DAMPING SEALS FOR TURBOMACHINERY

George L. von Pragenau
Systems Dynamics Laboratory
NASA Marshall Space Flight Center
Huntsville, Alabama 35812 USA

Abstract

The paper addresses rotor whirl stabilization of high performance turbomachinery which operates at supercritical speed. Basic whirl driving forces are reviewed. Stabilization and criteria are discussed. Damping seals are offered as a solution to whirl and high vibration problems. Concept, advantages, retrofitting, and limits of damping seals are explained. Dynamic and leakage properties are shown to require a rough stator surface for stability and efficiency. Typical seal characteristics are given for the case of the high pressure oxidizer turbopump of the Space Shuttle. Ways of implementation and bearing load effects are discussed. Feasibility was first proven analytically in a Center Director Discretionary Fund project in fiscal year 1981. The Rocketdyne Division of Rockwell International has since installed such seals in the high pressure oxidizer turbopump for development to raise the speed limit and thus achieve higher thrust for the Space Shuttle Main Engine.

Introduction

High performance turbomachinery operates supercritically, i.e. with speeds above rotor resonances (critical speeds). Thus more power is being packed into lightweight units unless speed must be limited to prevent a rotor from whirling and causing damage. The speed limit challenged many as well as the writer to find a solution, not without earnest prayer (which God answered), and not without the foundation layed by others such as Dr. Henry Black from Scotland, who is well remembered.*[1-4]

* Deceased January 17, 1980.

Constant Whirl Forces

Whirl is an orbital motion, superimposed upon the rotation of a rotor. Whirl forces are conceptually known, but accurate data are not readily available. Whirl forces (Fig. 1) are constant or increase with rotor motion.

Synchronous constant forces are caused by mass unbalance, runout rubbing, and seal runout. Subsynchronous constant forces are generated by rotating Coulomb friction forces in rotor joints.[5] The constant forces produce a forward whirl limit cycle response at synchronous or subsynchronous frequencies. Well balanced rotors with minute runout yield low vibrations, however, Coulomb friction in rotor joints can cause damaging rotor vibrations. Coulomb friction whirl can be reduced with lubricating the rotor joints and splines. The whirl can be avoided with tight but nonlubricated interference fits. Joint locking with a high rotor unbalance is not recommended.

Proportional Whirl Forces

Increasing whirl forces can be most destructive. The simplest representation of the forces is a proportional relationship which couples rotor deflections by skewed-symmetric matrices to transverse whirl forces. Rotor hysteresis,[6] seal gaps,[1] journal bearings,[7] turbine blade bypass,[8] and impeller interactions drive usually subsynchronous forward whirls. External rotor rubbing drives a backward whirl that for hard rubbing can become supersynchronous (Fig. 1).

Rotor Stabilization

The purpose of rotor stabilization is to negate the whirl regeneration of proportional forces while reducing the whirl limit cycle caused by constant forces. Rotor suspension damping, stiffness, and asymmetry are whirl stabilizers ordered after their efficiency. Natural damping is usually too low to prevent whirl in high pressure applications. Therefore, squeeze film dampers are used in turbomachinery
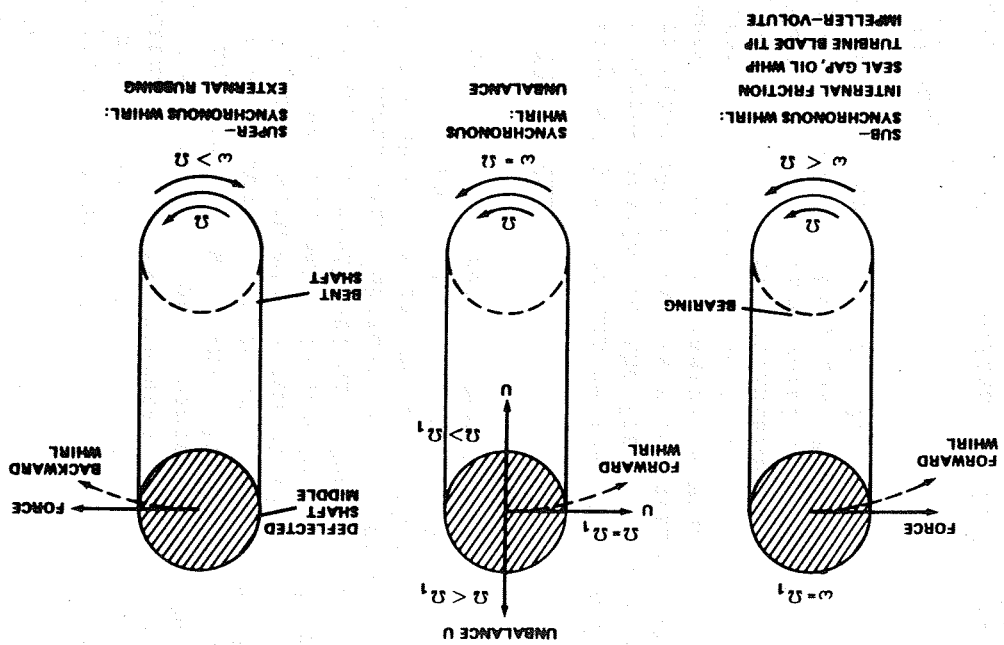
Figure 1. Principal Whirl Cases.

