds needs to be analyzed and experimentally verified. This study, together with the bearing structural analysis, provides guidelines for the design and imple-

Fig. 1 A sensor-integrated “smart” ball bearing
Structural Defect-Induced Vibration

In this section, the form of a defect signal is discussed. Several mathematical expressions related to the mechanics of rolling element bearings are introduced. These expressions are of fundamental importance to the development of a bearing vibration model. While a conventional condition monitoring system is essentially a separated data recorder and analyzer, the integrated sensor module studied here is an integral part of the modified bearing structure.

An analytical model for predicting the vibration spectrum of a conventional rolling element bearing having a point defect on its inner ring was developed by [23]. This model was experimentally verified by NASA researchers using nonlinear signal analysis techniques [24]. For the analysis, the defect frequency of a bearing is defined as the frequency at which the rolling elements impact a structural defect on the ring surface (race-way). At each impact, an impulsive input is applied to the bearing structure. The magnitude of the input is proportional to the load carried by the rolling elements, which is a function of the geometry of the bearing. For the presented study, a commercially available deep groove ball bearing (model 6220) was analyzed. Relevant geometry and dimensions are shown in Fig. 2.

Generally, the load distribution on a rolling element bearing, shown schematically in Fig. 3, is given by:

\[
q(\psi) = \begin{cases} 
q_{\text{max}}\left[1 - \frac{1}{2\epsilon}(1 - \cos \psi)\right]^n & \text{for } -\psi_i > \psi > \psi_i \\
0 & \text{elsewhere}
\end{cases}
\]  

(1)

where \(q_{\text{max}}\) represents the maximum load and \(\epsilon\) is the load distribution factor [25]. In a bearing with nominal diametral clearance, \(q_{\text{max}}\) can be approximated by:

\[
q_{\text{max}} = \frac{5F_r}{Z\cos \alpha}
\]  

(2)

where \(F_r\) is the applied radial load, \(Z\) is the number of rolling elements, and \(\alpha\) is the mounted contact angle (variations in clearance and load are discussed later). The value of the exponent \(n\) in Eq. (1) depends on the type of bearings involved. For roller bearings, \(n = 3/2\) and for ball bearings, \(n = 10/9\). The load distribution limiting angle \(\psi_i\), as is seen in many applications, then \(0 < \psi_i < 0.5\) and \(0 < \psi_i < 90\) deg. Here the angle \(\psi_i\) is given by:

\[
\psi_i = \cos^{-1}(1 - 2\epsilon)
\]  

(3)

To make the present analysis independent of the radial load, Eq. (1) was normalized:

\[
q_n(\psi) = \frac{q(\psi)}{q_{\text{max}}}
\]  

(4)

Assuming that the outer ring of the bearing is fixed (as illustrated in Fig. 1) and the inner ring rotates at a frequency \(f_i\), as is seen in many applications, then the frequency \(f_d\) at which an inner ring defect passes over a rolling element can be determined by:

\[
f_d = \frac{f_i}{2}(1 + \lambda)Z
\]  

(5)

where \(\lambda = d_b \cos \alpha / d_m\) is dependent on the bearing geometry, and \(Z\) is the number of rolling elements. An evaluation of eight different deep groove ball bearings with bore diameters in the range of 75 mm < \(d < 110\) mm has shown that the ratio \(f_d / f_i\), given by Eq. (5), is approximately equal to a constant of six. Therefore, for a bearing whose outer ring is fixed, a defect on the inner ring surface would pass over a rolling element at about six times the shaft rotation frequency. For the presented study, the outer ring of
a type 6220 ball bearing whose geometrical parameters are shown in
Fig. 2, was structurally modified [26]. Since the bore of this
bearing is 100 mm, the analysis presented is applicable to various
deep groove ball bearings with a bore diameter in the range of
75 mm < d < 110 mm. Using the given parameters, the ratio of
bearing defect frequency $f_d$ to bearing rotational frequency $f_1$ is
converted as $f_d/f_1 = 5.86$ from Eq. (5), and $\psi_d = 84$ deg from Eq.
(3). While a rolling element would pass over a point defect on the
inner ring approximately 6 times for each inner ring revolution
(assuming a fixed outer ring), a defect impulse occurs only when
the rolling element and defect collide within the load zone (i.e.,
when $-\psi_d < \varphi < \psi_d$). This means that the number of defect im-
 pulse per inner ring revolution can be approximated by:

$$N_i \approx \left| \frac{\psi_d}{180} \right| \left( \frac{f_d}{f_1} \right)^2$$

