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ROTORDYNAMIC ANALYSIS

FOUR PAD TILTING PAD JOURNAL BEARING DESIGN AND APPLICATION FOR MULTISTAGE AXIAL COMPRESSORS

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Four Pad Tilting Pad Bearing Design and Application for Multistage Axial Compressors

The advantages of operating multistage axial compressors on 4 pad tilting pad bearings are discussed and compared to other fixed bore and tilting pad bearing designs. These advantages include operation free of subsynchronous vibration and, with between pivot loading, placement of peak response speeds well outside of the operating speed range. Examples of analytical design studies of 3 actual rotor systems are presented and discussed to illustrate the design recommendations. Test stand results are also included for 3 axial compressors to help substantiate the analytical results.

Introduction

The selection and design of fluid film bearings for turbomachinery has become extremely important for both the original equipment manufacturer (OEM) and end-users. The OEM is concerned with the expedient testing, shipment and successful field operation of their product. The end-user has typically a product that is dependent on the satisfactory operation of the turbomachinery to either drive the process equipment or to compress, pump or otherwise perform its given process function.

Improvements in monitoring vibration levels have enabled the OEM and end-user to collect data on a large number of machines and thus to establish acceptance criteria for rotor synchronous and nonsynchronous vibrations. The occurrence of high synchronous response is usually the result of poor or improper balance or rotor build procedures but can also be traced to marginal bearing design in regards to effective system damping levels. Furthermore, marginal bearing design can be the cause of subsynchronous vibration either directly by bearing induced oil whirl [1-3] or indirectly by providing insufficient damping to counteract the destabilizing excitations of the rotor system [4-6]. These excitation mechanisms include steam whirl, aerodynamic excitation from turborotor stages or labyrinths, internal friction, entrapped fluids, piping resonance, acoustic standing wave phenomena in associated piping, loose shaft elements, loose bearing inserts or housings, aerodynamic excitation arising from turbulence or vortex shedding from piping or turborotor stages, and operation in a region close to or in a condition of surge [7-11]. Additional acceptance criteria prohibit peak responses near the machine operating range.

Calculation procedures and design guides for bearings are currently being developed to the point that satisfactory predictions can be made in the original design stages of new and/or improved rotor-bearing systems. This capability has rapidly increased the interest of the end-user in the analytical prediction of response sensitivity, peak response placement and stability in the early stages of new turbomachinery design. Rigid specifications have thus been prepared by most OEM's in regard to acceptable rotor dynamics behavior and equally strict specifications are appearing in current new orders and re-rate studies. Proper bearing design is essential to satisfy these specifications.

This paper considers the design and application of a 4 pad tilting pad bearing for heavy, low speed center-hung axial compressors whose shaft stiffness is in the same range as the bearing stiffness. This type of machine operates well below the first bending critical. However, the operating range is usually between the first and second rigid body criticals (cylindrical and conical modes). These modes are sometimes referred to as bearing criticals since the bearing stiffness and damping properties have a large influence on the frequency and amplification of the resonance speeds. Furthermore, if the bearing properties are asymmetric, distinct peak amplifications may be observed at different frequencies whose mode shapes correspond very nearly to the cylindrical mode. That is, the first rigid body mode may be observed twice due to the bearings' asymmetric properties. Clearly, since the bearings control these rigid body modes, bearing design is crucial in determining the performance of this class of turbomachinery.

The major advantage of the 4 pad tilting pad bearing is the symmetric stiffness and damping properties that result when loaded between pivots [4, 12]. With symmetric dynamic characteristics, the split first mode discussed above no longer exists. By removing a peak response from near the operating range, it is much easier to meet peak response placement critieria.

Although the speeds are relatively low for axial compressors, oil whirl may still be a problem with fixed lobe bearings. Additionally, at certain stator settings, strong

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Table 1 Summary of axial compressor characteristics

Machine Number	W_r		N	N_{cr}	L_b		K_s x 10^{-6}		$\bar{K}$*
	(lbs)	(N)	(RPM)	(RPM)	(in)	(cm)	(lb/in)	(N/cm)	
1	4739	21079	5500	6000	97	246	2.4	4.2	2.3
2	16210	72102	3600	4780	134	340	5.3	9.3	1.7
3	7118	31661	5500	7500	87	221	5.7	10.0	0.68

* At operating speed using average bearing stiffnesses

aerodynamic excitation forces may be imparted on the rotor. It will be shown that these difficulties may be overcome with the 4 pad bearing design.

The theoretical analysis considers 3 different axial compressors that range in speed from 3,600 to 5,500 rpm and in weight from 72,102 N (16,210 lb) to 21,079 N (4739 lb). Synchronous response and stability characteristics are compared for different bearing designs. These designs include a 3 axial groove bearing [13], a pressure dam or step journal bearing [14] and tilt pad bearings with 4 and 5 pads. The advantages of the 4 pad bearing will be demonstrated and some design guidelines discussed. Finally, test stand results are presented for axial compressors operating on both fixed lobe and 4 pad tilting pad bearings.

Theoretical Analysis

Table 1 summarizes the characteristics of the 3 axial compressors used in the analysis. The parameters for the different bearing designs considered are listed in Table 2. Speed dependent stiffness and damping characteristics are calculated as in reference [14] for the fixed lobe bearings and references [12, 15] for the tilting pad bearings. Synchronous characteristics are used for the tilt pad designs in the stability calculations. Also for stability, a transfer matrix approach similar to the method outlined in reference [16] is employed.

Figure 1 is a stability map for the lightest axial (W_r = 21,079 N, 4739 lb), machine number 1. This compressor has a journal diameter of 13.97 cm (5.5 in.) and an operating speed of 5500 rpm. All destabilizing forces are lumped into one parameter, aerodynamic cross-coupling, Q and placed at the rotor midspan location. This parameter is plotted against the real part of the eigenvalue or the growth factor. The stability threshold is indicated at a growth factor of zero while positive values are unstable.

Three bearings are considered: 3 axial groove, step journal and 4 pad tilting pad. The 3 axial groove design is predicted to be either on the stabiity threshold or unstable depending on the amount of destabilizing aerodynamic forces present in the compressor. The step bearing design, however, places the compressor into the stable region with a real root of -22 compared to $+4$ for the 3 axial groove bearing ($Q = 1.75 \times 10^4$ N/cm, 10^4 lb/in.). The 4 pad tilt pad bearing design with between pad loading further increases the compressor's stability margin. Two curves are shown in Fig. 1 indicating the extreme tolerance range on the bearing clearance and preload (Fig. 2).[1] Since damping increases with decreasing preload and/or decreasing bearing clearances, c_b, the M = 0.3, c_b = 0.0762 mm (3.0 mils) bearing provides a larger margin of stability compared to the $M = 0.5$, c_b = 0.127 mm (5.0 mils) 4 pad design. The compressor goes unstable with all 3 bearing designs for extremely large values of Q.