not only for vibration attenuation but also for whirl prevention.[4] However, such dampers are unsuitable for cryogenic applications. The difficulties prompted the development of damping seals.[9]

Damping seals are like annular seals, except for rough inner surfaces on stators (Fig. 2). Annular seals are a first step and were implemented to eliminate whirl from the high pressure turbopump (HPFTP) of the Space Shuttle Main Engine (SSME).[2,3] However, such seals can destabilize a rotor resonance below 1/2 the speed frequency. Damping seals improve this ratio to < 1/4, reduce leakage, and increase damping. Damping seals were recently installed in a high pressure oxidizer turbopump (HPOTP) for SSME development as shown in Figure 3.

Stability Criterion

The stability condition of a single mass rotor (Fig. 4) suffices to examine the seal effect. Also, more complex models are represented, considering one resonant mode that is only negligibly detuned by the seal. Equation (1) describes the internal seal whirl effect with the whirl factor Q proportional to the seal damping C. The Couette factor c of equation (2) is the ratio of circumferential bulk flow velocity v to rotor surface velocity w. Ω is the rotor speed in rad/s. Equation (3) is the characteristic equation of the model in Figure 4. Conformal mapping,[10] substituting with equation (1), and reordering produce the inequality (4) which sets the speed limit. K_y and K_z are the lateral suspension stiffnesses in y and z, m_s is the rotor mass, and s is the Laplace operator. Ω_1 in equation (5) defines the first critical speed or resonance.

$$Q = c\Omega C \tag{1}$$

$$c = v/w \tag{2}$$

$$K_y K_z + Q^2 + sC(K_y + K_z) + s^2[C^2 + m_s(K_y + K_z)]$$
$$+ s^3 2Cm_s + s^4 m_s^2 = 0 \tag{3}$$

