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EFFECT OF BALL-RACE CONFORMITY ON SPINNING FRICTION

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Cleveland, Ohio

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ABSTRACT

Tests were conducted in the NASA spinning friction apparatus with SAE 52100 steel specimens with ball-groove conformities of infinity (ball on a flat), 60, 55, and 51 percent. The coefficient of spinning friction decreased with increasing maximum Hertz stress to a minimum value of approximately 0.05. For a constant stress, the coefficient of spinning friction increased with decreasing conformity (increasing percent conformity). Spinning torque increased with increasing stress for each conformity tested. For a given stress, spinning torque decreased with decreasing conformity.

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SUMMARY

The NASA spinning friction apparatus was used to determine the effect of ball-groove conformity on spinning torque and the coefficient of spinning friction. Ball-groove conformity is the groove cross radius expressed as a percentage of the upper ball diameter. SAE 52100 1/2-inch (12.7-mm) diameter balls were loaded against and spun in cylindrical grooves having ball-groove conformities of infinity (ball on a flat), 60, 55, and 51 percent. The tests were run at maximum Hertz stresses from 48 000 to 384 000 psi (330×10^6 to 2650×10^6 N/m^2), a spinning speed of 1050 rpm, and room temperature (no heat added) with a synthetic paraffinic oil containing no additives as the lubricant.

The coefficient of spinning friction decreased with increasing maximum Hertz stress to a minimum value of approximately 0.05 for all conformities tested. For constant maximum Hertz stress the coefficient of spinning friction increased with decreasing conformity (increasing percent conformity). However, as the maximum Hertz stress decreased, the value of the coefficient of spinning friction for all of the conformities tested reached, and in one case exceeded, 0.5, that value generally given for SAE 52100 steel sliding unlubricated on SAE 52100 steel.

Spinning torque increased with increasing maximum Hertz stress for each conformity tested. However, torque decreased with decreasing conformity (increasing percent conformity) where stress was constant.

INTRODUCTION

For engineering applications requiring high efficiency and low power consumption, it is important that frictional losses in moving components be decreased. Rolling-element bearings are components of many engineering systems. To attain high bearing efficiencies, it becomes necessary to study closely factors such as bearing kinematics, contact stress, material deformation, and coefficients of sliding and spinning friction that affect bearing power loss (ref. 1).

The shape, size, and pressure distribution of the contact areas of two solid, elastic bodies have been derived mathematically by Hertz and are presented in reference 1. The contact area between a ball and race in a ball bearing is elliptical because of elastic deformation. The kinematics of component motion within a ball bearing has been studied analytically in references 2 and 3. Because of bearing geometry and associated kinematics, the balls in an angular-contact ball bearing have an angular velocity, called spinning, about an axis perpendicular to the contact area of the ball on either race, depending upon ball control (ref. 2). Ball control is the ability of either race to impose rolling without spinning on the balls as the bearing rotates about its axis. For ball bearings, ball control is a function of the relative size of the ball-race contact areas, the normal load in the ball-race contacts, the eccentricity of the contact ellipses, and the coefficient of spinning friction in the contact areas.

Ball spinning motion causes a spinning moment or torque about an axis normal to the contact area. The friction force associated with this moment results from Coulomb friction and/or fluid shear forces, depending on whether dry surface conditions or boundary or elastohydrodynamic lubrication exist at the contact interface. The resulting torque contributes to internal bearing losses.

Preliminary tests, reported in reference 4, were conducted under varying maximum Hertz stress to 136 000 psi (938×10^6 N/m^2) at a spinning speed of 950 rpm, with a 51-percent ball-groove conformity using a polyphenyl ether (5P4E), a highly purified naphthenic mineral oil, and a di-2-ethylhexyl sebacate (MIL-L-7808) as lubricants. It was found that the coefficient of spinning friction decreased with increasing maximum Hertz stress to an intermediate stress level and had a minimum value of 0.122, 0.089, and 0.050, respectively, for the lubricants listed previously. Spinning friction tests with a MIL-L-7808 lubricant reported in reference 5 compared favorably with those results of reference 4.

In addition to the lubricant and stress factors, the eccentricity of the contact ellipse, that is, the ratio of the major axis to the minor axis, may affect coefficients of spinning friction. The shape of the contact ellipse is affected by ball-race-groove conformity, that is, the groove cross radius expressed as a percent of the ball diameter.

An increase in contact stress enlarges the contact area, and thereby increases the maximum sliding velocity between the ball and the race in the contact where spinning occurs. From theory, this sliding velocity can affect the mode of lubrication present; that is, boundary, elastohydrodynamic, or mixed. The mode of lubrication present in the outer portion of the contact area may greatly affect the measured spinning torque and hence the coefficient of spinning friction.

The objectives of the research reported herein were to determine the effect of ball-race conformity on spinning torque and on the coefficient of spinning friction. In order to accomplish these objectives, tests were conducted in the NASA spinning friction appara-

tus at ambient temperature with four ball-groove conformities, those being infinity (ball on a flat), 60, 55, and 51 percent. Test conditions included a drive speed of 1050 rpm, maximum Hertz stresses from 48 000 to 384 000 psi (330×10^6 to 2650×10^6 N/m^2), and a synthetic paraffinic oil having no additives as the lubricant with SAE 52100 steel 1/2-inch (12.7-mm) diameter ball specimens. Spinning friction coefficients were calculated from measured torques due to spinning. The results were analyzed with respect to contact loads, Hertz stresses, and ball-groove conformity. All experimental results were obtained with lubricant from the same batch and specimens from the same heat of material.

