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A loose-ring lubricator model

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Abstract: The loose-ring lubricator is used as a pump within self-contained journal bearings to deliver lubricating fluid from a reservoir or sump to the bearing. It is desired to understand its behaviour sufficiently to be able to predict the performance of new designs. An analytical model is derived that uses the balance of forces acting on the ring to predict the ring speed. The delivered flowrate is then estimated using viscous lifting theory. The results compare favourably with experimental data and published empirical models. Improvements are also suggested to account for additional influences that might affect actual lubricator operation.

Keywords: self-contained lubricator, journal bearing, ring speed, viscous lifting, delivery flowrate

NOTATION

A	area (m^2)
b	width (m)
B	non-dimensional width $= b/[(Rh_0)^{1/2}]$
Ca	capillary number $= u\eta/\gamma$
C_f	drag coefficient
D	non-dimensional film thickness $= h(\rho g/\gamma)^{1/2}$
F	tangential force (N)
Fp	fluid property number $= \eta(g/\rho\gamma^3)^{1/4}$
g	gravitational acceleration (9.81 m/s^2)
h	film thickness (m)
i	ring immersion depth (m)
I_m	submerged fraction $= [\cos^{-1}(1 - i/r_r)]/\pi$
k_g	geometric factor
K	correction factor
L	length (m)
m	mass (kg)
n	number of contacts
P	radial force (N)
q	flow per unit width (m^2/s)
$\bar{q}$	flowrate (m^3/s)
r	radius (m)
R	reduced radius $= (r_s^{-1} - r_r^{-1})^{-1}$ (m)
Re	Reynolds number $= [u(2r)]/\nu$
s	ring thickness (m)
u	velocity
α	non-dimensional coefficient
β	withdrawal angle
γ	surface tension (N/m)
η	dynamic viscosity (Pa s)
θ	angle in ring coordinates
ν	kinematic viscosity (m^2/s)
ρ	density (kg/m^3)
ϕ	attitude angle
ϕ_r	ring taper angle

Subscripts

1, 2	sequential values
∞	asymptotic value
c	contact
f	film
F	tangential component
o	origin (or overall)
p	perimeter
P	radial component
r	ring
rec	recirculation
s	shaft (or sump)
x	horizontal component
y	vertical component

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1 INTRODUCTION

The loose-ring lubricator consists of a ring-shaped piece of material that rests on a horizontal shaft and is driven by friction as the shaft rotates (Fig. 1). The rotating ring raises a thin film of liquid from the sump and delivers a portion of the flow to the shaft and bearing. This lubricator has the advantages of small size, simple manufacture, low cost, high reliability, low power consumption

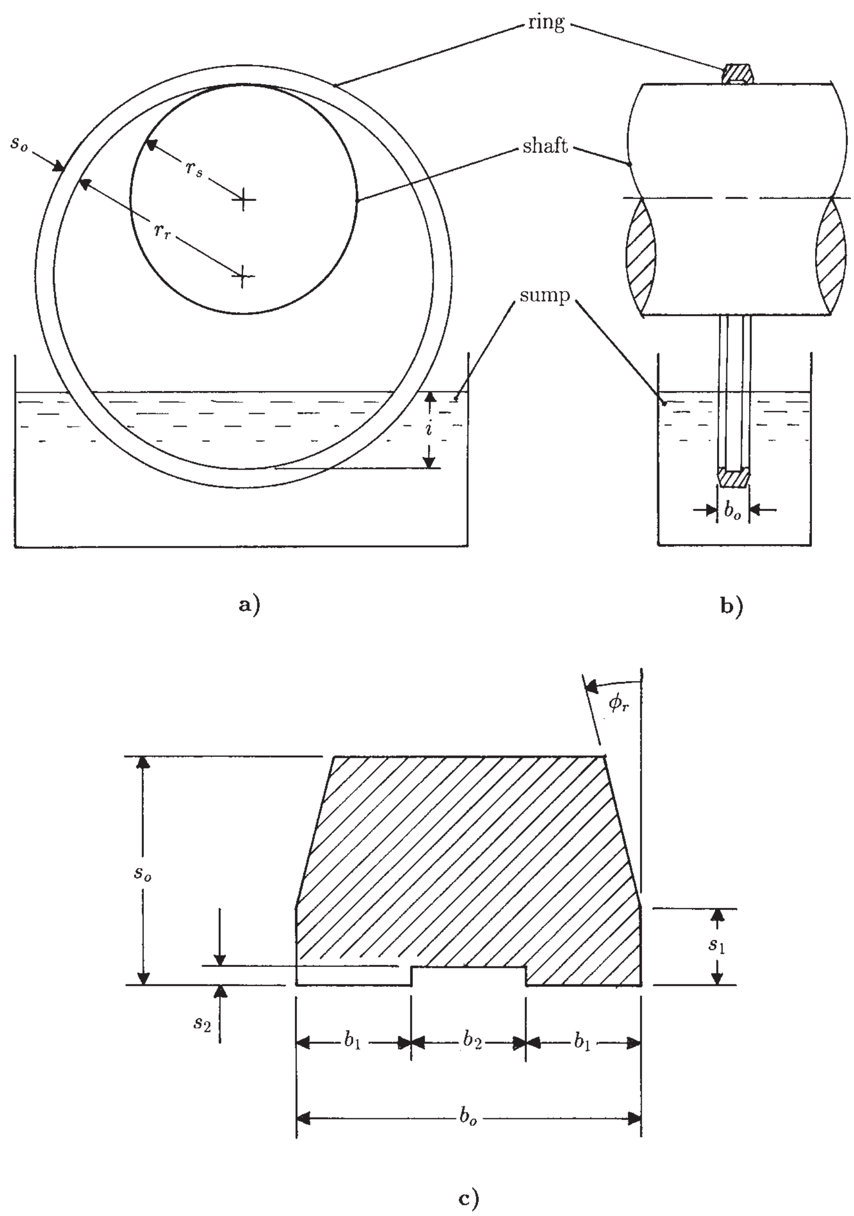

Fig. 1 Definition diagram of a loose-ring lubricator: (a) side view; (b) front view; (c) cross-sectional view

and self-contained operation. It is limited, however, in the capacity to deliver the high lubricant flowrates needed for some bearing designs, i.e. for cooling.

