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Research paper

On the apex seal analysis of limaçon positive displacement machines

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ABSTRACT

Rotary machines, and limaçon machines in particular, offer a better power to weight ratio compared to reciprocating machines; however, leakage due to improper apex and side sealing have prevented rotary machines from thriving. In this paper, a modelling approach is presented to analyse the vibration of apex seal during the machine operation and the power loss caused by the seal friction. The seal and spring are modelled as a spring-mass system in which the seal deformation is negligible. The seal-groove relative positions have then been categorised into nine different possible cases based on the number of contact points between the seal and the seal groove. A case study has been presented to demonstrate the reliability of the model.

1. Introduction

In fluid processing machines, leakage through gaps separating different machine components is a major problem which adversely impacts machine performance. In rotary machines, leakage occurs mainly through the rotor-housing clearance and side gaps. Amrouche et al. [1] argue that sealing problem is a major disadvantage of the rotary engines. In order to prevent fluid leakage, limaçon rotary machines are equipped with apex seals and side seals. During machine operation, variable forces from the pressure differential between the working chambers, combined with the elastic, inertial and frictional effects, excite vibratory modes into the seals. The resulting vibration or “lift-off” and “poor contact” has been noticed by Matsuura, Terasaki, and Watanabe [5] who investigated the behaviour of apex seal against the housing surface. Seal vibration will vary the loading pattern between the seal, seal groove and the machine housing which will result in power loss due to friction. On the other hand, Pennock and Beard [6] considered that the friction between the side seal and the machine rotor is insignificant and can be ignored. The authors then went on to investigate the effect of crankshaft speed fluctuations on apex seal forces and concluded that this effect is also insignificant. Of note is that Pennock and Beard [6] did not include the

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Nomenclature

p_1p_2	limaçon chord
$2L$	limaçon chord length
C	centre of gravity of the seal
X_oY_o	stationary frame
X_rY_r	frame fixed at the centre point of the rotor and moves with it.
X_1Y_1	rotating frame
$\hat{i}, \hat{j}$, and $\hat{k}$	unit vectors of the X_1, Y_1, and Z_1 axes
X_sY_s	the frame attached to the seal centre of gravity with the unit vectors $\widehat{Y_s}$ and $\hat{X_s}$
XY	a frame fixed to the groove. Also signifies the initial position of the seal-attached frame (X_sY_s) with unit vectors $\hat{Y}$ and $\hat{X}$
o	limaçon pole
O_g	the seal origin
θ	crank angle
r	radius of the base circle
m	centre point of the chord
R_h	housing radial distance
b	limaçon aspect ratio
H	machine's depth measured perpendicular to the page
ω	rotor angular velocity
L_i	distance from the rotor chord centre point, m, to the seal's centre of gravity
δ_s	initial deflection of seal spring
δ_w	initial deflection of housing due to contact with seal
k_s	spring stiffness
k_w	the equivalent spring stiffness of the limaçon housing wall
c, c_r, c_w, and c_s	damping coefficients (All typed in lowercase)
m_s	mass of the seal
F	force acting on the seal
P_a, P_b	pressure of the chamber above and below the rotor chord, respectively
A_p, A_f	areas of the seal that are exposed to the machine chambers
ρ_s	density of seal material
W_g	width of the seal groove
W_s	width of the seal
d_{si}	seal height ($i = 1,\ 2,\ or\ blank$)
C_r	rotor-housing clearance (Uppercase C)
z	portion of the rotor apex truncated to machine the seal groove
ζ_o	seal protrusion at the initial position
$x,\ \dot{x},\ \ddot{x}$	seal linear displacement, velocity, and acceleration along X_r-axis
$y,\ \dot{y},\ \ddot{y}$	seal linear displacement, velocity, and acceleration along Y_r-axis
$\varphi,\ \dot{\varphi},\ \ddot{\varphi}$	seal angular displacement, velocity, and acceleration
$s_i,\ c_i,\ i_i$	special points on the apex seal ($i = 1,\ 2,\ 3,\ \ldots$)
g_i	points on the seal groove ($i = 1,\ 2,\ 3,\ \ldots$)
B_s	seal depth measured perpendicular to XY plane

gas pressure difference in their study but suggested that such a problem along with different machine starting conditions should be further investigated.

The work by Handschuh and Owen [4] suggested that the power loss of rotary machines is drastically affected by the crankshaft rotational speeds and the friction drag coefficient of the apex seal. The seal and rotary machine performance have been investigated from the viewpoint of lubrication and the trajectory of the seal's centre of mass [3,15]. These authors concluded that the increase of the rotor rotational speed is likely to decrease the range of maximum to minimum oil film thickness and will decrease the apex seal vibration amplitude. An outcome which reduces the wear of seal and housing surface.

From a different perspective, Warren and Yang [14] have proposed a deviation-function method for designing the rotary machine based on the apex seal geometry. The authors suggest that this approach will help improve the sealing capability and effectiveness which would impact the machine performance favourably. On top of that, the newly designed housing reduces forces on apex seal and reduces the wear of the seal and the housing surfaces. Knowing that the apex seal is prone to damage and failure, Rose and Yang [9] took on the challenge of redesigning the seal. They presented two approaches, a wider apex seal and a multi-apex-seal, designed to provide more stable configurations and improve seal effectiveness.