APPARATUS AND PROCEDURE

Test Specimens

The upper test specimen is a conventional 1/2-inch (12.7 mm) diameter bearing ball made of SAE 52100 steel having a nominal Rockwell C hardness of 63. The lower

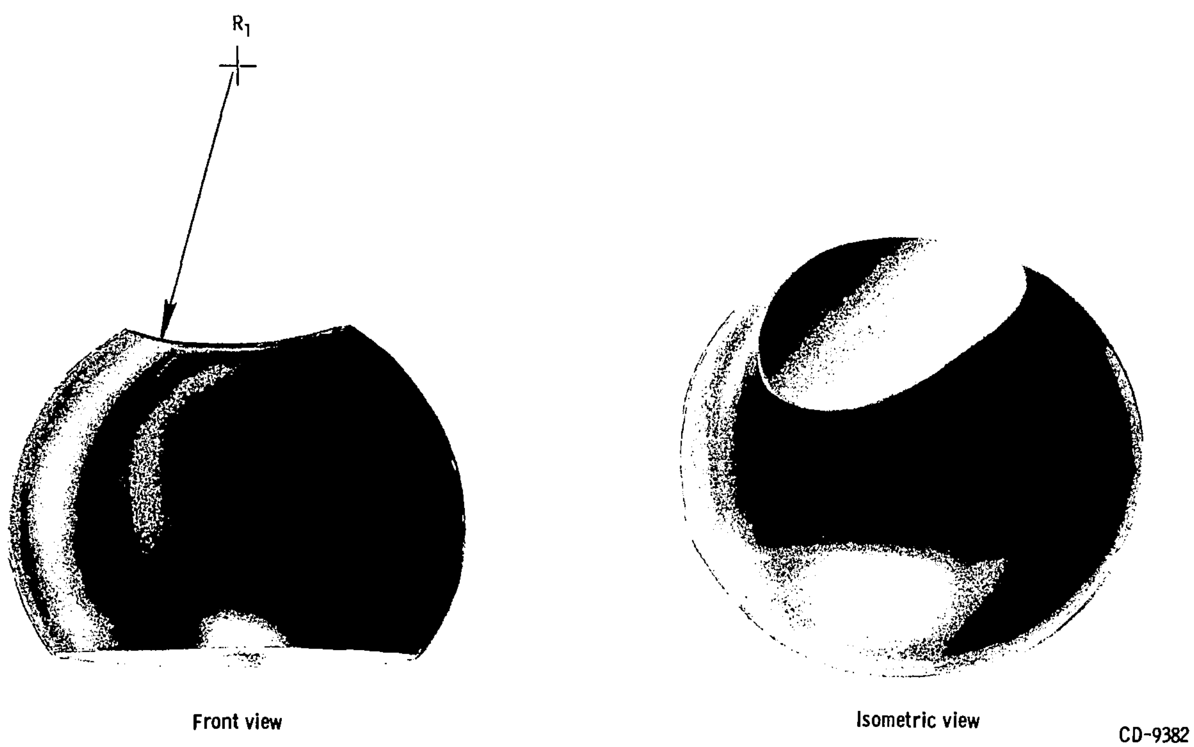


Figure 1. - Lower test specimen.

test specimen (fig. 1) which simulates a race groove of a bearing is a bearing ball of the same material from the same heat with a flat ground on it. A cylindrical groove having a radius R_1 is ground with its axis parallel to the flat. For the specimens used the radius R_1 was ground to conformities of 51, 55, and 60 percent of the upper ball diameter and infinity (ball on a flat). The groove radius expressed as the percent of the upper ball diameter is defined as the ball-groove conformity. The surface finish of the cylindrical groove was approximately 2 to 6 microinches (0.05 to 0.15 μm) rms.

Spinning Friction Apparatus

A spinning friction apparatus (see fig. 2) modified from that reported in reference 4 was used for the tests reported herein. The apparatus essentially consists of a turbine drive, a pneumatic load device, an upper and lower test specimen, a lower test-housing assembly incorporating a hydrostatic air-bearing and a torque-measuring system. In operation, the upper test specimen is loaded pneumatically against the lower test specimen through the drive shaft. As the drive shaft is rotated, the upper test specimen spins in the groove of the lower test specimen. This causes an angular deflection of the lower test specimen housing. This angular movement is resisted by a torsion wire. A mirror is attached to the bottom of the lower test housing. A light source in the torque-measuring device is reflected from the mirror back to a pair of photoelectric cells which center themselves on the reflected beam by means of a servomechanism thus providing an accurate measurement of the angular deflection of the lower test housing. This angular deflection is translated to a torque reading by means of a static calibration curve. During a test, the housing deflection is continuously recorded on a strip chart.

Operating Procedure

The lower test specimen was positioned in the lower housing, and the test lubricant was put on the specimen groove. The upper test specimen was set in the tungsten-carbide tip of the drive shaft and held in place by differential pressure between atmospheric and the vacuum in the line attached to the shaft. The drive shaft was lowered to bring the upper and lower test specimens in contact and the specimens were loaded. The vacuum line was then removed. Compressed air was fed to the drive turbine and the upper test specimen was driven with a given angular velocity. The excess lubricant within the contact area was collected in the centercup of the lower test housing.

The experimental value of spinning torque for a given test condition was read from the strip chart after a steady state value was reached.

