ENGINEER'S notebook

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A CLASSICAL SLEEVE BEARING INSTABILITY IN AN OVERHUNG COMPRESSOR

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PROBLEM DESCRIPTION

An old centrifugal compressor of the overhung type with a single impeller was put into service after modifications to the flow path. Because the rotating component modifications were small, the same bearings were used. The bearings were sleeve bearings, with the impeller end (non-drive end) bearing being a three-lobe bearing and the drive end bearing being a two-pad bearing with axial supply grooves. In particular, the bearing in the impeller end worked both as a bearing support and as an annular seal to contain the discharge pressure of the compressor. Figure 1 is a schematic of the compressor rotor and seal layout in the non-drive end. Table 1 illustrates the main dimensions of the bearings on both ends along with the carrying static loads.

During the starting procedure, a subsynchronous vibration appeared at approximately 5,200 rpm, as shown in Figure 2. The vibration was observed to be speed dependent at slightly more than half the running speed frequency.

ANALYSIS OF THE PROBLEM

Figure 3 depicts the bearing detailed geometry.

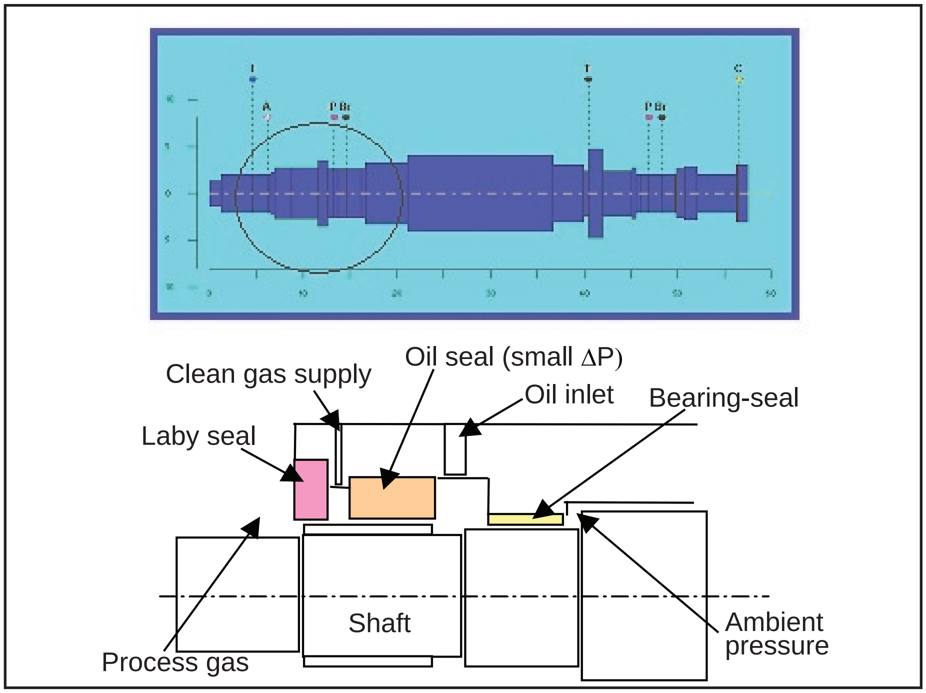

Figure 1. Compressor rotor and area of bearing-seal.

ABSTRACT

In a revamp overhung, rigid rotor compressor, an oil whirl instability appeared from the sleeve bearings. The peculiarity of the bearing arrangement in the impeller end is that the bearing also works as a seal. Bearing calculations and rotordynamic analyses predicted the threshold speed of instability. The analyses showed that the instability came from this bearing, even though the rotor had not crossed any bending critical speed. The axial pressure gradient had a small effect on the hydrodynamic film of the bearing because the flow is laminar. A solution based on a tilting pad bearing/seal was used to solve the problem, and the compressor achieved satisfactory operation at full speed.

Note the absence of supply grooves in the non-drive end (NDE) bearing because this element has to seal the oil pressure on one side. The initial analysis of the compressor with the existing bearings was challenging because the exact effect of the large axial pressure drop across the NDE bearing was unknown. Ignoring the axial pressure profile, the bearing can be analyzed with a bearing predictive code solving the Reynolds equation for the given lobe geometry [1,2]. The perturbation analysis of the Reynolds equation at the steady state operating condition renders the bearing rotordynamic force coefficients (stiffness and damping) used in the stability analysis.

