DEVELOPMENT OF A PNEUMATIC SENSOR

FOR MEASURING THE TORQUE OF

INSTRUMENT BALL BEARINGS

by

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I. INTRODUCTION

With the increased trend toward miniaturization and high speeds in electromechanical systems, miniature and instrument ball bearings are finding wide applications. One of the many important operational characteristics of these bearings is the friction torque. This torque becomes quite critical, particularly in space and military systems where limited power is available.

A survey of the commercially available torque testers revealed that most were designed only for determining starting torque and bearing irregularities. Since these testers assess bearing quality or rate bearings comparatively, they tell nothing about the torque to be expected from the bearing in its final application. Others which were capable of measuring the torque for various loadings had only a limited speed range and were quite expensive.

A few investigators have developed testers capable of measuring torques at high speeds. These testers have, however, required a mechanical contact for measuring the torque which could lead to inaccuracies. In no case have reports been made of the effects of varying radial and axial loads on torque for instrument ball bearings.

Accordingly, the objectives of this investigation were as follows:

1. To design and develop a noncontact torque sensor of high sensitivity which was capable of measuring torques of instrument ball bearings under combined radial and axial loading when operating at high speeds.
2. To determine the effects on torque when radial and axial loads were varied.

II. REVIEW OF LITERATURE

In an effort to standardize bearing torque testers, a committee sponsored by the American Ordnance Association and the Department of Defense established MIL-STD-206. This specification, "Friction Torque Testing for Instrument Ball Bearings," establishes functional and calibration requirements for torque transducers and for testing procedure criteria.

Bearings tested in accordance with this specification are required to be rotated through 360 degrees clockwise and then 360 degrees counterclockwise at a maximum speed of 1/2 rpm at ambient temperature. The bearing rotational axis is vertical with an axial load of 75 to 400 gm depending upon the bearing size.

A variety of running torque testers conforming to MIL-STD-206 have been developed and are employed by both bearing manufacturers and users.[2] The Fafnir automatic torque tester utilizes a rubber belt by which a one directional rotation is given the outer race for two revolutions at one rpm, while the inner race is constrained by a spiral spring. The relative angular rotation of one race to the other being indicative of the torque. This instrument is capable of testing bearings up to 1 3/16 O.D. with a torque range of ± 0.18 to ± 1.8 gm-cm. The Bendix Bearing Quality Recorder and the Miniature Precision Bearing Mark III torque testers are identical in operation but with different ranges of torque sensitivity. In both, a two rpm drive system is employed with the torque on the inner race opposed by a Microsyn generator. The Bendix instrument has a torque range from 50 to 20,000 mg-mm compared to 2,500 to 50,000 mg-mm for the MPB machine. Axial loads on the bearing range from 15 to 400 gm. Perhaps the most widely used torque tester in industry is the Sunshine Ball Bearing Torque Tester whose principle of operation is very similar to the Bendix and MPB instruments.

Essentially, the above instruments are used only for determining starting torque and diagnosing bearing characteristics and irregularities such as lubricant drag, ball and retainer interference, contamination, dents, scratches, eccentricity, etc. It should be noted that these testers do not measure the torque expected from the bearing in its final application. Consequently, these testers assess bearing quality or rate bearings comparatively.

In order to study and evaluate dry bearing materials, a dynamometer was developed for determining the friction torque (and coefficient of friction) of bearings under varying loads and speeds.[3] The test bearing was mounted at the center of a shaft which was supported by two split pillow blocks. The shaft was driven by a belt and pulley from a shunt-field DC motor with the rotational speed measured by a tachometer-generator attached at the opposite end of the shaft. A rod, which served as the torque arm, was attached to the underside of the bearing housing on which a weight pan was also mounted for loading the bearing radially. Beneath the weight pan a miniature precision bearing contained in a yoke served as the pivot point for another rod to which was affixed the core of a linear variable differential transformer whose output was indicative of the torque of the test bearing. Bearings tested by this dynamometer could be radially loaded only and run to a speed of 2,000 rpm.

The Sadamel Electro-Microdynamometer[4] was designed to operate on the principle of the galvanometer. The torque to be measured was applied to a shaft suspended by platinum wires from shockproof supports, which carried a mirror and a moving coil. The coil was free to rotate between a permanent magnet and its core. While the shaft was unloaded, a beam of light reflected off the mirror onto an indicating scale gave the angular displacement of the shaft relative to some arbitrary zero point. This reflection from the mirror was set to the zero point by adjusting a knob attached to one of the shaft suspension wires. When a torque was applied to the shaft, it was counter-balanced by the electromagnetic torque developed in the moving coil. The current required to null the system was thus an indication of the applied torque. This current was measured and indicated by a second light beam galvanometer which reflected onto a second scale calibrated in units of torque. This torque tester was capable of measuring torque to as low as 0.0001 gm-cm. Full scale torque ranges from 0.0003 to 3 gm-cm and 0.03 gm-cm are available with 0.5 percent accuracy depending upon the model. Bearings could be loaded axially up to 250 gm and rotated at speeds to 2,800 rpm.

