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THE INFLUENCE OF SWIRL BRAKES AND A TIP DISCHARGE ORIFICE ON THE ROTORDYNAMIC FORCES GENERATED BY DISCHARGE-TO-SUCTION LEAKAGE FLOWS IN SHROUDED CENTRIFUGAL PUMPS

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ABSTRACT

This paper reports on experiments conducted in the Rotor Force Test Facility at the California Institute of Technology to examine the effects of a tip leakage restriction and swirl brakes on the rotordynamic forces due to leakage flows on an impeller undergoing a prescribed circular whirl. The experiments simulate the leakage flow conditions and geometry of the Alternate Turbopump Design (ATD) of the Space Shuttle High Pressure Oxygen Turbopump and are critical to evaluating the pump's instability problems.

Results indicate the detrimental effects of a discharge orifice and the beneficial effects of adding swirl brakes. Plots of the tangential and normal forces versus whirl frequency ratio show a substantial increase in these forces along with destabilizing resonances when a discharge orifice is added. When swirl brakes are added, some of the detrimental effects of the orifice are reduced. For the tangential force, a significant reduction occurs and a destabilizing resonance appears to be eliminated. For the normal force, although the overall force is not reduced, once again a destabilizing resonance appears to be eliminated.

NOMENCLATURE

$[A^*]$	Rotordynamic force matrix
c	Cross-coupled damping coefficient, normalized by $\rho\pi\omega^2 R_2^2 L\varepsilon$
C	Direct damping coefficient, normalized by $\rho\pi\omega^2 R_2^2 L\varepsilon$
$F^*_x(t)$	Lateral horizontal force in the laboratory frame
$F^*_y(t)$	Lateral vertical force in the laboratory frame
F^*_{ox}	Steady horizontal force
F^*_{oy}	Steady vertical force
F^*_n	Force normal to whirl orbit
F_n	Force normal to whirl orbit normalized by $\rho\pi\omega^2 R_2^2 L\varepsilon$
F^*_t	Force tangent to whirl orbit
F_t	Force tangent to whirl orbit normalized by $\rho\pi\omega^2 R_2^2 L\varepsilon$

F_1, F_2	Lateral forces in rotating frame
H	Clearance between impeller shroud and housing
k	Cross-coupled stiffness coefficient normalized by $\rho\pi\omega^2 R_2^2 L\varepsilon$
K	Direct stiffness coefficient normalized by $\rho\pi\omega^2 R_2^2 L\varepsilon$
L	Axial length of impeller
M	Direct added mass coefficient normalized by $\rho\pi\omega^2 R_2^2 L\varepsilon$
Q	Volumetric leakage flow rate
R_2	Radius of impeller at leakage inlet
u_s	Mean leakage inlet path velocity of fluid
u_θ	Mean leakage inlet swirl velocity of fluid
$x^*(t)$	Horizontal displacement of impeller on its orbit
$y^*(t)$	Vertical displacement of impeller on its orbit
ε	Eccentricity of impeller's circular whirl orbit
Δp_o	Nominal pressure drop through tip discharge orifice
Δp_t	Overall pressure drop through leakage annulus
ρ	Density of leakage fluid
ϕ	Leakage flow coefficient, $u_s/\omega R_2$
ω	Main shaft radian frequency
Ω	Whirl radian frequency

INTRODUCTION

Previous experimental and analytical results have shown that discharge-to-suction leakage flows in the annulus of a shrouded centrifugal pump contribute substantially to the fluid induced rotordynamic forces (Adkins, 1988). Experiments conducted in the Rotor Force Test Facility (RFTF) at Caltech on an impeller undergoing a prescribed whirl have indicated that the leakage flow contribution to the normal and tangential forces can be as much as 70% and 30% of the total, respectively (Jery, 1986). Other experiments at Caltech have examined the rotordynamic consequences of leakage flows and have shown that the rotordynamic forces are functions not only of the whirl ratio but also of the leakage flow rate and the impeller shroud to pump housing clearance. The forces were found to be inversely proportional to the clearance. A region of forward subsynchronous whirl was found for which the average tangential force was destabilizing. This region decreased with flow coefficient (Guinzburg, 1992a). Also, it was demonstrated that leakage inlet (pump discharge) swirl can increase the cross-coupled stiffness coefficient (in some tests by over 100%) and hence increase the range of positive whirl for which the tangential force is destabilizing (Guinzburg, 1992b).

In recent experimental work, the present authors demonstrated that when the swirl velocity within the leakage path is reduced by the introduction of ribs or swirl brakes on the housing, then a substantial decrease in both the destabilizing normal and tangential forces could be achieved (Sivo, 1993).

Motivation for the present study is that previous experiments have shown that restrictions such as wear rings or orifices at pump inlets affect the leakage forces (Guinzburg, 1992a). Recent pump designs such as the Space Shuttle Alternate Turbopump Design (ATD) utilize tip orifices at discharge for the purpose of establishing axial thrust balance. The ATD has experienced rotordynamic instability problems and one may surmise that these tip discharge orifices may also have an important effect on the normal and tangential forces in the plane of impeller rotation. The present study simulates the ATD leakage flow conditions and determines if such tip leakage restrictions contribute to undesirable rotordynamic forces.

Additional motivation for the present study is that the widening of the leakage path annular clearance and the installation of swirl brakes in the ATD has been proposed to solve its instability problems. The present study assesses the effect of such a design modification on the rotordynamic forces.