Table 2 indicates that for machine number 1, the length to diameter ratio of the 4 pad tilt pad design is less than the L/D ratio for the fixed lobe cases (0.45 compared to 0.55). A decrease in bearing length is necessary for tilt pad bearings if they are to fit into the same housing due to the additional space needed for the shoe retainers and end seals. Figure 3 indicates that as L/D increases for the 4 pad tilting pad bearing, load between pads, damping increases and stiffness decreases. Thus the largest L/D ratio should be used in this case.

The second axial listed in Table 1 is the heaviest of three compressors weighing 72,102 N (16,210 lb) with a 20.32 cm (8.0 in.) journal. Also, the operating speed is lower at 3600 rpm. The operating Sommerfeld number is 0.24 which is much lower than 0.75 and 0.71 for machine numbers 1 and 3, respectively. The stability map for this axial is shown in Fig. 4. Similar results are depicted compared to Fig. 1. Again, the 3 axial groove bearing operates on the stability threshold while the step journal design moves the margin of stability into the stable region. The 4 pad tilting pad bearing further increases the stability margin. Two different preloads are

[1] In Figs. 2, 3, and 5 the same vertical axis is used for both stiffness and damping. Appropriate units and exponential multipliers are indicated on each axis. Proper curve identification is shown inside the axes.

Nomenclature

c = bearing radial clearance (L)

c_b = $R_v - R$, tilt pad bearing assembled radial clearance in line with a pivot (L)

c_p = $R_p - R$, pad radial clearance (L)

C_{xx}, C_{yy} = bearing principle damping coefficients (FTL^{-1})

D = $2R$, journal diameter (L)

h_1 = step journal bearing pocket clearance (L)

K_{xx}, K_{yy} = bearing principle stiffness coefficients (FL^{-1})

$\bar{K}$ = $(K_{xx} + K_{yy})/K_s$, stiffness ratio

K_s = $m_m\,\omega_{cr}^2$, shaft stiffness (FL^{-1})

K' = h_1/c, step journal bearing film thickness ratio

L = bearing axial length (L)

L_b = bearing span (L)

M = $1-(c_b/c_p)$, tilt pad bearing preload

m_m = modal mass (FT^2L^{-1})

N, N_s = shaft rotational speed (rpm), (rps)

N_{cr} = first rigid bearing critical speed (rpm)

Q = aerodynamic cross coupling (FL^{-1})

R = journal radius (L)

R_v = radius from bearing center to pivot, tilt pad bearings (L)

R_p = pad radius of curvature (L)

Re = $\dfrac{\rho V c}{\mu}$, bearing Reynolds number ($c = c_b$, tilt pad bearings)

S = $\dfrac{\mu N_s L D}{W}\left(\dfrac{R}{c}\right)^2$, Sommerfeld number ($c = c_b$, tilt pad bearings)

V = $R\omega$, velocity of journal (LT^{-1})

W = bearing external load (F)

W_r = total rotor weight (F)

X, Y = coordinate system for rotating journal in bearing housing

θ_s = location of step measured with rotation from positive X-axis (degrees)

μ = average fluid viscosity (FTL^{-2})

ρ = fluid density (FT^2L^{-4})

ω_{cr} = first rigid bearing critical speed (T^{-1})

Table 2 Summary of bearing parameters for the three axial compressors

Machine Number	Bearing Type	D (in)	D (cm)	L/D	c, c_b (mils)	c, c_b (mm)	S*	R_e*	Step Journal K'	Step Journal θ_s	Tilting Pad M	Tilting Pad Loading
1	3 Axial Groove	5.5	13.97	.55	3.0	.0762	.75	272				
	Step Journal	↓	↓	.55	3.0	.0762	↓	↓	4.7	120°		
	4 Pad Tilt	↓	↓	.45	3.0	.0762	↓	↓			.3	Between
	4 Pad Tilt	↓	↓	.45	5.0	.127	↓	↓			.5	Between
2	3 Axial Groove	8.0	20.32	.78	5.5	.1397	.24	442				
	Step Journal	↓	↓	.78	5.5	.1397	↓	442	4.1	120°		
	4 Pad Tilt	↓	↓	.70	6.0	.1524	↓	482			.2	Between
	4 Pad Tilt	↓	↓	.70	6.0	.1524	↓	482			.45	Between
3	3 Axial Groove	6.5	16.51	.77	4.5	.1143	.71	766				
	Step Journal	↓	↓	.77	↓	↓	↓	↓	4.6	120°		
	4 Pad Tilt	↓	↓	.60	↓	↓	↓	↓			.5	Between
	5 Pad Tilt	↓	↓	.60	↓	↓	↓	↓			.5	Between
	5 Pad Tilt	↓	↓	.60	↓	↓	↓	↓			.5	On

* At Operating Speed

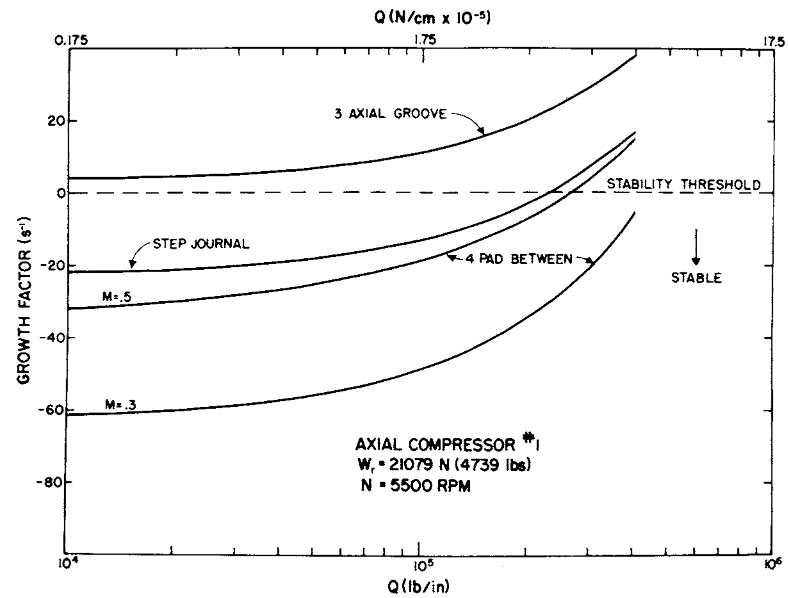

Fig. 1 Stability map, axial compressor number 1

shown, $M = 0.2$ and $M = 0.45$. The lowest preload provides the largest stability margin due to the increase in bearing damping.