With this bearing geometry, the compressor stability was verified to be in jeopardy for the maximum bearing clearance case close to the observed threshold speed of instability (see Figure 4). Note that the compressor goes unstable even though it hasn't crossed the first bending critical speed that is predicted to be approximately 13,700 rpm. The mechanism for the instability comes directly from the journal bearing dynamics. In particular, predictions show that the bearing with the lowest

Non-Drive End (NDE)	Drive-End (DE)
Diameter: 5.25 in	Diameter: 4.125 in
Length: 3.00 in	Length: 2.375 in
Type: Three-lobe (no axial feeding grooves)	Type: Two-pad with axial feeding grooves
Static load: 640 lb	Static load: 205 lb

Table 1. Compressor bearing dimensions - original built.

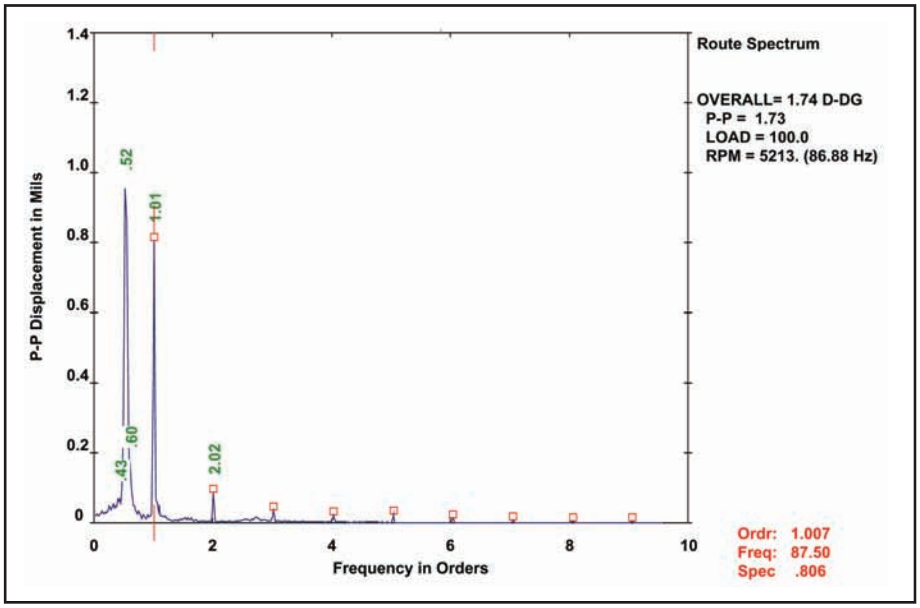

Figure 2. Frequency spectrum of vibration of compressor rotor on sleeve bearings.

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threshold speed of instability is the bearing in the impeller end (NDE, three-lobe bearing). This is expected because this bearing has the closest geometry to a cylindrical journal bearing.

In order to verify the predictions of the hydrodynamic bearing code for the NDE bearing, predictions of force coefficients from a specialized bearing-annular seal code used for cryogenic liquid bearings in high-pressure turbo-pumps [3] were compared with the original predictions (see Figures 5 and 6). These predictions show that the axial pressure distribution has some effect on the rotordynamic coefficients affecting stability (cross-coupled stiffness and direct damping).

In particular, cross-coupled coefficients increase because of the axial pressure gradient. The negative effect of this on the stability is offset by a corresponding increase in the direct damping coefficients, so that the net effective damping from the bearing is only slightly modified. The verification of the bearing force coefficients allows confirmation of the predictions of stability, as observed during the actual start-up of the compressor.

