D. D. Joseph

R. Bai

T. Y. Liao

A. Huang

Department of Aerospace Engineering
and Mechanics,
University of Minnesota,
Minneapolis, MN 55455

H. H. Hu

Department of Mechanical Engineering
and Applied Mechanics,
University of Pennsylvania,
Philadelphia, PA 19104-6315

Parallel Pipelining

In this paper we introduce the idea of parallel pipelining for water lubricated transportation of oil (or other viscous material). A parallel system can have major advantages over a single pipe with respect to the cost of maintenance and continuous operation of the system, to the pressure gradients required to restart a stopped system and to the reduction and even elimination of the fouling of pipe walls in continuous operation. We show that the action of capillarity in small pipes is more favorable for restart than in large pipes. In a parallel pipeline system, we estimate the number of small pipes needed to deliver the same oil flux as in one larger pipe as $N = (R/r)^\alpha$*, where r and R are the radii of the small and large pipes, respectively, and* $\alpha = 4$ *or 19/7 when the lubricating water flow is laminar or turbulent.*

1 Introduction

The use of water as a lubricant to reduce friction in pipelining of heavy crude oil is an old idea which has been used sporadically over the past half century (Joseph and Renardy, 1992). The essence of lubrication is that water forms an annulus around oil so that oil does not touch the pipe wall, thus reducing the drag. A surprising but robust feature of fluid dynamics is that such a flow configuration is achieved more often than a stratified configuration, even when the densities of the viscous oil and the less viscous water are different. The levitation of the oil core against gravity appears to involve lift forces which are not yet perfectly understood. However, the fluids in a stopped pipeline are bound to stratify, leaving oil on the wall. The restart of a stopped pipeline is a major obstacle to be overcome before any water lubrication system becomes operational. Different ideas have been brought forth to prevent oil from sticking to the pipe wall and to make startup easier. For instance, hydrophilic walls, like the wall of a cement-lined pipe, have been proposed and are under investigation currently. In this work we shall discuss the possible effects of the size of the pipe upon the startup problem of a lubricated pipeline. We shall show that small pipes promote the action of capillarity and lead to configurations which are better for restart. In a parallel system we seek to determine the number of small pipes that will deliver the same flux as in one larger pipe with the same pressure gradient. The solution of this problem depends on the nature of the flow, laminar or turbulent, and other factors. It takes many more small pipes to deliver the same fluxes as in one larger pipe. A parallel system can have major advantages over a single pipe with respect to the cost of maintenance and continuous operation of the system, to the pressure gradients required to restart a stopped system and to the reduction and even elimination of the fouling of pipe walls in continuous operation. The engineering decision about whether many small pipes are better than a large one depends on the conditions of application in which the underlying fluid mechanics is a major player.

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2 Number of Small Pipes Which Are Equivalent to One Larger Pipe

We are going to calculate the number of small pipes of radius r needed to get the same flux of oil as in one larger pipe of radius R, provided that the pressure gradient and input ratio of volume flow rates are the same in each and every pipe.

First, consider the case when the lubricating water flow is laminar, and no secondary motion is present. The velocity field has one component, the axial component W, which can be solved analytically or numerically (see Bentwich, 1965 and Huang et al., 1995). The total fluid flux can be calculated from

$$Q = \int_\Omega W d\Omega, \tag{2.1}$$

where the integral is over the cross section of the pipe. Since the velocity W is linearly proportional to R^2 and the pressure gradient along the axis of the pipe, and $d\Omega$ scales as R^2, it is readily seen that

$$Q = R^4 q, \tag{2.2}$$

where the function q does not depend on R. Hence if a set of smaller pipes of radius r is to be used to deliver the same flux as one single larger pipe of radius R, $N = (R/r)^4$ of them will be needed.

For turbulent flow, a more realistic calculation of the number of small pipes can be performed based on empirical relations put forward by Arney, et al. (1993). A theoretical justification for these relations based on $k - \epsilon$ modeling of turbulence in the lubricating water flow has been given by Huang, Christodoulou and Joseph (1995).