From Eq. (6), the test bearing will experience $N_i = 3$ defect im-
pulses per revolution. Since the maximum load $d_{max}$ occurs at $\psi$ = 0,
as shown in Fig. 3, an impact between a rolling element and
an inner ring defect at this location will produce a large impulse
and accordingly, a large sensor output. Impacts occurring at $\psi$ ≠ 0
will produce smaller sensor outputs. This is demonstrated by the
simulated sensor output $s(t)$ in Fig. 4, which shows the bear-
ing response to scaled defect impulses over 7 inner ring revolu-
tions (note that the effect of lubrication is not considered in the
presented analysis). The large sensor output shown at $t = 0$ is
cau sed by an impact at $\psi = 0$. The variation in magnitude of the
impulses over successive bearing rotations is caused by the fact
that one rotation of the inner ring does not necessarily advance the
rolling elements by an integer multiple of $2\pi/Z$. The angle be-
tween impacts $\psi_d$ is determined by the product of the rate at
which the inner ring rotates ($2\pi f_1$), as well as the period between
impacts (i.e., $1/f_2$, with $f_2$ given by Eq. (5)), which in turn is
related to the geometry of the bearing. Specifically, the resulting
angle between impacts is given by $\psi_d = 2\pi l/Z 2/1 + \lambda$. For the
bearing analyzed, 7 revolutions of the inner ring are required be-
fore the group of impulses is repeated (i.e., before another impact
 occurs at $\psi = 0$). The resulting vibration spectrum $S(f)$, as shown
in Fig. 5, is a series of pulses occurring at integer multiples of $f_1$.
The pulses vary in magnitude with local maxima at integer multi-
tiples of $f_2$. The main lobes of the spectrum are located at $nf_2$
($n = 0, \pm 1, \pm 2, \ldots$). As discussed previously, for an inner ring
frequency of $f_1 = 1.0$ Hz (i.e., 60 rpm bearing rotational speed)
and using Eq. (5), the defect frequency is $f_d = 5.86$ Hz. Since the ratio
$f_d/f_1$ is constant for deep groove ball bearings with a bore in the
range of 75 to 110 mm, each bearing in this dimensional range
would have a vibration spectrum similar to that shown Fig. 5.
Therefore, Fig. 5 presents a predicted, theoretical vibration spec-
trum for ball bearings within the specified range, each having an
inner ring defect. Collectively, Figs. 4 and 5 present the antic-
ipated defect signals in the time and frequency domains that the
integrated sensor module must detect.

The above-presented analysis was based on a deep groove ball
bearing with positive diametral clearance and a radial load. These
assumptions were used to determine the magnitude and timing of
impacts between a defect and the rolling elements. A change in
diametral clearance would change the domain and range of the
load distribution, and hence the magnitude of the impacts. How-
ever, the basic pattern of defect signal would remain the same.
This is because the basic mechanism behind a defect signal re-
mains the same: it is a sequence of impacts caused by an irregular-
ity on the rolling element/raceway interface. This mechanism is
present, even when the defect is located on a rolling element
instead of on the raceway. It further applies to the situation when
an axial load is added to the loading condition, although the spe-
cifics of the defect signal shape will vary. Furthermore, changing
from a ball bearing to a roller bearing would not change the basic
shape of a defect signal, as Eq. (1) applies to both roller and ball
bearings.