For studying torque characteristics of grease-packed R-2 and R-3 instrument ball bearings, J. C. Lawrence[5] developed a high speed torque tester. The tester consisted of a vertical shaft shouldered in steps to accept the inner races of R-2, R-3, and R-4 bearings and driven by an air motor. A 1-lb. hollow steel cylinder was constructed to fit over the outer race of the test bearing. By adding mercury to the hollow cylinder, axial loads up to 7 lb. were possible. A vertical slotted shaft extended from the top of the load cylinder to a blade which was attached to a spiral spring whose angular displacement was calibrated to measure torque.

The R-2 and R-3 test bearings were lubricated with three types of greases: MIL-G-3278, MIL-G-7118, and a silicone grease. Torque tests were then performed with the test bearings subjected to 1 and 3 lb. axial loads at the two speeds of 2 and 15,000 rpm. From the data obtained, the following conclusions were made:

1. Bearings lubricated with the grease conforming to MIL-G-3278 yielded the lowest torque, while those lubricated with the silicone grease gave the highest values of torque.
2. The observed torque level due to 1 and 3 lb. axial loads for silicone grease tested did not vary significantly.
3. R-2 bearings with 2-piece construction retainers displayed a decrease in torque with increasing grease content for all greases. If the major source of torque

in this region was due to retainer drag, then increasing the amount of lubricant may have resulted in better lubrication, thus causing a reduction in torque.

In order to study the effects of service environment on the performance of ball bearings, P. R. McCarthy of Gulf Research Corporation[6] developed a transducer capable of testing bearings at high speeds and high temperature under combined radial and axial loading. Radial loads were applied by wrapping a flexible steel strap around the bearing housing and a pulley located directly below the bearing housing. The pulley was attached to a beam pivoted at one end with the other end free to rotate. When weights were added to the free end of this beam, a radial load was applied to the test bearing. By means of another weight and lever system, a dowel was forced to make contact with the bearing housing cover plate causing an axial load to be applied to the test bearing. Bearings varying in size from 20 to 25 mm bore were tested to speeds of 45,000 rpm at temperatures ranging from ambient to 600 F.

Many investigations of the running torque characteristics of instrument ball bearings operating at high speeds with various types and amounts of lubricants have been conducted by H. H. Mabie.[7] With the inner race of the test bearing driven at the desired speed by an air turbine, the resistance torque acting on the outer race was determined by observing the angular rotation of a disk containing a pendulous mass which was attached to the bearing housing. In order to measure the angular displacement, marks in one degree increments were scribed around the periphery of the disk with the displacement being measured relative to a pointer located on the air turbine housing. Turbine speeds were determined by means of a light and photocell arrangement.

In Mabie's tests, torque characteristics for R-2, R-3, and R-4 instrument ball bearings lubricated with MIL-L-6085A oil and MIL-G-3278A grease were determined for speeds ranging from 1,000 to 40,000 rpm. In addition to varying the lubricant, Mabie also obtained torque data for bearings with different retainers. For R-2 bearings, crown, ribbon, phenolic, and PTFE retainers were used, while the R-3 and R-4 bearings were tested with ribbon and PTFE retainers.

Based upon the results of these tests, Mabie reached the following conclusions:

1. Good repeatability in data was obtained for oil and grease-packed bearings up to

a minimum speed of 30,000 rpm.

2. The quantity of lubricant appeared to be the controlling factor in obtaining repeatability of data and not the method of retainer control.
3. No change in the torque characteristics of the bearings on subsequent tests was observed as long as the quantity of oil was brought back to the original level before retest.
4. The torque-time relations for R-2 bearings were found to be essentially constant for both oil and grease lubricants. R-3 and R-4 bearings lubricated with oil showed a small decrease in torque for the time tests.

In studying the effects of varying axial loads on the torque characteristics of R-3 instrument ball bearings, Clarke[8] developed a tester which used SR-4 strain gauges mounted on a thin cantilever beam as the torque sensor. A balanced disk was attached to the bearing housing such that when the inner race of the bearing was rotated at the desired speed by an air turbine, the resulting torque caused the housing and disk to rotate. To restrain this rotation, the disk was equipped with an L-shaped pointer which contacted a small cup located near the end of the cantilever beam. By locating SR-4 strain gauges near the base of the beam, the measured strain due to beam deflection served as an indirect measurement of the torque.

Clarke's tests were concerned with testing R-3 bearings subjected to constant radial but varying axial loads, lubricated with MIL-L-6085A oil and MIL-G-3278A grease. All tests were conducted with a 47 gm radial load and 6, 12, 24, 47, 94 gm axial loads.