Define an average velocity

$$V = \frac{Q}{\pi R^2}, \tag{2.3}$$

and a Reynolds number

$$\mathrm{Re} = \frac{2\rho_c R V}{\mu_w}(1 + \eta^2(m - 1)), \tag{2.4}$$

where Q is the total flux, η (≤ 1) is the radius ratio of the concentric oil core to the pipe, m is the viscosity ratio of water to oil, and $\rho_c = (1 - \eta^2)\rho_w + \eta^2\rho_o$ is a composite density of water (ρ_w) and oil (ρ_o). The friction factor is defined as

$$\lambda = \frac{4Rf}{\rho_c V^2} \tag{2.5}$$

where f is the pressure gradient along the pipe.

A number of authors have attempted to correlate the friction factor λ with the Reynolds number Re, for example, Sinclair (1970) and Brauner (1991). For our purpose here we shall take a simple empirical relation put forward by Arney et al. (1993), which is

$$\mathrm{Re}^{1/4}\lambda = 0.316. \tag{2.6}$$

Substituting (2.3), (2.4), and (2.5) into the above correlation, we get

$$Q = R^{19/7}q$$

where q again is a function independent of R. Hence in turbulent flow, $N = (R/r)^{19/7}$ small pipes with radius r will be needed to deliver the same flux as in a single pipe of radius R.

Therefore, in general, the number of smaller pipes needed to deliver the same amount of total flux can be written as

$$N = (R/r)^{\alpha}, \tag{2.7}$$

where $\alpha = 4$ when the lubricating water flow is laminar, and $\alpha = 19/7$ when the water flow is turbulent. For example, the total flux in five pipes of radius 1 in. is the same as in one pipe of radius 1.495 in. in laminar flow, and the same as in one pipe of radius 1.809 in. in turbulent flow. This shows that a rather large number of small pipes are required to deliver the same flux as a single larger one.

However, if it is possible to get a more efficient lubrication with smaller water fractions in small pipes than in large pipes, then the number of small pipes needed would decrease. We can also decrease the number of small pipes by paying a penalty in the pressure gradient required for the parallel pipes. If the small pipes are efficiently lubricated, an increase in the oil flux can be achieved at a small cost.

3 Cost, Maintenance, and Continuous Operation of a Parallel Pipeline System

One of the major advantages of parallel pipelining is that the system can be kept in continuous operation without failures. It is unlikely that all of the parallel pipes in a system would fail at the same time. The repair of a single pipe in the system need not interfere with the continued operation of the others. It could be useful to have a water-filled pipe in the parallel system to press into service if one of the pipes in the parallel system fails. It is also easier to handle small pipes rather than large ones.

Another advantage of parallel pipelines is the possible reduction of the total cost in the pipeline system: initial, maintenance and operational costs. A rough quotation of cost of cement-lined pipe in 1993 is: \$5/ft for 2-in. pipe; \$8/ft for 3-in. pipe; and \$12/ft for 4-in. pipe. The initial cost of a parallel pipeline may be higher than a single one. However, the savings in the maintenance and operational costs may outweigh the initial cost. The engineering decision about whether many small pipes are better than a large one depends on the conditions of application and the required reliability of the system.

4 Effects of Capillarity in Lubrication and Restart of a Lubricated Pipeline

In a stopped lubricated pipeline, initially centered and continuous oil core may break up into drops or slugs under the action of capillarity, contact the pipe wall and spread. In this and the next two sections we are going to examine the effect of capillary force on the breakup and the spreading of the core.

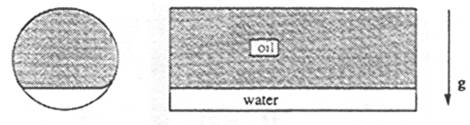

Fig. 1 Oil rises up in the pipe and sticks to the wall. We have a perfect vertical stratification. This situation is promoted by a large density difference, small interfacial tension, hydrophobic walls and large pipe radius. This configuration is not good for restart or continuous operation because the oil is sticking to the wall and water wedging between oil and wall is not possible.

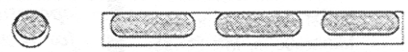

Fig. 2 Water-wedged rest configuration of oil slugs in water. This is promoted by small pipes, small density differences, large interfacial tension and hydrophilic walls (water-wet concrete walls, for example). Restart of this configuration is much easier than the restart of the completely stratified configuration shown in Fig. 1.