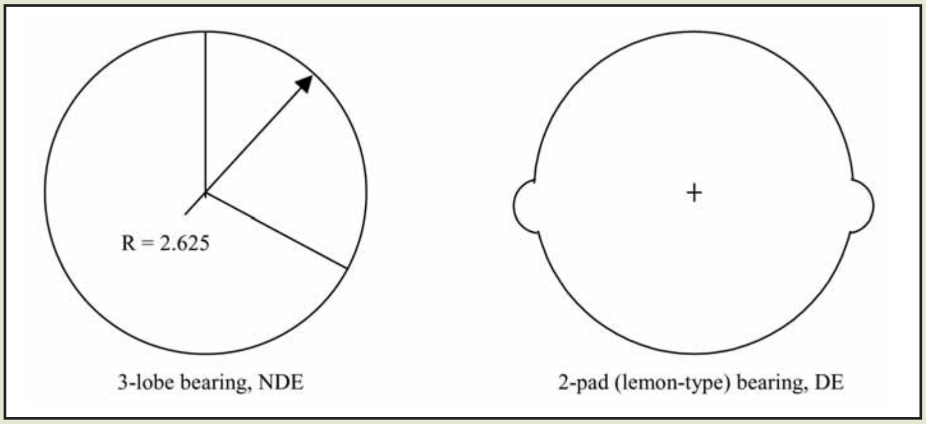

Figure 3. Geometry of original sleeve bearings of compressor rotor. Note that the NDE bearing has no axial grooves.

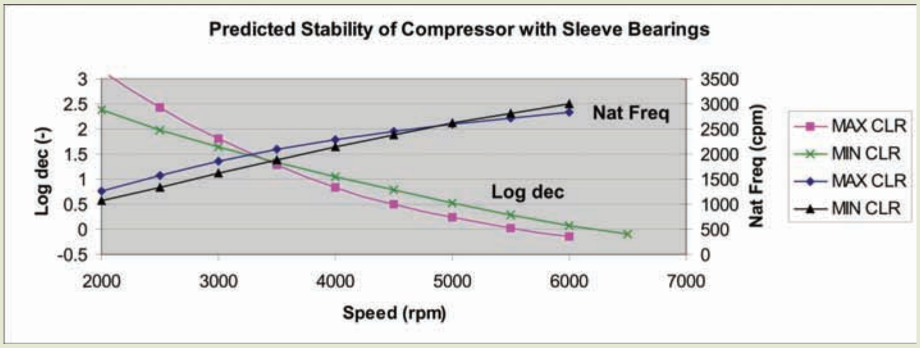

Figure 4. Predictions of stability indicator (log dec) and natural frequency as function of speed. Original bearing configuration.

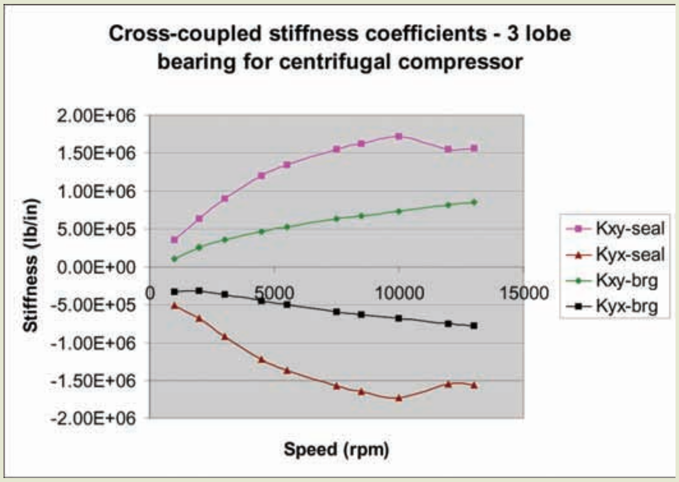

Figure 5. Cross-coupled stiffness coefficients for original NDE bearing of centrifugal compressor.

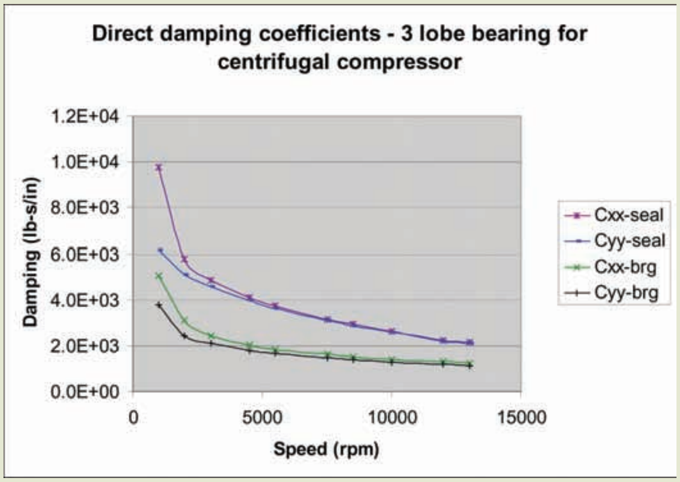

Figure 6. Direct damping coefficients for original NDE bearing of centrifugal compressor.