A number of investigators have reported their efforts to improve the capacity and robustness of the loose-ring or 'oil ring' design. The first well-known scientific study of oil rings was by Karelitz [**1**], who published data for several sizes of ring. He varied parameters such as ring immersion, lubricant viscosity, ring weight and cross-sectional geometry in an attempt to see what factors affected the ring speed and hence the delivery flowrate. His study was typical of the experimental/empirical approach adopted by subsequent workers.

The next known study was by Baudry and Tichvinsky [**2**], who observed that grooves cut into the ring inner surface markedly improved the delivery flowrate, a result independently observed by Baildon [**3**]. The former were the first to suggest that the behaviour of a loose-ring lubricator could be predicted using lubrication theory for the ring/shaft contact, along with appropriate data for skin friction in the sump and the work done in lifting the liquid. Their data were used by Hersey [**4**] to produce the first mathematical description of ring delivery flowrate. That same year a group of papers on ring and disc lubricators were presented at a conference on lubrication [**3**, **5**, **6**]. More experimental work appeared from time to time [**7**–**10**]. Lemmon and Booser [**11**] used dimensional analysis along with experimental data to produce empirical equations for ring speed and lubricant delivery flowrate.

The loose-ring lubricator problem was also taken up by researchers, for instance Keysell [**12**], who began to develop basic theoretical analyses. The first known attempt to include some form of viscous lifting in the analysis of a ring lubricator was by Mangles [**13**]. He supposed that the lifted flowrate might be related to the momentum thickness of the boundary layer attached to the ring. He obtained delivery flowrate predictions of the right order of magnitude but with the wrong functional dependence on speed. Davies and Smith [**14**] developed a basic analysis of viscous lifting, leading to the film thickness prediction

$$h = \left(\frac{2u_r\eta}{\rho g}\right)^{1/2} \tag{1}$$

on the lifting side of the ring. No corrections for the observed flow loss mechanisms were developed and so the delivery flowrate predictions were usually substantially in error when compared with their experimental data.

More recently a number of authors have considered the advanced development of self-contained lubricators [**15**–**18**]. A detailed thermal analysis of a ring-lubricated journal bearing was presented by Dowson *et al.* [**19**]. Unfortunately, there was no general analytical estimate of the ring flowrate and it had to be supplied experimentally. It was this problem that led to the present research.

The lack of detailed analysis of the loose-ring lubricator can be traced to the complexity of its operation. The usual approach has been to simplify the problem by dimensional grouping (Buckingham's pi theorem). This is somewhat arbitrary since there has been only a fuzzy understanding of the important primary quantities to begin with. The number of variables affecting oil rings is at least 11, i.e. three fluid properties η, ρ and γ, three ring dimensions r_r, b_o and s_o, ring mass m_r, ring immersion i, gravity g and two shaft variables r_s and u_s. Nonetheless, the authors have developed an analytical approach to all aspects of the problem that has yielded a first-principles model capable of predicting ring speeds and lubricant delivery flowrates. A key feature enabling the latter has been the use of viscous lifting theory developed by physicists and chemical engineers interested in coating processes. The cross-disciplinary nature of this fundamental physical process has been clearly revealed. The building blocks to enable the prediction of important variables have been available for some time but not recognized by plain bearing engineers.

2 ANALYSIS

2.1 Force balance

The dynamic forces acting on a loose-ring lubricator are indicated in Fig. 2 and consist of radial, P, and tangential, F, components. The forces and torques generated by gravity, the n ring/shaft contacts ($n = 1$ for no grooves) and the sump must balance for steady-state conditions, hence the equations

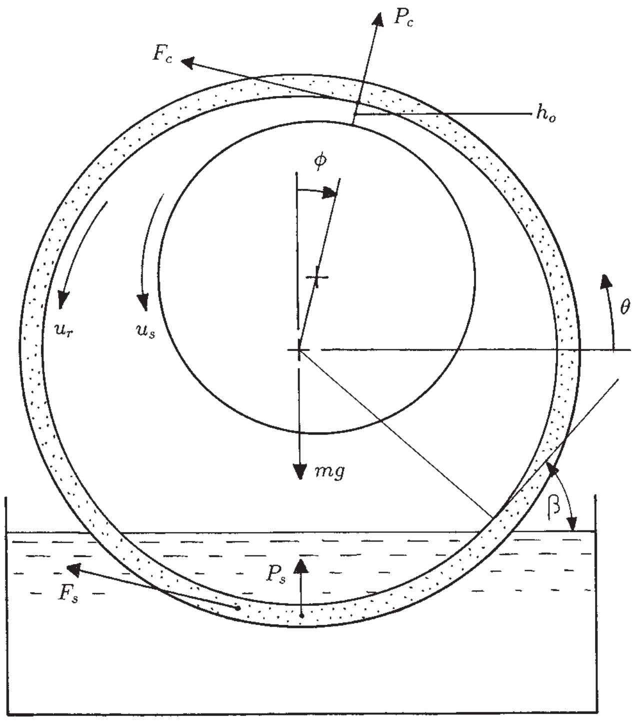

Fig. 2 Forces acting on the ring

$$\text{(Vertical)} \quad 0 = -F_{sy} + P_s - (m_r + m_f)g - nF_{cy} + nP_{cy} \tag{2}$$