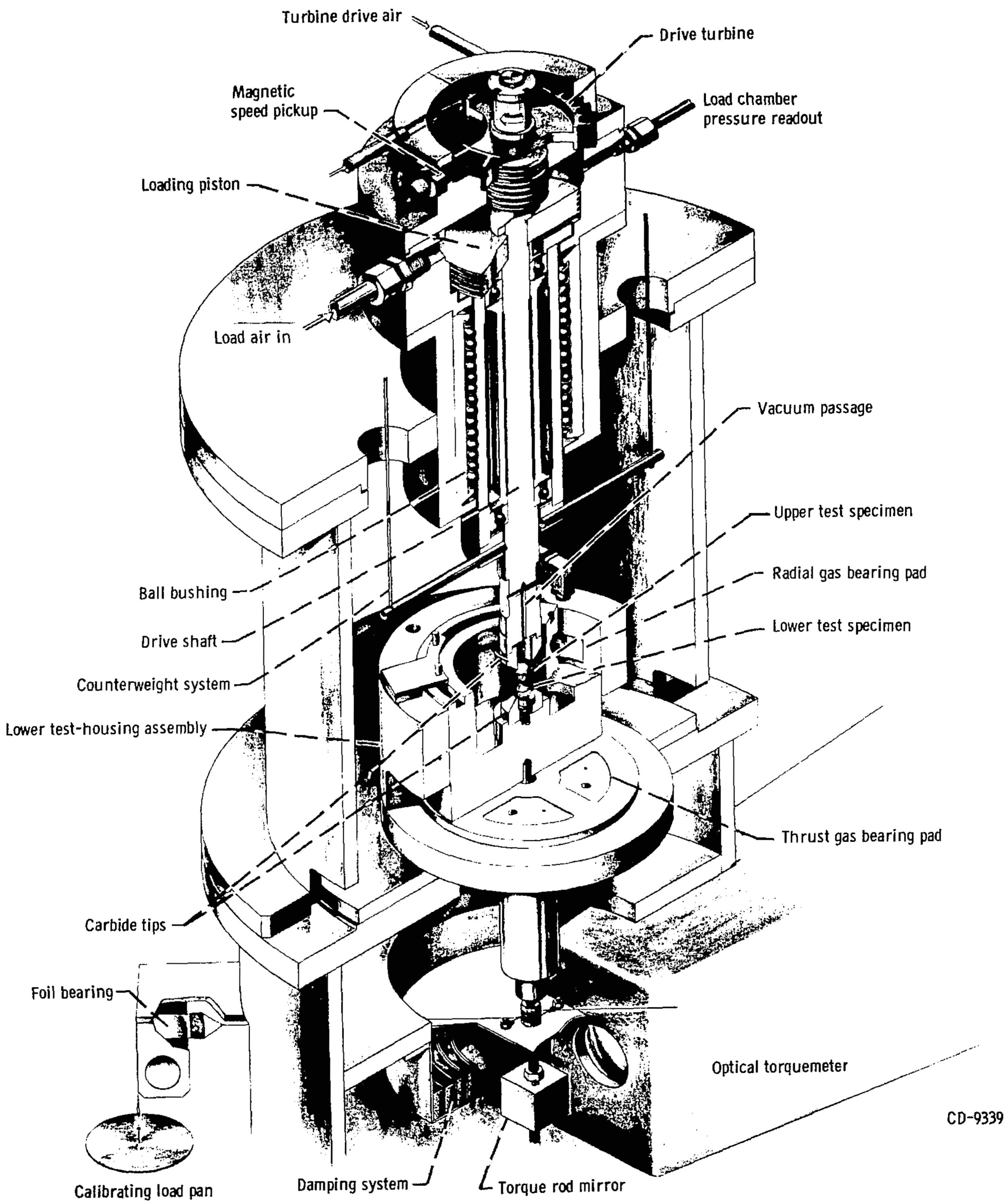


Figure 2. - Test apparatus.

The spinning friction coefficient f_s for each condition was calculated using the following equation from reference 6:

$$M_s = \frac{3}{8} f_s N a E(k)$$

where

M_s	measured spinning torque, lb-in. (N-m)
f_s	coefficient of spinning friction
N	normal load, lb (N)
a	major semiaxis of contact ellipse, in. (m)
E(k)	complete elliptic integral of second kind
k	$\left[1 - (b/a)^2\right]^{1/2}$
b	minor semiaxis of contact ellipse, in. (m)

RESULTS AND DISCUSSION

Groups of SAE 52100 steel 1/2-inch (12.7-mm) diameter balls were run in the NASA spinning friction apparatus against lower grooved test specimens of varying conformities. Standard test conditions included a drive speed of 1050 rpm, maximum Hertz stresses from 48 000 to 384 000 psi (330×10^6 to 2650×10^6 N/m^2), and ambient temperature (i.e., no heat added) with a synthetic paraffinic lubricant having no additives. The resulting torques due to ball spinning were measured, and the coefficients of spinning friction were calculated from the equation previously given,

$$f_s = \frac{8}{3} \frac{M_s}{N a E(k)}$$

Surface measurements of some lower test specimens were made on a random basis. The surface finishes measured varied from 2 to 6 microinches (0.05 to 0.15 μm) rms. These values included both surface roughness and surface waviness. Limited tests with specimens of different surface finishes indicated no measurable effect of surface finish on torque and thus coefficient of spinning friction under these test conditions.

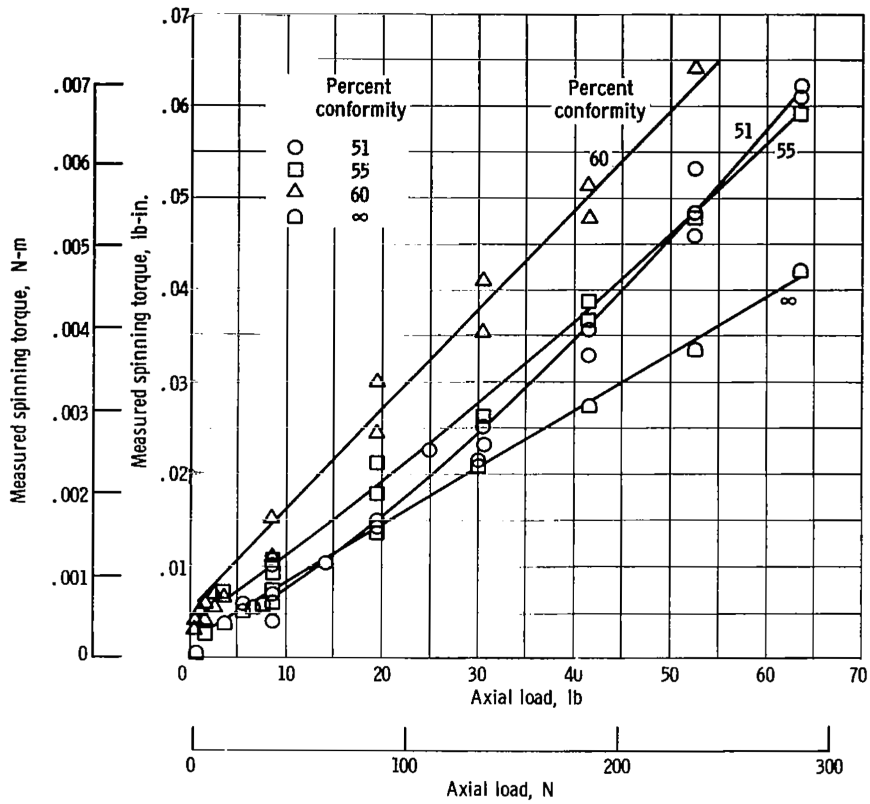


Figure 3. - Measured spinning torque plotted against applied axial load for ball-race conformities of 0.51, 0.55, 0.60, and ∞.