ROTORDYNAMIC FORCES

Figure (1) shows a schematic of the hydrodynamic forces that act on a rotating impeller whirling in a circular orbit. F^*_x and F^*_y are the instantaneous lateral forces in the laboratory frame. Ω is the whirl radian frequency and ω is the main shaft radian frequency. The eccentricity of the orbit is given by ε. The lateral forces are given in linear form as:

$$\begin{pmatrix} F^*_x(t) \\ F^*_y(t) \end{pmatrix} = \begin{pmatrix} F^*_{ox} \\ F^*_{oy} \end{pmatrix} + [A^*] \begin{pmatrix} x^*(t) \\ y^*(t) \end{pmatrix} \tag{1}$$

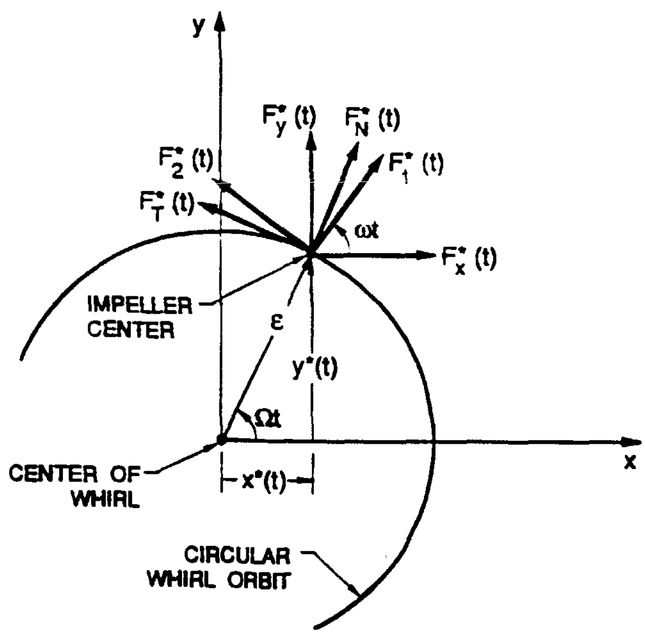


FIGURE 1. SCHEMATIC OF THE FLUID-INDUCED FORCES ACTING ON AN IMPELLER WHIRLING IN A CIRCULAR ORBIT

F^*_{ox} and F^*_{oy} are the steady forces which result from flow asymmetries in the volute. $[A^*]$ is the rotordynamic force matrix. It is a function of the mean flow conditions, pump geometry, whirl frequency ratio Ω/ω and if outside the linear range it may also be a function of the eccentricity ε. In the case of a circular whirl orbit:

$$x^*(t) = \varepsilon cos(\Omega t) \tag{2}$$

$$y^*(t) = \varepsilon sin(\Omega t) \tag{3}$$

The normal and tangential forces for a circular whirl orbit are given by (Jery, 1986 and Franz 1989):

$$F^*_n(t) = \frac{1}{2}(A^*_{xx} + A^*_{yy})\varepsilon \tag{4}$$

$$F^*_t(t) = \frac{1}{2}(-A^*_{xy} + A^*_{yx})\varepsilon \tag{5}$$

ROTORDYNAMIC COEFFICIENTS AND STABILITY

To study the stability of an impeller, it is convenient for rotordynamicists to fit the dimensionless normal force F_n to a quadratic function of the whirl ratio and to fit the dimensionless tangential force F_t to a linear function of the whirl ratio. The expressions are given by:

$$F_n = M\left(\frac{\Omega}{\omega}\right)^2 - c\left(\frac{\Omega}{\omega}\right) - K \tag{6}$$

$$F_t = -C\left(\frac{\Omega}{\omega}\right) + k \tag{7}$$

where the dimensionless coefficients are the direct added mass (M), direct damping (C), cross-coupled damping (c), direct stiffness (K), and cross-coupled stiffness (k). As can be seen from equation (7), a positive cross-coupled stiffness is destabilizing because it starts the forward whirl of the impeller since it is equal to the tangential force at zero whirl ratio. Also, from equation (6), a large negative direct stiffness is destabilizing because it promotes a positive normal force which increases the eccentricity of the whirl orbit.

A convenient measure of the rotordynamic stability is the ratio of the cross-coupled stiffness to the direct damping (i.e. k/C) termed the whirl ratio. This is just a measure of the range of positive whirl frequency ratios for which the tangential force is destabilizing.

TEST APPARATUS

The present experiments were conducted in the Rotor Force Test Facility (RFTF) at Caltech. The leakage flow test section of the facility is shown in Figure (2).