Figure 5 shows that for $S = 0.24$ and increasing L/D ratios, stiffness decreases while damping increases. The increase in damping is not as pronounced as in Fig. 3 ($S = 0.75$). However, the decrease in stiffness is still beneficial from a stability standpoint since softer bearings allow more bearing damping to combat the destabilizing aerodynamics inboard of the bearings [5]. Again, the largest L/D ratio should be chosen. Additionally, since this compressor is very heavy, concern for large bearing loads becomes important and the largest pad length possible is also advantageous from this viewpoint.

Theoretical response plots for the compressor supported by 3 axial groove and 4 pad tilting pad bearings are shown in Fig. 6. The curves are for the discharge end probe located 3 inches outboard of the bearing centerline. The response is relative to the support and 18 ounce-inches of unbalance is used to excite the compressor. The 4 pad bearing is extremely effective in reducing the synchronous vibration level at operating speed and in fact over the entire speed range shown.

The 3 axial groove response shows 2 peaks at 2000 rpm and 3500 rpm. Close examination of the mode shapes and the machine critical speed map indicates that both peak responses are first mode excitations due to the asymmetry in the 3 axial groove bearings. The 4 pad tilt pad bearing with between pad loading is symmetric and results in only one peak in the operating range at 2800 rpm. The response plot for the step journal bearing is not shown but is very similar to the 3 axial groove case with peak responses at 2000 and 3500 rpm.

Machine number 3 weighs 31,661 N (7118 lb) and has a journal diameter of 16.51 cm (6.5 in.). Figure 7 is a stability plot for this compressor. The stability analysis predicts that the axial may operate unstably with 3 axial groove bearings. The step bearing design again places the compressor into the stable region. Three different types of tilting pad designs are also considered: 4 pads with load between pivots and 5 pads with load on and between pivots. All tilt pad designs provide a large stability margin (growth factor between -50 and -60 for $Q = 1.75 \times 10^4$ N/cm, 10^4 lb/in.). The 5 pad between is slightly superior to the 5 pad on. The 4 pad between design provides a slightly larger stability margin for low values of Q but becomes less stable for larger Q values compared to the 5 pad designs. This is due to the lack of stiffness asymmetry with the 4 pad between bearing ($K_{xx} = K_{yy}$). Stiffness

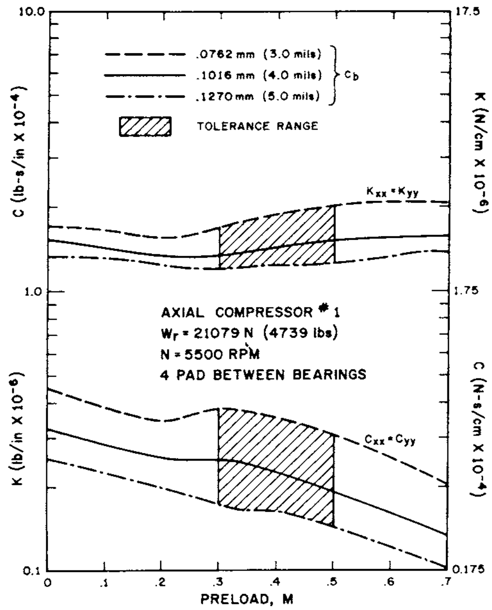

Fig. 2 Variation in stiffness and damping with preload and bearing clearance, 4 pad bearing, D = 13.97 cm (5.5 in.)

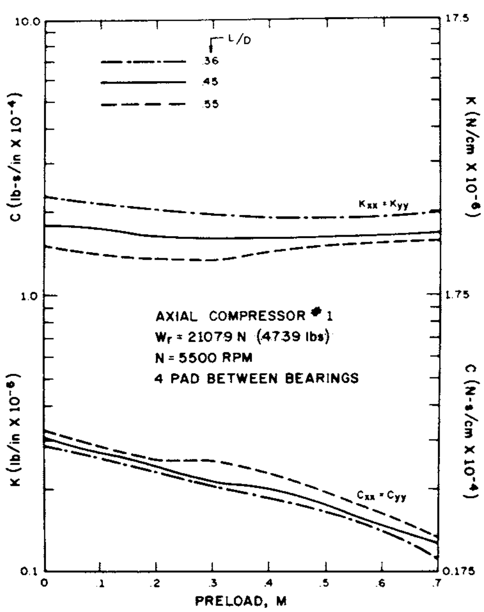

Fig. 3 Variation in stiffness and damping with preload and length to diameter ratio, 4 pad bearing, D = 13.97 cm (5.5 in.)

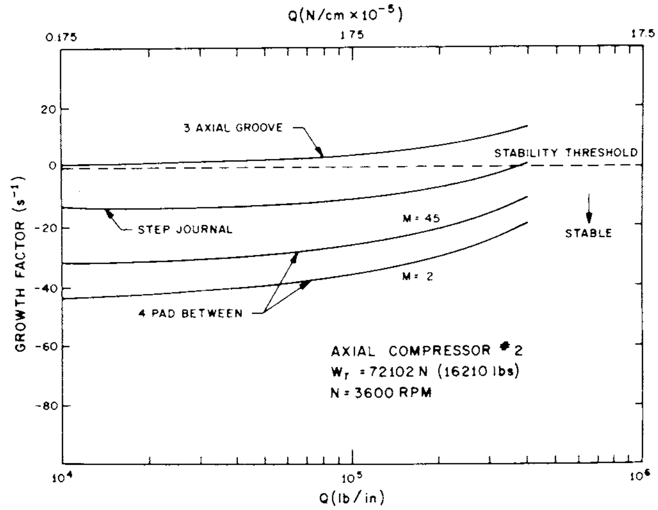

Fig. 4 Stability map, axial compressor number 2

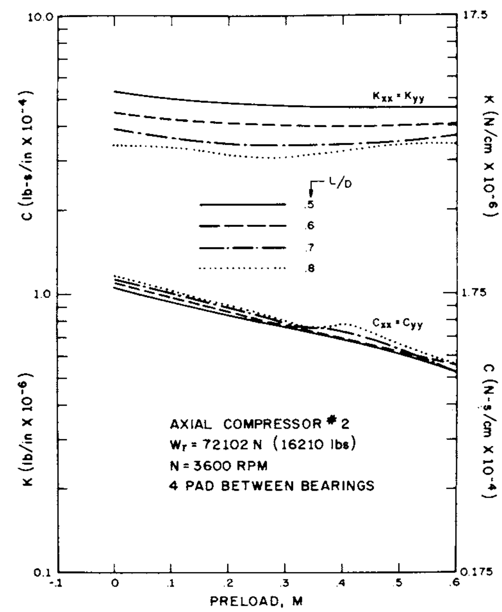

Fig. 5 Variation in stiffness and damping with preload and length to diameter ratio, 4 pad bearing, D = 20.32 cm (8.0 in.)