Based upon the results of these tests, the following conclusions were reached:

1. The effect of axial loading on a R-3 instrument ball bearing with 47 gm radial load is negligible until the axial load equals or exceeds the radial load.
2. Grease lubricated bearings had about twice the torque of bearings lubricated with an equal weight of oil.
3. No change in torque characteristics was observed for subsequent tests as long as the quantity of lubricant was brought back to the original level before retest.

As seen from the above review, many schemes have been employed for determining bearing torque characteristics. Each, with the exception of Mabie's tester, have required a mechanical contact for measuring the torque. Since such contacts can lead to inaccuracies in measurements, it is desirable to have a measuring technique which requires no contact but yet has very high sensitivity. In the following section, the development of such a transducer will be discussed.

III. DEVELOPMENT OF THE PNEUMATIC TORQUE TRANSDUCER

During the past twenty years, pneumatic techniques have been applied to many areas of precision engineering. Pneumatic comparators have been developed by Graneek, et al.[9,10] for gauging and surface finish measurement. When operated at their maximum sensitivity, these comparators were found to be capable of detecting changes in displacement and surface roughness of a few microns. The flapper-nozzle valve, which operates on the same principle as the above mentioned comparators, has wide application in the field of pneumatic feed-back control systems. Since this flapper-nozzle valve is the "heart" of the proposed torque transducer, consideration of its operational characteristics will be undertaken.

Operational Characteristics of a Flapper-Nozzle Valve

The flapper-nozzle valve, shown schematically in Figure 1, consists of two orifices in series. The upstream orifice, 0_1, is of constant area, A_1, while the downstream orifice, 0_2, has a variable cylindrical flow area, A_2, depending upon the diameter of the nozzle, d, and the distance, x, between the nozzle and flapper. This variable cylindrical flow area is shown in Figure 2. The area of orifice 0_2 can, therefore, be given as:

$$A_2 = \pi dx \qquad (1)$$

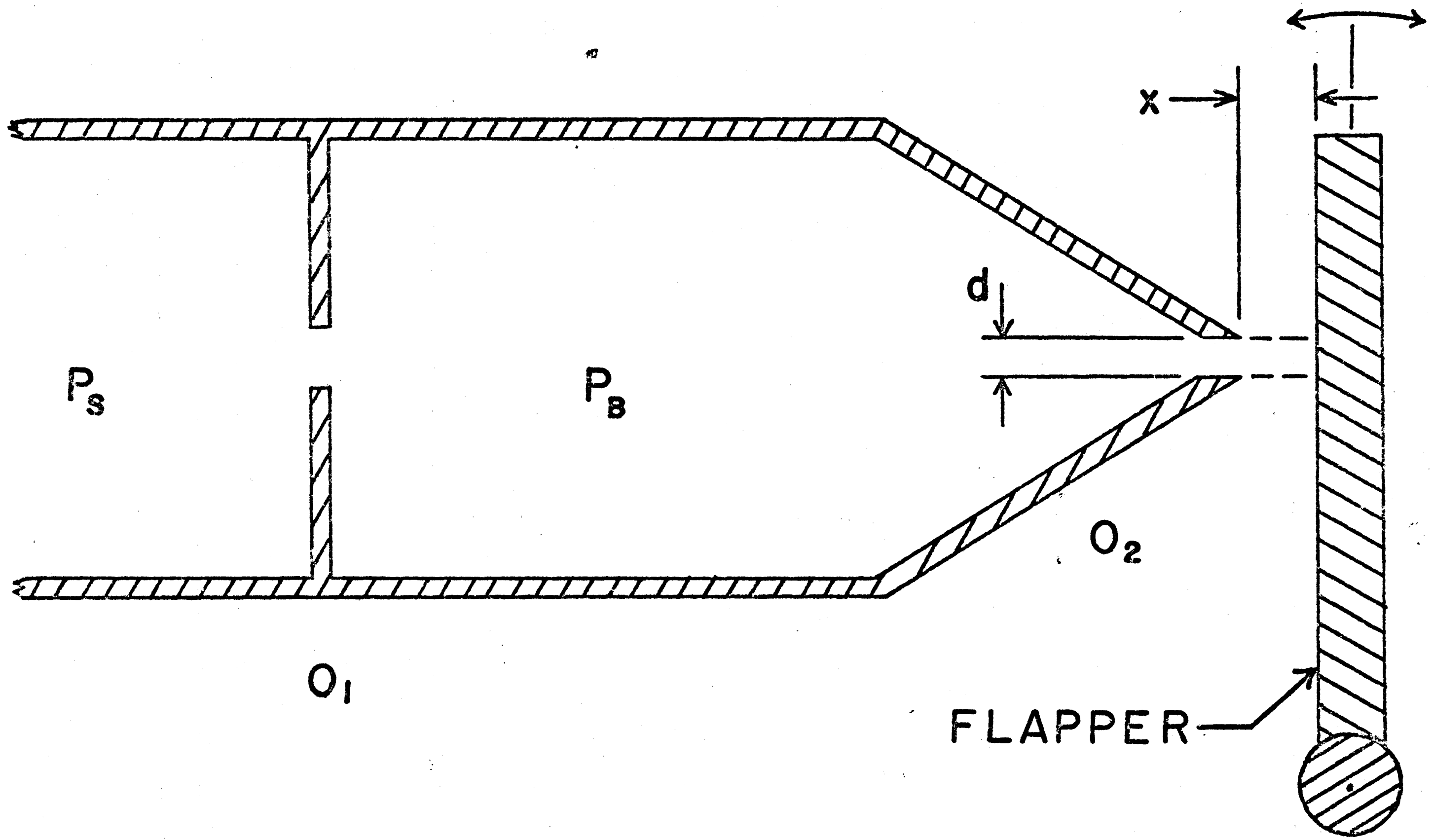

FIGURE 1. FLAPPER-NOZZLE VALVE

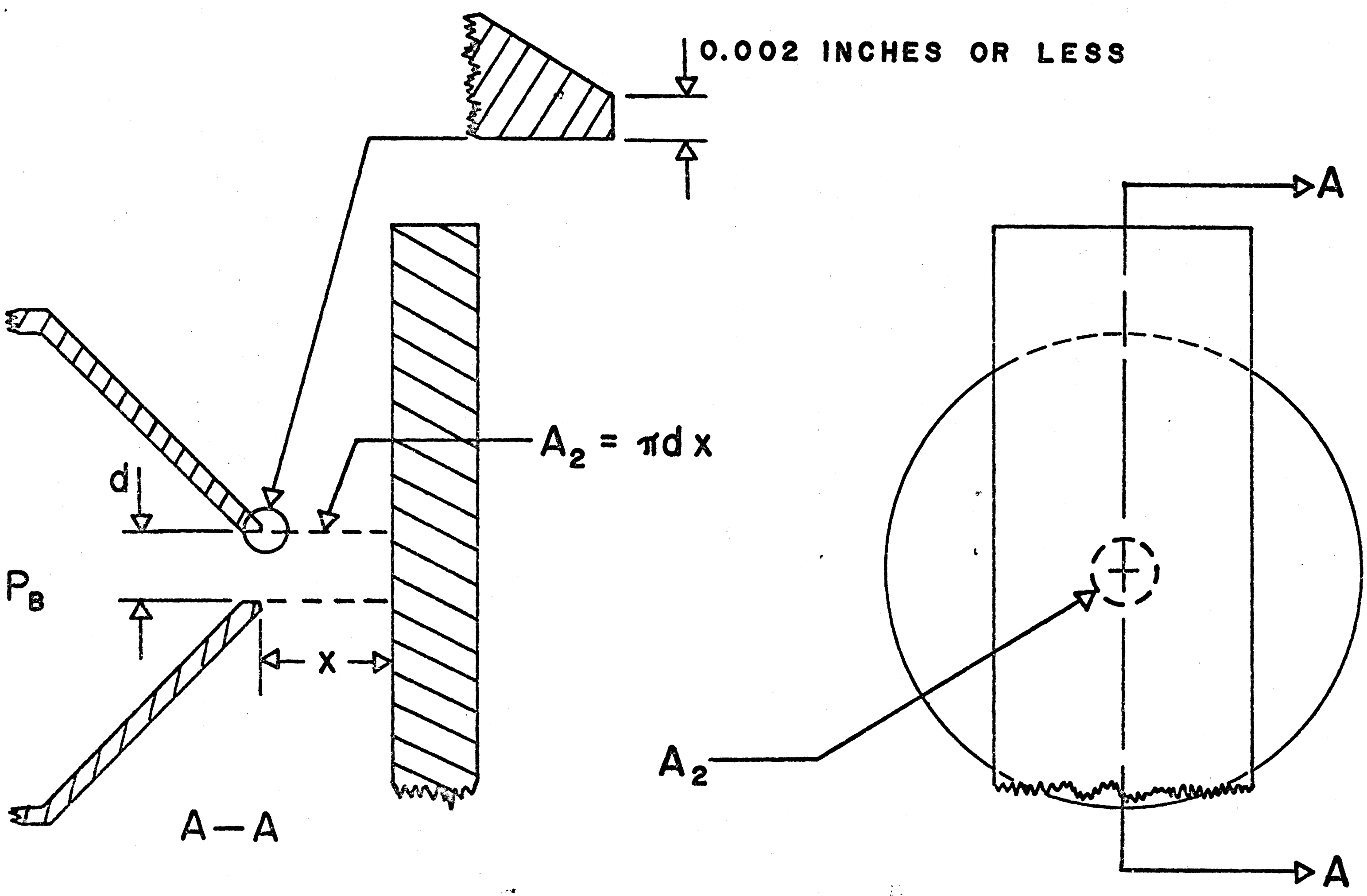

FIGURE 2. VARIABLE ORIFICE AREA, A_2

A gas, such as air, from a constant pressure source is first supplied to the orifice, 0_1, and escapes to the atmosphere through 0_2. P_s represents the constant supply pressure upstream from the orifice, 0_1, while P_b represents the back pressure existing between 0_1 and 0_2, both pressures being referred to atmospheric pressure. The back pressure P_b is dependent upon the resistance to flow through 0_2. A decrease in 0_2 provides greater resistance to flow, and, therefore, causes P_b to increase or vice versa. Obviously, the range of P_b is dependent upon P_s. When the flapper is completely against the nozzle ($A_2 = 0$), P_b equals P_s. When the flapper is removed such that $A_2 >> A_1$, P_b drops to a pressure of one atmosphere.