Structural Model of the Outer Ring

To monitor load and vibration within the bearing structure, a
piezoelectric sensor is embedded into a slot cut through the outer
ring. The sensor has solid contact with both the top of the slot and
the bearing housing. Each time a rolling element passes over the
slot, the sensor generates an electrical charge output that is pro-
portional to the load applied to the bearing, $F_r$. Since the outer
ring is structurally supported by the bearing housing, it can be
assumed to be rigid. The piezoelectric sensor can be modeled as a
spring with a stiffness constant $k$ that is related to its material
composition. As shown in Fig. 6, the section of the bearing outer
ring where the slot is cut can be modeled as a beam of varying
cross-section, with a spring support at the midpoint. To establish
the structural model, boundary conditions must first be assigned.
Since the ends of the beam are solidly connected to the surround-
ing bearing structure, which is directly supported by a rigid hous-
ing, clamped boundary conditions are considered appropriate.
This model is shown in Fig. 7. Furthermore, the segment of the
ring encompassed by the slot subtends an angle of $2 \sin^{-1}(L/D)$
= 2$\psi_{max}$ = 12 deg. The difference in ring thickness between
the center and the end of the slot is determined by:

$$\Delta = \frac{1}{2} (d_m + d_b - \sqrt{(d_m + d_b)^2 - L^2})$$

Using the parameters given in Fig. 2, the difference in ring thick-
ness was calculated to be $\Delta = 0.55$ mm. Since $\psi_{max} = 6$ deg is the
maximum angle at which the bearing load $q(\psi)$ is applied, and the
thickness difference $\Delta$ is less than 18 percent of the thinnest
section of the ring (calculated as $(D - 2h - d_m - d_b)/2 = 3.2$ mm), the modified section of the outer ring can be simpli-

Fig. 4 Bearing response to defect impulses over 7 revolutions of the inner ring

Fig. 5 Frequency spectrum of the bearing vibration due to an inner ring defect
fied to that of a beam with a constant cross-section and a vertically applied load, as shown in Fig. 8. The bearing load \( q(c) \) is determined by Eq. (1), and the location \( a \) where the load is applied is related to \( \psi \) by the expression:

\[
a = \frac{1}{2} (d_{a} + d_{b}) \sin(\psi)
\]

(8)

When \( a \leq L/2 \), the deflection of the sensor is given by:

\[
\delta_{a \leq L/2} = \frac{4qa^{2}(4a - 3L)}{192EI - kL^{3}}
\]

(9)

where \( E \) is the elastic modulus of the outer ring material, \( I \) is the area moment of inertia of the cross-section of the modified ring, and \( k = Y_{33}A/l \) is the stiffness of the piezoelectric sensor. The cross-sectional area of the sensor is defined as \( A \), the height is \( l \), and the elastic modulus of the piezoelectric material is \( Y_{33} \).

When \( a > L/2 \), the sensor deflection is given by:

\[
\delta_{a > L/2} = \frac{4q(L - a)^{2}(L - 4a)}{192EI - kL^{3}}
\]

(10)

This translates into a load on the sensor as:

\[
R_{s} = \frac{\gamma_{33}A \delta_{c}}{l}
\]

(11)

As a result, the electrical charge produced by the sensor is:

\[
Q = d_{33}R_{s}
\]

(12)

where \( d_{33} \) is the charge constant of the piezoelectric material. Using a charge amplifier with a gain \( G \), the voltage output produced by the embedded sensor is:

\[
V = GQ = Gd_{33} \frac{Y_{33}A \delta_{c}}{l}
\]

(13)

This voltage output is then a direct measure of the outer ring deflection due to a specific bearing load. Conversely, from the sensor voltage output, the load applied to the bearing can be determined to identify overloading conditions. Thus, the embedded sensor module can be used in principle for both diagnostic and prognostic purposes.

For an inner ring frequency of \( f_{i} = 11.7 \) Hz (corresponding to a bearing speed of 700 rpm), Eq. (13) produces the results as plotted in Fig. 9. Each ‘‘pulse’’ represents a rolling element passing over the sensor module. The width of each pulse is determined by the...
length of the slot, i.e., the length of the beam in Fig. 8. These pulses are relatively long in duration when compared to those caused by a bearing structural defect, as shown in Fig. 4. The high frequency nature of the defect-induced signals is characteristic in distinguishing them from “normal” load pulses caused by the rolling elements passing over the sensor module.