Capillarity works more strongly in small than large pipes. Another way to say this is that gravity is more important in large pipes. The capillary force can be used to keep oil from sticking to the wall. To fix the idea, let us suppose that there is about 20 percent water and 80 percent oil in the pipe, and there is no motion. The oil can be distributed in the pipe in many ways. In Fig. 1 we show a bad distribution. In Fig. 2 we show a good distribution in which water is wedged between the pipe wall and the oil. In Fig. 3 slugs developed in an actual experiment in which capillarity is sufficiently strong relative to gravity. In this experiment the ratio of the effects of gravity to those of surface tension is not large enough for oil to ball up to plug the pipe. If the pipe were larger or the surface tension were smaller, we would get a picture more like Fig. 1.

It is apparent that strong effects of capillarity, especially when combined with the antisticking effects of hydrophilic walls, like wet cement, are advantageous for the startup of a stopped line.

Hu and Joseph (1992) computed the size of slugs like those shown in Fig. 3 from an idealized problem, in which a motionless oil core of uniform radius is centrally located in a pipe filled with water and gravity is neglected. They then studied the linearized stability of this problem to disturbances proportional to $\exp(\sigma t)\exp(i\alpha x)$, where t is the time, x is the coordinate along the axis of the pipe, α is the wave number of the disturbance in the x direction and σ is the growth rate of the disturbance. The analysis is framed in Bessel functions and is essentially the same as one given previously in a Ph.D. thesis by Hammond (1982). The growth rate σ is real valued. They found that the presence of the pipe wall has an important role in determining the wavelength corresponding to the maximum instability and the maximum growth rate of the disturbances. As the fraction of the fluid in the annulus becomes smaller and smaller, the wave number of maximum instability increases and tends to a limiting value of $0.7/R_1$ where R_1 is the radius of the

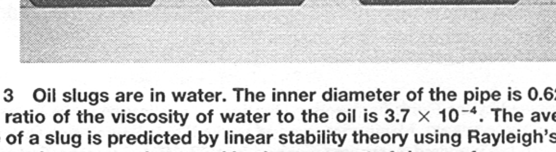

Fig. 3 Oil slugs are in water. The inner diameter of the pipe is 0.625 in. The ratio of the viscosity of water to the oil is 3.7×10^{-4}. The average size of a slug is predicted by linear stability theory using Rayleigh's idea of a maximum growth rate, taking into account of viscous forces, surface tension, volume fraction and pipe radius but neglecting gravity.

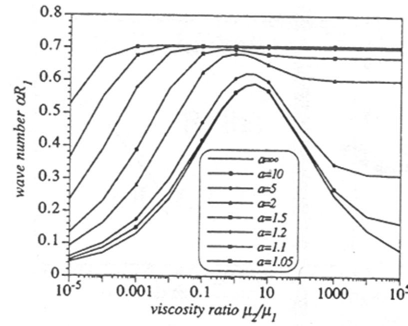

Fig. 4 Variation of the wave number corresponding to the maximum instability with the viscosity ratio (μ_2/μ_1) at different radius ratios ($a = R_2/R_1$), where μ_1 and μ_2 are the viscosities of the fluid in the core and in the annulus, R_1 and R_2 are the radii of the core and the pipe

core, see Fig. 4. The presence of the pipe wall reduces the growth rate of the disturbances (Fig. 5). The reduction is especially large when the more viscous fluid is in the annulus. When the annulus is very thin, the core may not break up during the time interval of interest since the growth rate is so small. Hu and Joseph (1992) reported that satisfactory agreement was exhibited between the theory and an experiment.

We get small slugs in small pipes; they scale with the pipe radius. Since gravity is neglected, the slugs will not flatten. In practice, the spreading property of the sessile drop, which is discussed in the next section, will spread the drop laterally in the pipe. Hence, for large pipes we can expect a stopped pipeline to look like Fig. 1, whereas for small pipes we might expect configuration shown in Fig. 2 and Fig. 3, which are good for restart.

5 Maximum Height of Sessile Drops

Once the lubricated pipeline stops, the drops or slugs of oil will float to the top section of the pipe, contact the pipe wall, spread and form the configuration shown in Fig. 2. Here we review the spreading property of a sessile drop on a planar surface.

