PUMP INSTABILITY PHENOMENA GENERATED BY FLUID FORCES

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Rotor dynamic behavior of high energy centrifugal pumps is significantly affected by two types of fluid forces: one due to the hydraulic interaction of the impeller with the surrounding volute or diffuser and the other due to the effect of the wear rings. In this paper, the available data on these forces is first reviewed. A simple one degree of freedom system containing these forces is analytically solved to exhibit the rotor dynamic effects. To illustrate the relative magnitude of these phenomena, an example of a multistage boiler feed pump is worked out. It is shown that the wear ring effects tend to suppress critical speed and postpone instability onset. But the volute-impeller forces tend to lower the critical speed and the instability onset speed. However, for typical boiler feed pumps under normal running clearances, the wear ring effects are much more significant than the destabilizing hydraulic interaction effects.

SYMBOLS

b_2	impeller exit width
c	damping coefficient associated with wear rings
$\bar{c}$	$= c/\omega$
$c*$	critical damping
c_g	general damping coefficient
F	applied force
H	radial clearance
$H*$	nominal radial clearance
j	$\sqrt{-1}$
k	stiffness coefficient
k_s	shaft stiffness
$K*$	non dimensional stiffness coefficient
m	mass
m_h, m_h'	volute-impeller interaction mass coefficient
m_L	Lomakin mass
R	unbalance response amplitude
r_2	impeller tip radius
t	time
U_2	impeller exit tip speed
X, Y	independent geometric coordinates
Δ	mass eccentricity
ρ	density

ω angular velocity
Ω whirl angular velocity
ω_n critical speed in air
ξ $c_g/c*$

Subscripts:

o   nominal or design condition

BACKGROUND

The speed and energy requirements of centrifugal pumps in critical service have steadily increased in the last one or two decades. This increasing power density has been contributing to reliability problems as well documented for example with the space shuttle engine pumps. Less well known, however, are the problems of pumps in services involving boiler feed, water injection, reactor charge, etc. In a number of cases, pump rotor dynamic phenomena appear to be at the root of field troubles. As a consequence, considerable research is being carried out at present to identify the rotor dynamic exciting forces and the corresponding response of the pump system.

It is now clear that in high energy centrifugal pumps, rotor dynamic behavior is significantly affected by the fluid dynamic phenomena arising in the gaps between rotating and stationary members. One class of such a rotor-stator interaction arises in the gap between the impeller and the surrounding stationary member. The stationary member may be a single or a double volute or a diffuser. The typical radial gap between the tongue of the volute or diffuser and the impeller periphery is between 2 and 10 percent of the impeller radius. The other class of interaction occurs in the close-clearance gaps whose function is to minimize the leakage of pumped fluid from high pressure to low pressure areas. These gaps, which are variously called as wear-rings or seal rings, have typical clearance to radius ratios of the order of 0.2%. The forces in these gaps are much stronger than those due to volute/diffuser-impeller interactions.

In this paper, the physical mechanism for the volute-impeller interaction is first described. Available data for estimating its magnitude are then reviewed. The response of a simplified rotating system subject to the volute-impeller forces as well as wear ring forces is then calculated using an analytical model. The amplitude of response and the onset of instability are predicted by this approach. The purpose of developing this model is more to derive a physical appreciation of the effects than to obtain accurate results for practical geometries. In this spirit, the method is applied to a multistage boiler feed pump, and the results reveal certain very interesting effects of the geometries typically utilized in such pumps.

VOLUTE IMPELLER INTERACTION EFFECTS

At the design point, the flow field in the volute is characterized by a free vortex velocity distribution in the radial direction. In the circumferential direction, the velocity and static pressure are uniform. As a consequence, the relative flow in the impeller is steady. If the impeller is

now displaced with respect to the geometric center of the volute, the flow area in the direction of motion is squeezed, and the tangential velocity is increased. At the diametrically opposite position, the area is increased and the velocity is decreased. Assuming no change in total head or losses, the differences in velocity on either end results in a difference of static pressure. In particular, the static pressure in the area in the direction of the displacement is reduced. Thus a force now is exerted in the same direction as the displacement, i.e., the force does not tend to restore the deflection. Hence the resulting stiffness is negative. Further, since the tangential velocities are now different, integration of the momenta around the circumference will lead to a different value than when the impeller is centered. In particular, if we only look along the direction in which deflection has taken place, the difference in tangential momenta will lead to a force at right angles to the deflection, thus generating the cross-coupling stiffness.

