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DRAG REDUCTION EXPERIMENTS WITH ROTATING CASINGS IN LIQUID RING VACUUM PUMPS

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ABSTRACT

Liquid ring vacuum pumps and compressors operate with a continuous high velocity circulation of liquid within a cylindrical casing. A large part of the power loss in liquid ring compressors can be attributed to the frictional drag of the fluid against the stationary wall. Experiments are described with a vacuum pump design in which the casing is supported on fluid film bearings and can spin freely. Testing of the device has indicated a marked increase in efficiency of the unit.

Comparisons of the actual liquid ring vacuum pump performance with and without a rotating casing are made. The actual results are compared to predicted results obtained from a computer model for liquid ring performance analysis. The basis for this model has been previously reported in (1). The model has been adjusted to account for the effect of the rotating casing wall.

NOMENCLATURE

\( A_s \) shear area \( m_r \) mass flow in radial direction
\( b \) axial length of body \( R \) casing radius
\( C_d \) wall drag coefficient \( r \) rotor radius
\( c_m \) max velocity outside rotor \( \text{Rey} \) Reynolds’ number
\( f_{\text{end}} \) loss due to end wall friction \( U \) rotor peripheral speed
\( f_m \) turbulence mixing loss \( v_r \) radial velocity inside rotor
\( f_s \) fluid shear loss \( v_{r0} \) radial velocity outside rotor
\( f_w \) outer wall friction loss \( w_L \) wall length
\( K_{\text{end}} \) exp. end wall constant \( \rho \) liquid density
\( K_w \) exp. wall friction constant \( \omega \) angular velocity
\( K_m \) exp. mixing loss constant \( \gamma \) kinematic viscosity
\( K_s \) exp. shear loss constant

INTRODUCTION

Liquid ring vacuum pumps and compressors are used in many industries because they are extremely reliable and versatile. They have unique advantages such as near isothermal compression, the ability to compress a wet gas, and the ability to handle liquid slugs along with the gas. These advantages have solidified their position over the years as the optimal choice for many significant and difficult industrial applications.
The principle of operation is explained with the aid of the cross section shown in Figure 1. The rotor is mounted with a center offset from the center of the body. The cone has passages communicating with the head to direct the gas and liquid in and out of the machine. Provision is also made for supplying a continual flow of liquid to the pump. When the rotor turns at sufficient speed, a liquid ring is formed and it spins at approximately the same rotational velocity as the rotor. Because of the eccentricity, the inner surface of the ring moves in and out relative to the rotor. In effect, it acts like a liquid piston that draws air into the rotor on the intake side, compresses the air in the compression zone, and pushes the air (and water) out on the discharge side. In the transition zone, the rotor is closest to the casing and any residual gas not purged out the discharge is carried to the inlet side.

In conventional designs the body or casing is stationary. Since the liquid ring circulates against the surface continually at relatively high velocity the fluid drag against this surface usually represents the largest loss in liquid ring pumps. A possible solution to this problem is to allow the casing to rotate. The incentive for improved efficiency can be understood even on the basis of one common application of liquid ring pumps - in paper mills. The large majority of paper machines in the world use liquid ring pumps to produce the vacuum used to de-water the paper sheet as it is formed. Large paper machines run with several thousand installed horsepower (KW) of liquid ring pumps. Obviously, an improvement in efficiency will have a positive impact on life cycle cost.

DESCRIPTION OF PROTOTYPE

A cross sectional view of the prototype used to conduct the experiments is shown in Figure 2. This is a heavily modified version of a standard or stock model AHC 80 pump manufactured by Nash. Only the rotor and shaft are standard the remaining parts being modified to allow the casing to spin freely on a fluid film. The rotor diameter is 7.76 inches (19.7 cm). The casing is constructed with an end wall on one side and removable cover plate on the other side for assembly around the rotor. It is supported in rotation by a series of hydrostatic bearings in the surface of the outer body. The bearing fluid is supplied to the bearings via the cylindrical body constructed from two concentric cylinders with a plenum between them. There are a series of hydrostatic fluid bearing recesses equally spaced around the inner diameter of the outer body assembly.