The working fluid is water. The main components of the test section apparatus consist of a solid or dummy impeller (or rotating shroud), a housing (or stationary shroud) instrumented for pressure measurements, a rotating dynamometer (or internal force balance), an eccentric whirl mechanism (not shown) and a leakage exit seal ring (or face seal). The solid impeller is used so that leakage flow contributions to the forces are measured but the main through flow contributions are not experienced. Figure (3) shows a close-up of the leakage path. The inner surface of the housing has been modified to accommodate meridional ribs or swirl brakes along the length of the leakage annulus. The ribs are each 3/16 of an inch wide and with a height, d, of 0.2 inch. 11 equally spaced ribs can be installed. At the entrance to the leakage path, an aluminum ring can be installed on the

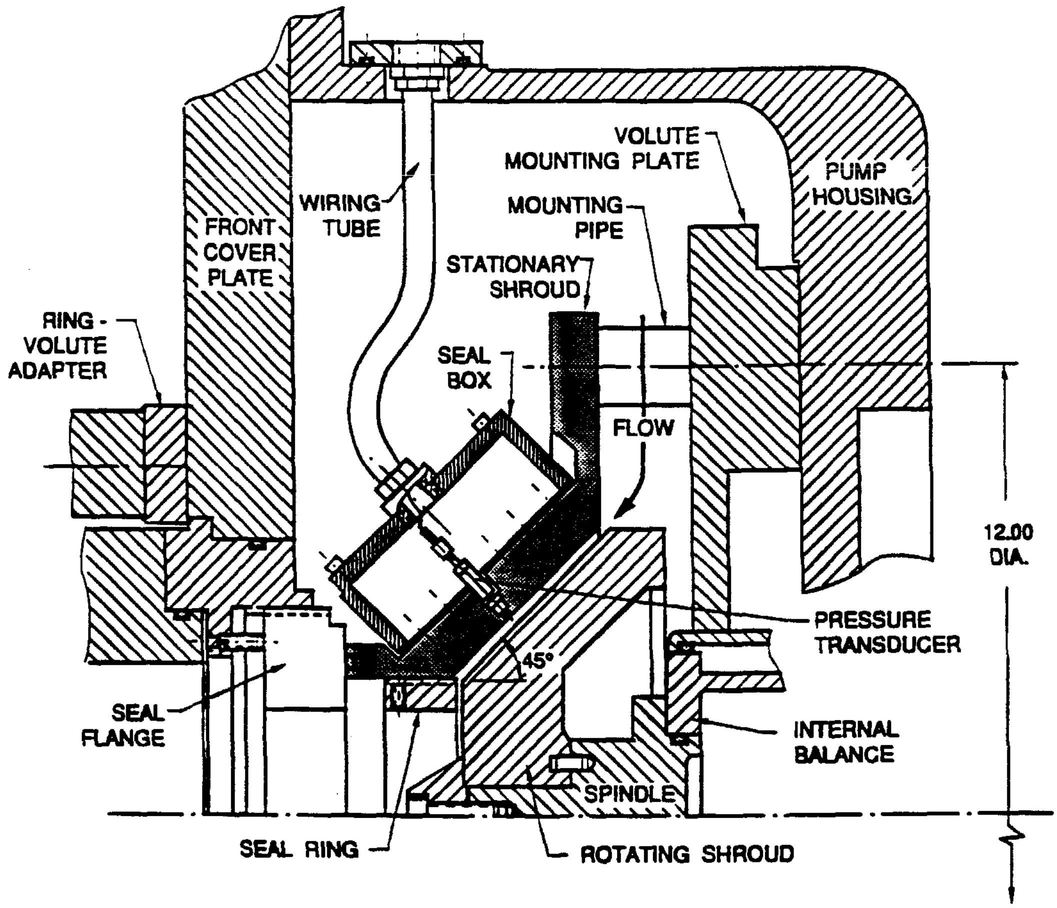


FIGURE 2. LEAKAGE FLOW TEST SECTION (ZHUANG, 1989)

housing and a plastic attachment can be installed on the dummy impeller. Together, these form a discharge corner seal which simulates the tip discharge orifice of the Space Shuttle ATD. To match the orifice geometry of the ATD with that of the experimental model, an orifice leakage gap, O, of 0.02 inches and an orifice leakage length overlap, C, of 0.01 inches were used (refer to Figure (3)). This orifice configuration is refered to as Orifice 2 by the authors. The leakage flow annulus between the impeller and housing is inclined at 45° to the axis of rotation. The nominal clearance between the solid impeller and the housing can be varied by axial adjustment of the housing. The flow through the leakage path is generated by an auxiliary pump. The solid impeller is mounted on a spindle attached to the rotating dynamometer connected to a data acquisition system which permits measurements of the rotordynamic force matrix. Jery, 1986 and Franz, 1989 describe the operation of the dynamometer. The eccentric drive mechanism imposes a circular whirl orbit on the basic main shaft rotation. The radius of the circular whirl orbit (or eccentricity) can be varied. The face seal at the leakage exit models a wear ring. The clearance between the face seal and impeller face is adjustable.