In order to determine the relation among spinning torque, conformity, and load, the data of figure 3 were obtained. In this figure, the measured spinning torque increases with applied axial load. However, no clear trend with conformity is seen.

Since the actual contact involved is one between two elastic bodies, a change in both contact area and shape occurs because of the change in conformity for a given load. A factor which is indicative of this change in contact area and shape, as well as load, is the calculated maximum Hertz stress. Figures 4(a) to (d) are plots of spinning torque against maximum Hertz stress for various conformities. Figure 4(e) shows that spinning torque increases with increasing Hertz stress. Further, the increase in spinning torque for a given increase in maximum Hertz stress is smaller for a higher percent conformity. Most important is that a clear trend of the effect of conformity and spinning torque is seen where torque decreases with increasing percent conformity for a given Hertz stress.

The coefficient of spinning friction was calculated from the measured spinning torque data. The calculated values of the coefficient of spinning friction are plotted as

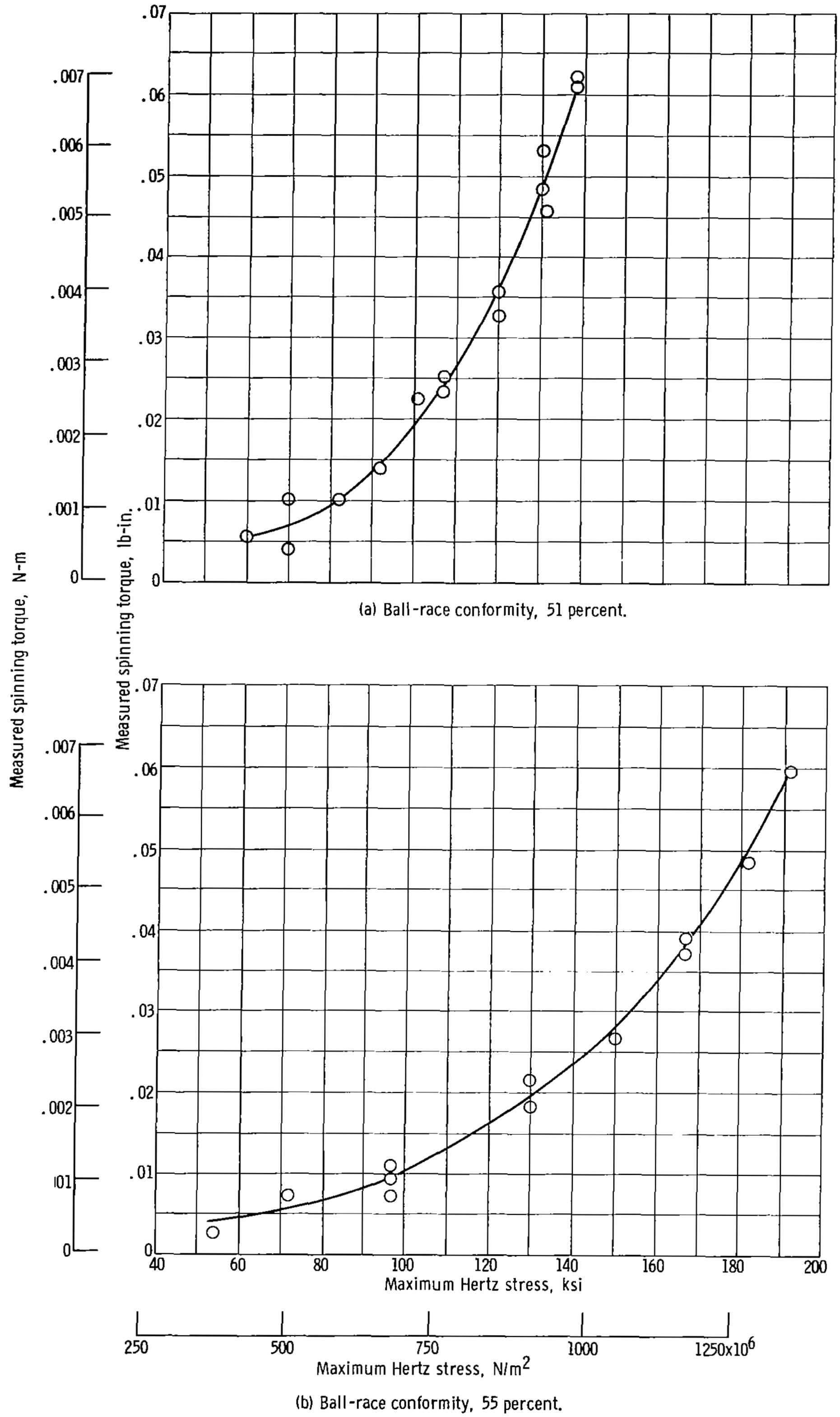
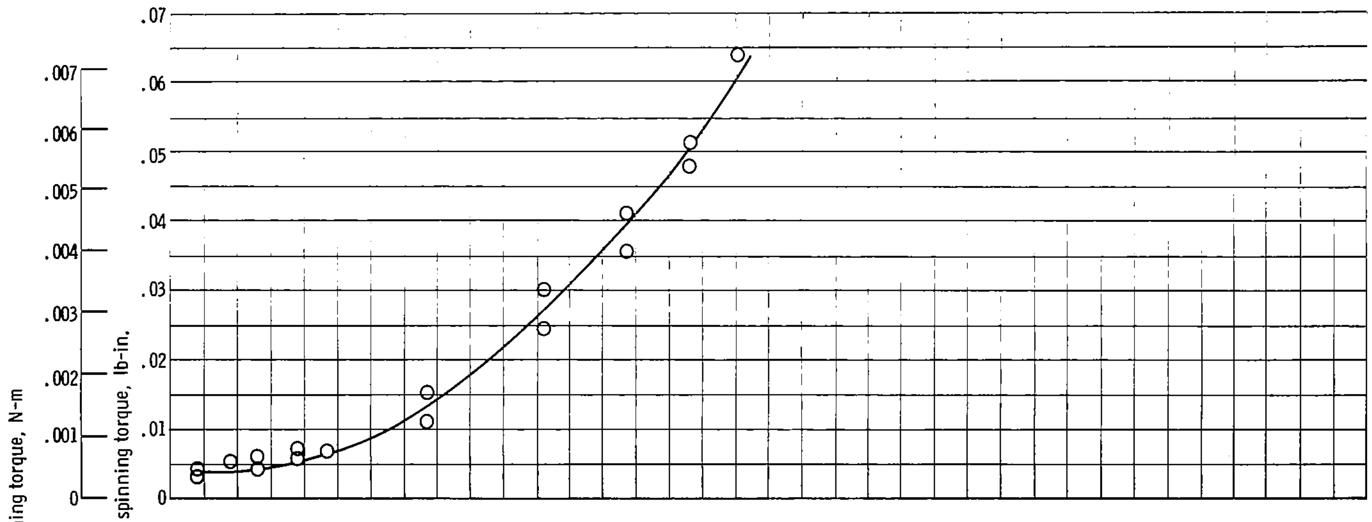
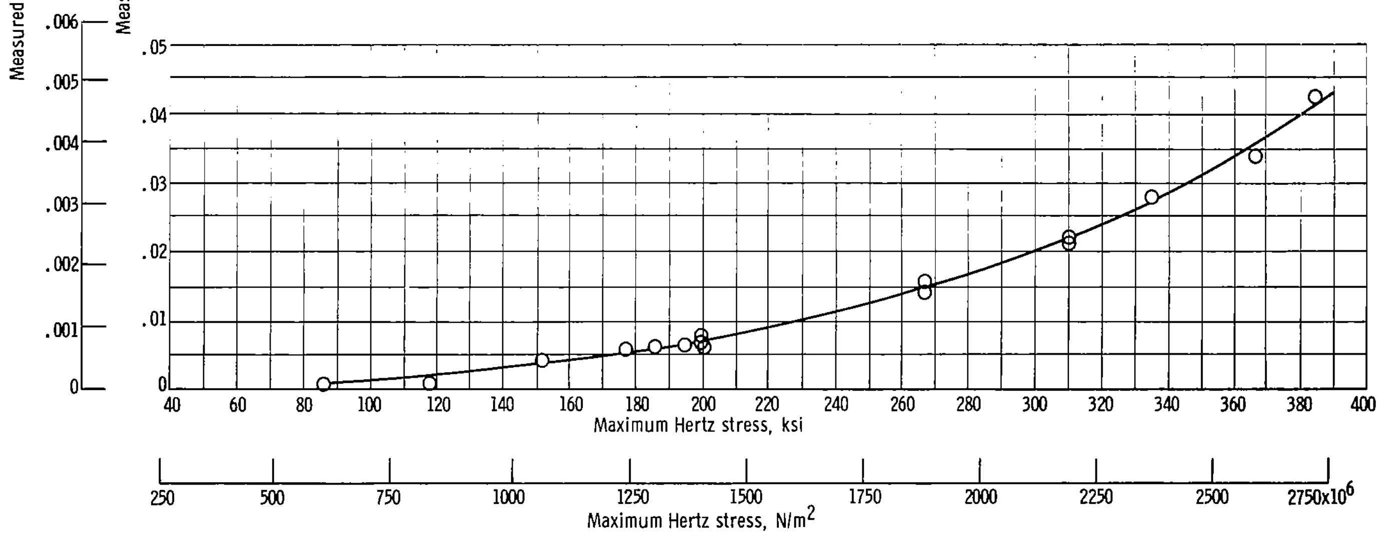


Figure 4. - Measured spinning torque as function of stress. Spinning speed, 1050 rpm; lubricant, synthetic paraffinic oil; temperature, room (no heat added).