Suppose a sessile drop is placed on a plane as in Fig. 6. The volume of the drop is increased holding the contact angle fixed. There is a volume when $h = h_{\max}$ such that further increase of

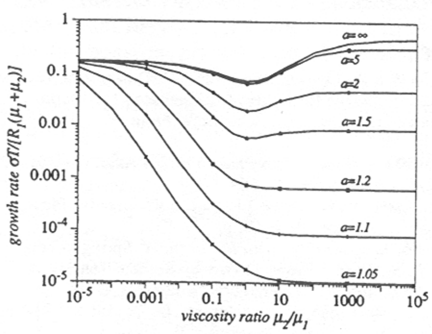

Fig. 5 Variation of the growth rate versus viscosity ratio at various radius ratios, where T is the interfacial tension

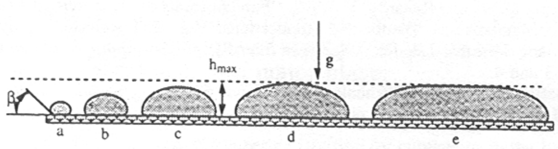

Fig. 6 If the volume of a sessile drop is increased while holding the contact angle fixed, a condition will be reached where the height of drop does not increase and the drop just spreads laterally

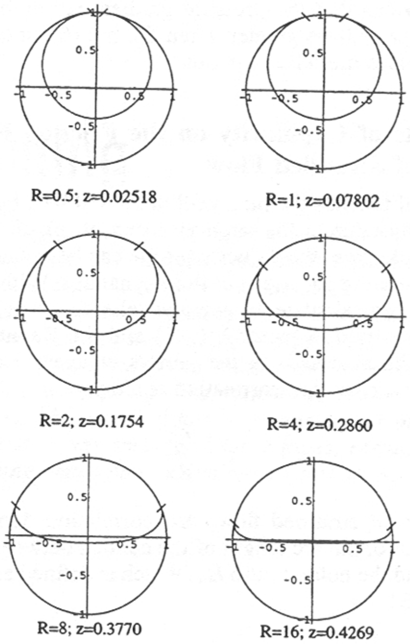

Fig. 7 Shapes of the oil core in different sizes of pipeline, where R is the pipe radius and z is the ratio of the arc length of contact between the oil and pipe wall to the circumference of the pipe. The pipeline is normalized by its radius. Density of water is 1.0 gm/cm^3, oil is 0.995 gm/cm^3. The value of the interfacial tension is 26.3 dyne/cm. The volume ratio is one to one. The contact angle of oil to the pipe wall is zero.

volume leads to lateral spreading rather than to an increase in the height of the drop. We can say that gravity prevents the drop height from increasing past $h_{\max}$. Finn (1986, p. 63) has shown that the maximum height of an axisymmetric sessile drop is

$$h_{\max} = \sqrt{\frac{2(1 + \cos \beta)T}{g \Delta \rho}} \tag{5.1}$$

where T is the interfacial tension, g is the gravitational acceleration, $\Delta\rho$ is the density difference of the fluids inside and outside the drop, and β is the contact angle of the drop with the solid surface as shown in Fig. 6. The maximum height $h_{\max}$ is a limiting value of the highest point on a drop of zero contact angle ($\beta = 0$) as the volume of the drop is increased to large values. The same Eq. (5.1) holds for a two dimensional sessile drop. The top of the drop will get flatter and flatter as the volume of the drop is increased. We call this a spreading property of sessile drops. Obviously we can put a sessile drop in a pipe.

6 Effect of Pipe Size on the Shape of Oil Core

In this section we examine the shape of a two dimensional sessile drop inside a pipe. Consider the case where the oil is lighter than the water and the volume ratio of oil to water is 1 to 1. Obviously increasing the diameter of the pipe increases the volume of the sessile drop and brings the spreading property into play. The effects of surface tension get less and less important. In Fig. 7 we have plotted the shape of the core in pipes of different diameter. The contact angle of the oil with the pipe wall is taken to be zero, which is appropriate for a hydrophilic wall, like the wall of a cement-lined pipe which absorbs water. There is an arc of contact of the oil bubble with the top wall. The length of the contacting arc increases with pipe radius because the buoyant weight of a larger bubble must be balanced by a bigger reaction force resulted from integrating the pressure inside the oil core over the contacting arc length.

