EXPERIMENTAL RESULTS CONCERNING CENTRIFUGAL IMPELLER EXCITATIONS

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INTRODUCTION

Based on what the authors have seen, the modern high performance centrifugal compressor is more likely to have problems with nonsynchronous rotordynamic instability than any other single class of machine. When designed for industrial application, these machines are typically multistage, with a rotor designed to operate at supercritical speeds, and supported on oil film bearings.

When such a compressor exhibits nonsynchronous whirl, the first troubleshooting efforts are usually aimed at modifying the oil film bearings. For example, a machine with cylindrical journal bearings may be modified to accept tilt-pad bearings, which have smaller cross-coupling coefficients (but also less damping).

The frequency with which these efforts fail suggests to the authors the existence of rotordynamic destabilizing excitations which are unique to centrifugal-type machines and which are not associated with the bearings.

EXPERIMENTAL OBSERVATIONS FROM A SIMPLE TEST RIG

To begin verification of this intuitive hypothesis, a simple test rig was set up in the Rotor Dynamics and Machinery Vibrations Laboratory at Texas A&M University. The objective was to observe the effect of working fluid, swirling in a housing, on the dynamics of an impeller with radial vanes.

Figure 1 shows the impeller and shaft removed from the rig, and figure 2 shows the impeller installed in the rig but without the fluid container. The impeller is supported vertically from a very flexible quill shaft in order to produce a low critical speed, and to allow the fluid dynamic effects on the impeller to predominate.

The shaft is supported from ball bearings, so that there is no possibility of "oil whip" from fluid film bearings as a destabilizing influence.

The impeller has been run both in the atmosphere (as in Figure 2), and submerged in working fluids contained in a cylindrical housing, open at the top. Variable speed is obtained with a DC gearmotor drive unit. The speed is measured with a proximity probe pulse tachometer and electronic digital counter.

The natural frequency of free vibrations was measured with a Selspot optical tracker, with the shaft displacement signal fed into a Nicolet 446 A frequency spectrum analyzer. The frequency spectrum shows the natural frequency in air to be 45 CPM, and the time trace digital memory feature of the Nicolet allowed a calculation of the logarithmic decrement which yielded $\delta = 0.0198$, or 0.315% of critical damping, a very lightly damped rotor. The damping comes from atmospheric drag, and from internal hysteresis in the shaft assembly. The latter is known to be destabilizing on forward whirl, if the internal damping is large enough relative to the external damping.

The first test observations from this rig were reported in reference [1]. Since that time, additional tests have been run with working fluids having different viscosity values, in order to investigate a hypothesis that viscous shear is the exciting force which drives the instability in this rig.

It was found that the threshold speeds of instability are sensitive to the fluid level in the container, so this was held constant for all tests reported here.

Rotating tests in the atmosphere (fluid container removed) showed the impeller to be stable up to a maximum speed allowed by the motor, except that the impeller executes a small backward triangular orbit (bounded) at speeds above 112 rpm. This shows that internal friction is not a significant destabilizing force in this rotor.

To investigate viscosity effects, two working fluids were used: (1) kerosene ($\mu \approx 1 \times 10^{-6}$ lb-sec/in^2) and (2) SAE 40 motor oil ($\mu \approx 100 \times 10^{-6}$ lb-sec/in^2).

With the impeller running in kerosene, the threshold speed of instability for forward whirl is 89 rpm, or twice the critical speed. Above this speed, the impeller executes forward whirl at its natural frequency with an amplitude that grows with time (until contact with the fluid container is made).

At speeds near the threshold speed, the whirling grows to a small amplitude and then dies out repetitively.

With SAE 40 oil, the threshold speed of instability for forward whirl is raised to at least 174 rpm (3.9 times the critical speed), which is the highest speed allowed by the greater horsepower requirements of the high viscosity fluid. (At this speed, with this fluid, the gearmotor draws over 100 amperes).

THEORETICAL CONSIDERATIONS

The test reported above shows that viscous forces are stabilizing rather than destabilizing in this apparatus. This rules out a model based on Reynolds' theory, such as treating the impeller as a large journal bearing or as a large thrust bearing.

