Table 4 Comparison of Analytical and Experimental Results

Test No.	Speed rpm N	External Load W_{ext}		Exp. shaft motion, Vert. (A1)		Horiz. (A2)		Min. film clear., Analytical		Experimental $h_{M_{exp}}$		Temperature Rise (Film) Analytical ΔT_F		Experimental ΔT_F	
		N	(lb)	μm	(mils)	μm	(mils)	μm	(mils)	μm	(mils)	°C	°F	°C	°F
L-4	5000	1,657	(370)	−63.8	(−2.51)	5.08	(0.2)	14.9	(.58)	12.7	(.50)	1.0	(2.1)	0	(0)
L-5	5000	2,038	(455)	−63.5	(−2.50)	3.30	(0.13)	12.8	(.50)	12.7	(.50)	1.18	(2.5)	0	(0)
L-6	5000	2,576	(575)	−65.8	(−2.59)	0	(0)	10.5	(.41)	10.5	(.41)	1.5	(3.1)	0	(0)
L-7	7500	2,576	(575)	- -	- -	- -	- -	16.0	(.63)	- -	- -	1.8	(3.8)	1.9	(4)
L-9	7500	3,181	(710)	−61.0	(−2.40)	16.8	(0.66)	13.7	(.54)	13.0	(.51)	2.0	(4.3)	1.9	(4)
L-10	7500	3,566	(796)	−63.5	(−2.50)	8.90	(0.35)	12.6	(.49)	12.2	(.48)	2.2	(4.6)	1.4	(3)
L-11	10000	2,576	(575)	−58.7	(−2.31)	33.0	(1.3)	21.8	(.85)	8.9	(.35)	2.5	(5.3)	2.8	(6)
L-14	10000	3,947	(881)	−54.7	(−2.15)	31.0	(1.22)	10.0	(.63)	13.5	(.53)	3.2	(6.9)	2.8	(6)
L-15	10000	5,318	(1187)	−55.4	(−2.18)	28.2	(1.11)	12.7	(.50)	14.2	(.56)	3.7	(8.0)	2.35	(5)

Film average density (ρ) = 9.836×10^2 Lg/m^3 (0.0355 lb/in^3)
Average bearing supply flow (Q_S) = 0.189 l/s (3 gpm)
Nominal water inlet temperature 57°C (135°F)

Comparisons were made between the analytical studies and the experimental data and are shown in Table 4. Temperature rise in the bearing film and minimum clearance were used for comparisons. The analytical temperature rise was based on the average film temperature difference between the leading and trailing edges of the pads. This was compared to the maximum temperature difference measured between the grooves of a pad. The temperature data compares favorably with the analytical predictions but are slightly higher generally. This means that reliable data can be obtained for flow and power loss using the design manual [1].

Film clearance comparisons are also illlustrated in Table 4. Some difficulty was encountered due to instrumentation drift and obtaining the actual zero reference. The assumption made here is the test L-6, which corresponds to the test that has the smallest clearance, would act most like a tilting pad. The minimum clearance was matched at that point. The correction for vertical motion (A1) data became 0.119 μm (0.468 mils) while the correction for the horizontal motion (A2) became 0.279 μm (1.1 mils). The vertical correction of 0.119 μm (0.468 mils) corresponds well with the 0.12 μm (0.5 mils) deflection of the elastomer when the shaft is in the rest position. The other assumption is that the pad overall deflection is small. This assumption is reasonable based on the data and the thermal expansion of the elastomer. The above assumptions are reasonable when comparing minimum film clearance but are not as valid when trying to determine shaft position and pad tilt. Based on these assumptions the comparison is very good with the exception of Test L-11, which shows the experimental value of the clearance to be smaller than the analytical, but also smaller than the more heavily loaded tests of L-14 and L-15.

In addition to the above testing, the bearing was also tested for certain off design conditions. This included starved condition, high water temperature, and dynamic unbalance loading. Under all test conditions the bearing exhibited good dynamic performance.

8 Conclusions

1. The journal bearing performed extremely well under all test conditions including some premature shutdowns due to test rig problems. These bearings were tested under off-design conditions including high temperature, insufficient supply flow (starved), relatively large steady state and dynamic loads.
2. Physical inspection of the bearings after testing indicated no measurable wear. The elastomer and carbon material maintained their original shape and properties.
3. Shaft lift-off occurs between the speeds of 200 and 400 rpm. This low lift-off speed should minimize bearing wear.
4. The journal bearing design criteria (Section 3) appear to be adequate to assure a good design.
5. Temperature rise comparisons between the testing and the analysis were reasonable. Power loss calculations for the bearings should therefore be reasonable.
6. Clearance comparisons between experiment and the analysis for the journal bearing are acceptable. Trends for the most part were consistent.
7. The larger the steady state load, the better the dynamic performance.
8. Compliant-mounted bearing offers the following advantage over a conventional tilting pad bearing:

- avoids fluttering of unloaded pad
- can absorb shock loading
- good tolerance to dirt ingestion
- good start and stop characteristics
- low probability of rotor destruction due to bearing seizure.

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Glenn K. Rightmire[1]

This paper presents a rather appealing concept of a hydrodynamic bearing. In the opinion of the discusser, it has interesting content, yet it conciseness prompts a number of comments and questions.

