MEASURING THE EFFECT OF STARTING AMOUNT OF LUBRICANT ON FRICTION OF STARVED EHL

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Abstract- A series of friction measurements were performed for compliant contact under starved lubrication. Three conditions of initial preparations of the lubrication condition were examined: heavy lubrication, light lubrication, and dry friction. The effects due to surface roughness, entrance angle, contact pressure, and sliding speed were also measured and discussed. The Stribeck diagrams formed with the measured data indicate significant influence on friction performance due to the starting amount of lubricant applied on the friction interface for starved EHL, even when the operating conditions are the same.

Index Terms- Elastomer, damper, initial condition, soft contacts

I. INTRODUCTION

In recent years, a new class of mechanical dampers have been gaining attention for its advantages of being low cost and free from oil leakage [1]-[2]. The so-called lubed-rubber damper (or surface effect damper) utilizing friction of lightly lubricated elastomer as the major damping force, generates damping force in accordance with input shock velocity; hence performs similar to a conventional fluid damper. With a little amount of lubricant on the elastomer surface, the damper does not require flooded damping fluid sealed inside. As a result, concerns about fluid leakage can be entirely eliminated, and the high manufacturing costs associated with the efforts for preventing fluid leakage can be removed as well. Successful applications of the lubed-rubber damper have been seen in some commercial products; however, promotion of such technology has not been successful due to lack of matured design methods and related tribology theories. Fig. 1 indicates a sample design of the lubed-rubber damper [1], [3]-[4]. The invention primarily consists of a compliant hollow rubber tube sliding inside a rigid sleeve. As shown in Fig. 1, the rubber tube is often connected to a moving shaft at one end (the bottom end in Fig. 1) and a small friction component at the other. The small friction component, designed to be oversized relative to the inner diameter of the rigid sleeve, can generate friction upon axial motion relative to the outer rigid sleeve. On the other hand, the hollow rubber tube is purposely designed to be slightly smaller than the inner diameter of the rigid sleeve to form an initial radial gap between the two surfaces. When the shaft pushes the hollow rubber tube upwards, the small friction component resists the motion of the hollow rubber tube; as a result, the axial compression causes it to expand radially. When the hollow rubber tube expands to an extent that contact between the rubber tube and the rigid sleeve takes place, friction between the rubber tube and the rigid sleeve is then initiated, which will then be utilized as the major source of energy dissipation. When the shaft moves in the opposite direction, namely, pulling the hollow rubber tube downwards, the small friction component still resists the motion of the hollow rubber tube; however, this time it helps to stretch the hollow rubber tube and causes it to shrink radially and cease to contact the rigid sleeve. As a result, no friction will be generated by the hollow rubber tube, and the total damping force will be much less than that in the upward direction. Such directional difference in damping force is critical to many damper and shock absorber designs. With this important feature, this new class of elastomeric friction damper can become a promising candidate for replacement of existing oil dampers.

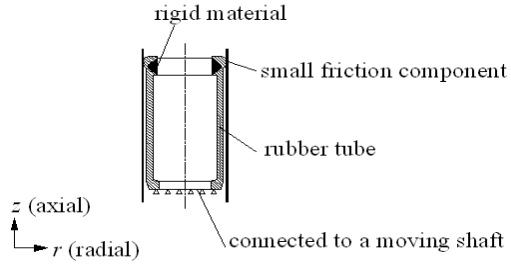


Fig. 1 Schematic of a typical lubed-rubber damper.

In a previous study regarding the friction along the rubber tube [3], it was found that the local friction increases rapidly along the rubber tube, resulting higher local friction near the bottom than at the top. This often leads to early breakdown due to excessive wear of the rubber tube near the bottom. Since the damping force comes from lubricated friction of soft elastomeric contact, obtaining basic knowledge about the friction characteristics of such atribological system is critical to the design practice of the product. One important difference in the current tribological system compared to many other lubrication systems is that a constant supply of lubricant does not exist to form a

thick lubrication film. For the lubed-rubber damper, usually an initial amount of grease is applied at the friction interface. When the damper starts to operate, the rubber tube as shown in Fig. 1 may push or squeeze excessive lubricant to both ends of its sliding track, leaving little yet unknown amount of lubricant at the friction interface for future service. The lubricant may find for itself steady existence at the interface as a function of operating parameters. However, the initial amount of lubricant applied on the friction interface may play a critical role in defining the friction performance as well. In the current study, we would like to focus on testing the lubrication system pertinent to the lubed-rubber damper to gain some insights about the effects of various parameters presumed most critical to the performance.