asymmetry is favorable to machine stability [4, 5, 9, 10].

Figure 8 compares the theoretical response curves for the compressor supported by the 3 axial groove, the step journal and the 4 pad between bearing design. The response, relative to the support, is for the bearing centerline location with 5 ounce-inches of unbalance. Figure 9 illustrates the absolute response at the compressor's center. Note that for the fixed lobe designs, a peak response is predicted to be just above 5000 rpm, very close to the operating speed of 5500 rpm. Another peak response is indicated at just below 2500 rpm. Again, the 4 pad bearing is effective in removing the peak response out of the operating range. Additionally, the sensitivity of the compressor is reduced with the 4 pad design over the fixed lobe cases.

All tilt pad designs are compared to the 3 axial groove bearing for machine number 3 in Fig. 10. The 5 pad designs increase the compressor's sensitivity at the unbalance criticals over the 3 axial groove case. The 5 pad on bearing being asymmetric shows a well damped peak response just below 3000 rpm as well as a larger peak at 4200 rpm. In fact, this 4200 rpm critical has the largest amplification of all bearing designs considered. The 5 pad between bearing shows only one peak response in the operating range since it is only slightly asymmetric. However, its peak is larger than the 4 pad between case.

Test Stand Results

This section presents test stand results for axial compressors similar to machine number 2. A response plot for a compressor operating on 3 axial groove bearings is shown in Fig. 11. From Fig. 6, peak responses are predicted at 2000 and 3600 rpm for this compressor. While the 2000 rpm critical does not amplify, Fig. 11 shows a peak at 3750 rpm which is well within the machine operating range.

Frequency scans are shown in Fig. 12 for the compressor of Fig. 11 with 3 axial groove bearings. Four scans corresponding to the 2 sets of probes are displayed at 3780 rpm. An extremely small 40 Hz (2400 cpm) component is evident for the inlet-horizontal probe. Scans at higher speeds

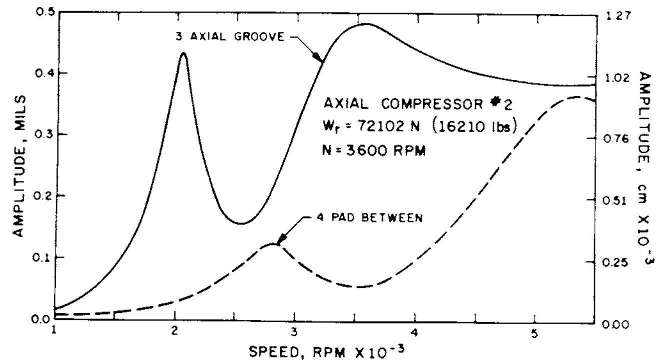

Fig. 6 Relative peak to peak response, probe location, axial compressor number 2, 3 axial groove and 4 pad bearings

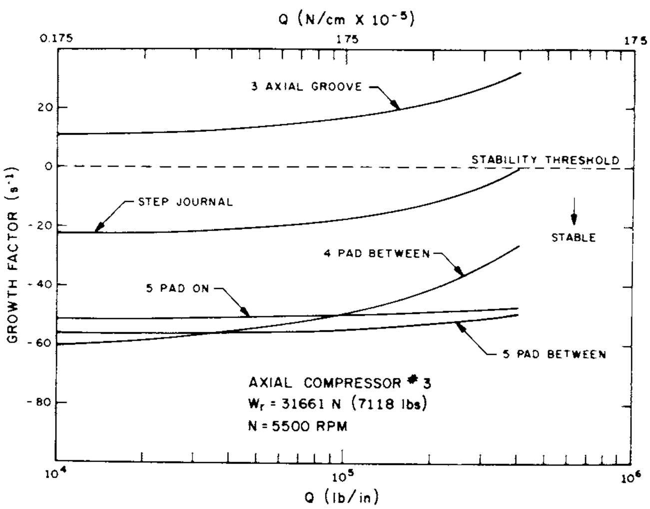

Fig. 7 Stability map, axial compressor number 3

were not recorded. A full rotor-bearing stability analysis for this compressor predicts a damped natural frequency of 2340 cpm (39 Hz).

Figure 13 illustrates a response plot for the same axial discussed in Figs. 11 and 12 but with step journal bearings. The filtered signal indicates a critical at 3500 rpm. The corresponding frequency spectrum is presented in Fig. 14 from 3500 to 4200 rpm. A small 33 Hz (1980 cpm) subsynchronous component is evident at trip speed (4200 rpm) at this particular stator setting. This component, bounded and within customer acceptance criteria, is a reexcitation of the first fundamental natural frequency (2000 rpm, Fig. 6) caused by a combination of oil whirl and aerodynamic excitation. The whirl ratio is 47 percent. The stability analysis predicts a damped natural frequency of 2360 cpm (39.3 Hz) which gives a whirl ratio of 56 percent.

A peak hold plot during an accel is shown in Fig. 15 for an axial on four pad tilting pad bearings. The general flow of the plot matches the analytical curve of Fig. 6 with a peak near the predicted value of 2800 rpm.

Figure 16 illustrates a frequency scan at 4100 and 4380 rpm for the same axial. Note the absence of a strong subsynchronous component. However, a small 45 Hz (2700 cpm) component is evident that corresponds to the predicted 2800 rpm first peak response speed.