LEAKAGE FLOW   u

SMOOTH
SURFACE

SEAL STATOR PART

v

u

TRIANGULAR
POCKETS FOR
ROUGH STATOR
SURFACE

w

NOTE: CLEARANCES
EXAGGERATED

SEAL ROTOR PART

STATOR

v

LOW LEVEL CIRCULATION
(COUETTE FLOW)
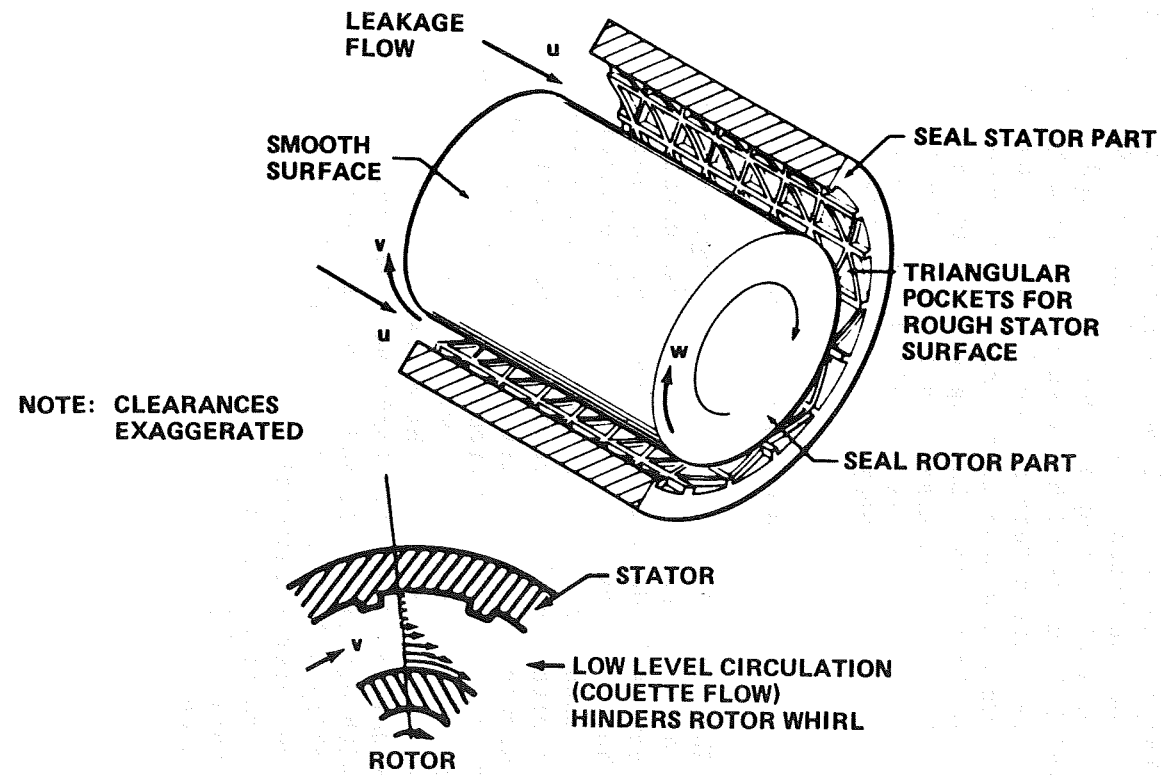HINDERS ROTOR WHIRL

ROTOR

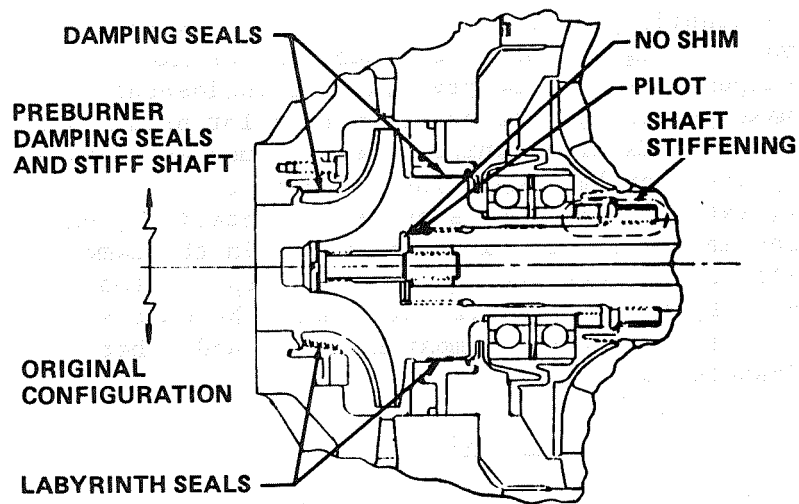Figure 2. Damping Seal Concept.

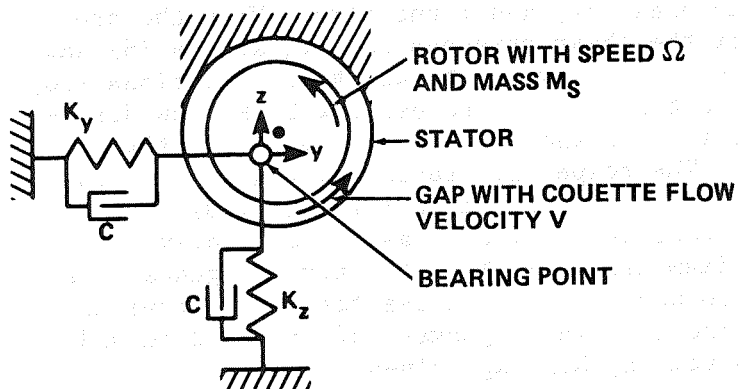Figure 3. HPOTP Damping Seal Retrofit.