It seems likely that an inviscid fluid analysis based on Bernoulli's theory would yield useful results if applied correctly. Visual observations indicate that the fluid swirls in a vortex centered around the container's geometric axis, regardless of impeller whirling.

There are several characteristics of centrifugal compressors which are not present in this test apparatus. Notably, they are: (1) high fluid pressures in the volute surrounding the impeller, (2) radial flow of the fluid, proportional to delivery rate, and (3) compressibility.

The last characteristic is minimal in some of the very high pressure machines which seem to be most subject to rotordynamic instability, since the fluids are practically incompressible at the working pressures encountered.

For an imcompressible fluid, variations in radial velocity can produce Coriolis forces on the impeller. Figure 3 shows a case in which radial velocity $\dot{r}$ of the impeller will slow down the fluid in the direction of $\dot{r}$, and allow an increase in fluid radial velocity on the opposite side of the impeller. The variation in fluid radial velocity around the impeller can be expressed as

$$\Delta V_R = \dot{r} \cos \theta \tag{1}$$

At each θ location there is a difference in tangential Coriolis force as compared to the $\theta + \pi$ location, due to ΔV_R. This force difference is

$$d(F_2 - F_1) = 4\rho\omega\Delta V_R \, (R_o - R_i)\left(\frac{R_o + R_i}{2}\right)ab \; d\theta \tag{2}$$

where ρ = fluid mass density

ω = rotational speed

R_o = impeller outside radius

R_i = impeller inside radius

a = impeller axial thickness

b = width between vanes

Substitution of equation (1) into equation (2) gives

$$d(F_2 - F_1) = 2\rho\omega(R_o^2 - R_i^2)b \; \dot{r} \cos \theta \; d\theta$$

which is normal to the vane located at θ, acting opposite to the direction of the vanes' tangential velocity for $-\pi/2 < \theta < \pi/2$, and in the same direction for $\pi/2 < \theta < 3\pi/2$.

The component normal to the whirl vector r, which is potentially destabilizing, is

$$dF_\phi = \cos\theta \; d(F_2 - F_1) \tag{3}$$

or

$$dF_\phi = 2\rho\omega(R_o^2 - R_i^2)ab\;\dot{r}\cos^2\theta\; d\theta$$

Integration around the impeller yields

$$F_\phi = 2\rho\omega(R_o^2 - R_i^2)ab\;\dot{r}\int_0^{2\pi}\cos^2\theta\; d\theta$$

or

$$F_\phi = 2\pi\rho\omega(R_o^2 - R_i^2)ab\;\dot{r}$$

which indicates a negative cross-coupled damping coefficient

$$C_{\phi r} = -2\pi\rho\omega(R_o^2 - R_i^2)ab \tag{4}$$

so that

$$F_\phi = -C_{\phi r}\dot{r}\ .$$

The effect of this coefficient should be to excite forward whirl, with an amplitude which would tend to grow and decay in a cyclic manner.

THE EFFECT OF LOAD

Many of the rotordynamic instabilities reported for centrifugal compressors are load dependent, or fluid pressure dependent. References [1] and [2] describe specific cases reported in the literature and/or observed by the authors.

Reference [3] defines a load-dependent cross-coupled stiffness coefficient which is currently used almost universally by rotordynamics consulting engineers for computerized stability analysis of all types of machines. A study of reference [3] reveals that the coefficient has real meaning only for axial flow machines.

References [2] and [4] describe a theory for load-dependent excitation in machines with very high torque loadings, and is applicable to centrifugal, as well as axial flow, compressors.

All of these theories are yet to be verified by experimental research or controlled tests in real machines.

CONCLUSIONS

The experimental observations reported here point strongly to the probability that centrifugal impellers produce rotordynamic destabilizing forces which are independent of the type of bearing used for rotor support.

A comprehensive and verified theory to explain and predict these forces is yet to be developed.