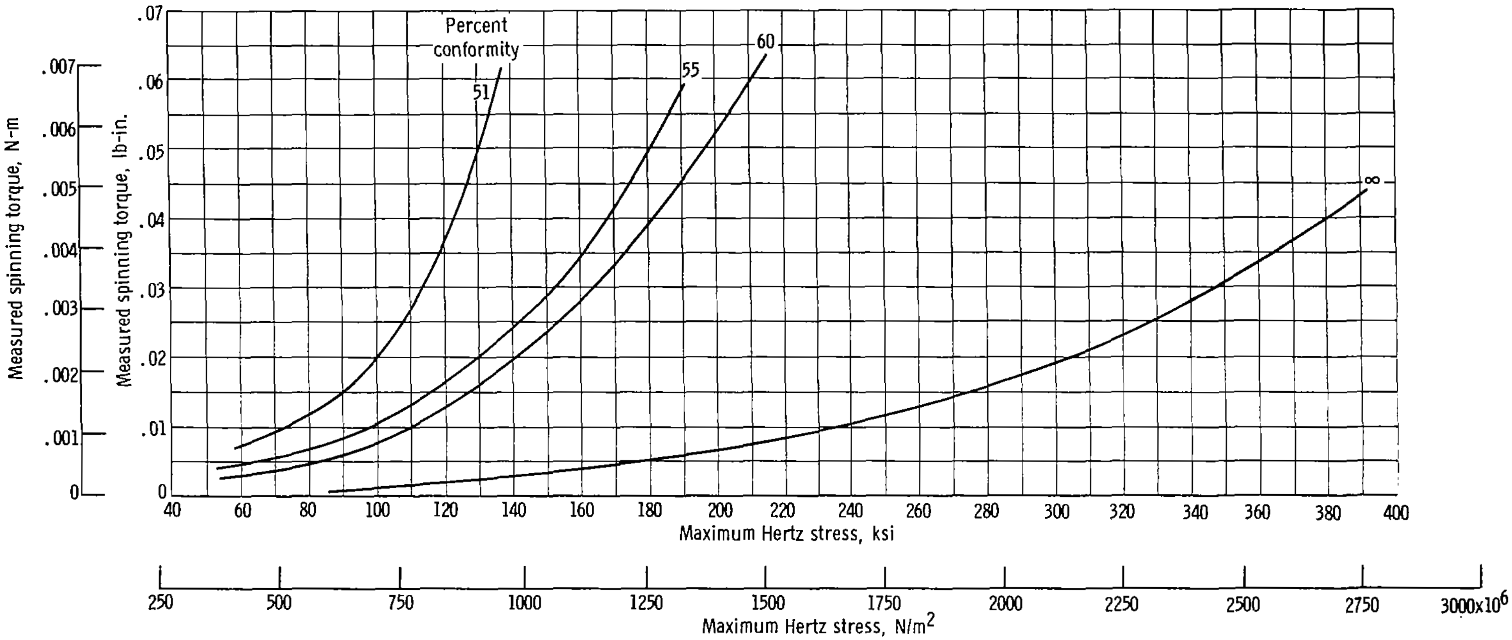


(c) Ball-race conformity, 60 percent.



(d) Ball-race conformity, infinite.

Figure 4. - Continued.



(e) Summary.

Figure 4. - Concluded.

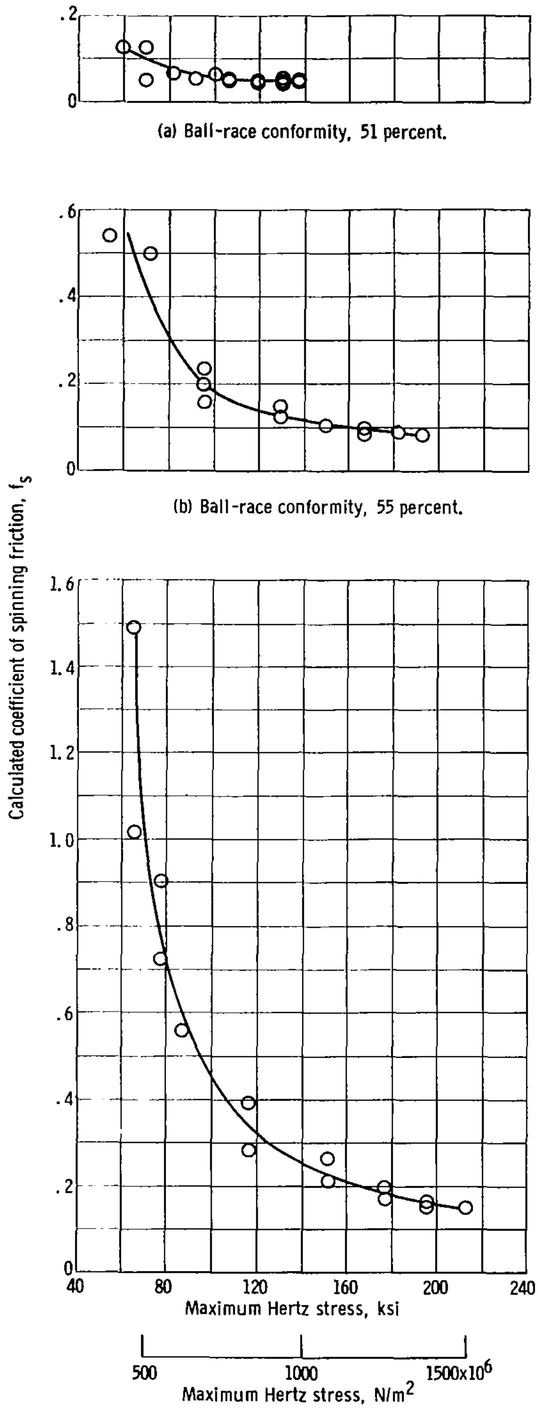
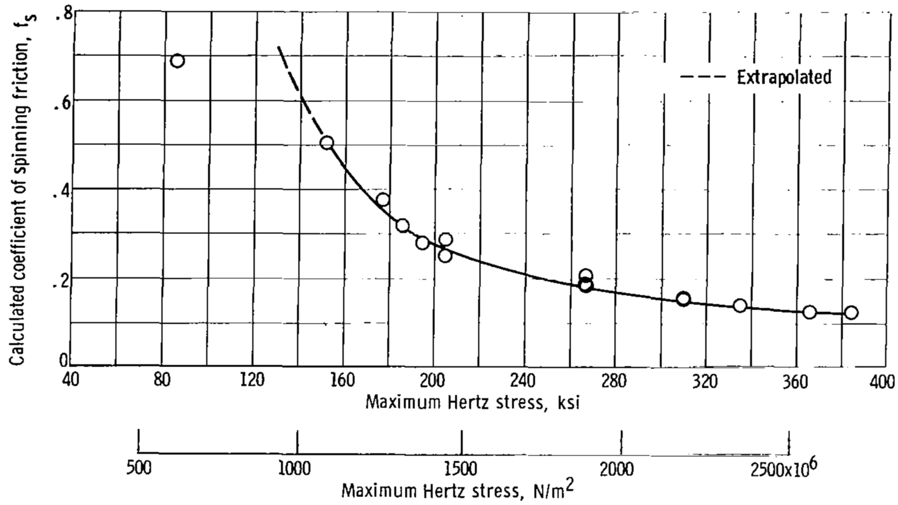


Figure 5. - Calculated coefficient of spinning friction plotted against stress. Spinning speed, 1050 rpm; lubricant, synthetic paraffinic oil; temperature, room (no heat added).