At off-design conditions, the flow field is no longer uniform, and the impeller relative flow is unsteady. The computation of the flow field for such situations is quite complex. Reference 1 contains a one dimensional finite element calculation method. It is evident that even a physical understanding of the flow mechanism when the impeller is moved off center is difficult. This phenomenon is being extensively studied at CalTech and test results for stiffness coefficients are reported in reference 2. The coefficients are non-dimensionalized as below:

$$K^* = k \; \frac{r_2}{\left(\frac{1}{2} \rho \; U_2^{\;2}\right) 2\pi r_2 b_2}$$

K* is considered to be a constant for a given geometry. Thus the dimensional stiffness k varies only with speed for a given geometry and fluid density. It is convenient to define the stiffness coefficients in the following way:

$$k_{xx} = - m_h \omega^2 = k_{yy}$$

$$k_{xy} = m_h{}' \omega^2 = - k_{yx}$$

Here the coefficients m_h and $m_h{}'$ have units of mass. Based on data of reference 2, as the direct stiffness is negative, m_h will be positive. The cross coupled terms are assumed to be equal and opposite. From the above:

and

$$m_h = K^*_{xx} \pi \rho r_2^{\;2} b_2$$

$$m_h{}' = K^*_{xy} \pi \rho r_2^{\;2} b_2$$

WEAR RING PHENOMENA

The dynamic effects induced by the leakage flow in close clearance annular fits have been extensively studied. A recent review of pertinent publications

can be found in reference 3. Based on these studies, the dynamic coefficients of wear rings can be represented as below:

$$k_{xx} = k_{yy} = m_L \omega^2$$

$$c_{xx} = c_{yy} = 2\,\bar{c}\omega$$

$$k_{xy} = -k_{yx} = \bar{c}\omega^2$$

$$c_{xy} = c_{yx} = m_{xx} = m_{yx} = m_{yy} = m_{xy} = 0$$

Here m_L and $\bar{c}$ have units of mass. The equations for computing these may be found in reference 4.

SIMPLIFIED ANALYSIS

To appreciate the significance of the fluid dynamic phenomena, a simplified rotor dynamic analysis of a single degree of freedom system can be carried out. The derivation below closely follows the analysis of H. Black (ref. 5) except that the volute-impeller interaction effect is now included. Assuming that the fluid forces can be represented as described earlier, we can obtain:

$$
\begin{bmatrix} m & o \\ o & m \end{bmatrix}
\begin{bmatrix} \ddot{x} \\ \ddot{y} \end{bmatrix}
+
\begin{bmatrix} k_s + m_L\omega^2 - m_h\omega^2 & \bar{c}\omega^2 + m'_h\omega^2 \\ -(\bar{c}\omega^2 + m'_h\omega^2) & k_s + m_L\omega^2 - m_h\omega^2 \end{bmatrix}
\begin{bmatrix} x \\ y \end{bmatrix}
$$

$$
+
\begin{bmatrix} 2\bar{c}\omega & o \\ o & 2\bar{c}\omega \end{bmatrix}
\begin{bmatrix} \dot{x} \\ \dot{y} \end{bmatrix}
=
\begin{bmatrix} F_x \\ F_y \end{bmatrix}
$$

The response of the rotor system to unbalance can be obtained by assuming circular whirling.

$$r = x + jy = R\,e^{j\omega t}$$

$$m\,\ddot{r} + 2\,\bar{c}\omega\dot{r} + \left\{ k_s + m_L\omega^2 - m_h\omega^2 - j\,(\bar{c}\omega^2 + m'_h\omega^2) \right\} r$$

$$= m\,\Delta\omega^2\,e^{j\omega t} \tag{1}$$

The absolute value of the amplitude of response is:

$$|R| = \frac{m\Delta}{\sqrt{\left\{\dfrac{k_s}{\omega^2} - (m - m_L + m_h)\right\}^2 + (\bar{c} - m'_h)^2}} \tag{2}$$

Critical speed is obtained when $|R|$ reaches a maximum value i.e., when:

$$\omega = \sqrt{\frac{k_s}{m + m_h - m_L}}$$

It is clear from the above that the negative direct stiffness of the volute-impeller interaction leads to a reduction in critical speed. Further, if the Lomakin mass is greater than the sum of the geometric mass and the volute-impeller interaction effect, there is no critical speed.