The casing and removable end plate is shown in Figure 3. It resembles an open "canister" with a cover and in fact that terminology will be used in the paper.
The outer body assembly is shown in Figure 4. Some detail of the individual hydrostatic bearing recesses can be seen.

The end plate containing the cone port is shown in Figure 5. The axial thrust on the canister is directed towards this end; it is also fitted with a series of hydrostatic bearings to maintain the axial position of the canister.

These drawings and pictures show the prototype in its "final" configuration. It arrived in that state through a series of modifications. In the initial configuration (not shown) the rotating...
canister was constructed without end walls, i.e., it was a simple cylinder or sleeve. In that configuration the end plates did not include the hydrostatic bearing structure since there was essentially no axial thrust acting on the sleeve.

The bearing fluid in initial experiments was water, which is the typical liquid used in liquid ring pumps. The spent bearing water simply ran into and mixed with the liquid ring water. Experiments were also run with air as the radial bearing fluid. The spent air also flowed into the pump where it combined with the air volume being compressed by the machine.

**TEST RESULTS**

As mentioned the initial configuration of the prototype incorporated a simple sleeve without end walls. The bearing fluid was water with an anti-rust preservative (tradename "Panol"). Figure 6 illustrates the increase of efficiency (in relative terms) obtained in this first series of tests. The efficiency term refers to actual volume pumped (ACFM or M³/min) divided by the shaft power consumption (BHP or kW) of the unit. All comparisons are made at 10 in. Hg absolute (0.34 Bar). The sleeve is supported on the fluid film structure and is driven by the drag exerted by the liquid ring; hence the relative velocity is decreased between the liquid ring and the casing wall resulting in a reduction in drag losses. At a mid-range speed the increase is approximately 12 per cent over the baseline curve. The baseline is the standard production pump constructed with the same rotor and overall parameters.

The prototype was then modified to the configuration illustrated in Figure 2. The rotating sleeve was fitted with end walls; hence the outer periphery and end walls exposed to the liquid ring could rotate due to the drag of the liquid ring. The result of this test is also shown on Figure 6. As can be seen there was a further large increase in efficiency. At a mid-range point the improvement is 43 per cent compared to the baseline. The speed of the canister was measured to be 40 to 42 per cent of the rotor speed.

The prototype was then modified to allow operation with pressurized air as the radial bearing fluid (the end wall bearing was maintained with water flow). The modification consisted primarily of the addition of restricting orifices to each of the bearing recesses and a reduction of canister running clearance. The changes were made to optimize the bearing parameters for air. The air was supplied to the outer body plenum at approximately 80 psig (6.5 Bar). The spent bearing air flowed into the compression section of the pump where it mixed with and was discharged with the pumped volume of air. The air bearing allowed essentially drag free rotation of the canister.

The results with air operation are shown in Figure 7. The top curve represents the ultimate efficiency obtained in this configuration. The gain in efficiency was a remarkable 66 per cent above the same reference point. Also, as can be seen the air operated unit was able to run at more than twice the speed of the baseline unit with improved efficiency. The rotational speed of the canister was measured at 83 to 88 per cent of rotor speed. Given the nearly frictionless support of the canister, the rotating speed is probably representative of the overall average liquid ring speed.
Figure 6 - Efficiency comparison of sleeve versus canister

Figure 7 - Comparison of water and air operated bearing

COMPARISONS OF ACTUAL RESULTS TO THE MODEL

A computer model for predicting the performance of liquid ring pumps has been previously reported (1). The reader may refer to the referenced article for a full description, but briefly described the liquid ring compressor is represented as a series of adjacent control volumes...
which completely define the space occupied by gas or liquid within the rotor and body. In the tangential direction stations are established at a sufficient number of locations to describe a full revolution. At each of these stations, the fluid losses are calculated and mass and energy balances observed.