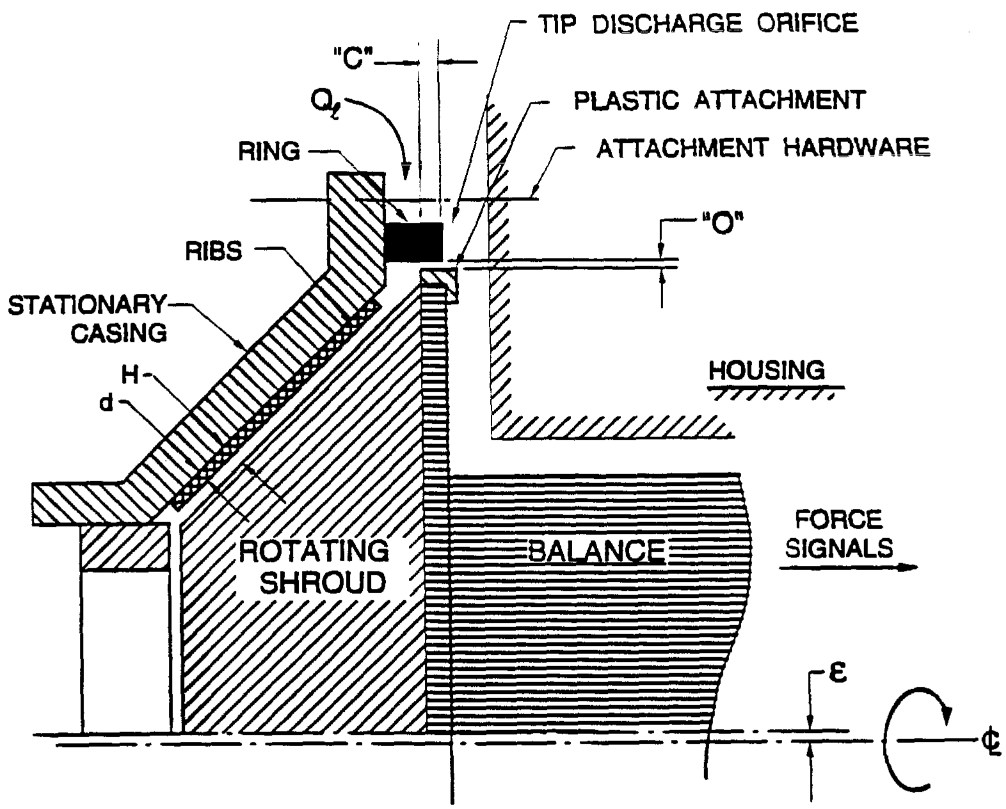


FIGURE 3. CLOSE-UP OF LEAKAGE FLOW PATH AND TIP DISCHARGE ORIFICE

TEST MATRIX

This experiment was designed to measure the rotordynamic forces due to leakage flows for a geometry simulating the Space Shuttle ATD over a range of subsynchronous whirl frequency ratios.

All tests were conducted at a shaft speed of 2000 RPM and a whirl eccentricity ε of 0.01 inches. Tests were performed for whirl frequency ratios in the range $-0.7 \leq \frac{\Omega}{\omega} \leq +0.7$ at 0.1 increments or finer.

For tests without brakes, a nominal annulus clearance H of 0.1 inches was used. For tests with brakes, H was 0.3 inches. In this manner, H - d was maintained at 0.1 inches for all tests to conform with the proposed design modification of the ATD. 11 brakes were used.

For the ATD, the leakage flow coefficient, ϕ, at operating speed is approximately 0.021. This is closely matched experimentally with a leakage flow rate of 10 GPM, yielding a ϕ of 0.0215, and hence results for tests at this flow rate are presented.

To properly model the effect of the tip discharge orifice of the ATD it is important to match the ratio of the orifice pressure drop to the overall leakage pressure drop (i.e. $\Delta p_o/\Delta p_t$) with that of the ATD. For the ATD, this pressure ratio is approximately 0.102. To matched this pressure ratio at 10 GPM the experimental model's leakage exit face seal was adjusted, but it was found that doing so did not change the pressure ratio much. A face seal clearance of 0.05 inches yielded an orifice pressure ratio of 0.109, and hence was found to be satisfactory. Tests were also performed at a tighter face seal clearance of 0.02 inches, which yielded an orifice pressure ratio of 0.098. This was done to assess the effect of the face seal. The tighter face seal clearance is also typical of the present Space Shuttle HPOTP.

RESULTS FOR TESTS TO DETERMINE THE EFFECT OF A DISCHARGE ORIFICE

Figure (4) shows comparison plots of the normalized tangential and normal rotordynamic forces versus whirl frequency ratio with and without a tip discharge orifice at 2000 RPM, 10 GPM and the wide face seal clearance of 0.05 inches. By looking at the normal force plot, one can immediately see that the discharge orifice causes a substantial increase in the force. Also, a destabilizing "resonance" appears to be produced at the positive whirl frequency ratio of 0.1 when the orifice is installed. Looking at the tangential force plot, one can again see an increase in the force when an orifice is installed. There also appears to be a resonance near a whirl frequence ratio of 0.1, although it could not be fully captured due to the inability to test at the low whirl ratios between 0 and 0.075.