The relationship between P_b/P_s and A_2/A_1 can be determined experimentally for constant values of A_1 and P_s. Figure 3 illustrates the shape of such a characteristic curve. From this curve it can be seen that for values of P_b/P_s between points A and B, the curve follows the straight line equation:

$$P_b/P_s = C_1 - (C_1/C_2)(A_2/A_1) \qquad (2)$$

If the flapper position x (or A_2) can be arranged in such a manner so as to depend on some quantity such as force or torque, then corresponding changes in that quantity may be determined by measuring corresponding

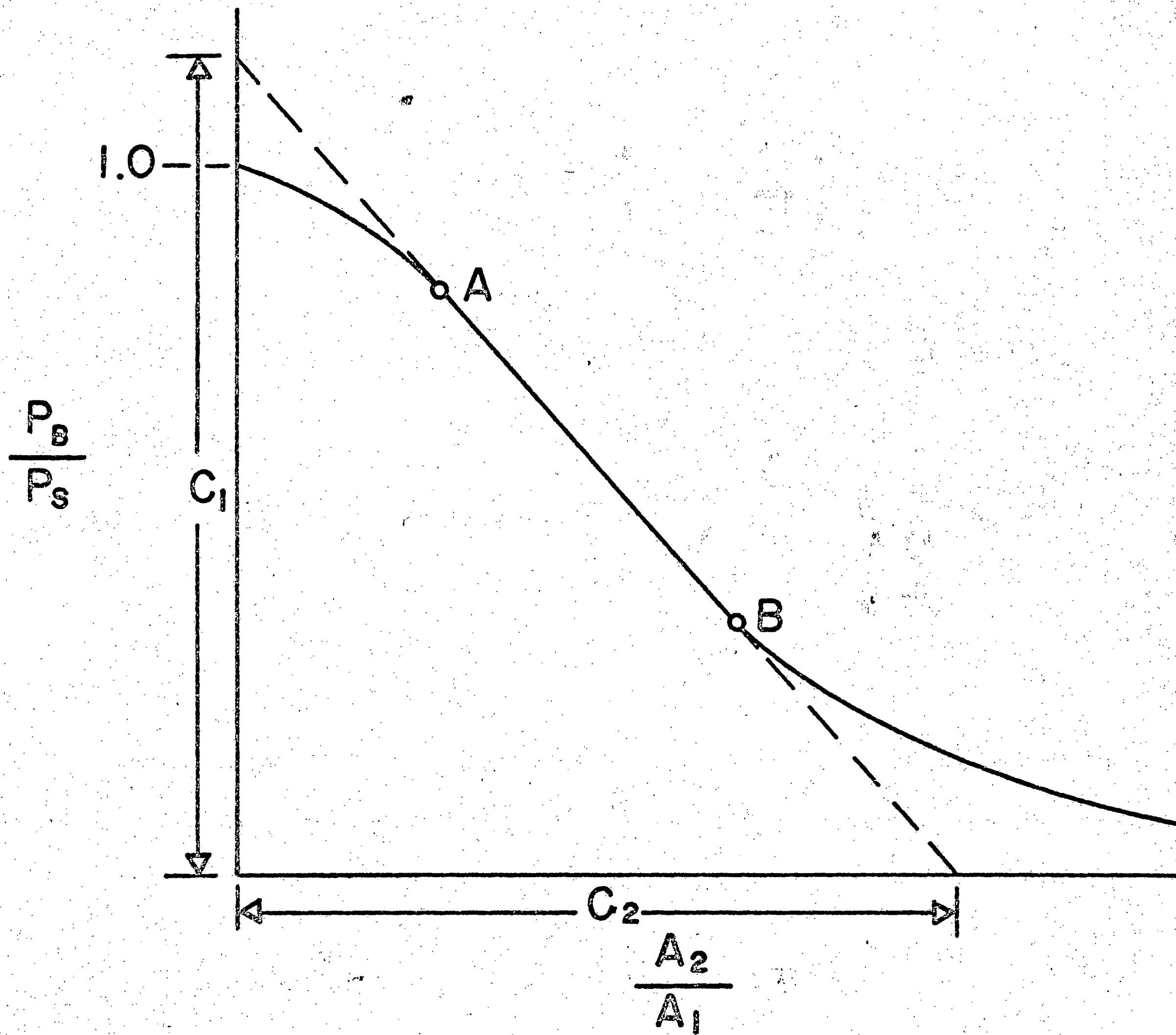

FIGURE 3. CHARACTERISTIC CURVE FOR TWO ORIFICES IN SERIES

changes in P_b.