A second interesting point that may be observed is that the critical speed is independent of damping! The peak response under damped conditions occurs at the undamped natural frequency. Of course, this is true only for the case that the damping coefficient is proportional to the speed.

The stability of the rotating system can be analyzed by assuming circular whirling

$$r = R\, e^{j\Omega t}$$

The solutions will be more realistic if we assume a generalized damping coefficient c_g

$$c_g = \xi\, c^* = 2m\xi\omega_n$$

when ω_n is the critical speed in air

$$\omega_n = \sqrt{\frac{k_s}{m}}$$

For stability analysis, the right hand side of the force equilibrium equation (1) can be assumed to be zero. Then we can get:

$$m\,\ddot{r} + (2m\xi\omega_n + 2\bar{c}\omega)\,\dot{r} + \left\{k_s + (m_L - m_h)\,\omega^2 - j\,(\bar{c}\omega^2 + m_h'\omega^2)\right\}\, r = 0$$

$$- \Omega^2 + (2\xi\omega_n + 2\frac{\bar{c}}{m}\omega)\, j\, \Omega +$$

$$\left(\omega_n^2 + \frac{m_L - m_h}{m}\omega^2 - j\, \frac{\bar{c} + m_h'}{m}\omega^2\right) = 0$$

By separately equating the real and imaginary parts to be zero, we get:

$$\Omega^2 = \omega_n^2 + \frac{m_L - m_h}{m}\omega^2 \tag{3}$$

$$\Omega = \frac{\dfrac{\bar{c} + m_h'}{m}\omega^2}{2\xi\omega_n + \dfrac{2\bar{c}}{m}\omega} \tag{4}$$

The onset of instability as defined by equation (3) and (4) can be analytically expressed for two simplified cases:

Case (i): $\xi = 0$ Equation (4) yields

$$\Omega = \frac{\omega}{2}\left[1 + \frac{m_h'}{\bar{c}}\right]$$

Substitution in equation (3) yields:

$$\omega_{onset} = \frac{2\omega_n}{\sqrt{(1 + \dfrac{m_h'}{\bar{c}})^2 - 4\,\dfrac{m_L - m_h}{m}}}$$

In the absence of the volute-impeller interaction (i.e., $m_h' = m_h = 0$), the onset of instability occurs at greater than twice the air critical speed. When m_h' and m_h are not zero, the instability onset threshold is reduced.

Case (ii): In the absence of wear ring effects ($\bar{c} = m_L = 0$), and when the predominant diagonal stiffness is that due to the shaft elasticity (m_h is much less than k_s/ω^2), the damping needed to avert instability can be derived from equation (3) and (4) as below:

$$2\xi = \frac{m_h'\,\omega^2}{k_s}$$

i.e. since $m_h'\omega^2$ is the cross coupling stiffness due to volute-impeller interaction, instability can arise when the critical damping ratio is less than one half the ratio of cross-coupling to direct stiffness. This particular result was stated by Prof. T. Caughey of California Institute of Technology.

In the general case, when the simplifying assumptions of case (i) and (ii) are not made, equations (3) and (4) describe a quartic which can only be solved numerically.

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NUMERICAL EXAMPLE

To illustrate the significance of the various fluid dynamic effects, the response of a boiler feed pump rotor has been worked out. It should be understood that the purpose is more illustrative than precise. Accurate response calculations should, of course, be done only by using a computer program which represents the geometry faithfully. Complete details of the calculation may be found in reference 4. The major reults alone are reported here.

The pump contains six impellers in series. There are ten wear rings, a center stage piece and a balance sleeve. In these calculations, the balance sleeve effect is ignored because of its proximity to the end bearing. The response of the pump to unbalance as given by equation (2) is shown in Figure 1. With nominal clearance, the response is very flat with a theoretical critical speed at about 14000 RPM. If the balance sleeve effect had been included, the critical speed would have been completely suppressed. With increasing wear in the clearances, peak response is observed and the critical speed moves closer to the air critical speed.

The centerstage piece plays a dominant role in the response picture. Figure 2 shows the results when the effect of the centerstage is ignored. (In some pump designs when the impellers are aligned in the same direction, there is no need for a centerstage piece). It can be seen that the response pattern is strongly indicative of lightly damped critical conditions.