Figure 4. Simple Rotor Model.

$$\Omega < [(K_y + K_z)/2m_s + (K_y - K_z)^2/4C^2]^{1/2}/c \qquad (4)$$

$$\Omega_1 = [(K_y + K_z)/2m_s]^{1/2} \qquad (5)$$

The stability limit of inequality (4) is high for low c values, high rotor resonances, and asymmetric rotor suspensions. Asymmetry can be coincidental which makes symmetry a safer assumption for predicting speed limits. External whirl effects, such as turbine blade bypass, can be lumped into Q by raising c. Conversely, external damping, such as from bearings, can be lumped into C by reducing c to maintain the same Q. Thus external effects are equivalently represented internally by the Couette factor c which becomes a measure of stability for comparing seals and other rotor dynamic components.

Flow Friction

The flow velocities relative to the rotor and the stator differ because of rotation. The velocities describe bulk flow so turbulent flow can be treated similar to pipe flow. Equations (6) and (7) describe shear stresses τ_r and τ_s respective to the rotor and stator. f_r and f_s are respectively the rotor and stator surface frictions which generally differ. u is the axial bulk flow velocity and ρ is the fluid mass density. The seal gap pressure gradients are given in equations (8) and (9) for the axial and circumferential directions x and q respectively. h is the gap thickness, p the pressure, and t the time. Note the projections of the shear stresses from equations (6) and (7). The Reynolds numbers R_r and R_s in equations (10) and (11) are taken in the respective bulk flow directions of the rotor and the stator. μ is the fluid viscosity. The respective rotor and stator friction factors f_r and f_s of equations (12) and (13) are Moody's approximation with 2h as hydraulically equivalent pipe diameter D.[11] The effective roughnesses k_r and k_s reach pocket depth values for well circulating pockets. The triangular pockets of Figure 2 should be best for obtaining high k_s values.

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$$\tau_r = \rho[u^2+(v-w)^2]f_r/8 \tag{6}$$

$$\tau_s = \rho(u^2 + v^2)f_s/8 \tag{7}$$

$$-\partial p/\partial x = \rho\{uf_r(u^2+(v-w)^2)^{1/2} + uf_s(u^2 + v^2)^{1/2}]8h$$

$$+ \rho \; du/dt \tag{8}$$

$$-\partial p/\partial q = \rho[\;(v-w)f_r(u^2+(v-w)^2)^{1/2} + vf_s(u^2 + v^2)^{1/2}]/8h$$

$$+ \rho \; dv/dt \tag{9}$$

$$R_r = \rho 2h[u^2+(v-w)^2]^{1/2}/\mu \tag{10}$$

$$R_s = \rho 2h(u^2 + v^2)^{1/2}/\mu \tag{11}$$

$$f_r = 0.0055 + 0.55(k_r/100h + 1/R_r)^{1/3} \tag{12}$$

$$f_s = 0.0055 + 0.55(k_s/100h + 1/R_s)^{1/3} \tag{13}$$

Couette Flow

The rotor generates a Couette flow which is established downstream of the seal inlet according to equation (14) which follows from equation (9) for the case of negligible circumferential pressure gradients and flow accelerations. Equations (15) and (16) are the asymptodes for u = 0 and u = ∞ which enclose the Couette factor c for all surface friction ratios. Figure 5 shows c as a function of surface friction ratio and the axial velocity ratio u/w. c is low for high stator friction. The friction ratio is approximately the cubic root of the roughness ratio, as the equations (12) and (13) indicate. Therefore, large roughness ratios are required.