II. BACKGROUND

Hsu and others [5] summarized that under the effect of fluid lubricant, there are four mechanisms contributing to the friction between polymer and metal; they are deformation, adhesion, Schallamach waves, and fluid shear. Deformation represents the energy dissipation associated with the hysteresis of elastomer undergoing cyclic deformation when the elastomer conforms to whatever slides over it.

The hysteretic behavior of elastomer is a complicated function of the magnitude, the direction, and the frequency of the strain cycles that the elastomer experiences [6]-[8]. However, if the bulk material is not undergoing massive cyclic deformation, the friction attributed to the hysteresis would be comparatively much smaller than the other sources [9]. Adhesion is the irreversible bonding force between two surfaces when the two surfaces come very close to one another [10]. As a major source of dry friction, the effect of adhesion decreases dramatically when the lubricant enters the interface between the friction pairs [11]-[13].

It was also found that the coefficient of friction between elastomeric pairs decreases with increasing normal load, and increases with relative speed [14]-[17]. Schallamach waves usually happen at the friction interface between compliant rubber and a hard surface under high speed without lubrication. The contact surface appears partially separated due to elastic waves taking place on the rubber surface [18]. Barquins and Roberts observed that when Schallamach waves happen, the friction force drops dramatically and becomes much less sensitive to speed, temperature, and normal loading [19].

When there is sufficient fluid lubricant at the interface between the two bodies moving relative to one another, the friction force in-between primarily is determined by the fluid shear of the lubricant. The pressure (P), speed (V), thickness (h), thickness at the maximum pressure (h_m), and viscosity (□), are governed by the Reynolds ' equation [20]-[22]:

$$\frac{dP}{dx} = 6\eta V \frac{h - h_m}{h^3} \quad (1)$$

Fluid shear force can be derived from the above equation to predict the frictional resistance. It is found that for metal lubrication, the coefficient of friction can be summarized as a function of a duty parameter -・V/W [22], in which W represents the normal load. When plotting the friction coefficient versus the duty parameter, one may find that the coefficient of friction has a single minimum value which is of the order of 0.001, and increases both ways as the duty parameter becomes larger or smaller. This is the well-known Stribeck Diagram originated by the experiments carried out by Richard Stribeck. For larger values of the duty number, the coefficient of friction increases along a straight line that would pass through the origin, clearly indicating the effect of fluid shear. However, the line does not really pass through the origin when the duty number becomes small; instead, the friction coefficient increases rapidly and reaches a maximum at a very small duty number [23]. This is believed to be due to the high friction between the metal surfaces, where massive contact starts to take place when the fluid film no longer effectively separates them. However, the characteristics of lubricated rubber do not appear to be the same. It was reported that the presence of roughness anisotropy strongly modifies the classical shape of the Stribeck curve asa consequence of local micro-EHL conditions which occur at the contact interface [20].

Although the friction damper depicted in Fig. 1 usually adapts lubricant for surface protection, the availability of lubricant is not maintained during the course of service, since there is no mechanism to constantly supply lubricant to the interface. Initially, a certain amount of lubricant is applied at the interface; but then the device is set to operate and no lubricant will be supplied to the interface hereafter. The amount of lubricant in service is then decided by self-balance among the possible key parameters such as contact pressure, speed, fluid viscosity, and perhaps the surface roughness, etc. Beside all that, it was observed that even if one places a large amount of lubricant at the interface initially, most of it will be squeezed or pushed out of the track during operation. On the other hand, if originally only very little amount of lubricant is introduced to the interface, that little amount of lubricant might remain on the track, and be effective in providing a certain level of lubrication. Nevertheless, the nature of the two tribological systems may be quite different. It is thus the main goal of this investigation to gain some insights of the effect of the initial application amount of lubricant on the coefficient of friction of lubricated elastomer.

III. FRICTION MEASUREMENTS

A. Testing Apparatus

A test rig as shown in Fig. 2 was designed for friction measurement. The apparatus consists of a pair of stationary elastomer specimens and a moving central plate, which slides against the elastomer specimens. The dynamic tester used here is capable of generating linear motion for the plate at various stroke lengths and speeds. The normal contact force can be adjusted by changing the weights applied at the two holders. When the plate is driven by the moving shaft and sliding against the elastomer specimens, the load cell under the elastomer specimens measures the resisting force, which reflects the friction force between the friction pairs.

