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ANALYSIS AND DESIGN OF A UNIFORM-CLEARANCE, PUMPING-RING ROD SEAL FOR THE STIRLING ENGINE

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ANALYSIS AND DESIGN OF A UNIFORM-CLEARANCE,

PUMPING-RING ROD SEAL FOR THE STIRLING ENGINE*

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SUMMARY

A uniform-clearance pumping ring, as opposed to the conventional taper-clearance one, is described. The uniform-clearance concept eliminates complex elastohydrodynamic problems and enables a simple analytical treatment to be made. An analytical expression is derived for the pumping rate showing the effect of various design parameters on the pumping ring's performance. An optimum clearance is found by which the pumping rate is maximized and a numerical example is presented to demonstrate the potential of the uniform-clearance design.

INTRODUCTION

Due to the energy crisis of the 1970's, a great deal of interest has been shown in development of the Stirling engine as an alternate to

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the internal combustion engine. One of the major problems in the

Stirling engine development is the sealing of the high pressure working

fluid (ref. 1). Towards this end several reciprocating seals are used,

with various amount of success, including a pumping ring which forms

the external seal (ref. 2). The pumping ring (fig. 1) is a cylindrical

bushing forced by some means against the reciprocating rod. The ring

is either pressed conically or given an inside radial taper to provide

a converging gap in the pumping direction. The inside radial taper is

considered to be the key factor for successful operation of the pumping

ring. As the rod translates in the pumping direction, large hydro-

dynamic pressures are developed in the thin tapered fluid film between

the ring and the rod. The hydrodynamic forces expand the ring and allow

fluid to be dragged through the clearance space. On the return stroke,

the hydrodynamic effect reduces the pressure allowing the ring to scrape

on the rod and prevent fluid from leaking back.

Performance analysis of the pumping ring described above is a for-

midable task. The pressures developed in the clearance between the ring

and the rod depend on the shape of the clearance. However, due to ring elasticity the clearance depends on the hydrodynamic pressure generated by the rod motion. Hence, the problem is an elastohydrodynamic one for which the hydrodynamic and elasticity governing equations have to be solved simultaneously. Only few attempts have been made so far to solve this difficult problem (refs. 3 to 6). These solution methods tend to be either restrictive or very complex.

The essence of pumping effect in any pumping-ring seal is simply a cyclic clearance variation which permits large amount of viscous-shear flow during the pumping stroke and restricts the leakage during the return stroke. Hence, the key factor in pumping ring design should be the proper clearance variation rather than the means by which it is achieved. The radial taper pumping ring utilizes a relatively simple hydrodynamic mechanism to generate the clearance variation. Unfortunately, this simplicity is overridden by the complexity of calculating clearance variation, making performance prediction very questionable.

This paper presents the analysis and design of a uniform-clearance pumping ring. The clearance variation is accomplished through means

other than hydrodynamic pressures, hence, eliminating the complexity of

the elastohydrodynamic problem. One possible mechanism is shown in

Figure 2 where an annular cavity on the back of the ring is properly

pressurized in some cyclic mode synchronized with the rod motion. By

maintaining a uniform radial clearance between the ring and the rod, and

thus eliminating the hydrodynamic pressures, the design and performance

analysis of the pumping ring become simple.

NOMENCLATURE

a   rod stroke

C   radial clearance

C_{in} initial clearance

C_p^* optimum clearance, equation (8)

E   modulus of elasticity of ring material

h   ring wall thickness

L   ring length

P   preload maximum pressure

p   cavity pressure

p_c clamping pressure

p_w working fluid pressure

Q   pumping rate

Q' net flow per cycle

$\bar{Q}$ dimensionless pumping rate, $6Q/\omega RaC_p^*$

q   flow rate per unit of circumferential leng:h

R   rod diameter

r_i, r_o ring inner and outer radii

T   cycle period

t   time

V   rod velocity

δ radial deflection

μ viscosity

ν Poison ratio

σ_t, σ_r tangential and radial stresses

ω reciprocating frequency

Subscripts:

p   pumping stroke

r   return stroke

PUMPING RATE ANALYSIS

Consider the pumping ring of Figure 2, the instantaneous flow rate

per unit of circumferential length into the high pressure region is:

$$q = \pm V \frac{C}{2} - \frac{c^3}{12\mu} \frac{P_w}{L}$$

(1)