REFERENCES

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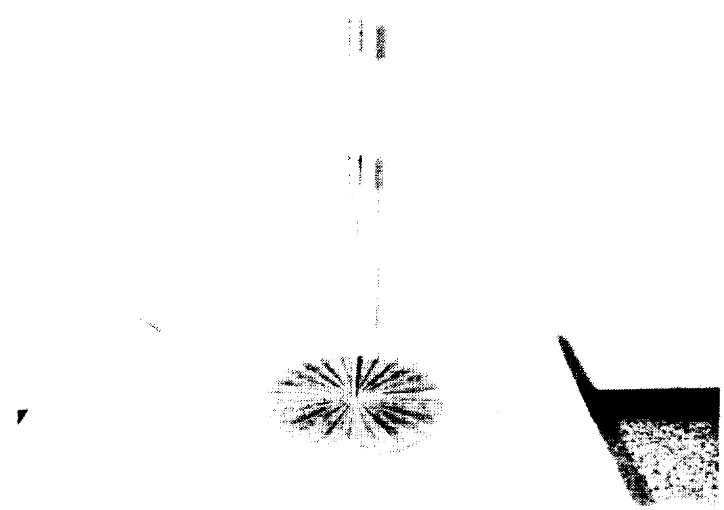

Figure 1. Test Rig Impeller and Shaft

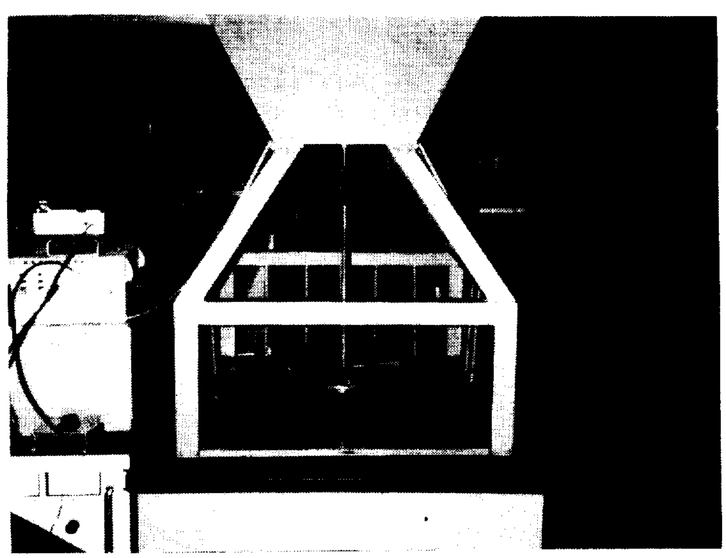

Figure 2. Assembled Test Rig Without Fluid Container

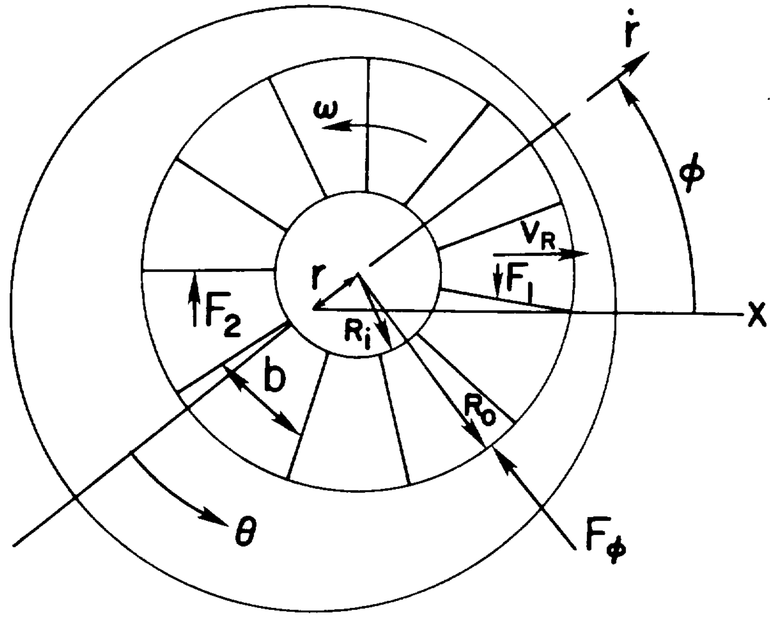

Figure 3. Coriolis forces generated by radial motion of impeller