Conclusions

Three axial groove and step journal bearings have asymmetric stiffness and damping properties and therefore produce a split or double first critical speed. For axial

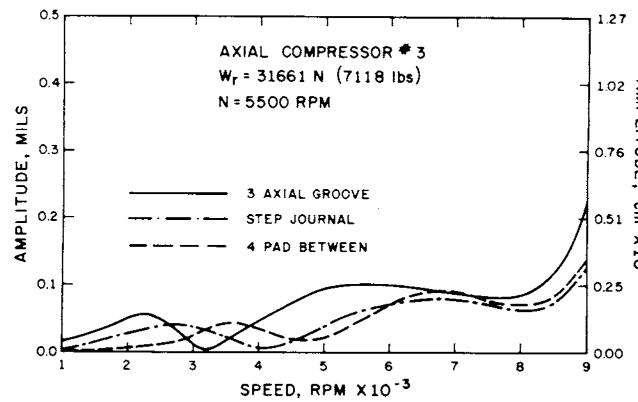

Fig. 8 Relative peak to peak response, bearing centerline, axial compressor number 3, 3 axial groove, step journal, and 4 pad bearings

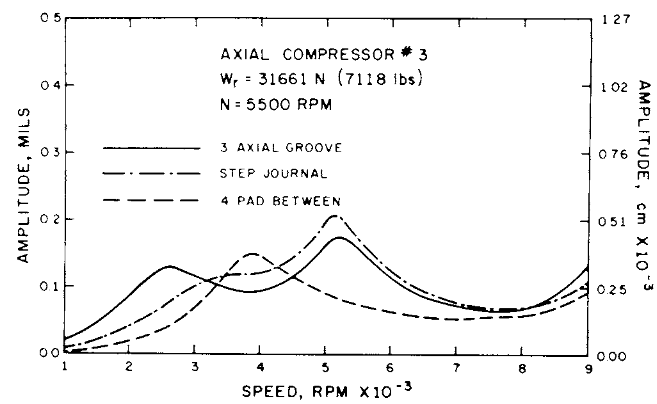

Fig. 9 Absolute peak to peak response, compressor center, axial compressor number 3, 3 axial groove, step journal, and 4 pad bearings

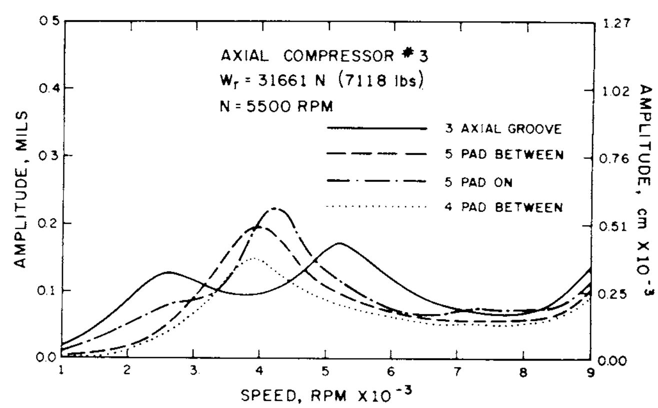

Fig. 10 Absolute peak to peak response, compressor center, axial compressor number 3, 3 axial groove, 4 and 5 pad bearings

compressors and other similar machines that operate between the first and second rigid body modes, this double peak makes it more difficult to meet peak response placement criteria.

Additionally, an axial compressor operating on 3 axial groove bearings may be subject to oil whirl and/or aerodynamic induced instabilities. A step journal bearing improves the stability characteristics of the compressor but, as seen from the test stand results, may still produce a small, bounded subsynchronous component at high speeds (trip speed) and at certain stator settings. Furthermore, the split first critical remains due to the bearings' asymmetric dynamic properties.

The 4 pad tilting pad bearing, however, produces a single first peak response due to its symmetric characteristics when loaded between pivots. Consequently, it becomes much easier