where the $+V$ and $-V$ are for the pumping stroke or the return stroke,

respectively. Designating the pumping and return strokes by the sub-

scripts p and r, respectively, the clearance during the pumping stroke

will be C_p and during the return stroke C_r. The net flow into the high

pressure side is

$$Q' = 2\pi R \int_{0}^{T/2} (q_p + q_r)\ dt$$

(2)

where T the cycle period is

$$T = \frac{2\pi}{\omega}$$

(3)

and ω is the reciprocating frequency. Substituting equation (1) in

equation (2) and noting that

$$\int_{0}^{T/2} V \, dt = a \tag{4}$$

where a is the rod stroke. The net flow into the high pressure side

during one complete cycle is

$$Q' = 2\pi R \left[\frac{a}{2}(C_p - C_r) - \frac{p_w}{12\mu L} \frac{T}{2}(C_p^3 + C_r^3) \right] \tag{5}$$

Dividing equation (5) by the period T and substituting equation (3),

we finally have the pumping rate in the form

$$Q = \frac{\omega R}{2} \left[a(C_p - C_r) - \frac{\pi p_w}{6\mu L\omega}(C_p^3 + C_r^3) \right] \tag{6}$$

An optimum clearance C_p can be found which will maximize the pumping

rate for any preselected C_r. Differentiating equation (6) with respect

to C_p and equating to zero we find

$$a - \frac{\pi p_w}{2\mu L\omega} \, C_p^2 = 0 \tag{7}$$

thus

$$C_p^* = \left(\frac{2\mu L\omega a}{\pi p_w} \right)^{1/2} \tag{8}$$

Equation (8) provides the optimum pumping-stroke clearance C_p, which

ensures maximum pumping rate for the pumping ring of a specified C_r.

This optimum clearance depends on the stroke, a, and reciprocating fre-

quency, ω, of the rod; on the length, L, of the pumping ring; and on the

viscosity, μ, and pressure, p_w, of the working fluid. It is clear from

equation (6) that a clearance C_p too small or too large can alter

drastically the pumping rate. If C_p is close to C_r, the pressure in-

duced leakage can overcome the viscous shear pumping and the high pres-

sure oil leaks through the gap. Also, when increasing C_p the pressure

induced leakage increases as a cubic function of C_p while the viscous

shear pumping increases only linearly. Hence, a value of C_p may be

reached where again the high pressure oil leaks to the low pressure side.

The preceding discussion demonstrates the importance of an accurately

controlled pumping-stroke clearance, C_p. It is difficult to achieve fine

control by hydrodynamic pressures associated with the tapered pumping

ring design (refs. 3 and 4). In fact, the hydrodynamic force may even

expand the ring too much thereby destroying the whole sealing effect.

Equation (6) can be arranged in the form

$$Q = \frac{R}{2} \omega a C_p \left\{ 1 - \frac{C_r}{C_p} - \frac{\pi p_w}{6 \mu L \omega a} C_p^2 \left[1 + \left(\frac{C_r}{C_p} \right)^3 \right] \right\} \qquad (9)$$

which after using the relation

$$C_p = C_p^* \frac{C_p}{C_p^*} \qquad (10)$$

and substituting equation (8) becomes

$$Q = \frac{\omega R a C_p^*}{6} \bar{Q} \qquad (11)$$

where $\bar{Q}$, the dimensionless pumping rate, is

$$\bar{Q} = 3 \frac{C_p}{C_p^*} - \left(\frac{C_p}{C_p^*} \right)^3 - 3 \frac{C_r}{C_p^*} - \left(\frac{C_r}{C_r^*} \right)^3 \qquad (12)$$

Equation (11) can be expressed in terms of all the design parameters

again using equation (8), hence,

$$Q = \frac{R}{6} (\omega a)^{3/2} \left(\frac{2\mu L}{\pi p_w} \right)^{1/2} \bar{Q} \qquad (13)$$

As can be seen from equation (13), the main factors affecting the pumping

rate are the rod frequency ω and stroke a, and the ratios C_p/C_p^* and

C_r/C_p^*. These two clearance ratios are actually the essence of the uniform-

clearance pumping-ring design. From equation (13) it is clear that in

order to have a positive pumping rate the dimensionless parameter $\bar{Q}$

must be positive; that is,

$$3 \frac{c_p}{c_p^*} - \left(\frac{c_p}{c_p^*}\right)^3 - 3 \frac{c_r}{c_p^*} - \left(\frac{c_r}{c_p^*}\right)^3 > 0 \qquad (14)$$

For an optimum design when $c_p = c_p^*$ the solution of equation (14) gives

$$\frac{c_r}{c_p^*} < 0.596$$

Hence, in addition to designing an optimum pumping-stroke clearance c_p^*, as given by equation (8), the return-stroke clearance c_r should be limited by $c_r < 0.596 \, c_p^*$.