The outer wall friction loss $f_w$ is calculated based on flat wall correlations as from Schlicting (2). The wall friction in the body has unique aspects and it has been found necessary to apply the experimental factor $K_w$ based on testing done at Nash.

$$f_w = K_w \rho \omega_1 b C_d c_m^3$$

The drag coefficient $C_d$ is calculated based on Schultz-Grunow as reported in White (3):

$$C_d = 0.427 / (\log \text{Rey} - 0.407)^{2.64}$$

This model was modified to simulate operation with a rotating casing with ends by replacing the velocity term $c_m$ with the term $\delta c$ calculated as the difference between the maximum ring velocity and the casing peripheral speed. The casing peripheral speed measured in the actual tests was used in these calculations.

The end wall friction loss is calculated using an equation originally developed for a disk rotating in an infinitely extended body of fluid (4).

$$f_{end} = K_{end} \pi (R^2 - r^2) \rho \omega^{0.7} \delta c^{2.6} \gamma^{0.2}$$

where $K_{end}$ is an experimental constant that has been adjusted for liquid ring pumps.

The above two losses appear to be dominant in conventional liquid ring operation.

The model results for the standard pump and the air operated prototype are shown in Figure 8. Here it can be seen that the model predicts very well for the standard pump, but it predicts considerably higher efficiency than was observed for the rotating casing. For instance at the reference point the model efficiency is 24 per cent higher than actual observed efficiency. Part of the difference is the loss of pumped volume due to the bearing airflow that is not accounted for by the model.

Figure 9 illustrates the very large reduction of wall friction losses due to the casing rotation as predicted by the model. Note that the wall friction loss shown is the total of outer and end wall losses.

The model calculates two other significant losses termed as mixing and shear losses and described more fully in (1). Briefly, the injection of higher momentum fluid from the rotor into the outer ring to accelerate the ring liquid results in a loss of kinetic energy and irreversible losses. The mixing loss is modeled as follows:

$$f_M = K_M m_t c_m (v_t - v_{r0})$$
Figure 8 - Comparison of actual results with model

Figure 9 - Model predictions of compression work and wall friction loss

Figure 10 - Model predictions of mixing and shear loss
As can be seen, the mixing loss is proportional to the product of the liquid ring velocity and the difference of radial velocities within and outside the rotor periphery. The radial velocities are taken relative to the rotor center.

Apart from the mixing loss, it is assumed that a viscous shear loss also occurs because the velocity gradient of the flow at the rotor periphery can be quite large. Especially in the compression and discharge zones, the velocity of the liquid ring outside the rotor periphery may fall below the rotor tip speed because of the conversion of velocity head to pressure rise. The viscous shear loss is represented by:

$$f_s = K_s \rho A_s U^s \text{abs}(U - c_m)$$

The shear loss is proportional to rotor peripheral area, peripheral speed raised to the power s, and the absolute value of the difference between the rotor peripheral speed and the speed of the liquid outside the rotor. The constant $K_s$ and exponent s have been determined from experimental analysis.

Figure 10 shows model predictions for the mixing and shear losses for the case of the rotating casing and standard pump. As can be seen the losses are greatly reduced for the case of the rotating casing. This is probably due to the overall reduction in velocity differentials between the liquid within and outside the rotor periphery.

CONCLUSIONS

The model results appear to confirm the large efficiency gains that were observed in the test program. Clearly the concept of a rotating casing in a liquid ring pump adds complexity and cost to the basic liquid ring pump design. Also, there are significant design and reliability issues in scaling the concept up to larger machines in industrial service. However, these tests and analysis have confirmed that the potential exists for moving liquid ring pumping speeds and efficiencies to a higher plane.

REFERENCES