The rate of change of P_b with A_2 is given by:

$$dP_b/dA_2 = -(C_1/C_2)(P_s/A_1) \qquad (3)$$

and is greatest when A_1 is least. The largest value of A_2/A_1 in the linear range occurs at B. For maximum sensitivity A_1 should be chosen to satisfy:

$$A_1 = A_2max/(A_2/A_1)B \qquad (4)$$

where A_2max may be arbitrarily selected as long as it remains within the sensitive range of the jet.

The speed of response improves by increasing P_s and A_1 or by diminishing the volume between the two orifices 0_1 and 0_2. Since high sensitivity and quick response place conflicting demands on the optimum size of 0_1, A_1 is generally determined by sensitivity requirements. Improved response is then obtained by reducing the volume to a minimum and making P_s as large as possible.[9]

Shearer, et al.[11] determined the force acting on the flapper for sharp-edged nozzles as a function of the distance x between the flapper and nozzle. The sharp-edged nozzle, shown in Figure 2, was machined so that there was a flat surface of 0.002 in. or less on the metering edge. The equation for the force on the flapper, assuming incompressible flow, is given by:

$$F = \pi r^2 P_b [1+(2C_d x/r)^2] \qquad (5)$$

Where r = nozzle radius

P_b = back pressure

C_d = discharge coefficient of orifice 0_2.

In the above equation, the second term in the bracket may be neglected when x is small compared to r.

Norwood[12] determined experimentally that the force acting on the flapper was approximately equal to the pressure times the projected area of the nozzle. This relation, given by equation (6), held as long as the gas flow was parallel to the flapper after impingment.

$$F = \pi r^2 P_b \qquad (6)$$

Equation (6) verified that the force was independent of flapper-nozzle separation when x/r was less than one. It was found, however, that once the pressure drop across the nozzle became large, a particular position of the flapper caused an abrupt increase in the force on the flapper. Shadow graphs indicated that when this phenomenon occurred, the flow was no longer parallel to the flapper but had shifted back along the outer sides of the nozzle. This additional force on the flapper was due to momentum and was found to be approximately proportional to the flow.

<u>Flapper-Nozzle Valve as a Torque Sensor</u>

Consider a torque from an external source to be applied to the flapper. If the supply pressure, P_s, is high enough, equilibrium of the flapper will be established with an equal but opposite sense torque. This equilibrant torque consists of the force on the flapper due to jet impingment times the distance between the point at which the force may be considered to act and the pivot point of the flapper. Since it was shown in the previous section that the flapper force is proportional to the back pressure, P_b, changes in P_b correspond to a proportional change in the externally applied torque. This discussion of the flapper-nozzle valve illustrates the principle of operation of the proposed instrument ball bearing torque sensor.

Application of the above principle for determining bearing torque characteristics proved to be relatively simple. A hollow cylinder was machined so that only a light push was required to insert the bearing. Since all the bearings to be tested were of stainless steel, this bearing housing was made of the same material to allow for thermal expansion and contraction. To insure against slippage between the bearing and the housing, a cover plate, which was fixed to the housing, was designed to bear on the outer race of the bearing. Thus, when the inner race of the bearing was turned at a desired speed, the housing would rotate with the outer race. A groove was machined on the outer surface of the housing, parallel to the bearing axis, in which an aluminum plate was pressed. With a means of supporting and rotating the inner race of the bearing, the housing-plate-combination becomes the flapper with the bearing serving as the flapper pivot point.

To oppose the bearing torque acting on the flapper, prepurified nitrogen was supplied to an orifice and nozzle in series with diameters of 0.015 and 0.052 in., respectively. An aluminum block was machined with passageways which allowed the nitrogen, entering the block at a constant pressure, to flow first through the orifice and then to the nozzle. The orifice, which resembled a bolt, was machined such that the nitrogen was required to enter along the bolt axis and exit into another passageway, perpendicular to the axis, leading to the nozzle. O-rings between the orifice and block prevented any leakage. Since it was desirable to have the capability of moving the nozzle toward and away from the flapper, the nozzle was affixed to a cantilever beam located on top of the aluminum block. This beam, set in brass guides, could be moved relative to the flapper by a power screw arrangement. In order to have both an adjustable position and nitrogen flow to the nozzle, a flexible piece of tubing was used to connect the nozzle to the passageway in the aluminum block. For measuring the back pressure between the orifice and nozzle, a statis pressure tap was machined in the block perpendicular to the passageway leading from the orifice to the nozzle. Through a calibration procedure, measured values of the back pressure could then be related to the torque acting on the flapper. Figure 4 shows the bearing housing-nozzle arrangement for sensing the bearing torque.