(d) Ball-race conformity, infinite.

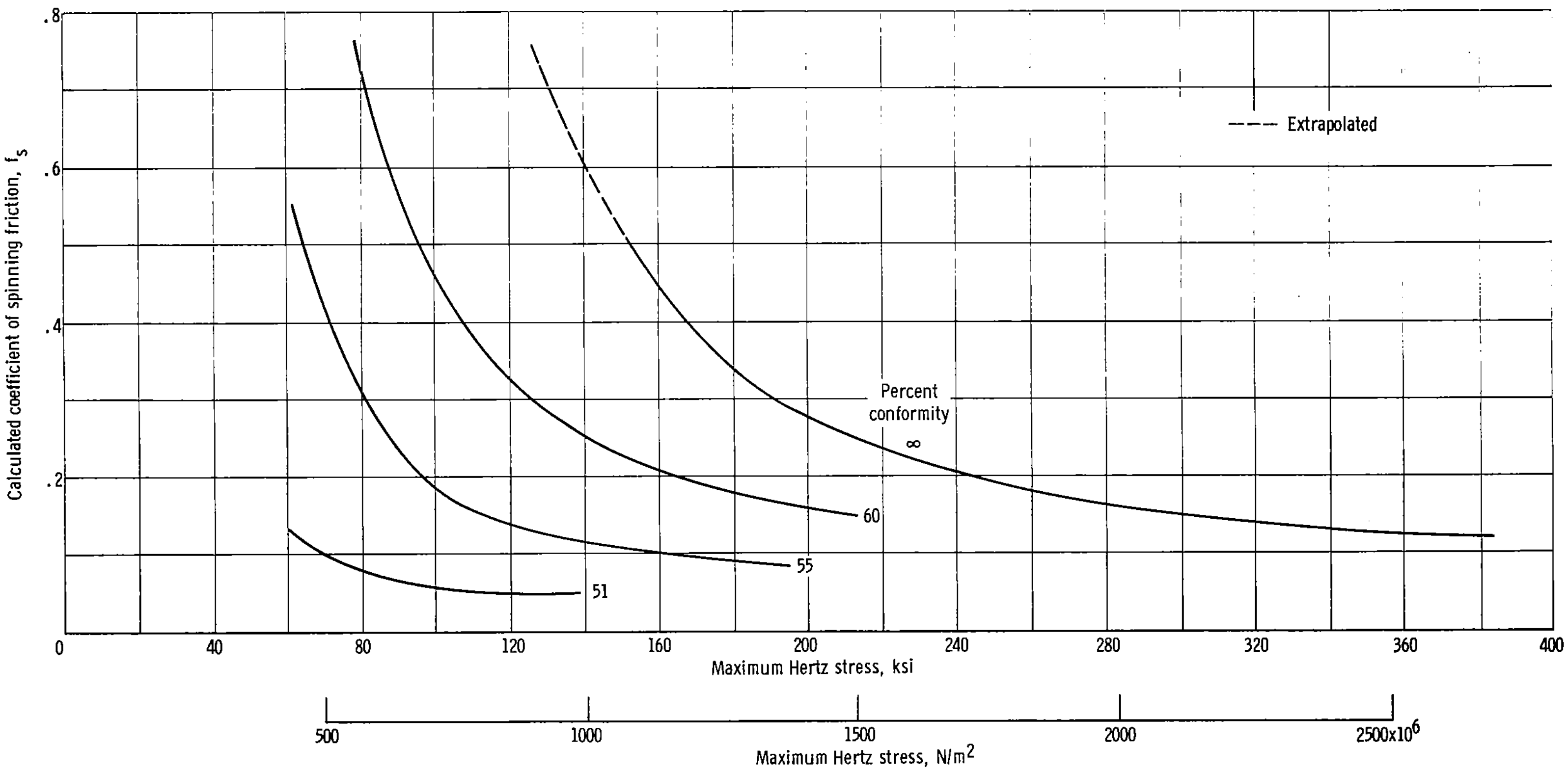
Figure 5. - Continued.

a function of maximum Hertz stress with percent conformity as a parameter in figure 5. It is seen, in general, that the coefficient of spinning friction decreases with increasing maximum Hertz stress. These results verify those reported initially in reference 4.

In order to better understand the effect of conformity on the coefficient of spinning friction, values of friction coefficient from the curves of figure 5 were plotted against the reciprocal percent conformity, for various values of Hertz stress, in figure 6. From this figure it can be seen that, regardless of the value of the contact stress, the coefficient of spinning friction approaches a minimum value of 0.05 at approximately a 51-percent conformity. Further, the magnitude of the difference of the coefficient of spinning friction for different contact stresses becomes larger as the percent conformity increases (i.e., the reciprocal of conformity decreases.)

For comparison with these results, it was necessary to establish a base line of results with unlubricated surfaces. For SAE 52100 steel sliding on SAE 52100 steel in a dry air environment, a coefficient of sliding friction of 0.5 was reported in reference 7. For steel-on-steel surfaces cleaned by outgassing at approximately 1832° F (1273 K) in vacuum, friction coefficients as high as 3.5 have been measured (ref. 8). These surfaces in a vacuum environment are considered free of ordinary contaminants and a large percentage of surface oxides.

Limited unlubricated tests were conducted with the four conformities. The specimens were ultrasonically cleaned in ethyl alcohol and vacuum dried for 24 hours prior to testing. Test conditions included maximum Hertz stresses ranging from 29 000



(e) Summary.

Figure 5. - Concluded.