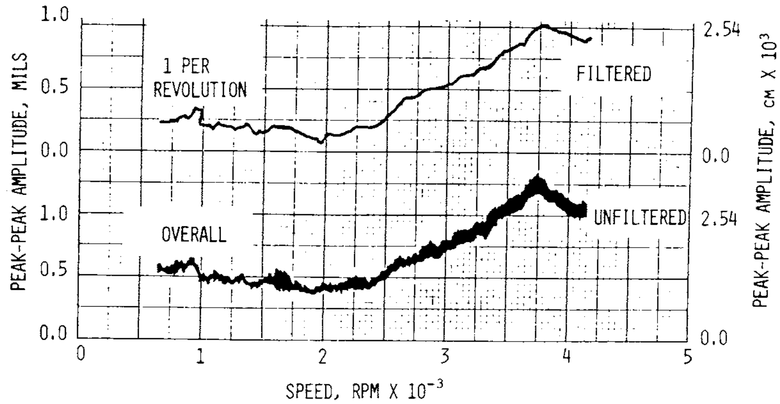

Fig. 11 Axial compressor speed-amplitude plot, 3 axial groove bearings

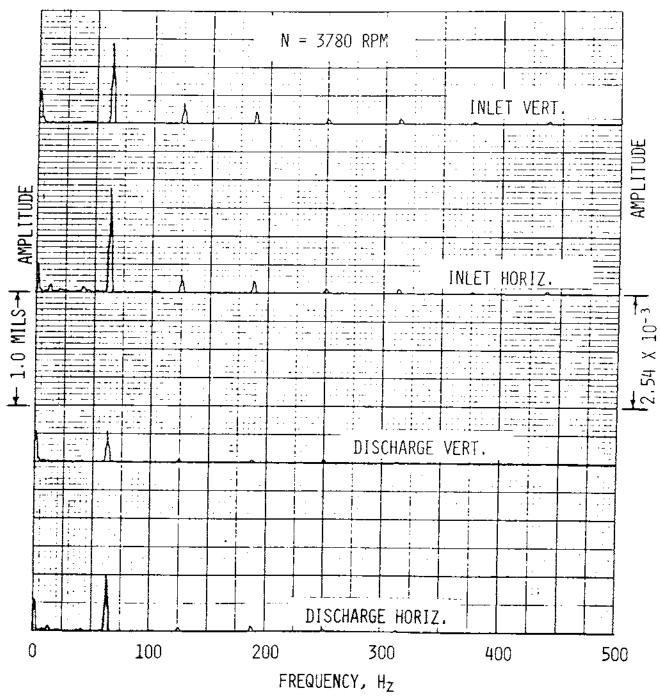

Fig. 12 Axial compressor frequency spectrum, 3 axial groove bearings

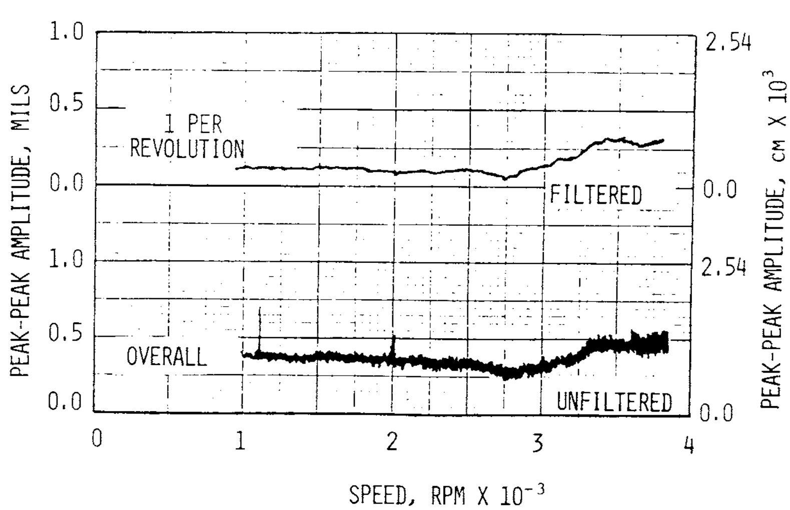

Fig. 13 Axial compressor speed-amplitude plot, step journal bearings

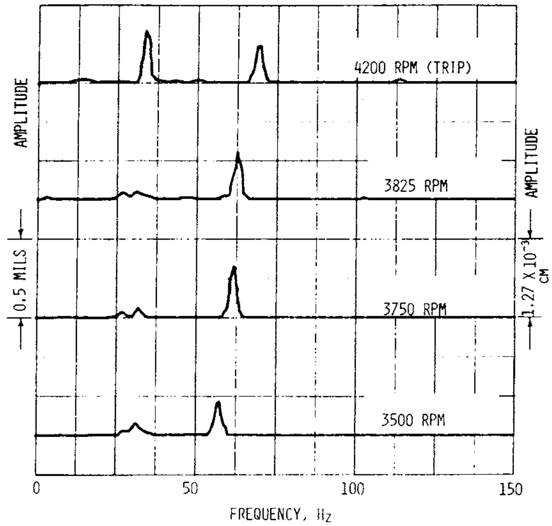

Fig. 14 Axial compressor frequency spectrum, step journal bearings

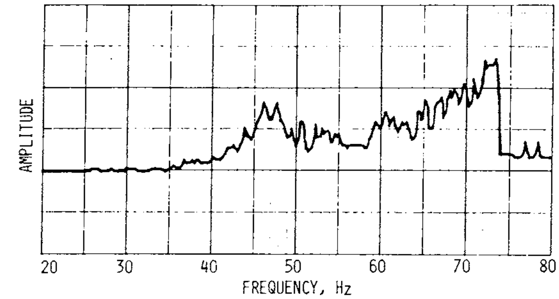

Fig. 15 Axial compressor peak hold plot, 4 pad tilting pad bearings

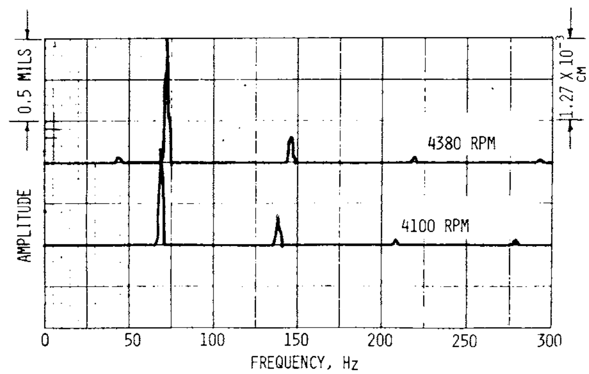

Fig. 16 Axial compressor frequency spectrum, 4 pad tilting pad bearings

to satisfy peak response acceptance criteria for axial compressors operating on 4 pad bearings. Other advantages of the 4 pad between pivot loaded bearing include excellent load carrying capacity, reduced synchronous vibration levels and stable operation.

Increasing the pad length provides increased damping with reduced stiffness for the 4 pad tilting pad bearing in the moderate Sommerfeld number range (between $S = 0.1$ and 1.0). Thus, 4 pad bearings should be designed with the longest pad length possible for favorable synchronous and subsynchronous vibration characteristics. The long pad length design also provides good load carrying capacity. Decreasing preload also increases damping.

The 5 pad bearings with on and between pivot loading also provide a wide stability margin. Futhermore, being only slightly asymmetric, they are effective in eliminating the split first critical for the theoretical analysis of axial compressor number 3. However, the resulting peak response vibration level is the largest compared to all other designs considered. This high response at the resonance speed for 5 pad bearings has also been observed experimentally in reference [17].

Test stand results are in good agreement with the theoretical analyses. Peak responses fall very close to the predicted values. Also, the frequency spectrum plots show an instability frequency at the predicted first fundamental natural frequency. For the cases presented, the 4 pad tilting pad bearing is successful in removing the peak response from the operating speed range. Furthermore, essentially no subsynchronous vibration is evident at trip speed.

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R. D. Flack[2]

Drs. Nicholas and Kirk are to be commended for their excellent paper. The paper represents very well how modern theoretical techniques can be used in the design of new turbomachines or retrofit machines in the field that may encounter difficulties. The information that they present on the effect of some bearing geometries on both the stability and the criticial speed unbalance response is certainly a welcomed contribution to the literature. Much of the previous literature has dealt only with the stability of rotor bearing systems when different bearing geometries were compared.

At the verbal presentation of this paper the authors presented results for two additional axial compressors which were fitted with tilting pad bearings. These results further demonstrated the predictions presented in the paper. The present comment is in the form of a request that the authors consider sharing these results in their response to this comment.

Anant Pal Singh[3]

The authors, with the aid of computerized calculation procedures for the rotor-bearing system dynamics and some test data, have presented results and conclusions which may be useful. Questions and remarks concerning this paper are as follows:

(1) The calculations of Sommerfeld number S* at operating speeds for the same journal diameter but different L/D ratio for the considered bearings cannot be identical as shown in Table 2 for three machines.

(2) The use of the largest possible L/D ratio for the tilting pads as recommended by the authors in this paper is not a normal practice and may lead to nonoptimum design. One must also consider the effect of increased L/D ratio on frictional horsepower loss, the rate of oil side-flow and space considerations.