$$(v-w)f_r[u^2+(v-w)^2]^{1/2} + vf_s(u^2+v^2)^{1/2} = 0 \tag{14}$$

$$u = 0: \quad c = 1/[1+(f_s/f_r)^{1/2}] \tag{15}$$

$$u = \quad : \quad c = 1/(1+f_s/f_r) \tag{16}$$

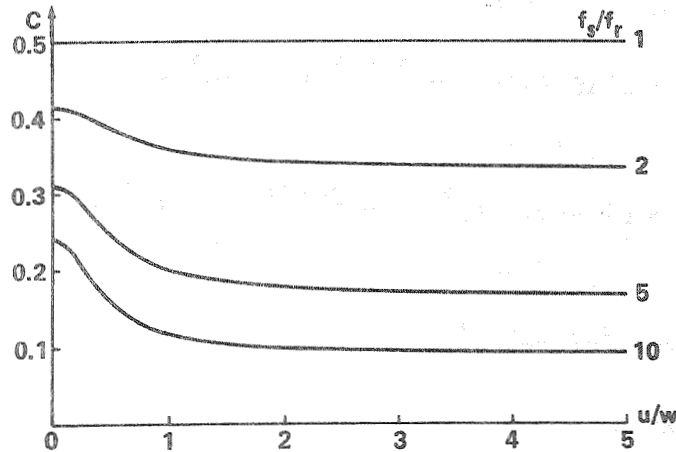
Figure 5. Couette factor $c = v/w$ versus the
friction and axial velocity ratios.

Dynamic Seal Parameters

The integral dynamic effect of a seal is described
by stiffness K, damping C, mass m, whirl coupling
$c\Omega C'$, and gyro coupling $c\Omega m'$ as shown in vector equa-
tion (17).

$$\begin{bmatrix} F_y \\ F_z \end{bmatrix} = \begin{bmatrix} K+sC+s^2m & c\Omega(C'+sm') \\ -c\Omega(C'+sm') & K+sC+s^2m \end{bmatrix} \cdot \begin{bmatrix} y \\ z \end{bmatrix} \tag{17}$$

The detailed derivations are given in reference 9,
except for minor differences in rearranging frequency
square terms herein. Some parts of the derivation are
repeated for continuity. Equations (18) through (22)
define the dynamic parameters and their response to
the whirl frequency ω. The time constants are small.
The basic stiffness K_o, damping C_o, and induced mass
m_o are given in equations (23) through (25) for inlet
loss f_1; resistivities F, G, and H; dimensions h, L,
and r of seal gap, seal length, and seal radius; axial
velocity u; and mass density ρ. The resistivities are

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given by the equations (26) through (32). The time constants are defined by the equations (33) through (36).

$$K = [K_o - (c\Omega)^2 m_o]/(1 + \omega^2 \tau^2) \tag{18}$$

$$C = [C_o(1 + \omega^2 \tau \tau_a) + m_o \omega^2 (\tau - \tau_b) - \tau K_o]/(1 + \omega^2 \tau^2) \tag{19}$$

$$m = [m_o(1 + \tau \tau_b(\omega^2 + (c\Omega)^2)) - C_o(\tau - \tau_a)]/(1 + \omega^2 \tau^2) \tag{20}$$

$$C' = [C_o(1 + \omega^2 \tau \tau_b) + 2m_o \omega^2 (\tau - \tau_b)]/(1 + \omega^2 \tau^2) \tag{21}$$

$$m' = [2m_o(1 + \omega^2 \tau \tau_b) - C_o(\tau - \tau_a)]/(1 + \omega^2 \tau^2) \tag{22}$$

$$K_o = (\pi r/h) u^2 (1 + f_1) \tau L(F + H)/2 \tag{23}$$

$$C_o = (\pi r/h)(\tau/\tau_a) L^3 (F + G)/12 \tag{24}$$

$$m_o = C_o \tau_o \tau_a/\tau_b \tag{25}$$

$$F = \rho[f_r(u^2 + (v-w)^2)^{1/2} + f_s(u^2 + v^2)^{1/2}]/8h \tag{26}$$

$$G = \rho u^2 [(f_r - g_r)/(u^2 + (v-w)^2)^{1/2}$$
$$+ (f_s - g_s)/(u^2 + v^2)^{1/2}]/8h \tag{27}$$