In addition to the three-lobe bearing, analysis of the two-pad bearing on the drive end of the machine reveals that the onset speed of instability for this bearing is higher than the onset for the impeller end bearing. This is important to note because analysis of the problem focuses on the three-lobe bearing for this reason.

Figure 7 shows the predicted onset speed of instability for each bearing as a function of the rotating speed under the static carrying loads. Clearly, the three-lobe bearing on the non-drive end becomes unstable earlier than the drive-end bearing. Note also that the bearing onset speed of instability information from Figure 7 helps in confirming the source of the instability predicted in the stability map of Figure 4. It also shows why the compressor becomes unstable before reaching the classical "oil whip" at twice the rotor-bearing first flexible natural frequency.

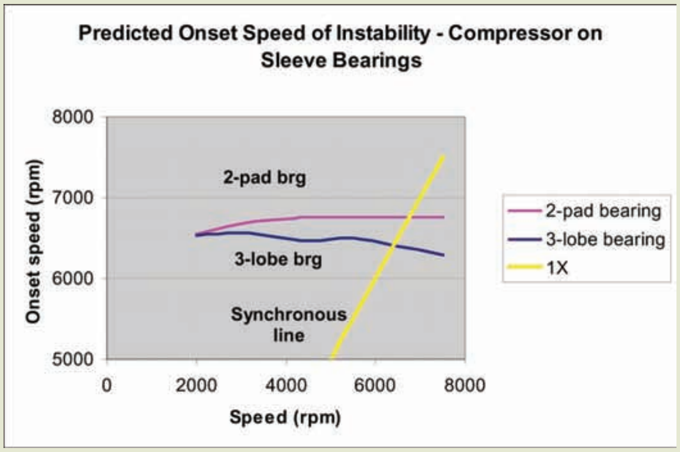

Figure 7. Predicted onset speed of instability of original centrifugal compressor sleeve bearings under carrying static loads.

SOLUTION TO THE PROBLEM

The instability problem is handled by the use of tilting pad bearings. Care must be exercised on the NDE bearing because it still must take an axial pressure gradient. Based on the manufacturer's experience with tilting pad seals, the solution involves a tilting pad bearing in the high-pressure section of the bearing cavity and a multiple-land babbitt seal to break down the oil sealing pressure. Figure 8 shows a schematic of the bearing-seal arrangement. The bearings are four-pad, load-between pad bearings with medium preload.

It is not unusual to have a bearing/seal assembly that combines tilting pads with oil-film ring seals. Tilting pads have been applied to the outer oil-film rings of the casing end seals of centrifugal compressors for many years. This configuration is used to eliminate or prevent detrimental subsynchronous vibration. The consistent reliability of units using the design has led to tilt pad seals becoming standard in many types of compressors with oil-film seals [4-12]. More than 600 compressors with tilting pad seals have been installed around the world in many different services. The design has been used at sealing pressures up to 21,475 kpa (3,100 PSIG), and for speeds up to 22,800 rpm. The first installation was in 1972 [5, 6]. For case histories of change-outs from ring seals to tilt pad seals see the papers [5-8, and 11]. Use of the tilt pad bearing programs in the analytical modeling of tilt pad seals is discussed in [12]. The most recent and extensive discussion of the analytical modeling of tilt pad seals is given in [11].

Figure 9 shows predicted stability of the compressor with the new tilting pad bearings and seal arrangement. Both bearings (DE and NDE) were changed to tilting pad bearings to enhance the stability margin.