It is obvious that the pressure gradient required to restart a stopped pipe will be greater when the arc of contact is larger. So small pipes are advantageous.

7 Effects of Capillarity on the Friction Factor and Holdup of Stratified Flow

When oil touches the pipe wall there will be a huge decrease in the oil flux due to the large viscosity of oil. If the oil-water interface is known, the velocity profile can be obtained by solving the governing equations of fluid dynamics. With the velocity profile, we can calculate the average velocity. Then a correlation between the friction factor λ (2.5) and the Reynolds number (2.6) can be obtained. In the perfect concentric core-annular laminar flow case this correlation is simply

$$\lambda = \frac{64}{\mathrm{Re}}.$$

In the case of stratified flow, this correlation depends on the viscosity ratio, the arc length of the contact between the oil and the wall and the holdup ratio H_w, which is defined as (see Arney et al., 1993)

$$\frac{V_w}{V_w + V_o}$$

where V_o and V_w are the volumes of oil and water in the pipe, respectively. A related quantity is the so-called input ratio of water

$$C_w = \frac{Q_w}{Q_w + Q_o}$$

where Q_o and Q_w are the oil and water flux, respectively. From Section 6 we know that we get shorter arc length of contact in smaller pipes. This in turn will give a higher value of oil flux as shown in Fig. 8, where the dimensionless oil flux q_o is plotted as a function of parameter $A = [(\rho_w - \rho_o)gR^2/T]$ for a viscosity ratio of 50, contact angle of zero degree and holdup ratio of 0.5. However, oil barely moves in the pipe when its viscosity is large. Thus the oil flux is much smaller than that of the water, which results in a value of input ratio of water C_w very close to one, as shown in Fig. 9. In the figure C_w is computed for different values of holdup ratio H_w and the length of contact. Therefore the total flux does not change dramatically as the contact length gets smaller. This means that the averaged velocity or the friction factor is not sensitive to the change of the pipe radius.

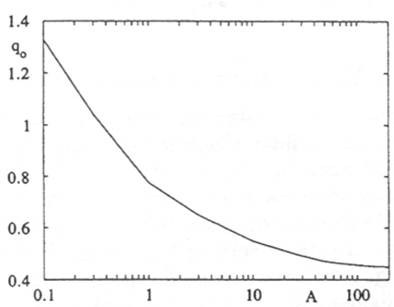

Fig. 8 The (dimensionless) oil flux as a function of parameter A, which is proportional to R^2. The viscosity ratio is 50, the contact angle is zero degree and H_w = 0.5.

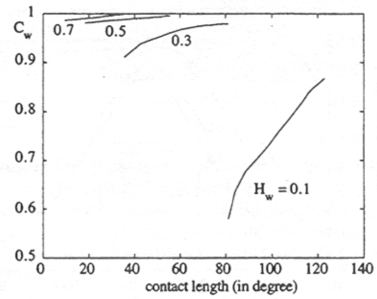

Fig. 9 The input ratio of water C_w as a function of the contact length for different values of H_w. The viscosity ratio is 50.

8 Conclusions

In a parallel pipeline system, we calculated that $N = (R/r)^\alpha$ small pipes are needed to deliver the same oil flux as in one larger pipe, where r and R are the radii of the small and large pipes respectively, and $\alpha = 4$ or 19/7 when the lubricating water flow is laminar or turbulent. In practice, it is possible to get a more efficient lubrication with small pipes than in large pipes, thus the number of small pipes needed would decrease.

We show that strong effects of capillarity, especially when combined with the anti-sticking effects of hydrophilic walls, create a good configuration in which water is wedged between the pipe wall and the oil, and are advantageous for the startup of a stopped pipeline.

We calculated the shape of an oil core in pipes of different diameter. The relative arc length of contact increases with pipe radius, requiring a larger pressure gradient to restart a stopped line. So again small pipes are advantageous.

Therefore, a parallel system can have major advantages over a single pipe with respect to the cost of maintenance and continuous operation of the system, to the pressure gradients required to restart a stopped system and to the reduction and even elimination of the fouling of pipe walls in continuous operation.

Acknowledgments

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