$$\text{(Torque)} \quad 0 = -F_s \times (r_r + s_o/2) + nF_c \times r_r \tag{3}$$

where subscripts c and s refer to the contact and sump and subscript y indicates evaluation of the vertical force component. The attitude angle, ϕ, is a measure of the dominance of the ring weight compared with the other forces. This may be evaluated by taking moments about the ring/shaft contact, noting that the mass of the lifted oil is small compared with that of the ring:

$$\text{(Torque)} \quad 0 = -F_s \times 2r_r + m_r g \times r_r \sin\phi \tag{4}$$

$$\phi = \sin^{-1}\frac{2F_s}{m_r g} \approx \frac{2F_s}{m_r g} \tag{5}$$

The model below is valid for $\phi \to 0$, i.e. the sump drag is much less than the ring weight. To simplify the problem, the following assumptions are made:

1. Finite width lubrication theory holds for the ring/shaft contact.
2. Relieved sections of the contact (grooves) generate negligible forces.
3. No solid–solid friction occurs in the contact.
4. Boundary layer friction essentially dominates the sump drag; i.e. the sump recirculation speed, u_{rec}, is much slower than the ring speed, u_r.

5. Sump buoyancy, P_s, lifted film weight, $m_f g$, and surface tension, γL_p, forces are all negligible.
6. The liquid properties are constant; i.e. isothermal conditions exist.

This last condition does not particularly limit the usefulness of the analysis, even if the ring operates in a non-isothermal environment. The ring has both greater contact area and contact time per revolution with the sump than with the shaft. Thus the entire ring and entrained fluid film will be essentially at sump temperature, even if the shaft is not. Only for a large temperature difference between shaft and ring would the isothermal assumption be expected to fail.

The various forces acting on the ring require detailed analyses in themselves, as outlined in the following sections. The ring speed model is then derived from equations (2) and (3) using the predictions for the contact forces and sump drag. Finally, viscous lifting theory is presented and applied to the ring inner surface.

2.2 Ring/shaft contact

The ring/shaft contact may be considered to be a hydrodynamically lubricated contact, assuming that no solid–solid friction occurs. This will be the case above a minimum shaft speed, when the hydrodynamic film becomes thick enough to separate completely the ring and shaft surfaces. An analysis of infinitely wide cylinders in lubricated contact has been presented by Dowson and Higginson [20]. With the addition of finite width correction terms, K, the forces acting on the ring for a single ring/shaft contact of width b_1 may be written as

$$\bar{P}_c = 2.447 K_P \frac{R}{h_o} \tag{6}$$

$$\bar{F}_c = -2.301 K_F \left(\frac{R}{h_o}\right)^{1/2} - 3.490 \left(\frac{u_r - u_s}{u_r + u_s}\right)\left(\frac{R}{h_o}\right)^{1/2} \tag{7}$$

where $\bar{P} = P/[\eta b_1(u_r + u_s)]$ is the non-dimensional radial force, $\bar{F} = F/[\eta b_1(u_r + u_s)]$ is the non-dimensional tangential force, including the web traction downstream of the nip, $R = (r_s^{-1} - r_r^{-1})^{-1}$ is the reduced radius of the contact and h_o is the minimum film thickness. The coefficients are upper limiting values for flooded lubrication conditions at high ratios of R/h_o, i.e. $R/h_o > 10^4$.

The finite contact width, b_1, implies that side leakage will be significant. Corrections to the above equations have been published by Dowson and Whomes [21] on the basis of numerical solution of the full Reynolds lubrication equation [22]. Their side leakage correction data (Figs 3 and 4) have been curve-fitted to yield

$$K_P = w_P\left(\frac{B^2}{9.79}\right) + (1 - w_P)\left(0.959 - \frac{1.98}{B}\right), \quad w_P = \left(1 + \frac{B^6}{100}\right)^{-1} \tag{8}$$

$$K_F = w_F\left(\frac{B^2}{16.57}\right) + (1 - w_F)\left(0.974 - \frac{3.86}{B}\right), \quad w_F = \left(1 + \frac{B^5}{500}\right)^{-1} \tag{9}$$