If the curvature of the modified bearing outer ring is not neglected for the analysis, then the moment of inertia of the outer ring geometry, the moment of inertia of the outer ring cross-section at each individual location is a function of the rolling element position $a$. The thinnest and thickest sections of the modified outer ring (shown in Fig. 10) are determined as:

$$h_1 = \frac{D}{2} - h - \sqrt{(d_m + d_h)^2 - a^2}$$

and

$$h_2 = \frac{D}{2} - h - \sqrt{(d_1 + d_h)^2 - a^2}$$

Upon analysis of the outer ring geometry, the moment of inertia of the cross-section where the slot is cut, is derived as:

$$I_0 = \frac{1}{3} wh_2^3 - 2wh_2(h_2 - h_1) \left[ \frac{5\pi}{16} - \frac{2}{3} \int_{a}^{\psi} h_2^2 + \left( \frac{2}{3} \int_{a}^{\psi} h_1 h_2 + \frac{\pi}{16} h_1^2 \right) \right]$$

(16)

where $h_1$ and $h_2$ are given by Eqs. (14) and (15), respectively. The centroid of the cross-section is located at:

$$y_c = \frac{3wh_2^2 - w_g(h_2 - h_1)[(3\pi - 4)h_3 + 4h_1]}{6wh_2 - 3\pi w_g(h_2 - h_1)}$$

(17)

and the area of the cross-section is:

$$A_c = wh_2 - \frac{1}{2} \pi w_g(h_2 - h_1)$$

(18)

The moment of inertia about the centroid is then:

$$I_0 = \frac{1}{3} wh_2^3 - 2wh_2(h_2 - h_1) \left[ \frac{5\pi}{16} - \frac{2}{3} \int_{a}^{\psi} h_2^2 + \frac{2}{3} \int_{a}^{\psi} h_1 h_2 + \frac{\pi}{16} h_1^2 \right]$$

(19)

where $y_c$ is given by Eq. (17).

Assuming that the curvature of the nonprismatic beam shown in Fig. 7 is determined by the moment of inertia and bending moment at $a$, then

$$\frac{d^2 y}{da^2} = \frac{1}{E} \frac{M(a)}{I_0(a)}$$

(20)

where $a$ is a function of $\psi$ as given in Eq. (8). The slope and deflection are then given by:

$$\theta(a) = \frac{dy}{da} = \frac{1}{E} \int_{a}^{\psi} M(a) I_0(a) \, da + C_1$$

(21)

$$\phi(a) = \int_{a}^{\psi} \theta(a) \, da + C_2$$

(22)

where $C_1$ and $C_2$ are constants of integration determined by the boundary conditions. With the moment of inertia $I_0$ given by Eq. (19) and $h_1$, $h_2$ and $y_c$, given by Eqs. (14), (15) and (17), respectively, the outer ring structural model becomes mathematically quite complex (the problem is statically indeterminate) when compared to the simple model given by Eq. (13). To reduce the computational load while maintaining the specific features of the bearing structure, a finite element (FE) model was developed to calculate the outer ring deflection $\phi(a)$ and slope $\theta(a)$.

Finite Element Model of the Outer Ring

Technically, the complete bearing structure can be modeled using a complex three-dimensional finite element model. However, by observing the nature of the boundary conditions and loads on the outer ring, it was determined that the FE modeling process could be greatly simplified by making use of the symmetry of the system. With respect to the boundary conditions, the surfaces of the bearing’s outer ring are fixed as indicated in Fig. 1. The loads on the bearing structure are applied to the outer ring through small ellipsoidal contact areas between the rolling elements and the outer ring groove. Assuming a pure radial load, the maximum of
the resulting Hertzian stress distribution is located at the base of the outer ring groove. Since this stress distribution is symmetric about the plane which divides the outer ring through the base of the groove, the strain normal to this plane, $\varepsilon_\perp$, is zero. Taking advantage of this feature, the FE model was constructed using four-node quadrilateral plane strain elements. A diagram of this type of element is shown in Fig. 11. The shape functions are:

$$ u = \frac{1}{4} [u_x(1-s)(1-t) + u_y(1+s)(1-t) + u_z(1+s)(1+t) ] + u_z(1-s)(1+t) + u_x(1-s^2) + u_y(1-t^2) $$

and

$$ v = \frac{1}{4} [v_x(1-s)(1-t) + v_y(1+s)(1-t) + v_z(1+s)(1+t) ] + v_z(1-s)(1+t) + v_x(1-s^2) + v_y(1-t^2) $$

where $u$ and $v$ are the displacements along the $z$ and $y$ directions, respectively. Shown in Fig. 12 is the mesh of plane-strain elements used to model the bearing. For the steel outer ring, structural elements were used. Considering the electro-mechanical interaction associated with the piezoelectric sensor, coupled-field elements were used for sensor output analysis. The constitutive equations for the piezoceramic material are:

$$ \{T\} = [C] \{\varepsilon\} $$

and

$$ \{D\} = [D] \{\varepsilon\} $$

where $\{T\}$ is a stress vector, $\{D\}$ is an electric flux density vector, $[C]$ is an elasticity matrix, $[\varepsilon]$ is the piezoelectric matrix, and $[\varepsilon]$ is the dielectric matrix, $\{\varepsilon\}$ is a strain vector, and $\{E\}$ is an electric field vector. The shape function for the electric field variable is analogous to Eq. (23). The steel and piezoceramic material properties are given in Table 1.

For both the simplified and true-geometry outer ring structural models presented above, the output of the embedded load sensor was assumed to be due only to a rolling element passing directly over the slot. However, for the finite element analysis, the effect of all loaded rolling elements was taken into account. All the rolling elements were modeled as compressive forces acting on the outer ring. The magnitude of each force was calculated by Eq. (1), and the location was determined by calculating the position of each rolling element based on the rotational speed of the inner ring. In Fig. 13, the sensor output predicted by the finite element model and that by the simplified beam model are compared. Since the FE model considered the effect of adjacent rolling elements that were not located directly over the slot, an increase of the width of each sensor output pulse was observed from the FE model. This means that the load sensor begins to respond to a rolling element before it actually runs over the slot. Due to the structural rigidity of the bearing, the actual difference between the FE and simplified analytical beam models is very small, especially in terms of the maximal signal amplitude ($<2$ percent). The agreement between these models verify the assumptions that were used to derive the simplified beam model. While either model can be used to analyze the sensor output of the bearing, the simplified beam model is a better design tool, since the bearing and sensor parameters are stated explicitly in Eqs. (9), (10), and (13). For example, these equations can be used to design an embedded sensor with an output range that satisfies requirements of the data acquisition hardware of other supporting electronics.

### Table 1 Material properties used for modeling the bearing vibrations

<table>
<thead>
<tr>
<th>Material: Piezoceramic type PXE-5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density: $\rho = 7700 \text{ kg/m}^3$</td>
</tr>
<tr>
<td>Poisson’s Ratio: $\nu = 0.31$</td>
</tr>
<tr>
<td>Elastic constants: $Y_{11} = 69 \text{ GPa}$, $Y_{33} = 106 \text{ GPa}$</td>
</tr>
<tr>
<td>Charge constants: $d_{33} = 374 \times 10^{-12} \text{ m/V}$, $d_{31} = -171 \times 10^{-12} \text{ m/V}$</td>
</tr>
<tr>
<td>Voltage constants: $g_{31} = 24.8 \times 10^3 \text{ V/m/N}$, $g_{33} = -11.4 \times 10^3 \text{ V/m/N}$</td>
</tr>
<tr>
<td>Relative permittivity: $\varepsilon = 1700$</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Material: Steel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density: $\rho = 7860 \text{ kg/m}^3$</td>
</tr>
<tr>
<td>Poisson’s Ratio: $\nu = 0.33$</td>
</tr>
<tr>
<td>Elastic constant: $Y_{11} = 200 \text{ GPa}$</td>
</tr>
</tbody>
</table>
Results Comparison