$$H = \rho[h_r(u^2 + (v-w)^2)^{1/2} + h_s(u^2 + v^2)^{1/2}]/8h \tag{28}$$

$$g_r = h_r/(1 + R_r k_r/100h) \tag{29}$$

$$g_s = h_s/(1 + R_s k_s/100h) \tag{30}$$

$$h_r = (0.55/3) \cdot (k_r/100h + 1/R_r)^{1/3} \tag{31}$$

$$h_s = (0.55/3) \cdot (k_s/100h + 1/R_s)^{1/3} \tag{32}$$

$$\tau = \rho L / [\rho u (1+f_1) + L (F+G)] \tag{33}$$

$$\tau_a = \rho L / [\rho u (1+f_1)(4+6\rho u / L (F+G) + L (F+G)] \tag{34}$$

$$\tau_b = \rho L / [4\rho u (1+f_1) + L (F+G)] \tag{35}$$

$$\tau_o = \rho / (F+G) \tag{36}$$

The flow induced masses in equation (17) are dipol masses that are negligible against the rotor mass. The whirl coupling is indeed proportional to the Couette factor c as shown earlier in equation (1). Table 1 gives the dimensions of diverse inlet and outlet seals of the HPOTP preburner pump impeller (Fig. 3). Table 2 lists the analytical results for labyrinth, damping, and smooth annular seals. Table 2 shows that the leakage rates of labyrinth and damping seals are comparable, while the smooth annular seals leak 1.6 times more. Tests by Rocketdyne produced the same ratio. The whirl ratio of damping seals is < 1/4 but 0.5 for the other seals. Damping seals have the highest damping while the smooth annular seals show the highest stiffness.

Table 1. HPOTP Preburner Pump Seal Dimensions for Inlet and Outlet at Full Power 30,367 RPM

PBP Seals Dimension	Gap mils1	Diam. inch	Leng. inch	Pocket mils 2	Press psi 3
Inlet Laby.	6	2.85	0.63	230	2006
Inlet Damp.	6	2.898	0.66	10(20)	2006
Inlet Smooth	6	2.898	0.66	0.032	2006
Outlet Laby.	4	3.345	0.87	80	6505
Outlet Damp.	4	3.3371	0.88	10(20)	6505
Outlet Smooth	4	3.3371	0.88	0.032	6505

NOTE: 1 = radial gap, 2 - effective pocket depth (actual depth), 3 = pressure difference.

Table 2. HPOTP Preburner Pump Seals' Dynamic for
Inlet and Outlet at Full Power 30,367 RPM

PBP Seals Dimension	Stif. Klb/in	Whirl Klb/in	Damp. Lbs/in	Ratio c %	Leak Lb/s
Inlet Laby.	2	2	1	50	4.94
Inlet Damp.	106	46	57	23	5.18
Inlet Smooth	151	105	53	50	7.90
Outlet Laby.	20	36	23	50	5.13
Outlet Damp.	447	281	388	22	5.09
Outlet Smooth	935	544	302	50	8.51

NOTE: % ratio times maximum speed must be below 1st
critical speed. Analysis is for 0.25 inlet
loss, 2.59E-8 psi-s lox viscosity, and
$1.05E-4 \text{ psi}(s/in.)^2$ lox density.

Seal Installation

Figure 3 illustrates a retrofitting case where
inlet and outlet labyrinth seals were replaced by
damping seals to inhibit a 450 Hz whirl at 30,000
RPM. Damping seals fit always into the space of
labyrinth seals. Leakages are comparable, but stiff-
ness and damping are much higher as seen in Table 2.
Both changes inhibit whirl, lower vibrations, and
increase critical speed. Damping seals seem to be
more effective than squeeze film dampers, simpler, and
smaller. Damping seals are suggested instead of
smooth annular seals to reduce leaks and to suppress
whirl.

The high stiffness of damping seals must be,
however, considered when replacing labyrinth seals.
Eccentricities between bearings and seals can cause
unexpected side loads.

Soft Seals

The side loads can be reduced with small eccen-
tricities, if tight tolerance costs are not prohibi-
tive. Soft seals are another concept worth

considering. Damping seals become softer as rough-
ness is increased. Axial flow is thus reduced and the
fluid compressibility much enhanced. Another way is
the insertion of a labyrinth seal downstream to shorten
a damping seal without changing leakage.

Hard Seals

The hard seal approach takes advantage of high
seal stiffness and uses the seals for supporting the
rotor. Critical speeds are so increased and loads of
pump and turbine discharges are shared by the seals.
Tight tolerances (eccentricities) can be circumvented
by loose fittings for the bearings' outer races. The
bearings support the rotor at startup and cutoff when
the seals are unpressurized or soft. At operation
speed the seals are stiff and can take over. Good
bearing life is expected because the bearings float
mostly during their operation in a loadless condition.
Outer race spinning due to floating can be prevented
with antirotation devices.

Conclusion

Damping seals are new technology for improving
turbomachinery operations. The shown properties
encourage many applications from retrofitting machines
that are beset by whirl problems to new and more
efficient designs. We are grateful and do not take
this progress for granted – be blessed the same way.

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