The maximum pumping rate can be achieved (see eqs. (12) and (13)) when $c_p = c_p^*$ and $c_r = 0$. However, to avoid rubbing which can result in excessive friction loss and wear, it is preferable to operate the pumping ring with a ratio $c_r/c_p^* > 0$. Figure 3 presents the effect of the return-stroke clearance ratio c_r/c_p^* on the pumping rate. As can be seen from Figure 3, the ultimate maximum pumping rate $\overline{Q}$ corresponding to $c_r/c_p^* = 0$ and $c_p/c_p^* = 1$ is $\overline{Q} = 2$. Increasing the return-stroke clearance ratio to $c_r/c_p^* = 0.1$ reduces the maximum pumping rate by only 15 percent to $\overline{Q} = 1.7$.

DEFORMATION ANALYSIS

As was shown in the previous sections, the pumping capacity of the uniform-clearance ring is based on maintaining different radial clearances during the pumping and return strokes. To that end, the ring has to be deflected so as to create this necessary difference. If initially the radial clearance between the ring and rod is C_{in} then the required radial deflections δ_p and δ_r corresponding to the pumping and return strokes, respectively, are

$$\delta_p = C_p - C_{in} \tag{15}$$

$$\delta_r = C_r - C_{in} \tag{16}$$

A radial deflection of the ring can be achieved by subjecting it to a uniform external pressure (see fig. 2). Under such conditions the tangential and radial stresses at the inner radius of the ring, r_i, are

$$\sigma_t = - \frac{2pr_o^2}{r_o^2 - r_i^2} \tag{17}$$

and

$$\sigma_r = 0 \tag{18}$$

The radial deflection is

$$\frac{\delta}{r} = \frac{\sigma_t}{E} - \nu \frac{\sigma_r}{E} \tag{19}$$

Hence, by equations (17), (18) and (19) the radial deflection of the

inner radius of the ring is

$$\delta = - \frac{p}{E} \frac{2r_o^2 r_i}{r_o^2 - r_i^2} \tag{20}$$

Using equations (15), (16) and (20) we can calculate the required cavity

pressures p_p and p_r for the pumping and return strokes, respectively.

Thus,

$$p_p = \frac{EC_p}{2r_i} \left(\frac{C_{in}}{C_p} - 1 \right) \left[1 - \left(\frac{r_i}{r_o} \right)^2 \right] \tag{21}$$

$$p_r = \frac{EC_p}{2r_i} \left(\frac{C_{in}}{C_p} - \frac{C_r}{C_p} \right) \left[1 - \left(\frac{r_i}{r_o} \right)^2 \right] \tag{22}$$

From equations (21) and (22) it can be seen that the cavity pressures are

mostly affected by the ratios C_{in}/C_p, C_r/C_p and r_i/r_o. The required

pressures can be reduced by making $C_{in} = C_p$; in which case the pumping-

stroke cavity pressure is zero, and the return-stroke cavity pressure is

$$p_r = \frac{EC_p}{2r_i} \left(1 - \frac{C_r}{C_p}\right)\left[1 - \left(\frac{r_i}{r_o}\right)^2\right] \qquad (23)$$

When the engine is not running the pumping ring has to function as a static seal. This means that the ring has to be clamped on the rod so that the clearance between them becomes zero or there is even an interference fit. To achieve this, the cavity pressure during stand still has to be increased to a value that will make $\delta > -C_{in}$.

Hence, from equation (20) and for $C_{in} = C_p$

$$p_c > \frac{EC_p}{2r_i} \left[1 - \left(\frac{r_i}{r_o}\right)^2\right] \qquad (24)$$

For simplicity of the pressurization system, the working-fluid pressure, p_w, can serve as the clamping pressure p_c provided that $p_w > p_c$.