To further verify the theoretical results obtained from the analytical and FE models, an experimental investigation was conducted using the test bearing shown in Fig. 2. A bearing test bed was designed and fabricated for the experiments, as shown in Fig. 14. The inner ring of the modified bearing was mounted on a 100 mm shaft and secured with an end cap. A flat ring was used to secure the outer ring in a housing. With the shaft mounted in the chuck of an industrial lathe, a threaded rod was inserted through a frame, screwed into the top of the housing and secured with a nut. The frame was supported by the lathe. Turning a nut at the top of the rod compressed a calibrated spring and applied a radial load to the bearing. With the spring stiffness constant \( k_s \) (lb/in) and the number of threads per inch on the rod \( N_t \) known, the spring deflection, and therefore the magnitude of radial load, were determined. The free end of the shaft was supported with a live center, and the lathe was used to spin the shaft at various rotational speeds while the housing remained fixed. Data from the embedded sensor was recorded using a 166 MHz laptop computer, which was equipped with a PCMCIA-based data acquisition system.

The cross-sectional area of the piezoelectric sensor was \( A = 41.4 \text{ mm}^2 \) and the height was \( l = 2.5 \text{ mm} \). The sensor was made out of type PXE-5 piezoceramic. The corresponding material properties are given in Table 1. The frequency at which the rolling elements pass over the slot where the sensor is located is given by:

\[
f_{d_0} = f_i \frac{1}{2} (1 - \lambda) Z \tag{26}
\]

where \( f_i \) is the shaft frequency, \( Z = 10 \) is the number of rolling elements, and \( \lambda \) is defined by \( \lambda = d_b \cos \alpha/d_m \). For a typical experimental run with a lathe setting of 700 rpm, \( f_i = 11.7 \text{ Hz} \), resulting in \( f_{d_0} = 68.5 \text{ Hz} \), as per Eq. (5), and \( f_{d_m} = 48.5 \text{ Hz} \). Since \( Z = 10 \), the rolling elements in this bearing are each separated by 36 deg. The corresponding arc length is 36 deg \( \times \pi/180 \text{ deg} \times D/2 = 56.5 \text{ mm} \) which is approximately three times the length \( l \) of the slot. The experimental results are shown in Fig. 15. A comparison with the predicted results from the FE and simplified analytical models shows that the theoretical and experimental studies are in good agreement. To quantify this observation, the root-mean-square (RMS) error \( e_{rms} \) was evaluated:

\[
e_{rms} = \sqrt{\frac{1}{N} \sum (V_{exp} - V_{FE})^2}
\]

where \( V_{exp} \) is the sensor output recorded from the experiment, \( V_{FE} \) is the sensor output predicted by the finite element model, and \( N \) is the number of data points recorded. From the data plotted in Fig. 15, the value of \( e_{rms} \) is calculated as \( e_{rms} = 0.54 \text{ Volts} \), which is less than 17 percent of the maximum recorded sensor output. This error is attributed to two reasons. First, unlike the sensor output predicted by the FE model, the experimental data did not
reach exactly the zero output line due to vibrational noise, which was observed from the machine environment during the experiments. Because of the high sensitivity of the piezoelectric sensor, the structure-borne vibration of the machine was contained in the sensor output. Secondly, in the FE model, the housing was assumed to be infinitely rigid and in solid contact with the outer ring. However, the experimental housing has a certain degree of flexibility since it was made of aluminum which is an elastic material. The output of the sensor is affected by this flexibility because the sensor is a structural link between the top of the outer ring slot and the housing. The largest error between the measured and predicted sensor outputs occurs when a rolling element is located near the edge of the slot (i.e., the location of a rolling element is approximately \( \phi_{\text{max}} \)). This confirms that the embedded sensor actually starts responding to a rolling element load before the rolling element itself is exactly over the slot. The smallest errors occur either when a rolling element passes over the center of the slot (i.e., when the sensor output is dominated by the rolling element load) or when the rolling elements are located as far as possible from the slot (i.e., when the sensor output is dominated by noise from structure borne vibration). This agreement between the analytical, FE, and experimental techniques further verifies the simplified beam model as an accurate and efficient design tool.