To obtain the instability onset, equations (3) and (4) were numerically solved for this case by M. L. Smith of Byron Jackson. His results are shown in figure 3 as a plot of the instability onset speed as a function of wear ratio for different values of ξ. It can be seen that even with no damping, the system is inherently stable as long as the clearance does not open up to more than twice its nominal value. At the operating speed of 5700 RPM, the system does not become unstable until the clearance opens up to about 3.8 times its nominal value. Of course, the pump hydraulic performance would have greatly degraded by then. Normally pump clearances are restored well before this condition is reached. With expected ξ values between 0.1 and 0.3, the system exhibits excellent stability.

The effect of the centerstage piece is illustrated in figure 4. At 5700 RPM, the system becomes unstable with a clearance ratio of 2.6. The volute impeller interaction has a strong effect on instability. Figure 4 shows that when this effect is neglected, the instability threshold is pushed up to very high speeds or extremely large clearances.

A very recent publication (ref. 6) has indicated that volute-impeller interaction produces strong damping and hydrodynamic mass coefficients. A strong point is also made in that publication that all of these effects should be properly included for accurate prediction of rotor dynamic behavior.

CONCLUSIONS

Several interesting conclusions can be made as a result of the analysis presented in this paper.

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1. The direct hydraulic stiffness in the volute-impeller area can be thought of as a mass, which adds to the geometric mass.

2. The direct stiffness in wear rings acts as a negative mass, subtracting from the geometric mass. Critical speed can be completely eliminated if the negative mass is sufficiently large.

3. If the only damping in the system is due to the wear rings (i.e., proportional to speed), the peak-response frequency is independent of damping!

4. In the absence of damping and volute impeller forces, the instability onset speed is at least twice the air critical speed. The cross coupling stiffness of the interaction matrix can bring the onset speed closer to air critical speed.

5. The magnitude of the interaction forces appear to be small for the boiler feed pump studied. Only when the wear ring clearances are significantly opened up, does the instability onset appear to be a threat to rotor dynamic reliability.

REFERENCES

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6. Jery, B.; Brennen, C. E.; Caughey, T. K.; and Acosta, A. J.: Forces on Centrifugal Pump Impellers. 2nd International Pump Symposium, Texas A&M University, April 1985.
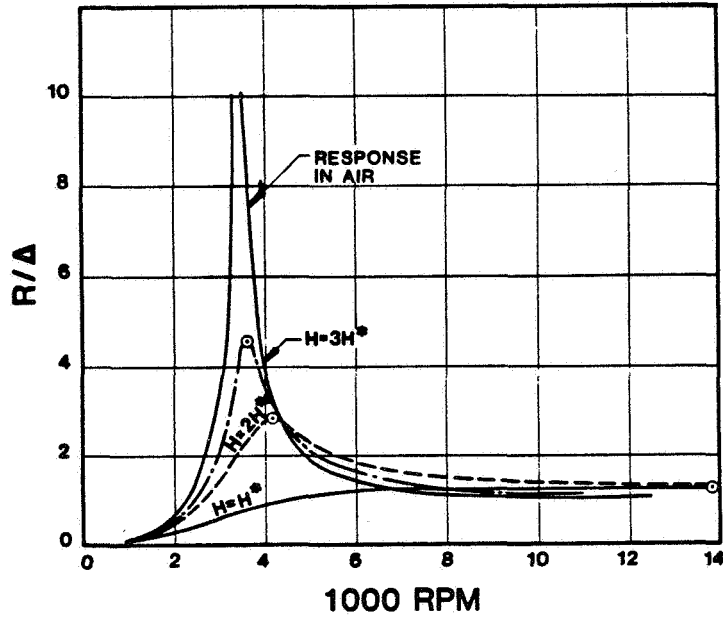
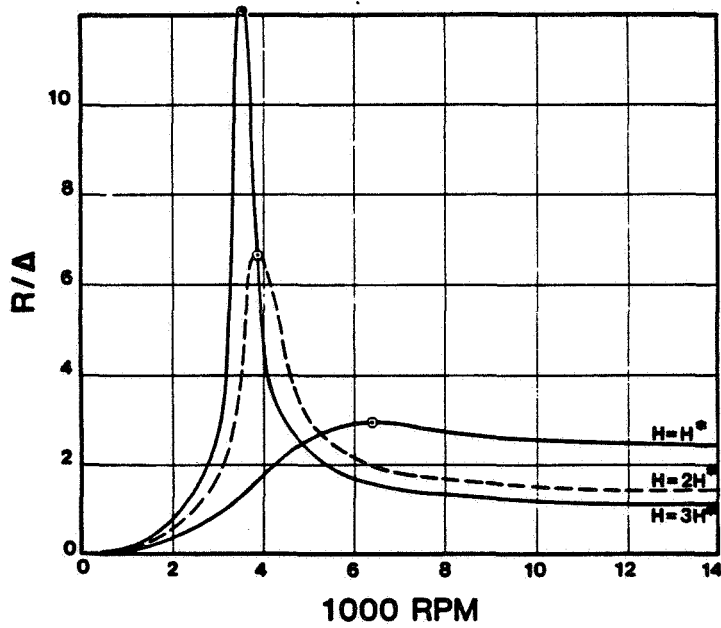Fig. 1: Response of volute pump to unbalance for various clearances.