Flapper Stabilizer

At a given speed, the torque of ball bearings exhibit a random time history rather than a constant value. Due to the fluctuation in torque, average values are difficult to determine. In attempting to damp these oscillations, another plate was pressed into the bearing housing 180° from the flapper. Three brass shims, to act as damping vanes, were inserted into this plate parallel to the direction of rotation

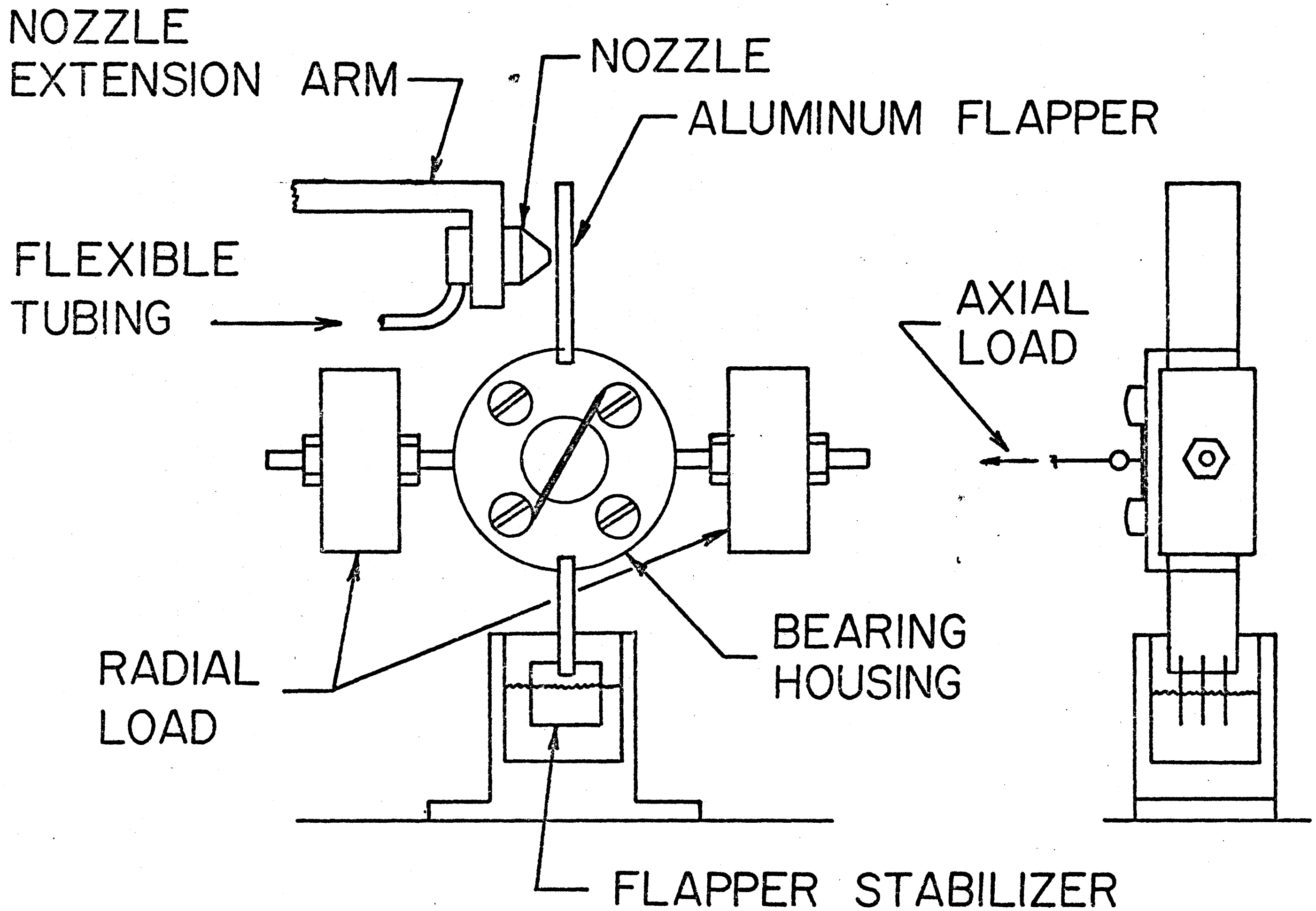

FIG. 4 BEARING TORQUE SENSOR CONFIGURATION

of the bearing. By immersing these vanes in a viscous fluid, the shearing action between the vanes and the fluid served to damp the torque fluctuations. Tests were conducted to determine the effectiveness of several fluids as dampers. Glycerine, 50-weight motor oil, and silicone oils of varying viscosity were used. Due to the high sensitivity of the measuring technique, a silicone oil, with a viscosity of 60,000 centistokes, was found necessary to effectively damp the oscillations. The arrangement of the damping vanes, located directly below the bearing housing, is shown in Figure 4.