To verify that Eq. (26) correctly gives the frequency at which the rolling elements pass over the slot, the spectrum of the sensor output was calculated and plotted in Fig. 16. In the upper portion of the figure, several important features of the normalized spectrum have been indicated. Two of these features are inherent characteristics of systems involving rotating machinery and sensitive electronics. The first feature is a small peak located at the shaft speed \( f_i = 11.7 \) Hz. This was caused by a slight imbalance of a rotating component (e.g., the lathe chuck). Second, the frequency of the power supply for the computer and lathe motor was 60 Hz, and electrical coupling between the power supply and the sensor electronics produced a 60 Hz component in the sensor signal. The most distinct feature in the spectrum is a large peak caused by the rolling elements passing over the load sensor at the frequency \( f_{d_o} \).

![Fig. 15 Predicted and experimental sensor outputs](image1)

![Fig. 16 Spectrum of the experimental data](image2)
Evaluating Eq. (26) with the parameters given in Fig. 2 and \( f_v = 11.7 \text{ Hz} \), the roll-over frequency is \( f_{d_k} = 48.5 \text{ Hz} \), which agrees with the experimental results shown in Fig. 16(a). The harmonics of \( f_{d_k} \) are also very prominent features of the spectrum. According to the vibration model described in the section “Structural Defect Induced Vibration,” an inner ring defect would produce a spectrum with lobes centered at the defect frequency \( f_{d_k} = 68.5 \text{ Hz} \), as shown in Fig. 5. The spectrum shown in Fig. 16(a) has what appears to be a very small lobe at 68.5 Hz, despite the fact that the test bearing had no inner ring defect. As stated above, the sensor signal was contaminated with structure-borne noise. Since this type of noise is generally uncorrelated (as opposed to the vibration caused by a defect), the lobe at 68.5 Hz should not appear in the auto-correlation spectrum, which is shown in Fig. 16(b). Indeed, the auto-spectrum has no lobe at 68.5 Hz. The main feature of the auto-spectrum is a peak at \( f_{d_k} = 48.5 \text{ Hz} \), the frequency at which the rolling elements pass over the load sensor. The harmonics of \( f_{d_k} \) are much less pronounced, as are the peaks at the shaft speed and power supply frequency.

Conclusions

The structural integration of a load sensor into the outer ring of a rolling element bearing provides an effective means for assessing the time-varying load conditions within the bearing structure. The developed beam and finite element models were able to predict the output of an outer ring-embedded load sensor module quite accurately, thus providing valuable input to the sensor design. Results of the study have shown that a bearing defect produces a pattern of signals which is not repeated between successive revolutions of the inner ring. It has also been shown that an impact between a defect and a rolling element is a relatively high frequency phenomena when compared to the pulse caused by a rolling element passing over the sensor module. The basic shape of the pulse is given by Eqs. (9), (10) and (13), which are not sinusoidal expressions. Thus, in the frequency domain, the load measurement pulse has several harmonics that can bury the signature of a defect, and with the addition of structure-borne noise, a defect might not be discovered by simply looking at the spectrum of the sensor signal. However, it has been shown that calculating the auto-correlation spectrum effectively compensates for the effect of the load pulse harmonics and structure-borne noise. This emphasizes the importance of combined time and frequency domain analysis of the sensor signal to accurately assess the condition of a bearing. With the combination of a well-designed sensor, an efficient signal processing algorithm and a knowledge of bearing mechanics, the condition of a bearing can be established, as demonstrated by the analytical and experimental results from this study. These results have shown that an integrated sensor is able to provide effective bearing condition monitoring through dynamic load and vibration measurement.

Future Work

Future research will focus on developing advanced and efficient signal processing techniques using wavelet transformation and neural-fuzzy networks to relate signal features to specific bearing faults. Also, further experimental investigations will be conducted on an improved bearing test bed that allows for a much wider range of operating conditions to be studied. The combined theoretical and experimental work will establish a strong basis for the development of an integrated on-line bearing condition monitoring system to provide early defect warning capabilities to a wide range of rolling element bearings and manufacturing equipment.

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References