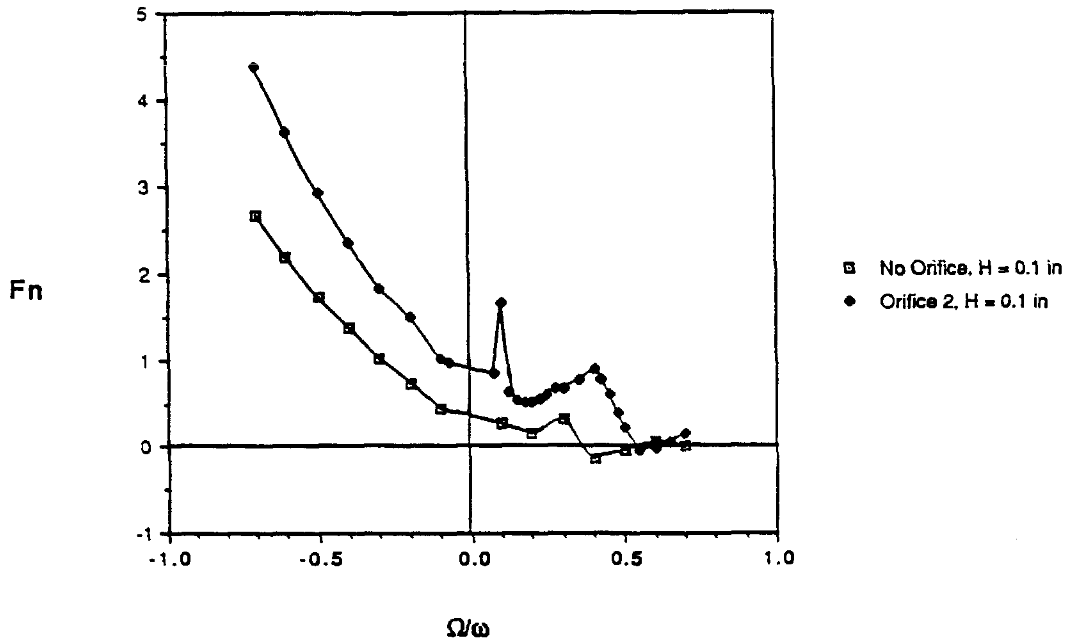
Figure (5) shows the same plots as Figure (4) although for tests at the tighter face seal clearance of 0.02 inches. Once again, the normal force plot shows a substantial increase in the force when the orifice is added. There also appears to be a resonance at 0.1 with the orifice. Looking at the tangential force plot, things appear to be less clear, although there is a destabilizing increase in the force between whirl ratios of 0.1 to 0.35. There appears to be no resonances, but this may be due to the fact that there is no data between whirl ratios 0 to 0.1.

RESULTS FOR TESTS TO DETERMINE THE EFFECT OF SWIRL BRAKES

Figure (6) shows comparison plots of the normalized tangential and normal rotordynamic forces versus whirl frequency ratio with the tip discharge orifice installed and with and without swirl brakes at 2000 RPM, 10 GPM and the wide face seal clearance of 0.05 inches. By looking at the normal force plot, one can see that although the force is not reduced for positive whirl ratios, the destabilizing resonance at the whirl ratio of 0.1 appears to be eliminated when brakes are installed. Looking at the tangential force plot, a significant reduction in the force is obtained and the resonance at the whirl ratio of 0.1 is eliminated. A resonance, however, is created at a whirl ratio of 0.4, but is not large enough to become destabilizing.

Figure (7) shows the same plots as Figure (6) although for tests at the tighter face seal clearance of 0.02 inches. Once again, for the normal force, the resonance at the whirl ratio of 0.1 is eliminated when brakes are installed. For the tangential force, there is a reduction in force between whirl ratios of 0.15 to 0.35 when brakes are installed. Once again, though, a non-destabilizing resonance is created at a whirl ratio of 0.4.

Comparison Plot (2000 RPM, Face Seal = 0.05 in, 10 GPM)
No Brakes



Comparison Plot (2000 RPM, Face Seal = 0.05 in, 10 GPM)
No Brakes

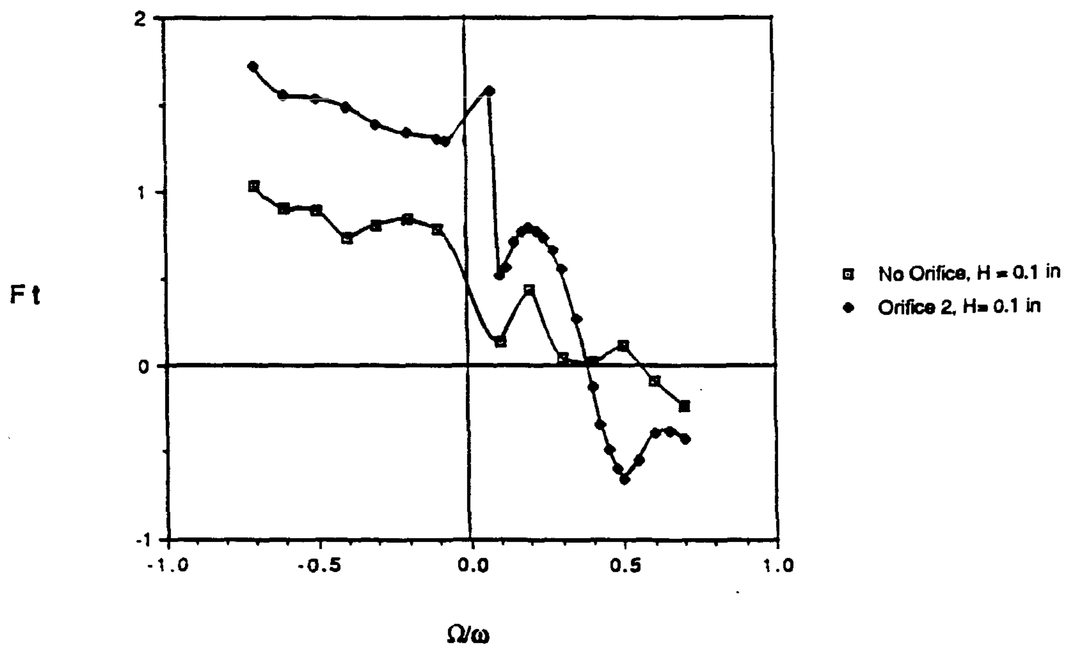
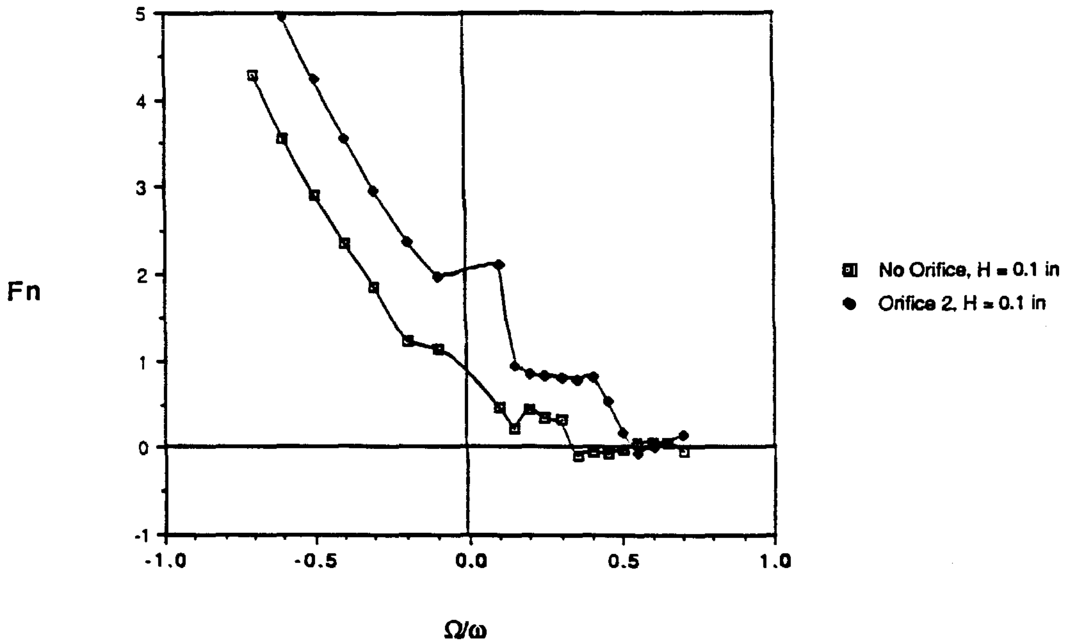