The preceding deflection analysis was based on the assumption that the ring has a plain cylindrical bore. However, after assembling in the engine, an initially plain cylindrical ring is distorted by the internal hydrostatic pressure distribution as shown in Figure 4(a). Thus, to ensure a plain cylindrical bore after assembly, the ring must be preloaded

by an external load that produces an equal, but opposite, deflection

to that generated by the internal pressure. Since the internal pres-

sure is not uniform but drops linearly from p_w to zero, the external

load would have to result in a similar linear drop from some value P

to zero as shown in Figure 4(b). The pressure P can be calculated

from the requirement of zero radial deflection at the inner radius of

the ring under the combined action of both P and p_w. If P and

p_w were uniform, the tangential and radial stresses of the ring inner

radius would be (ref. 7)

$$\sigma_t = \frac{p_w r_i^2 - P r_o^2 - (P - p_w) r_o^2}{r_o^2 - r_i^2} \tag{25}$$

$$\sigma_r = \frac{p_w r_i^2 - P r_o^2 + (P - p_w) r_o^2}{r_o^2 - r_i^2} \tag{26}$$

Substituting equations (25) and (26) in equation (19) and requiring

$\delta = 0$ at $r = r_i$, we find

$$(p_w r_i^2 - P r_o^2)(1 - \nu) - (P - p_w) r_o^2 (1 + \nu) = 0$$

or

$$P = \frac{p_w}{2} \left[1 + \nu + \left(\frac{r_i}{r_o}\right)^2 (1 - \nu) \right] \tag{27}$$

The linear drop of the external load from P at the high pressure side to zero at the low pressure side can be obtained by means of a conical press or shrink fit. In this case the outer radius, r_o, in equations (17) through (24) is that of the external member pressed or shrink fitted on the ring.

NUMERICAL EXAMPLE

The following numerical data is used for a pumping ring design example

rod radius, R $-------------$ 2 cm

rod stroke, a $------------$ 5 cm

ring length, L $---------$ 2.5 cm

engine frequency, ω $--------$ 377 rad/sec (3600 rpm)

oil pressure, p_w $--------$ 10 MPa

oil viscosity, μ $----------$ 5 mPa.s (AE 30 at 112° C)

ring modulus of elasticity, E $---$ 111 GPA (bronze)

The optimum pumping stroke clearance is calculated from equation (8) and is

$$c_p^* = 12.25 \text{ }\mu m$$

Selecting a clearance ratio $C_r/C_p^* = 0.1$ the pumping rate is found

from equations (11) and (12)

$$Q = 1.31 \text{ } cm^3/sec = 78.50 \text{ } cm^3/min$$

Selecting initial clearance $C_{in} = C_p^*$ the required cavity pressure

during the pumping stroke is (eq. (21)) $p_p = 0$. The cavity pressure

required for $C_r = 0.1 \text{ } C_p^*$ during the return stroke is calculated from

equation (23) and the clamping pressure p_c from equation (24). Table

1 presents values of the pressures p_r and p_c calculated for various

wall thickness values of the composite ring. As can be seen from Table

1 a ring having wall thickness of 3 mm will require a cavity pressure

$p_r = 7.5$ MPa during the return stroke, and a clamping pressure $p_c =$

8.3 MPa (which is less than p_w) when the engine is not running.

The ring can be made by press-fitting an outer member with wall

thickness of 1 mm on an inner member having a wall thickness of 2 mm.

The conical press fit will require a maximum pressure P which, by

equation (27) and for an inner member radius ratio of $r_i/r_0 = 20/22$, is

$$P = 9.4 \text{ MPa}$$

The composite ring is subjected to an oscillating load because p_r is being applied at the engine frequency. The maximum tangential stress generated by p_r at the inner radius of the ring is, by equation (17),

$$\sigma_t = -2p_r \frac{r_o^2}{r_o^2 - r_i^2} \tag{28}$$

Using the values $p_r = 7.5$ MPa, $r_i = 2$ cm, and $r_o = 2.3$ cm, we find from equation (28)

$$\sigma_t = -61.5 \text{ MPa} \tag{29}$$

In addition to this repeated stress, the inner member of the ring is loaded by P and p_w. The maximum stresses at the inner radius due to these pressures are calculated from equations (25) and (26) where now $r_o = 2.2$ cm. The result is

$$\sigma_t = -16 \text{ MPa} \tag{30}$$

$$\sigma_r = -7 \text{ MPa} \tag{31}$$

Hence, the tangential stress amplitude is -30.75 MPa and the mean tangential stress is -44.75 MPa. The radial stress amplitude is zero and the mean radial stress is -7 MPa. The mean and alternating

Von Mises stresses are, therefore (ref. 8) 48.6 MPa and 7 MPa, respectively. Available fatigue test data of copper-base alloys indicate bending fatigue-strength to tensile-strength ratio of about 0.35 which for hard bronze means fatigue strength of about 145 MPa. Since the pumping ring is under compression, it seems that the design is safe.