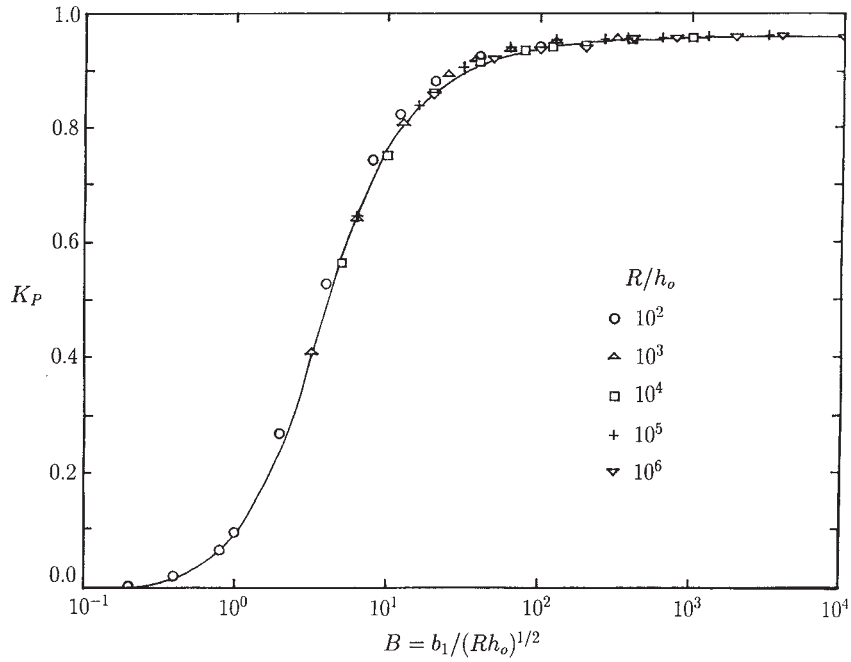

Fig. 3 Radial force side leakage correction factor K_P [21]

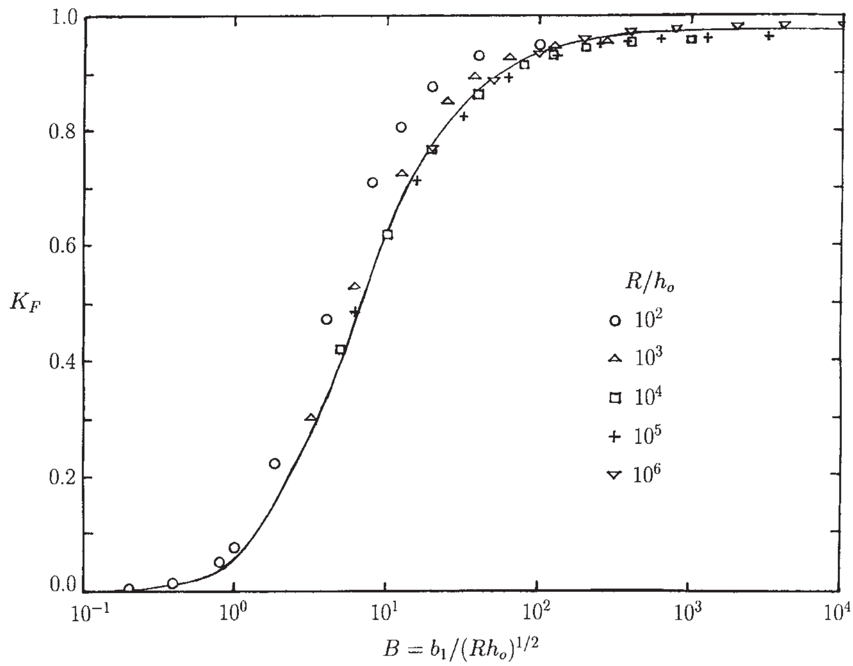

Fig. 4 Tangential force side leakage correction factor K_F [**21**]

where $B = b_1/(Rh_o)^{1/2}$ is the non-dimensional contact width and w is a weighting function with the range $0 < w < 1$. As $w \to 1$ ($B < 1$) equations (8) and (9) reduce to expressions for 'short' contacts and for $w \to 0$ ($B > 10$) they reduce to expressions for 'wide' contacts. As there is some dependence on R/h_o, the curves were fitted to data corresponding to the larger R/h_o values, i.e. $R/h_o \to \infty$. Technically, a correction should also be applied to the second term of equation (7). However, the exact coefficient for the 'short' contact is 3.332, requiring a correction of less than 5 per cent.

2.3 Sump model

For convenience, the sump supplying fluid to the rotating ring may be approximated as an infinite reservoir. For this condition, the fluid drag on a body may be written as

$$F = C_f \rho \frac{u^2}{2} A \tag{10}$$

where C_f is the drag coefficient, ρ is the fluid density, u is the bulk flow speed and A is the surface area. Neglecting the curvature of the ring surfaces, the flat plate boundary layer drag coefficient may be used, namely

$$C_f = 1.328 \left(\frac{\nu}{Lu}\right)^{1/2}, \qquad u = u_r \left(\frac{r_r + s_o/2}{r_r}\right) \tag{11}$$

where L is the plate length, ν is the fluid kinematic viscosity and u is the average ring surface speed. Defining I_m as the fraction of the ring circumference submerged in the sump and L_p as the perimeter length of the ring cross-section, the quantities L and A become

$$L = I_m(2\pi)(r_r + s_o/2) \tag{12}$$

$$A = L \times L_p \tag{13}$$

$$L_p \approx 2\left[(s_o - s_1)\left(\frac{1 - \sin\phi_r}{\cos\phi_r}\right) + s_1 + b_o\right] \tag{14}$$

$$I_m \approx \frac{\cos^{-1}(1 - i/r_r)}{\pi} \tag{15}$$

Upon substitution, the drag force becomes

$$\begin{aligned} F_s &= 0.664\left(\frac{\nu}{Lu}\right)^{1/2} \rho u^2 L \times L_p \\ &= 0.664(\nu L)^{1/2} \rho L_p u^{3/2} \\ &= 0.664(2\pi r_r I_m \nu)^{1/2} \rho L_p \left(1 + \frac{s_o}{2r_r}\right)^2 u_r^{3/2} \end{aligned} \tag{16}$$