FIGURE 4. EFFECT OF TIP DISCHARGE ORIFICE (WIDE FACE SEAL).

Comparison Plot (2000 RPM, Face Seal = 0.02 in, 10 GPM)
No Brakes



Comparison Plot (2000 RPM, Face Seal = 0.02 in, 10 GPM)
No Brakes

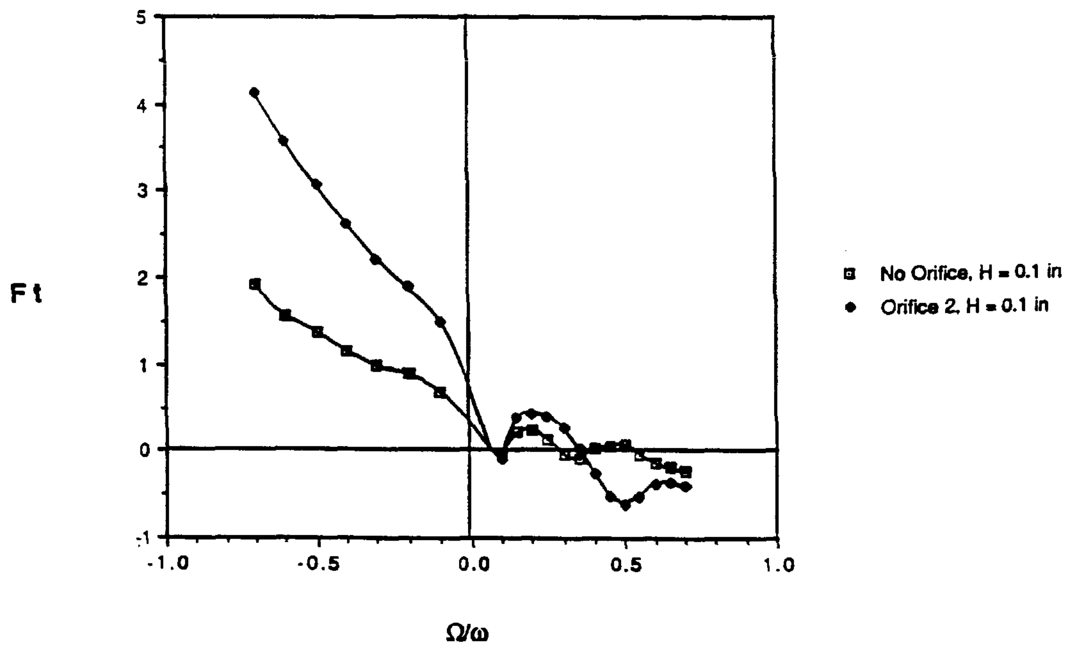
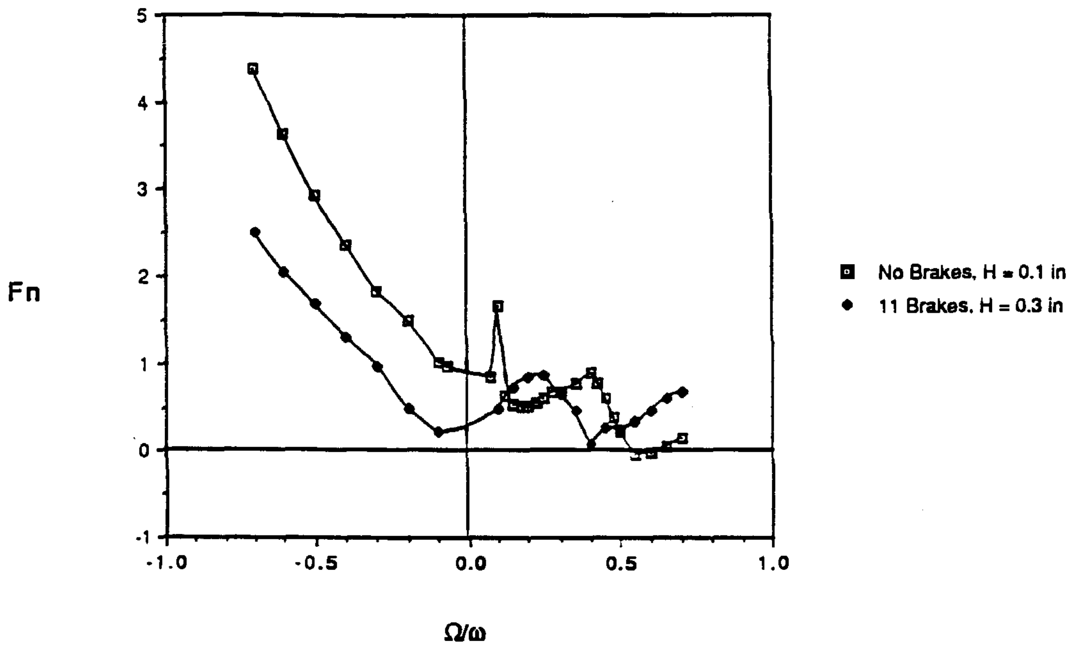