RESULTS AND DISCUSSION

The optimum design resulted in a pumping-stroke clearance $C_p^* =$ 12.25 μm. For this clearance a pumping rate $Q = 78.5$ cm^3/min can be obtained when the return-stroke cavity pressure is $p_r = 7.5$ MPa which ensures clearance ratio $C_r/C_p^* = 0.1$.

Any change in p_r will, of course, change the pumping rate due to the effect of p_r on the clearance ratio C_r/C_p^*. The pumping-stroke clearance C_p may also differ from the optimum value of C_p^* due to manufacturing tolerances. Hence, it is of interest to examine off-design pumping rates for various values of pumping-stroke clearances C_p and return-stroke cavity pressures p_r. For an easy

comparison equations (11), (12) and (23) can be rearranged in the form

$$Q = \frac{\omega R a C_p^*}{6} \left[3\frac{C_p}{C_p^*} - \left(\frac{C_p}{C_p^*}\right)^3 - 3\frac{C_r}{C_p^*} - \left(\frac{C_r}{C_p^*}\right)^3 \right] \qquad (32)$$

and

$$\frac{C_r}{C_p^*} = \frac{C_p}{C_p^*} - \frac{2r_i P_r}{E C_p^* \left[1 - \left(\frac{r_i}{r_o}\right)^2 \right]} \qquad (33)$$

Where C_p^* is the optimum pumping-stroke clearance given in equation

(8) as

$$C_p^* = \left(\frac{2\mu L \omega a}{\pi P_w}\right)^{1/2}$$

For any ratio C_p/C_p^*, equation (33) is used first to calculate C_r/C_p^*

as function of both C_p/C_p^* and P_r. The result is then used in equ-

ation (32) to calculate the pumping rate.

Figure 5 presents pumping rates, for the pumping ring of the

numerical example, as a function of the cavity pressure P_r at various

pumping-stroke clearance values. The clearances C_p and C_r are

normalized by C_p^* where $C_p^* = 12.25$ μm. As can be seen from Figure

5, at each given pumping-stroke clearance, C_p, there are two distinct

cavity pressures p_r corresponding to the cases $Q = 0$ and $C_r = 0$, respectively. If the return-stroke cavity pressure, p_r, is below the minimum value for $Q = 0$, then the ratio of the return-stroke to pumping-stroke clearances C_r/C_p becomes too large, the pressure induced flow overcomes the viscous-shear pumping and the working fluid starts leaking through the clearance gap. Increasing the pressure p_r reduces the return-stroke clearance C_r and, therefore, the clearance ratio C_r/C_p. The result is an increase in the pumping rate. The pressure p_r and, hence, the pumping rate can be increased up to the point where C_r vanishes. At this point the clamping pressure p_c is reached and further increase of p_r will not affect the pumping rate.

The benefit of optimizing the pumping-stroke clearance is obvious. Trying to keep a pumping rate of 78 cm^3/min, which in the design example is obtained when $C_p/C_p^* = 1$ and $C_r/C_p^* = 0.1$, becomes impossible if the ratio C_p/C_p^* is either reduced or increased by about 30 percent with respect to its optimum.

Figure 5 can be used as a design chart for pumping rings activated

by external pressure and having the design requirement of the example.

Similar charts can, of course, be plotted for different designs. How-

ever, for a more general use one would prefer to use Figure 3. As was

mentioned in the Introduction, the essence of any pumping ring is the

cyclic variation of its clearance rather than the means by which this

variation is achieved. Hence, the pumping rate depends completely on

the ratios C_p/C_p^* and C_r/C_p^* regardless of the mechanism causing the

ring deflection. Once the designer has used equation (8) to calculate

C_p^* and he has any possible mean to control C_p and C_r, Figure 3 can

be used to determine pumping rates at any known combination of the

ratios C_p/C_p^* and C_r/C_p^*.