2.4 Ring speed model

Using the above models, it is possible to derive an expression for the ring speed, u_r, for a given shaft speed. This is a necessary step before the lubricant delivery flowrate can be predicted. Neglecting the vertical components of the contact traction and sump drag, equation (2) reduces to

$$m_r g = nP_{cy} = 2.447 K_P \frac{R}{h_o} \eta n b_1 (u_r + u_s) \tag{17}$$

and, after rearrangement, gives the film thickness ratio

$$\frac{R}{h_o} = \frac{m_r g}{2.447 K_P \eta n b_1} \frac{1}{(u_r + u_s)} \tag{18}$$

Substituting in equation (7) gives the traction force

$$F_c = -2.301 K_F \left(\frac{R}{h_o}\right)^{1/2} \eta n b_1 (u_r + u_s) - 3.490 \left(\frac{R}{h_o}\right)^{1/2} \eta n b_1 (u_r - u_s)$$

$$= -k_{c1} \frac{u_r + u_s}{(u_r + u_s)^{1/2}} - k_{c2} \frac{u_r - u_s}{(u_r + u_s)^{1/2}} \tag{19}$$

Noting that $F_s = k_s u_r^{3/2}$, the ring speed is found from equation (3):

$$k_s u_r^{3/2} \times r_r \left(1 + \frac{s_o}{2r_r}\right) = -\left(k_{c1} \frac{u_r + u_s}{(u_r + u_s)^{1/2}} + k_{c2} \frac{u_r - u_s}{(u_r + u_s)^{1/2}}\right) \times r_r \tag{20}$$

$$u_r^2 k_s \left(1 + \frac{s_o}{2r_r}\right) \left(1 + \frac{u_s}{u_r}\right)^{1/2} = -[k_{c1}(u_r + u_s) + k_{c2}(u_r - u_s)]$$

Rearranged into an approximate quadratic form, the equation for ring speed becomes

$$u_r^2 + u_r \left(\frac{k_{c1} + k_{c2}}{k'_s}\right) + u_s \left(\frac{k_{c1} - k_{c2}}{k'_s}\right) = 0$$

$$u_r^2 + 2K_1 u_r + K_2 u_s = 0 \tag{21}$$

$$u_r = K_1 \left[-1 + \left(1 - \frac{K_2 u_s}{K_1^2}\right)^{1/2}\right]$$

where

$$K_1 = \frac{(K_F + 1.517) k_g}{2[(1 + u_s/u_r) K_P]^{1/2}} \tag{22}$$

$$K_2 = \frac{(K_F - 1.517) k_g}{[(1 + u_s/u_r) K_P]^{1/2}} \tag{23}$$

$$k_g = \frac{0.8835}{L_p [1 + s_o/(2r_r)]^3} \left(\frac{m_r g n b_1}{r_r \rho I_m}\right)^{1/2} \tag{24}$$

The ring speed is found iteratively starting with $u_r = u_s$ and the corrections K are adjusted for each iteration.

2.5 Lubricant flowrate

The most general viscous lifting theory available is that of Soroka and Tallmadge [**23**] and Tallmadge [**24**]. They relate the lifting speed to the film thickness by

$$Ca = K_3 D_\infty^{3/2} + K_4 D_\infty^2$$

or

$$D_\infty = \left(\frac{Ca}{K_3 D_\infty^{-1/2} + K_4}\right)^{1/2} \tag{25}$$

$$K_3 = \frac{[2(1 - \cos\beta)]^{3/4}}{3\alpha^{3/2}} \tag{26}$$

$$K_4 = \left[1 + \frac{1}{2} \exp\left(\frac{-5.13 Fp^2}{Ca^{4/3} D_\infty}\right)\right] \sin\beta \tag{27}$$

where $Ca = u\eta/\gamma$ is the capillary number (non-dimensional speed), $D_\infty = h_\infty(\rho g/\gamma)^{1/2}$ is the non-dimensional asymptotic film thickness, $Fp = \eta(g/\rho\gamma^3)^{1/4}$ is the fluid property (inertia) number and β is the withdrawal angle (Fig. 2). The coefficient α was constant in the original theory ($\alpha = 0.64304$) but was later allowed to vary according to the strain rate of the lifting process [**25**]. The following curve-fitted relation accurate to ± 0.09 per cent was developed by Innes [**26**]:

$$\alpha = -0.02971 + (0.45248 + 0.16692 K_5)^{1/2} \tag{28}$$

$$K_5 = \frac{3Ca - D_\infty^2 \sin\beta}{[3(Ca - D_\infty^2 \sin\beta)]^{1/3}} \tag{29}$$

which must be used iteratively with equation (25). The withdrawal angle β for the ring inner surface may be related to the ring immersion by

$$\beta = \cos^{-1}\left(1 - \frac{i}{r_r}\right) \tag{30}$$

which is less than 90° for a film on the top side of the lifting surface. Finally, the flowrate per unit width is calculated from the film thickness using the well-known expression for slow viscous flow:

$$q = u h_\infty - \frac{\rho g h_\infty^3}{3\eta} \sin\beta \tag{31}$$

The oil raised from the sump by viscous forces on the inner surface of the ring provides the inlet flow to the conjunction between each land on the ring and the normal line contact between the ring and the shaft. Of the total flow entering the conjunction, some will pass through the nip to form a fluid film, while the remainder will emerge as side leakage flow.