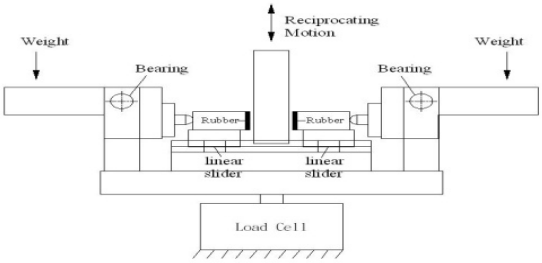


Fig. 2 Schematic of the apparatus of the friction experiment.

B. Design of the testing specimen

As the friction coefficient can be largely affected by the contact pressure [14], control over the contact pressure in the experiment is critical, and the elastomer specimen needs to be designed to have a contact pressure distributionas uniform as possible. For the above reason, the elastomer specimen was determined to be a thin elastomer layer bonded to a metal back plate as shown in Fig. 3. The elastomer specimen is made of silicone rubber with a hardness of Shore A 35. Since the elastomer is relatively compliant, the true contact area can significantly differ from the nominal area under the applied load during testing. Finite element analyses were carried out to calculate the true contact areas at various normal loads to provide a database of the true contact area for the friction measurement. ANSYS 10.0 [26] was used for the finite element calculation with the Hyperelastic elements (PLANE183) for the elastomer and the contact elements (TARGE169 and CONTA172)for friction modeling. Neo-Hookean material law was adapted with a shear modulus of 0.5 MPa and a bulk modulus of infinity, and the nonlinear solution procedure was activated with multiple sub-steps for accurate solution. The results are shown inFig. 4.

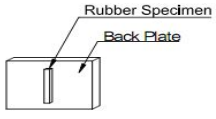


Fig. 3 The testing specimen.

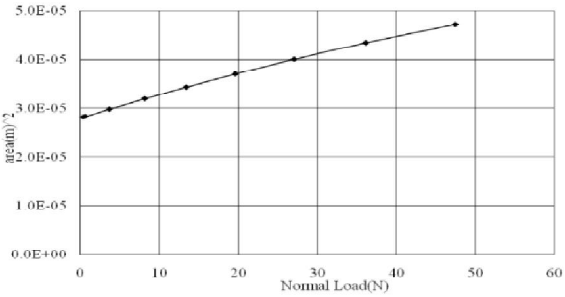


Fig. 4 Calculated true contact areas of the rubber specimen under various normal loads.

C. Testing Plan

Three types of methods for initial application of the lubricant were defined：1. heavy lubrication, 2. light lubrication, and 3. dry friction. Under " heavy lubrication, " initially a large amount of grease lubricant (with a height greater than 2 mm) is applied at both surfaces of the elastomer specimen and the central plate before testing. This condition represents the maximum amount of lubricant one can apply on the friction surface. As for the "light lubrication," initially a large amount of lubricant with a height greater than 2 mm is applied at the surface of the elastomer; however, excessive lubricant is immediately removed from the surface using a steel rod pressing and sliding against the elastomer surface prior to testing as depicted in Fig. 5. A fixture with a recess was made to hold the specimen, so when the 8mm steel rod gently slides at 4 mm/s against the rubber surface, the compression imposed on the rubber is maintained at a constant value of 10%. The resulting elastomer surface looks still somewhat oily; however, the lubricant left on the elastomer surface is obviously much less than the "heavy lubrication." As for the "dry friction," no lubricant was applied at the friction interface.

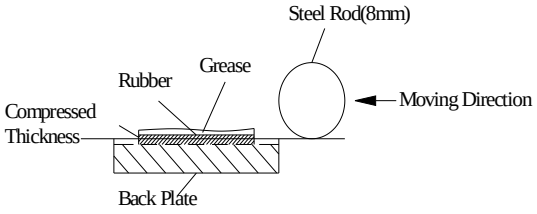


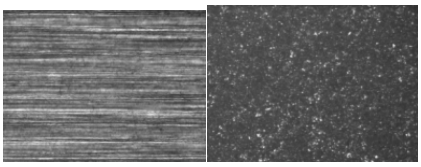
Fig. 5 Demonstration of "light lubrication" showing how to remove excessive lubricant for preparation.

The stroke length of the reciprocating motion is fixed at ± 12 mm [1]; the input waveform of the motion is triangular. Three frequencies were tested, resulting three testing speeds: 2 mm/s, 4 mm/s, and 8 mm/s. The grease used in the tests is Fluorosilicone grease with a Saybolt viscosity of 495.4 cSt at 100℉,thickened with PTFE. The dynamic viscosity is 0.713 Pa·s. The moving plate is made of 6061 aluminum alloy with an average surface roughness of 0.4 ·m along the friction direction.