FIGURE 5. EFFECT OF TIP DISCHARGE ORIFICE (TIGHT FACE SEAL).

Comparison Plot (2000 RPM, Face Seal = 0.05 in, 10 GPM)
Orifice 2



Comparison Plot (2000 RPM, Face Seal = 0.05 in, 10 GPM)
Orifice 2

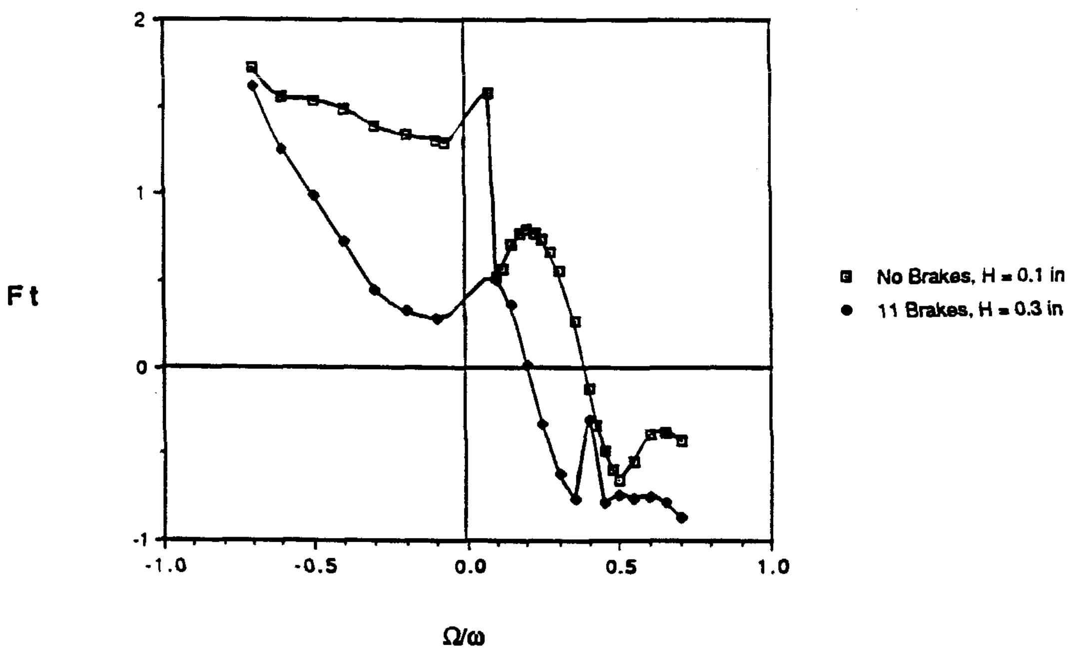
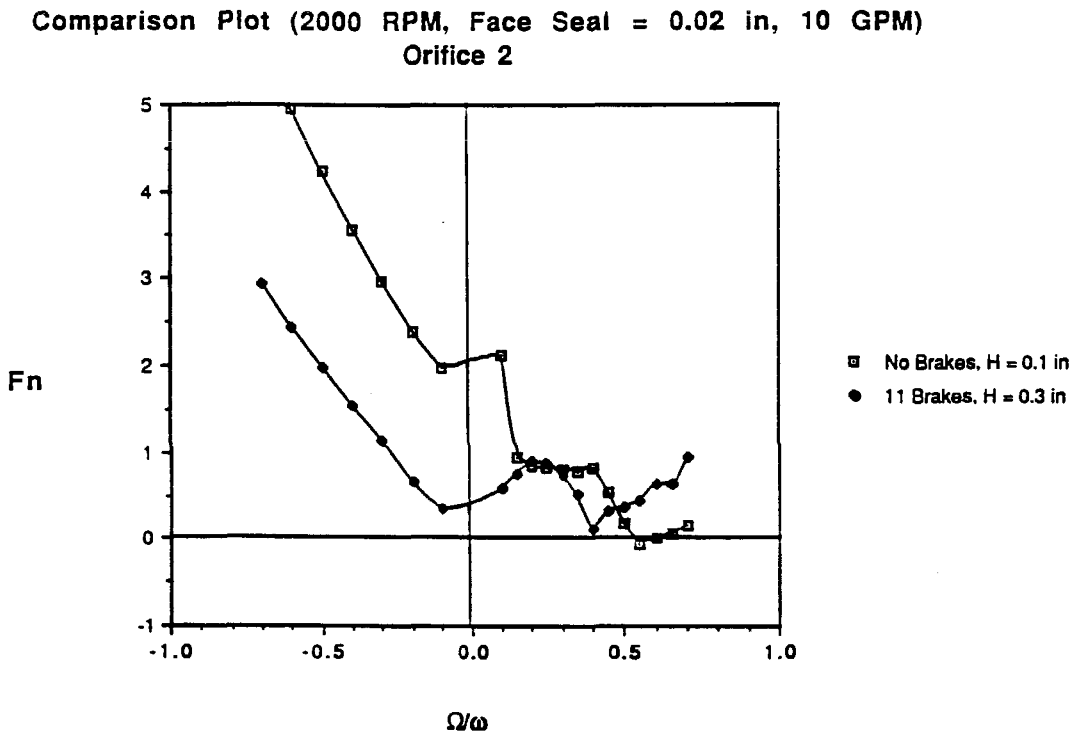