An important result of Figure 3 is the lower and upper limits of

C_p/C_p^* at given ratios of C_r/C_p^*. At $C_r = 0$, for example, C_p/C_p^*

for a positive pumping rate is limited by the value $C_p/C_p^* < \sqrt{3}$. As

C_r/C_p^* increases the range of permissible deviation from the optimum

design $C_p = C_p^*$ is narrowing down. At $C_r/C_p^* = 0.3$, for example, the

operating pumping-stroke clearance must be greater than $0.3\ C_p^*$ but

less than $1.55 \, C_p^*$. This result underlines the importance of controlling the pumping and return-strokes clearances and contradicts the assumption presented in a recently published work (ref. 9) that the higher the value of C_p, the better. In fact in the hydrodynamic pressure activated ring the pumping-stroke clearance may become too large causing failure due to leakage.

CONCLUDING REMARKS

A controlled cyclic variation of pumping ring clearance is the key factor in its successful operation. The radial taper pumping ring utilizes a relatively simple mechanism to cause clearance variation, however, this mechanism involves complex elastohydrodynamic effects. The uniform-clearance pumping ring, on the other hand, needs a somewhat more elaborate auxiliary mechanism but its performance analysis is straightforward.

A uniform-clearance pumping ring for the Stirling engine was analyzed and its capacity demonstrated through a design example. It was found that pumping rates can be maximized by optimization of the

pumping-stroke clearance. It was also found that if the ratio of

return-stroke over pumping-stroke clearance exceeds a critical

value or if the pumping-stroke clearance is too high, the pumping

ring starts to leak.

An external pressure variation synchronized with the rod motion

was suggested for the design example. However, any other means that

can serve the purpose of controlling cyclic clearance variation will

do as well.

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Table 1. - Return Stroke Cavity Pressure p_r and

Clamping Pressure p_c for Various Wall Thickness Values h

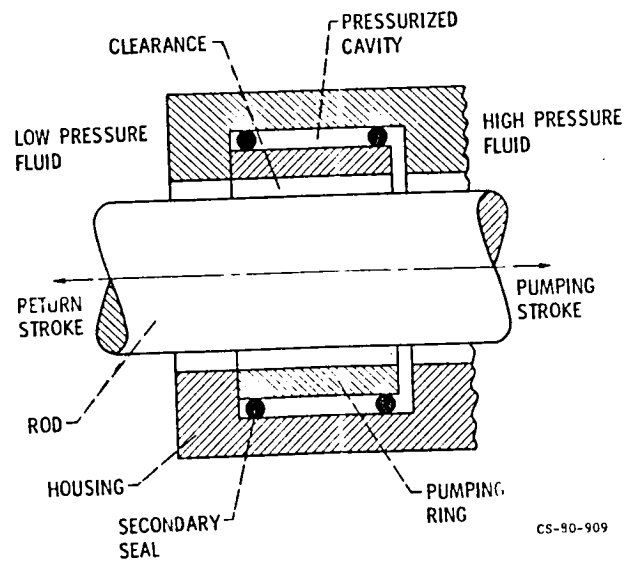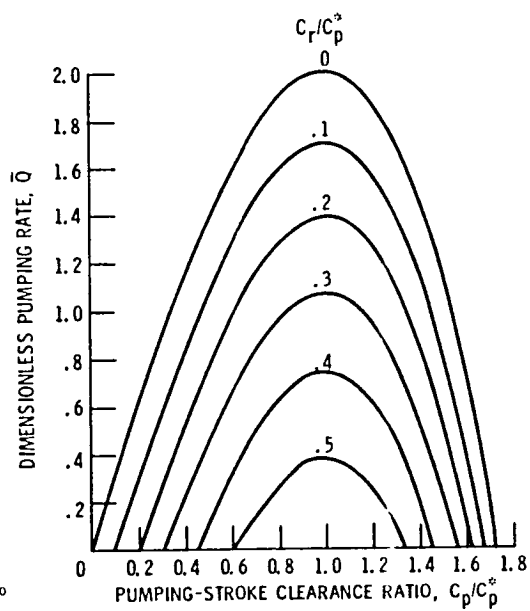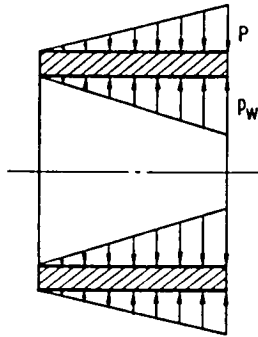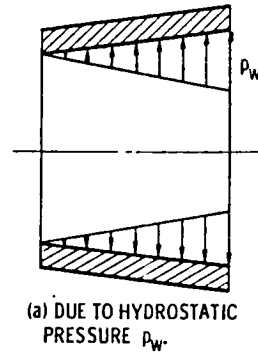| Wall thickness h mm	Return stroke cavity pressure p_r MPa	Clamping pressure p_c MPa
2	5.3	5.9
3	7.5	8.3
4	9.3	10.6
5	11.0	12.2
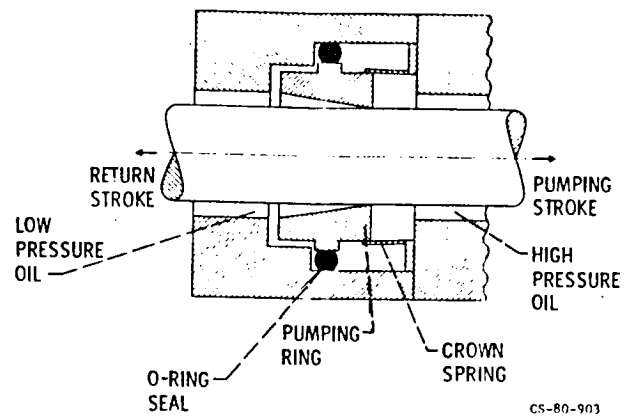
Figure 1. - Pumping ring assembly.