Fig. 2: Response as in Figure 1, except that the center stage piece effect is ignored.
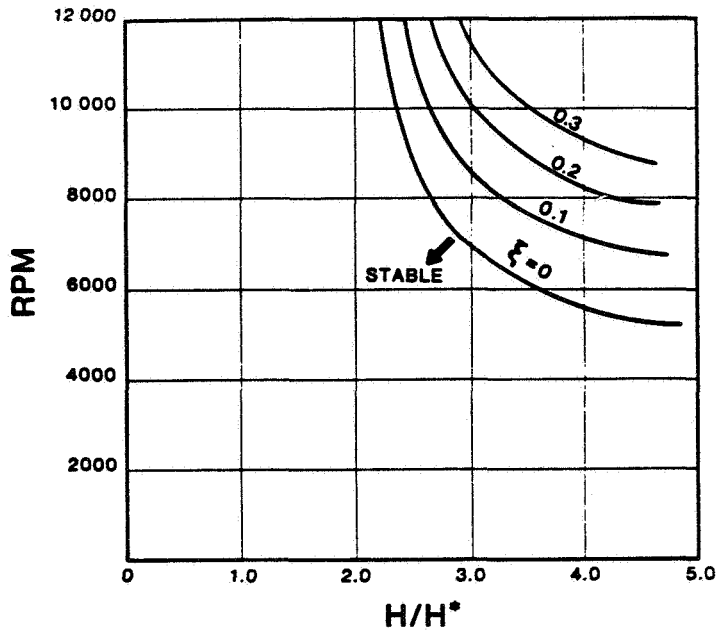
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Fig. 3:   Stability boundary of volute pump as a function
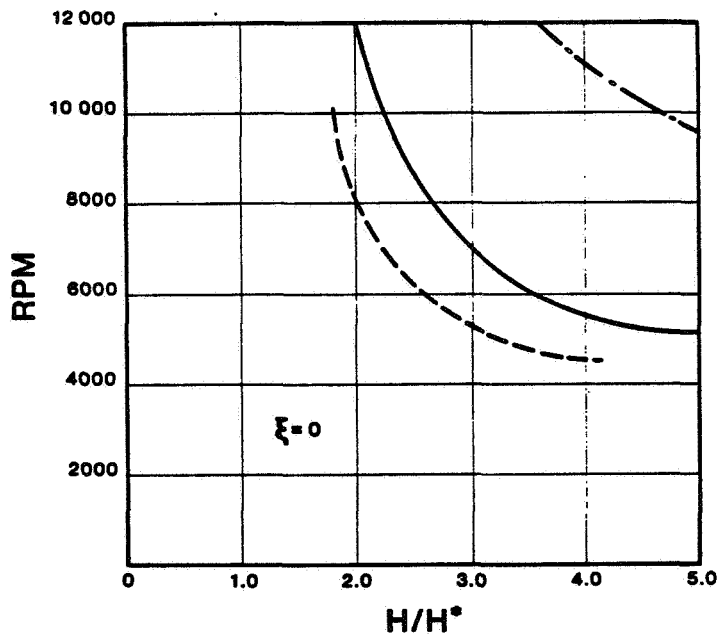of clearance and damping.

Fig. 4:   Stability boundary; ————— : Volute pump, —————— :
Volute pump with the center stage effect ignored,
— — — — —: Volute pump with hydraulic interaction
effects ignored.

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