FIGURE 6. EFFECT OF SWIRL BRAKES (WIDE FACE SEAL).



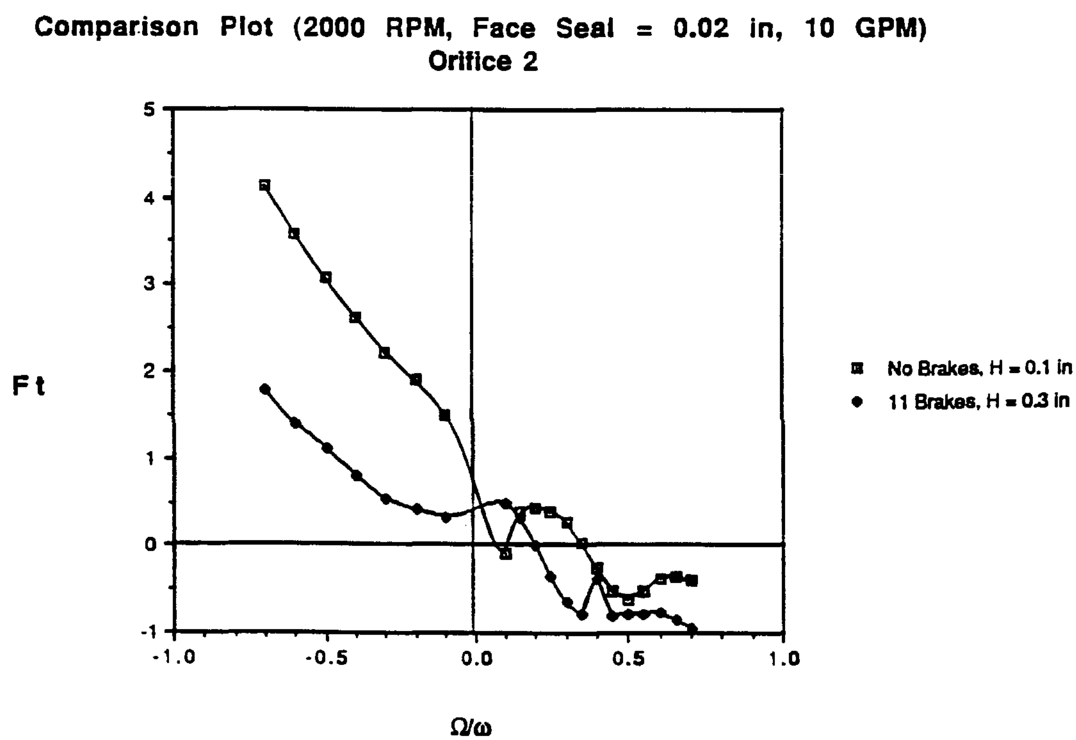


FIGURE 7. EFFECT OF SWIRL BRAKES (TIGHT FACE SEAL).

CONCLUSIONS

The present research has clearly shown that the effect of a tip discharge orifice of the type used for axial thrust balance in the Alternate Turbopump Design (ATD) of the Space Shuttle High Pressure Oxygen Turbopump on the rotordynamic forces due to leakage flows is destabilizing. Besides increasing both the normal and tangential forces, destabilizing resonances are produced when such an orifice is used.

The present research also has shown that the design modification of widening the leakage path annular clearance and installation of 11 swirl brakes in the ATD would reduce some of the detrimental effects of the orifice.

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