(3) It is difficult to compare and reach any definite conclusions for the comparative performance of 4 pad and 5 pad designs with the limited results observed in Figs. 7 and 10. No test stand results have been given or discussed for the 5 pad design.

(4) The effects of bearing asymmetry ratio (Kxx/Kyy) on the stability characteristics of any tilting pad journal bearing as reported by references [5] and [10], and the additional reference below, contradict the conclusions drawn in this paper.

Additional Reference

18 Nicholas, J. C., Gunter, E. J., and Barrett, L. E., "Tilt Pad Bearing Design," Report No. UVA/464761/MAE 78/145, University of Virginia, Charlottesville, Va., Mar. 1978.

Authors' Closure

The authors would like to thank the discussers for their interest in the paper. In response to Dr. Flack's comments, Figs. 17 through 21 illustrate the vibration response of 2 additional axial compressors similar to the compressors discussed in the Test Stand Results section. Both compressors are supported on the 4 pad tilting pad design discussed in the paper. Figure 17 shows the response plot for one compressor with an extremely low response level that is typical of a well balanced axial. The first peak response appears to be at 2900 rpm. Recall that the predicted first critical is 2800 rpm. The corresponding frequency spectrum at 3300 rpm is shown in Fig. 18. This axial operated at trip speed (4200 rpm) without a subsynchronous component.

Results from another axial on 4 pad bearings are plotted in Figs. 19 through 21. Figure 19 shows a test stand decel response at the 4 probe locations with the peak amplitude

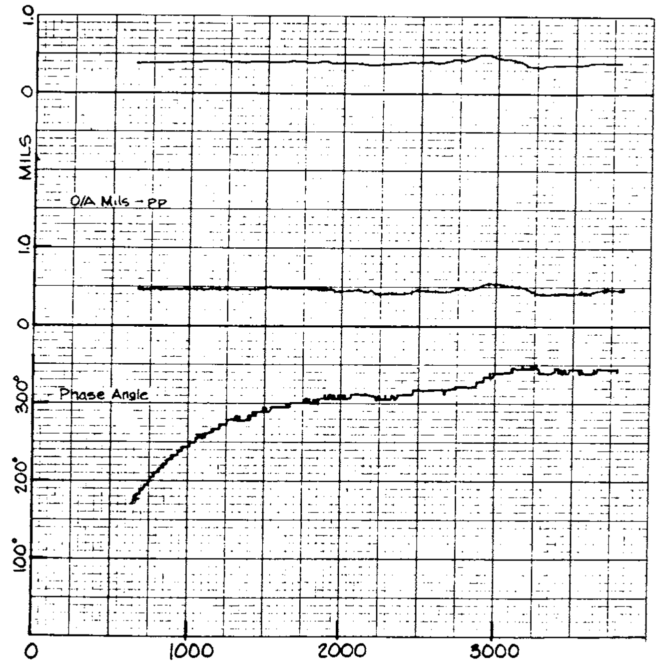

Fig. 17 Axial compressor speed-amplitude and phase angle plots, 4 pad tilting pad bearings

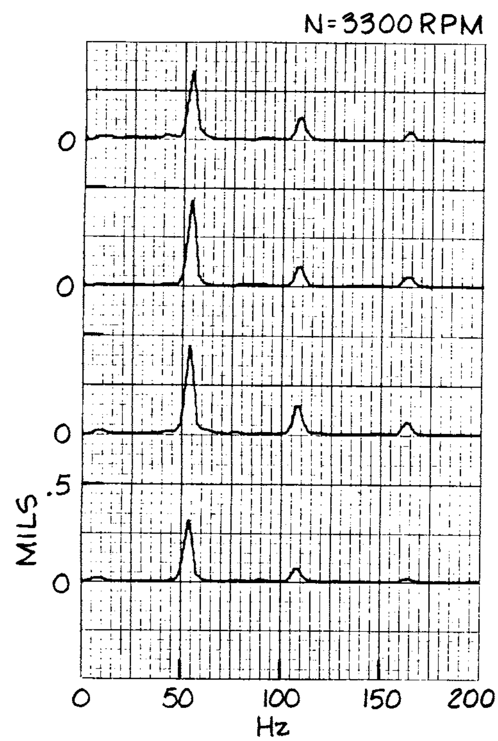

Fig. 18 Axial compressor frequency spectrum, 4 pad tilting pad bearings, N = 3300 rpm

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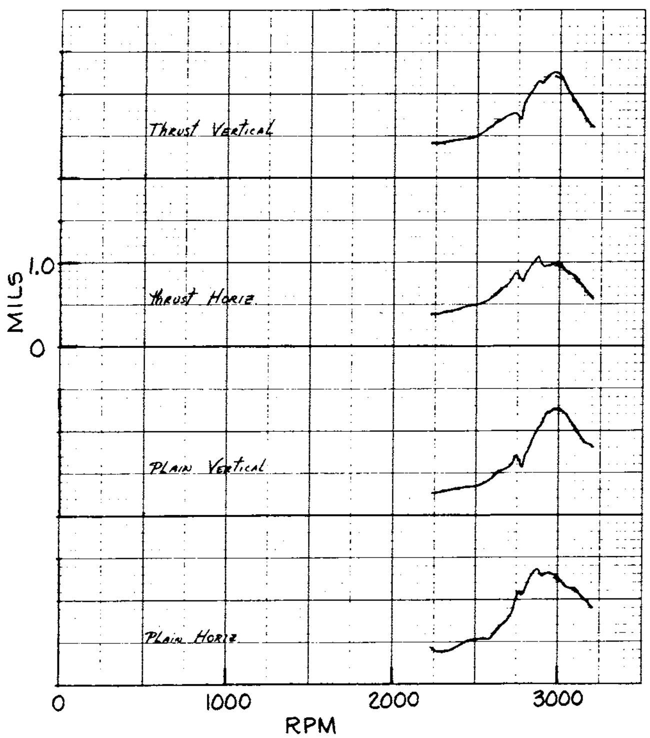