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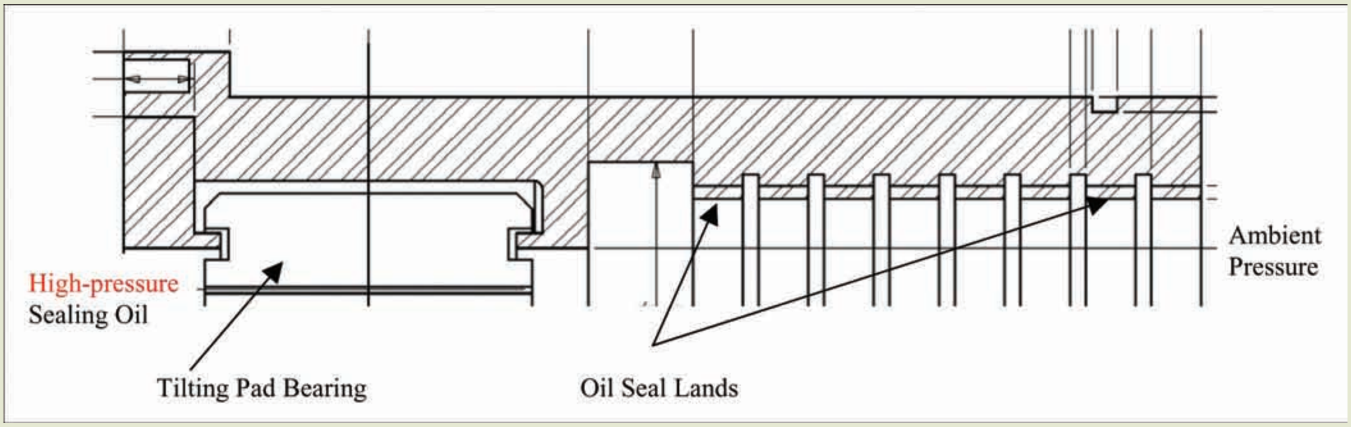

Figure 8. Schematic of final bearing-seal configuration. Tilting pad bearing with babbitted land, fixed geometry oil seal.

After the change of bearings, the compressor performed as expected, with moderate levels of vibration. Figure 10 shows vibration measurements taken during commissioning. No sign of subsynchronous vibration is present in the operating speed range.

SUMMARY AND REMARKS

This paper presents the dynamic analysis of an overhung compressor that presented an instability caused by the original sleeve bearings. Being overhung means that there is more uneven loading on the bearings when compared to most beam-style compressors. Interestingly, in this particular case, the more heavily loaded bearing goes unstable first because it contains no axial grooves to break-up the circumferential oil flow.

The compressor is a stiff shaft compressor, with the rigid bearing first critical speed twice the maximum continuous speed, and the synchronous first critical speed predicted to be above the running speed range. The rigid body modes were predicted to be below the running speed range and to be very well damped, with amplification factors under 2.0. Yet the compressor went unstable because of a sharp reduction in available system damping when the bearings lost the load capacity at their own onset speed of instability. This was without "locking on" to the classical first critical speed. What is unusual about this case study is that there is instability without the bearing whirl frequency locking onto the first critical speed.

The bearing that was installed on the high-pressure side of the compressor actually worked as a seal. Substitution of this bearing by a tilt pad seal/bearing solved the root problem and allowed satisfactory operation of the compressor.

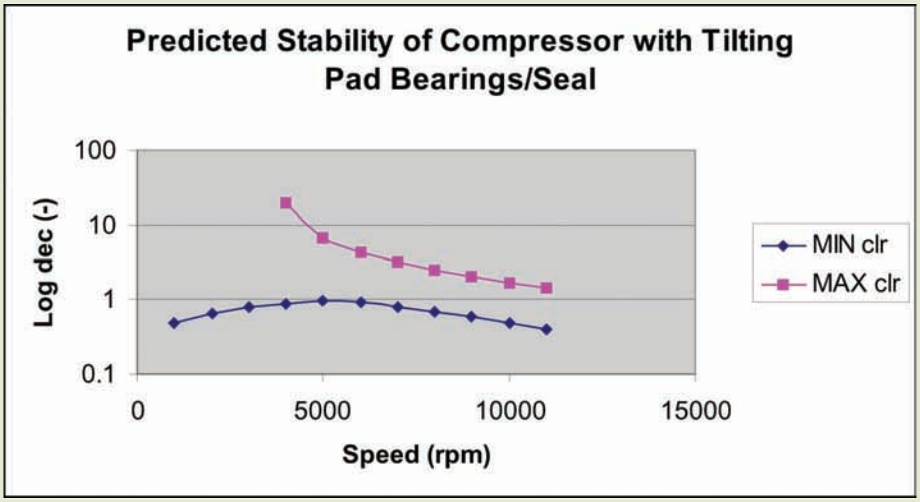

Figure 9. Predictions of stability indicator (log dec) as functions of running speed for compressor on new tilting pad bearings and multi-land oil seal.