It is noted that equation (25) is a ‘deep bath’ theory that does not hold for withdrawal from very shallow or restricted baths, i.e. where boundary layer physics or the presence of an opposing wall pre-meters the flow. Furthermore, it does not consider variations in the film thickness near the edges of the lifting surface. For the purposes of predicting the delivery flowrate, only the ring inner surface is considered and flow losses due to centrifugal migration and drainage are neglected.

3 DISCUSSION

3.1 Comparison with experiment

The theoretical ring speed predictions are compared with experimental data of Keysell [**12**] in Fig. 5 and of Mangles [**13**] in Fig. 6. Their particular ring and shaft geometries were analysed using the data in Table 1. Their experiments were for isothermal conditions and represent two different ring/shaft/fluid systems. The empirical ring speed formula of Lemmon and Booser [**11**] is also shown for comparison:

$$Re_{\mathrm{ring}} = 1.44\left(\frac{i}{2r_{\mathrm{r}}}\right)^{-0.2}(Re_{\mathrm{shaft}})^{0.80} \tag{32}$$

where $Re = u(2r)/\nu$ is the Reynolds number. The ring speed predictions by both the empirical formula and the first-principles model generally agree with the data to within 5 per cent, i.e. the probable measurement error.

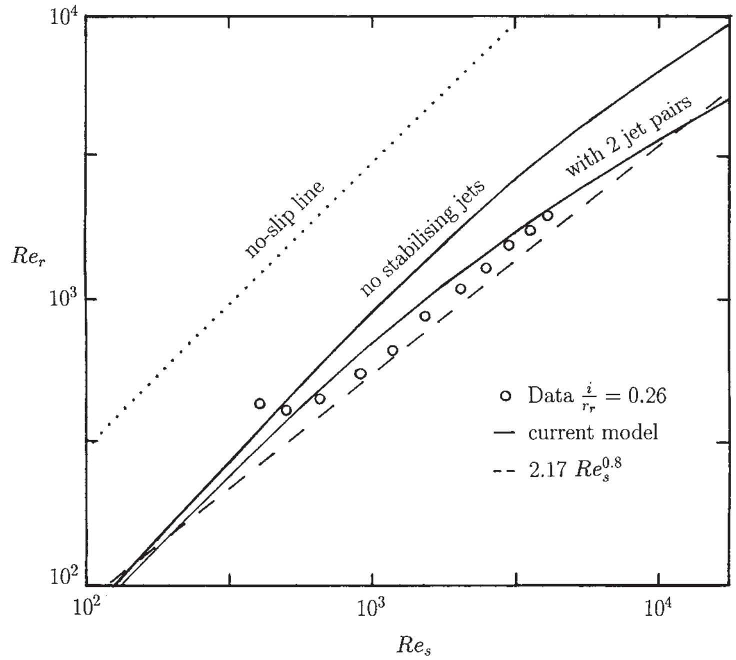

Fig. 5 Comparison of theory with the ring speed data of Keysell [**12**]

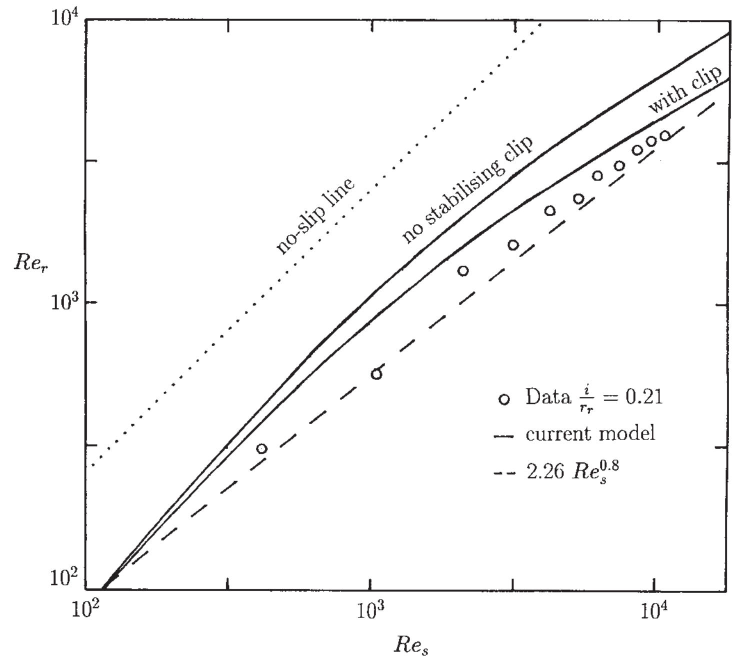

Fig. 6 Comparison of theory with the ring speed data of Mangles [**13**]

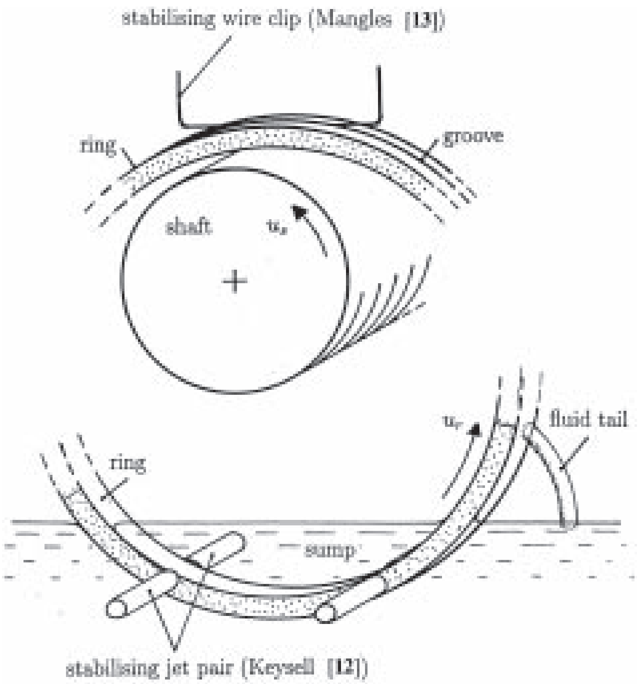

Fig. 7 Ring stabilizers used by Keysell [**12**] and Mangles [**13**]