Fig. 19 Axial compressor speed-amplitude decel, 4 pad tilting pad bearings

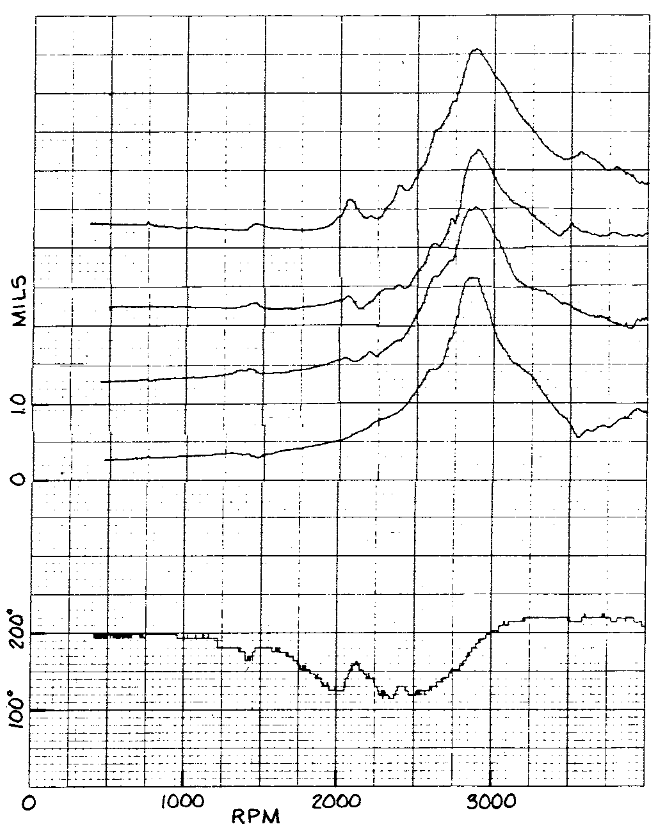

Fig. 20 Axial compressor speed-amplitude accel, 4 pad tilting pad bearings

located at between 2850 rpm and 3000 rpm. An accel for the same axial is shown in Fig. 20 with the first critical located at 2900 rpm. The large vibration amplitude was subsequently corrected by rebalancing. The corresponding frequency spectrum is illustrated in Fig. 21. Again, this compressor

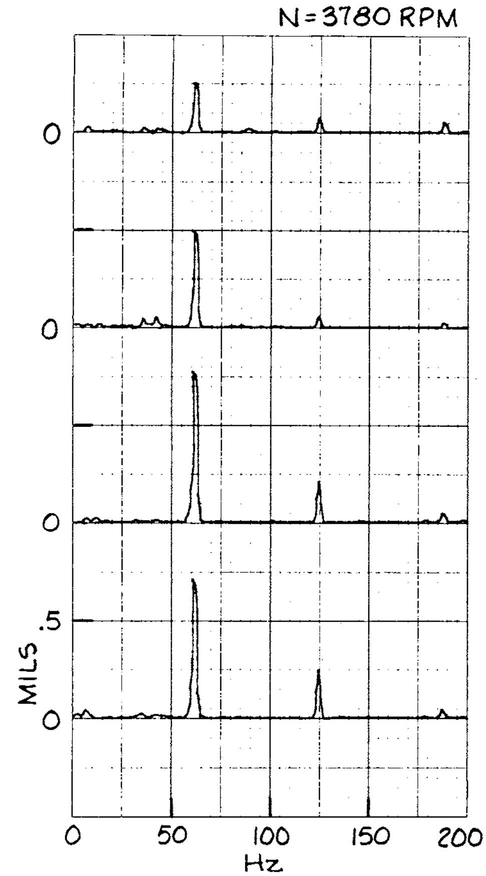

Fig. 21 Axial compressor frequency spectrum, 4 pad tilting pad bearings, N = 3780 rpm

operated at trip speed on the test stand free of subsynchronous vibration.

In response to the comments made by Mr. Singh, the Sommerfeld numbers indicted in Table 2 are in error for the tilting pad designs. However, they are correct for the fixed lobe bearings. Corrected Sommerfeld numbers for the tilt pad bearings are S = 0.63 and S = 0.23 for machine number 1, S = 0.18 for machine number 2 and S = 0.83 for machine number 3. The main point of the Sommerfeld number calculation is to show that the axial compressors analyzed here operate in the moderate Sommerfeld number range between S = 0.1 and S = 1.0. This fact remains valid.

The conclusions in this paper do not contradict the reported beneficial stabilizing effects of stiffness asymmetry from references [5, 10, 18] as stated by Mr. Singh. This point is addressed by the authors in the discussion of Fig. 7. As stated in the discussion, the 4 pad tilting pad bearing provides a slightly larger stability margin for low values of aerodynamic cross-coupling, Q, but becomes less stable for larger Q values compared to the 5 pad designs. This is due to the lack of stiffness asymmetry with the 4 pad between bearing with $K_{xx} = K_{yy}$. Furthermore, the 5 pad on bearing is slightly less stable than the 5 pad between bearing but the curves cross over for Q values greater than 5×10^5 lb/in. This is not shown in Fig. 7 as the plots stop at $Q = 4 \times 10^5$ lb/in. For stiffness asymmetry to be benficial, sufficient destabilizing excitations are necessary (i.e., large Q values for tilt pad bearings). For small Q values, stiffness asymmetry becomes less important and other parameters control the stability level such as shaft stiffness and bearing damping.

A major conclusion of this paper is that increasing pad length provides increased damping with reduced stiffness for the 4 pad bearing in the moderate Sommerfeld number range.

Mr. Singh's statement concerning this important design consideration improperly quotes and takes out of context the authors conclusions. The optimum bearing length is determined from overall analysis results, space limitations (addressed by the authors in the fourth paragraph of the Theoretical Analysis section) and practical design constraints. Pad geometries that result in square projected pad areas are preferred but aspect ratios of 0.5 to 2.0 are considered viable design alternatives. If vibration or critical speed placement are not potential problems, then the bearing designer will, without question, consider horsepower loss and oil flow. Otherwise, optimum effective damping is the most important bearing design concept.

Finally, Mr. Singh finds it difficult to compare the 4 pad to the 5 pad bearing. It is not a major contention of this paper to show all the advantages of a 4 pad design over a 5 pad design. For this application, the 4 pad design does show analytically a lower response level compared to a 5 pad design. This has been substantiated experimentally in reference [17] as stated in the conclusions. A peak response that existed in the operating speed range of an axial compressor had to be eliminated or moved. The authors analyzed the problem and decided to use a 4 pad bearing to obtain symmetric dynamic properties. The bearing was applied successfully to 3 axial compressors, removing the peak response from the operating range. Test stand results of a 5 pad bearing are not discussed since the 5 pad design was not considered beyond the design analysis study.

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