First of all, the bearing was designed to accomodate

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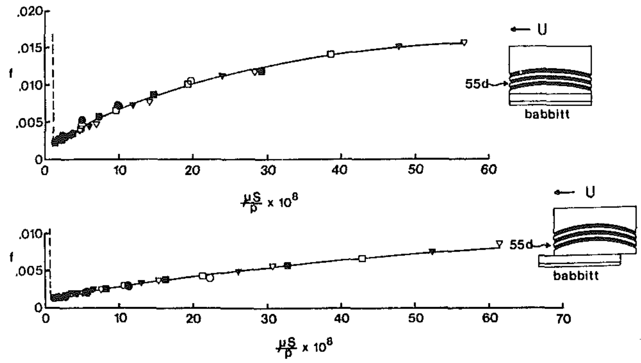

Fig. 7 Friction characteristics for the centered versus offset swing pad bearing

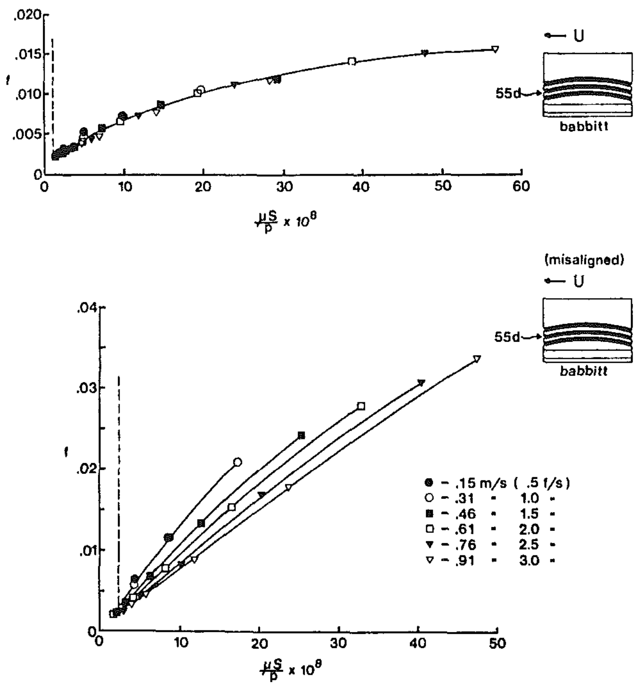

Fig. 8 Friction characteristics for the aligned versus misaligned swing pad bearing

rotation in either direction, so the "centrally pivoted" arrangement was the obvious way to implement it. However, the attendant frictional performance is poorer. Fig. 7, a plot included in a recent report on a swing pad study for the David Taylor R. & D. Center, bears this out. Here the shoe support is quite similar, although the undercut incorporated in the subject paper could act to improve those characteristics somewhat.

From the experience gained in the swing pad study, the discusser is skeptical about the pitch and roll accomodations claimed by the authors to be intrinsic in the rubber backed shoe – from the standpiont of friction. Figure 8 is offered to illustrate that concern.

The authors prescribe a load orientation, directed between the pads; this orientation does not give rise to optimum stiffenss. Is their specification based upon the optimization of other parameter?

Concerning the values obtained for minimum film thickness, the least favorable agreement between prediction and experimental results occurs at the lower loads and at the higher speeds. The poorer correlation in the former case seems to imply inadequate pitch accomodation through a lesser hydrodynamic moment, hence acting like a sticky pivot. How do the authors reconcile this fact with their assumption that the elastomer pitch stiffness can be considered to be negligible? Is its magnitude then small enough to preclude whirl? Moreover, are there explanations for the poorer agreement at higher speeds?

It is claimed in the paper that the bearing is tolerant to the ingestion of dirt. Could further justification of this claim be made?

Lastly, would the authors please address the following questions:

1. What were the physical properties of the elastomer employed in this study?

2. What influence are the time dependent properties of rubber expected to have on this type of bearing design?

3. What advantages does the elastomer backed bearing have over an equivalent spring mounted system?

Authors' Closure

The authors appreciate the effort that the discusser put into his comments on this paper. However, it is the opinion of the authors that the discusser totally missed the application that was described in the paper. More specifically the following comments are made:

The compliant mounted bearings are applied to high speed turbo-machinery in which the bearing fluid film is operating in a highly turbulent region. In this high speed operating region, where full lubrication is achieved the friction coefficient is not significantly effected by pivot location. This conclusion would also be true for the swing pad bearing, if the swing pad bearing were operating in this high speed environment.

If the rubber backing of the compliant mounted bearing is designed according to the specification described in Section 3.7 of this paper, the pitch and roll accomodations are clearly intrinsic. Since the discussion did not see this advantage in the swing pad bearing, this might be one of the advantages of a compliant mounted bearing over a swing pad bearing.

The load orientation (directed between pads) certainly does give rise to optimum stiffnesses. A three pad compliant mounted bearing that is unpreloaded has zero stiffness orthogonal to the applied load if the load orientation is directed into a pivot. Figure 2 of the paper can be used to illustrate this point. If the bearing is unpreloaded (an advantage of a compliant mounted bearing) and the load is directed into the pivot location of pad 1, then pads 2 and 3 would be unloaded and contribute nothing to the stiffness of the bearing. Since pad 1 only has stiffness in the direction of load, there would be no stiffness in the direction orthogonal to the load. This would clearly be an unstable operating situation.

The discusser's comments concerning the testing, where the comparison between experiment and analysis was poor, are well founded. If the compliant mounted bearing is not designed according to the criteria set in Section 3 of this paper then poor experimental comparison can result and possibly a poor bearing design. Indeed, it was the intention of the testing to examine some of the criteria and not to test an optimumly designed bearing.

Since the bearing pads can move radially and pitch they are considerably more tolerant to dirt, than the standard tilting pad or fixed geometry bearings.

A standard neoprene rubber was used for the tests.

The time dependent properties of rubber do not have a significant effect on the performance of the bearing during operation. There is a set that the rubber takes due to the static loading which should be taken into consideration when setting the clearance of the seals or blade in the particular turbomachine application.

The inherent damping of the elastomer makes it far more advantageous over an equivalent spring mounted system. Also the cost of manufacturing should be cheaper for an elastomer design as opposed to the spring design.