Table 1 Lubricator dimensions and fluid properties for the experiments of Keysell [**12**] and Mangles [**13**]

Parameter	Keysell	Mangles
Ring weight $m_r g$ (N)	2.5	8.04
Ring radius r_r (mm)	76.2	95
Ring width b_o (mm)	9.53	18
Ring width b_1 (mm)	9.53 ($n = 1$)	6 ($n = 2$)
Ring thickness s_o (mm)	6.35	10
Ring thickness s_2 (mm)	0	0.7
Taper angle ϕ_r (deg)	0	10
Immersion factor i/r_r	0.26	0.21
Shaft radius r_s (mm)	25.4	37.5
Oil density ρ (kg/m^3)	865	870
Oil surface tension γ (N/m)	0.031	0.031
Oil viscosity ν (m^2/s)	40×10^{-6}	85×10^{-6}

Some low speed data may be affected by solid–solid friction in the contact, which would tend to increase the ring speed. The no-slip line indicates the maximum possible ring speed under dry friction conditions. The ring speed model is surprisingly robust, so that neglecting the pumping effort and sump recirculation appears to cause the ring speed to be overpredicted by only a few per cent.

In order to obtain better agreement, allowances were made for the extra drag forces present during the experiments, notably owing to ring stabilizers (Fig. 7). These resulted in lower ring speeds than would otherwise be expected. The two pairs of stabilizing jets used by Keysell were estimated from equation (5), using his experimental data, to have roughly tripled the sump drag owing to the formation of new boundary layers. This was readily accounted for by substituting $k'_g = k_g/3$ in equations (22) and (23). Similarly, the stabilizing wire clip of Mangles essentially doubled the drag ($k'_g = k_g/2$).

The uncorrected flowrate predictions are plotted along with the experimental data in Fig. 8 (Keysell) and Fig. 9 (Mangles). Keysell used a semicircular scoop below the shaft to collect the free flow of oil from the journal. The ring was stabilized by two double-acting hydrostatic bearings acting on its submerged portion. Guards were used to collect oil thrown off by centripetal forces and care was taken to ensure that these did not influence the ring flow. No scrapers or guides that may have restrained the axial movement of the ring were employed. Mangles, on the other hand, used a journal scraper to remove oil adhering to the journal and a ring guide to provide a narrow slot through which the ring had to pass and which prevented the oil deposited on the shaft from being thrown off and lost through the rear of the ring slot. Clearly the theoretical/experimental correlation will

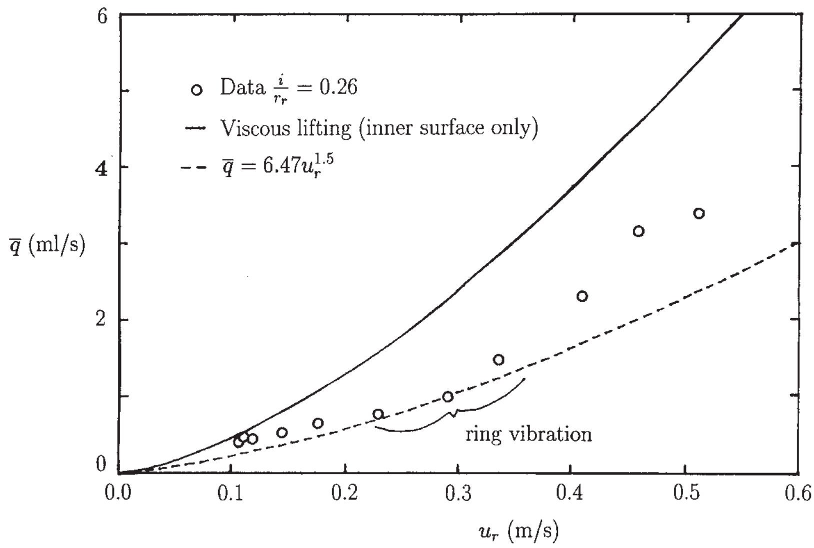

Fig. 8 Flowrate predictions and the data of Keysell [**12**]