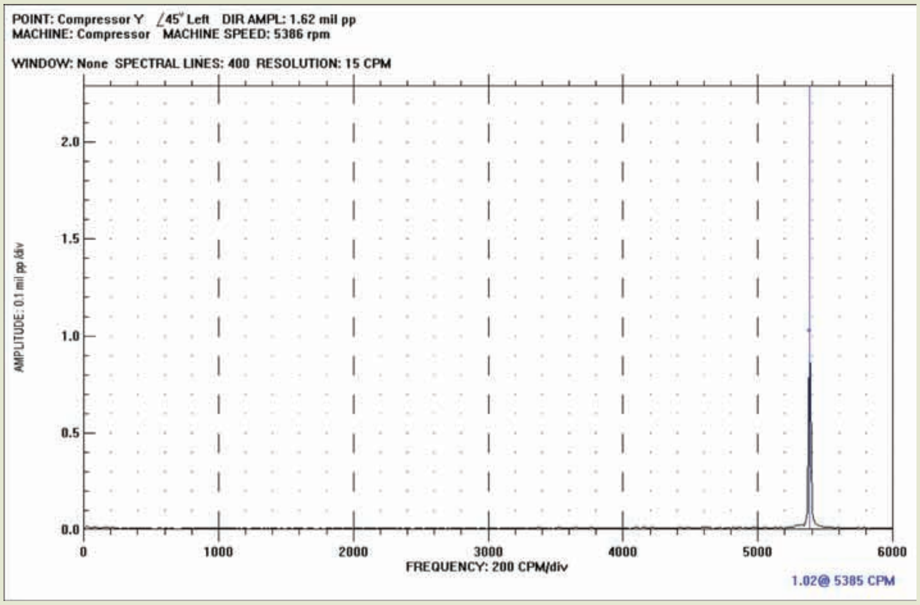

Figure 10. Compressor spectrum after change to tilting pad bearings.

ACKNOWLEDGMENTS

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BIOGRAPHIES

Oscar De Santiago is a senior rotordynamics engineer at Dresser-Rand in Olean, NY with more than 10 years experience in the turbomachinery industry. His professional interest is in current advances in tribology as applied to the rotordynamics of turbomachinery. Oscar began his career in Mexico as an analyst engineer of bearings in steam turbines, compressors, and gearboxes for the oil and sugar cane industries. He joined Dresser-Rand in 2002 and works in the rotordynamics group after spending one and a half years as a product design engineer. His current responsibilities include review and approval of rotordynamic designs of centrifugal compressors, turbo expanders, and power turbines, among others. He is also responsible for research and development efforts in the areas of magnetic bearings and other advanced rotor support systems. Oscar holds M.S. and Ph. D. degrees from Texas A&M University, and has published several technical papers on squeeze film dampers, advanced bearings, and parameter identification methods, among others. Oscar enjoys family life and is a (big time) soccer fan.

Ed Memmott is a principal rotor dynamics engineer and has been with Dresser-Rand since 1973. He has written 16 other papers on rotordynamics (most of them on high-pressure centrifugal compressors), and has given short courses on rotordynamics. In May 2007 at the ASME Turbo Expo 2007 in Montreal, he and Krish Ramesh presented the one-day short course, "Rotordynamic Analysis of Centrifugal Compressors and API 617 Review." In October 2005 at the 23rd Machinery Dynamics Seminar of the CMVA in Edmonton, he presented a three-hour short course, "A Review of the Dynamics Paragraphs of API 617 for Centrifugal Compressors." He was on the API subcommittee that wrote the dynamics paragraphs of the Seventh Edition of API 617, and on the task force that wrote the Second Edition of API 684, "API Standard Paragraphs Rotordynamic Tutorial." He belongs to the ASME, the Vibration Institute, the CMVA, and the MAA. He received an A.B. from Hamilton College (Phi Beta Kappa), an A.M. from Brown University, and a Ph.D. from Syracuse University, all in the field of mathematics. ■