IV. RESULTS AND DISCUSSION

A. Surface roughness

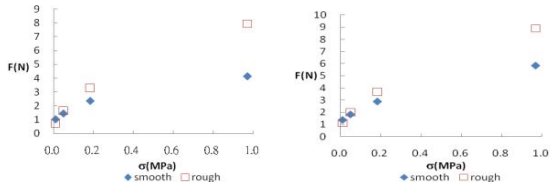
When very limited amount of lubricant is used, it is possible that solid contact happens vastly. In this scenario, surface roughness may play an important role in affecting the friction characteristics. Two different levels of surface roughness of the central plate were prepared with the roughness average (R_a) measured using a profilometer. Fig. 6(a) shows the CCD image of a smoother plate with a R_a of 0.4□m; Fig. 6(b) the image of a rougher plate, with a R_a of 1.98□m. Note that the roughness was measured in the same moving direction in the friction measurements.



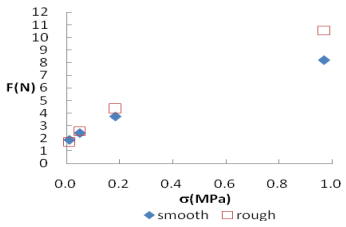
(a) smooth (b) rough

Fig. 6 CCD images of the central plates

In Fig. 7, the measured friction forces are shown at various contact pressures. Under initial heavy lubrication, one can see that the friction force associated with a rougher surface is in general greater than that of the smoother one. The effect magnifies particularly at higher pressures. This is probably due to the increase of total contact area becoming much greater for the rougher surface than the smoother one at high contact pressures [14], [25]. It could also be due to the greater surface deformation which results from the elastomer encountering the rough surface of the central plate. Either way, it suggests that even under the heavy lubrication, solid contact seems inevitable and the solid surfaces are not fully separated by a layer of lubricant with sufficient thickness.



(a) speed = 2mm/s (b) speed = 4mm/s



(c) speed = 8mm/s

Fig. 7 Effect of surface roughness on the friction force (heavy lubrication).

B. Entrance angle

In the hydrodynamic lubrication theory, the entrance angle of the fluid plays an important role in the formation of the fluid film and thus determines the friction force of the tribological system [22]. To study the effect of the entrance angle, two types of elastomer specimens were created – one with a perpendicular front edge, and the other with a curved edge of 1 mm radius (see Fig. 8).



(a) Perpendicular front edge (b) Curved front edge

Fig. 8 Elastomer specimens for testing of the effect of entrance angle.

The tests were carried out for both heavy and light lubrications at a sliding speed of 4mm/s. Fig. 9shows the data of light lubrication, where the friction forces of both cases are almost identical at every contact pressure. This result is not surprisingsince the amount of lubricant is very low, the curved front edge does not facilitate any more lubricant into the friction interface as it may if the lubricant is flooded. On the other hand, the data of heavy lubrication (Fig. 10) show a significant difference. It is interesting to note that much less difference is seen at higher contact pressures.

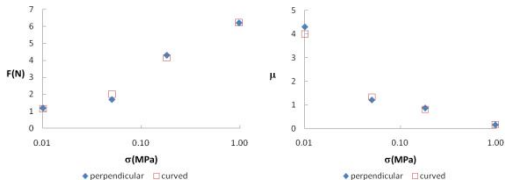


Fig. 9 Comparison of the entrance angle effect under light lubrication.

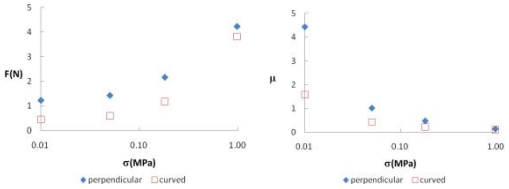
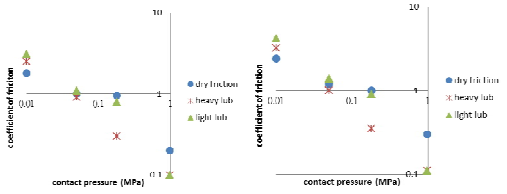


Fig. 10 Comparison of the entrance angle effect under heavy lubrication.