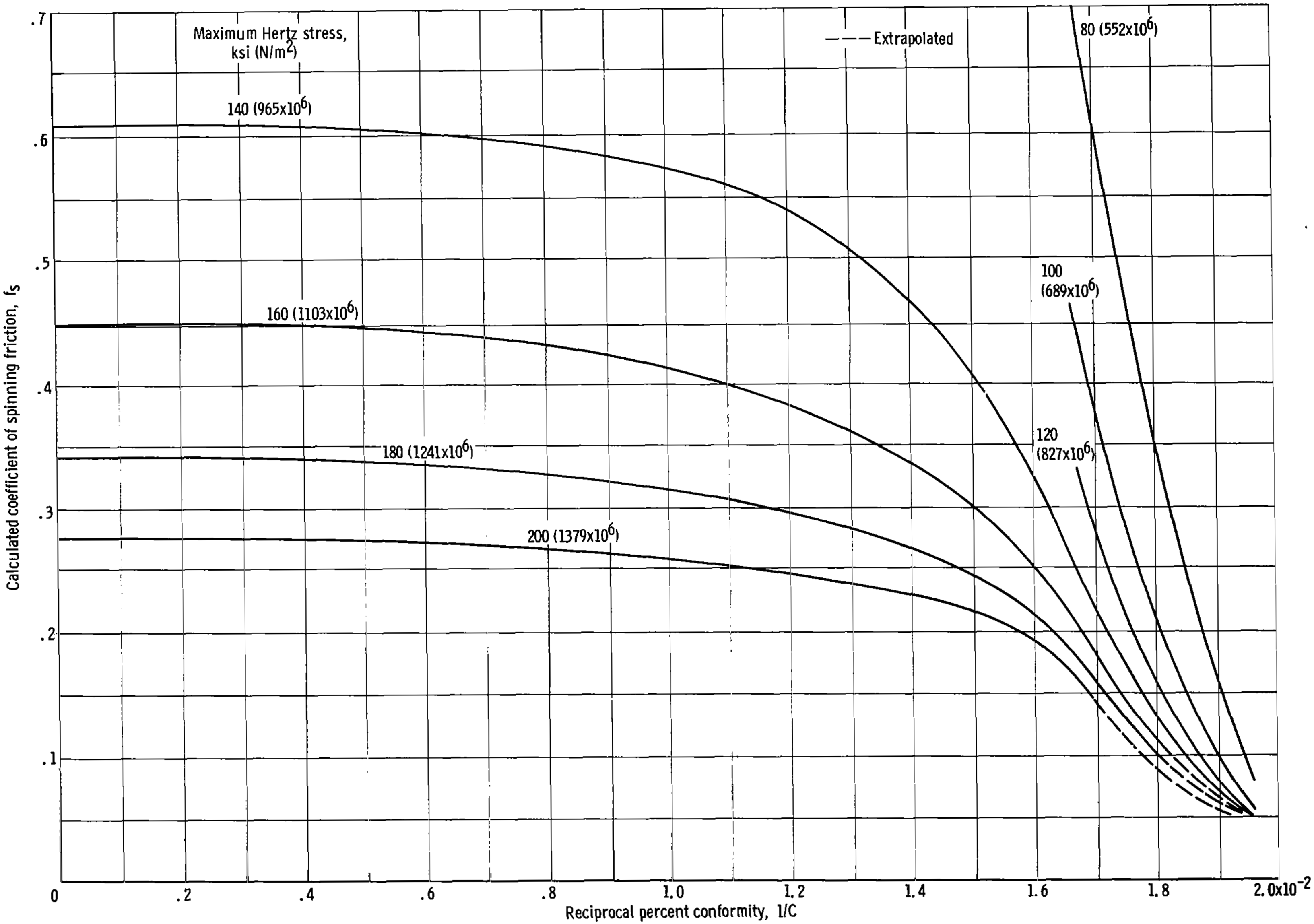


Figure 6. - Coefficient of spinning friction plotted against reciprocal conformity for various maximum Hertz stresses (from fig. 5).

to 147 000 psi (200×10^6 to 10105×10^6 N/m^2), normal loads from 0.7 to 3.2 pounds (3.1 to 14.2 N), with no heat added (room temperature) at a spinning speed of 1050 rpm. The calculated coefficients of spinning friction from these experimental results varied from approximately 2.5 to 5.0 and appeared to be independent of load and conformity.

It is commonly accepted that static and kinetic friction coefficients are independent of contact area, load, or stress. With a liquid lubricant present, this phenomenon would not be expected to hold for spinning friction because of elastohydrodynamic effects. Increases in contact stress enlarge the size of the contact area and thus increase the peripheral tangential velocity at any point on the perimeter of the contact zone. It is speculated that mixed lubrication occurs, that is, a combination of boundary and elastohydrodynamic lubrication, in the region of decreasing coefficient of friction in figure 5. Where the coefficient of spinning friction approaches a minimum of 0.05, it is probable that elastohydrodynamic lubrication prevails. Where the coefficient of friction is 0.5 or greater, it can be reasonably assumed that boundary lubrication prevails.

SUMMARY OF RESULTS

The NASA spinning friction tester was used to test SAE 52100 steel 1/2-inch (12.7-mm) diameter balls spun against race-groove specimens having ball-race conformities of 51, 55, and 60 percent and infinity (ball on a flat). The nominal hardness of the test specimens was Rockwell C 63. Test conditions were as follows: a maximum Hertz stress of 48 000 to 384 000 psi (330×10^6 to 2650×10^6 N/m^2); a spinning speed of 1050 rpm; room temperature (no heat added); and a synthetic paraffinic oil with no additives as the lubricant. Spinning torques were measured and the coefficients of spinning friction calculated. The following results were obtained:

1. The coefficient of spinning friction decreased with increasing maximum Hertz stress to a minimum value approaching 0.05 for all the conformities tested under lubricated conditions.

2. For a constant Hertz stress, the coefficient of spinning friction increased with decreasing conformity (i.e., increasing percent conformity).

3. As maximum Hertz stress decreased, the values of the coefficient of spinning friction for all of the conformities approached 0.5 under lubricated conditions.

4. Spinning torque increased with increasing maximum Hertz stress for each conformity tested.

5. For a given maximum Hertz stress, spinning torque decreased with decreasing conformity (i.e., increasing percent conformity).

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, April 3, 1968,
720-03-01-01-22.

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