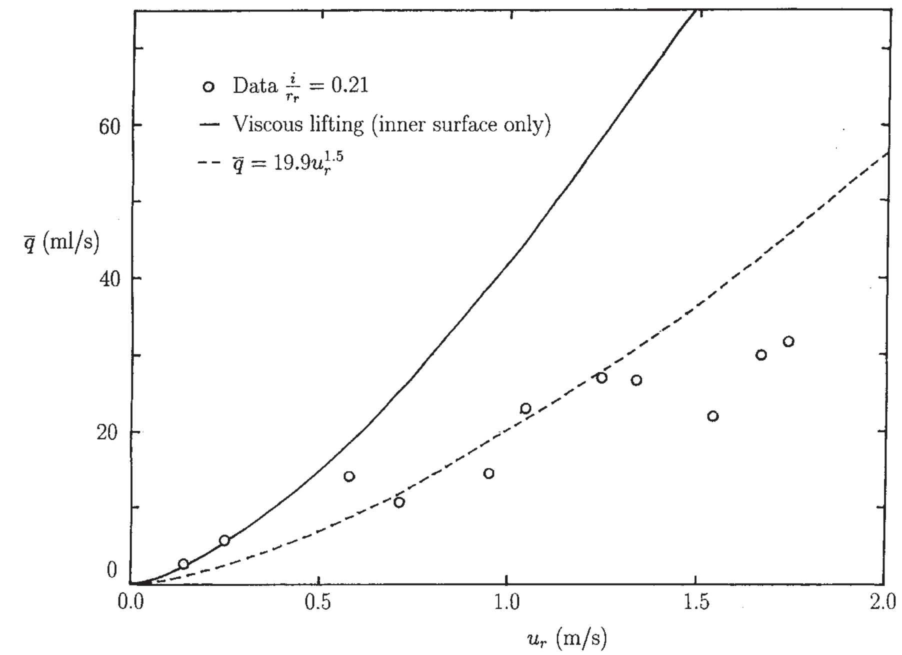

Fig. 9 Flowrate predictions and the data of Mangles [**13**]

be affected by the efficiency of the empirical oil collection methodologies.

In both cases, shown in Figs 8 and 9, the ring inner surface alone appears to have lifted more than sufficient fluid to account for the observed flowrates. Ring vibration reported by Keysell appears to have reduced the delivery flowrate for the indicated speeds (Fig. 8). The empirical formula of Lemmon and Booser [**11**] may be written as

$$\bar{q} = 0.49 \times 10^{-6} b_o \nu^{0.65} u_r^{1.5} \tag{33}$$

where $\bar{q}$ is in m^3/s, b_o is in m, ν is in m^2/s and u_r is in m/s. The predictions agree more closely with the experimental data than the uncorrected viscous lifting theory, although errors of 50 per cent or more are evident. However, viscous lifting theory shows the correct functional dependence with speed, which is encouraging since such consistent agreement has not previously been reported.

3.2 Further refinements

The loose-ring lubricator model presented here is quite basic and will require several refinements to yield accurate predictions in a number of cases. These are briefly discussed below under the subheadings of flow losses, sump effects, transient and dynamic effects and thermal effects.

3.2.1 Flow losses

The detailed flow on all of the lubricator surfaces needs to be considered more carefully in order to improve the accuracy of both the ring speed and the flowrate predictions. Under high speed conditions, centrifugal migration and drainage occur, resulting in a general movement of fluid from the inner to the outer ring surface. A fluid tail forms where the ring exits the sump (Fig. 7), representing considerable flow loss. At sufficiently high speeds it breaks up into droplets, some of which may be captured and used to augment the delivery flowrate. No attempt was made to analyse the tail, although this appears to be achievable. The use of an energy conversion equation similar to that for orifice discharge may be necessary to relate subambient pressures at the ring outer surface to fluid motion in the tail. Once the flow loss is known, the pumping effort can be calculated accurately and hence both the ring speed and delivery flowrate will be predicted with greater accuracy.

3.2.2 Sump effects

The ring model used averaged quantities such as the boundary layer length L and immersed fraction I_m. For very shallow immersions the boundary layer length would have to be considered separately for each ring surface. A complication arises when considering the flow losses and fluid migration mentioned previously. The ring re-enters the sump wet, requiring the boundary layer length to be adjusted to account for the incoming flowrate. Furthermore, not all of the fluid within the boundary layer will necessarily be lifted out of the sump, giving rise to a measurable recirculation flowrate. Both a 'wet ring' and 'recirculation' correction would be expected to reduce the sump drag and hence cause an increase in ring speed. However, at high speeds the pressure variation along the curved boundary layer could cause separated flow and/or vortex flow. This would lead to extra energy dissipation not accounted for by the current model and would be expected to cause a decrease in ring speed.

3.2.3 Transient and dynamic effects

The steady-state assumption needs to be relaxed and a dynamic analysis developed. The time-varying phenomena may include both fluid transients and mechanical transients, for instance the startup transient considered by Brockwell and Kleinbub [**18**]. These may, of course, be linked, such as for spray formation due to ring vibration.

Owing to their similar geometry, it would be expected that certain similarities exist between the dynamic behaviour of ring lubricators and hydrodynamic journal bearings. Real rings are known to be subject to phenomena such as swinging and lateral drifting (along the shaft) [**17**]. While the steady-state ring speed predictions appear to be robust, there remains some uncertainty about their general validity until further investigations are made.

3.2.4 Thermal effects

The thermal environment of the ring lubricator must inevitably receive attention. The temperature difference between shaft and ring is neglected in the analysis above for the reasons stated in Section 2. However, the variation in viscosity across the film in the ring/shaft contact might be expected to lead to greater slip for a hot shaft compared with the isothermal case. Such analysis is left for future study.

4 CONCLUSIONS

A loose-ring lubricator model has been developed based on a first-principles approach. Three-component models were incorporated into the analysis to provide contact force, sump drag and lifted flowrate predictions. The ring speed is calculated assuming steady-state balance between the opposing torques generated by shaft traction and sump drag. Comparison with detailed experimental data shows that the model predicts ring speed with an accuracy similar to that of published empirical formulae. In the absence of any flow loss corrections, viscous lifting theory was found to overestimate the experimental flowrate values. A number of refinements are suggested to enable the model to be more robust, particularly under high speed conditions in the presence of flow losses, ring dynamic effects and thermal effects. A careful case-by-case approach must still be taken until the model is well proven.

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