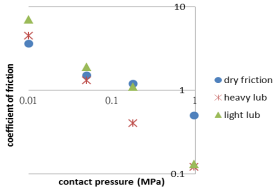
C. Contact pressure

The friction measurement was performed at four contact pressure levels – 0.01MPa, 0.05MPa, 0.18MPa, and 0.97MPa. Before each test, the finite element results in Fig. 4 were used to determine the normal loads needed to apply on the test rig. As shown in the test results of Fig. 11 comparing heavy lubrication with light lubrication, constant discrepancy is observed between the two groups. Light lubrication generates higher friction than heavy lubrication for most of the cases. Note that the operation conditions

are the same for the two groups, while the only difference is the initial amount of lubricant applied at the friction interface. Interestingly, at the lowest contact pressure tested, light lubrication generates the highest friction force among all testing cases, rather than the dry friction. Although there is no proof at this moment, it is possible that such a phenomenon may be attributed to the Schallamach waves associated with dry friction at low contact pressure [18]. A little lubricant might be sufficient to stick the rubber surface to its sliding counterpart and hence prevent Schallamach waves from happening. It is interesting to note that although the light lubrication condition consistently generates higher friction than heavy lubrication, they become very close at high contact pressure, in our case, 0.97 MPa. This suggests that the effect of initial amount of lubricant becomes much less critical at high operating pressure, as contact pressure dominates the amount of lubricant retained on the friction interface. It is also interesting to note that at high pressure, the dry friction is still significantly higher than the other lubricated cases. It seems that although the high operation pressure conditions the lubricated surface and equalizes the heavy and the light lubrication cases, there is still effective amount of lubricant left on the friction interface to create hydrodynamic lubrication to distinguish it from dry friction. Overall, the range of coefficient of friction associated with light lubrication is greater than all the other cases, implying that if one needs to enlarge the range of coefficient of friction, light lubrication may be a good choice.



(a) 2 mm/s(b) 4 mm/s

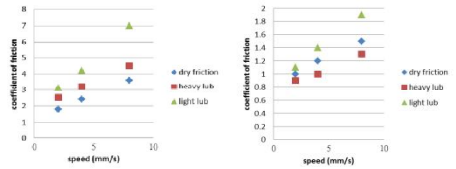


(c) 8 mm/s

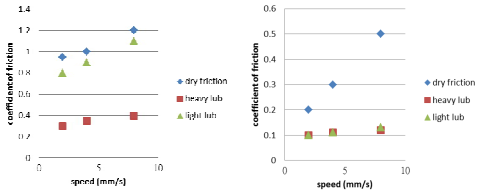
Fig. 11 Measured coefficient of friction showing the effect of contact pressure.

D. Sliding speed

Fig. 12 shows the effect of sliding speedon the friction coefficient. Dry friction and light lubrication somehow appear to have a stronger speed effect than the heavy lubrication case as shown in Fig. 12. It is most representative by the data of 0.97 MPa contact pressure in Fig. 12(d), where the dry friction coefficient increases significantly with increasing speed, while the lubricated friction coefficient barely changes. The enhancement of friction due to speed increase for dry friction may be attributed to the combined effects of viscoelasticity and adhesion of the elastomer. As for the lubricated cases, it is possible that at high pressure, most of the grease has been squeezed out of the friction interface, with only some PTFE solids lying between the two surfaces and effectively reducing the friction [10].



(a) 0.01 MPa(b) 0.05 MPa



(c) 0.18 MPa(d) 0.97 MPa

Fig. 12 Effect of sliding speed on the coefficient of friction.

CONCLUSIONS

As shown in the current work, the starting amount of the lubricant imposes significant effects on the friction performance, even when the operating conditions are the same. To summarize, it can be demonstrated with a Stribeck diagram shown in Fig. 13.Although the operating parameters (speed and pressure), surface, and the viscosity of the lubricant are the same, the light lubrication consistently provides higher friction than the heavy lubrication.

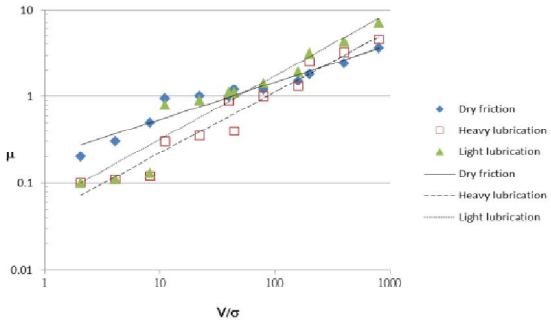


Fig. 13 Coefficient of friction vs. V/□.

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