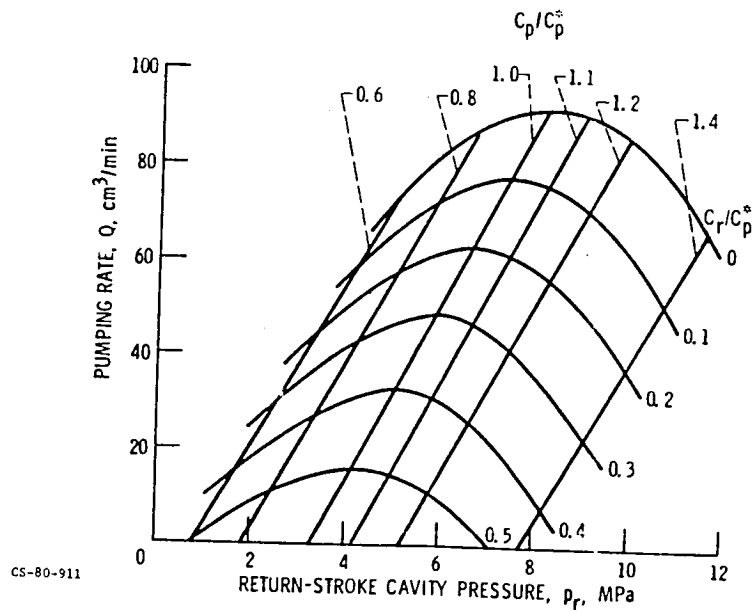Figure 2. - Uniform-clearance pumping ring.

Figure 3. - Dimensionless pumping rate $\overline{Q}$ as function of pumping-stroke clearance ratio C_p/C_p^* for various return-stroke clearance ratios C_r/C_p^*.

CS-80-910

(a) DUE TO HYDROSTATIC
PRESSURE p_w.

(b) DUE TO COMBINED HYDRO-
STATIC PRESSURE p_w AND
PRELOAD P.

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Figure 4. - Cylindrical ring
deflection.

Figure 5. - Pumping rate, Q, as function of cavity pressure, p_r, for
various clearance values, C_p/C_p^*.

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15. Supplementary Notes

16. Abstract

A uniform-clearance pumping ring, as opposed to the conventional taper-clearance one, is described. The uniform-clearance concept eliminates complex elastohydrodynamic problems and enables a simple analytical treatment to be made. An analytical expression is derived for the pumping rate showing the effect of various design parameters on the pumping ring's performance. An optimum clearance is found by which the pumping rate is maximized and a numerical example is presented to demonstrate the potential of the uniform-clearance design.

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