Speed Source

The author was fortunate to have access to an air turbine developed by H. H. Mabie [7] for driving R-3 instrument ball bearings to speeds of 40,000 rpm. This Pelton wheel type turbine, also used by Clarke[8] in his investigations, had been proven to be both smooth running and safe at high speeds. A schematic of the Mabie turbine is shown in Figure 5.

Speed Measurement

The rotational speed of the air turbine was measured by a light and photocell arangement. Since it was the same arrangement used by Clarke[8], only a brief description will be given. The measuring tech-

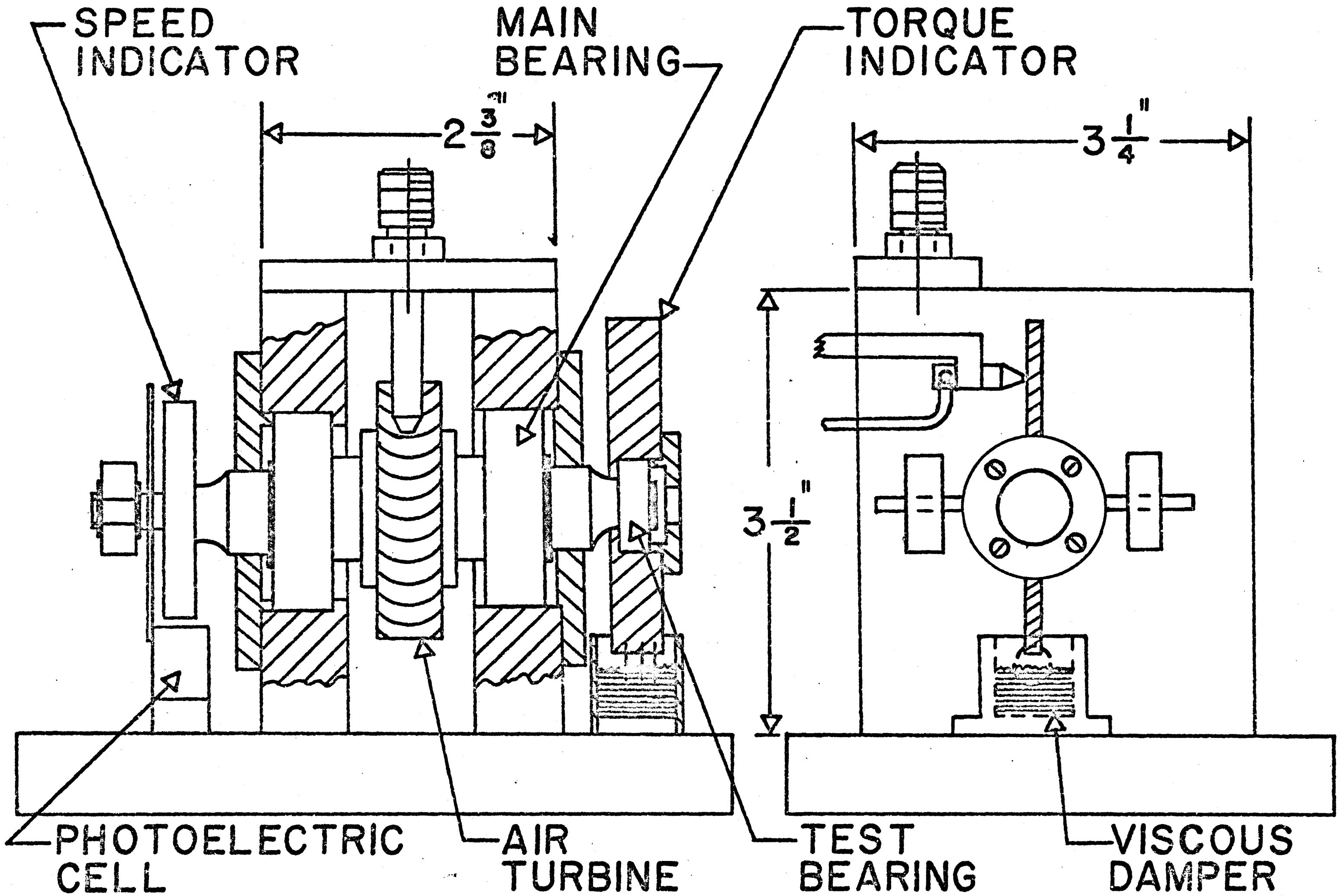

FIGURE 5. HIGH-SPEED TORQUE TESTER

nique depended upon a disk located on the left end of the turbine shaft as shown in Figure 5. The circumference of this disk was arranged with alternating light and dark regions. The pulses from the photoelectric cell, which was positioned so as to detect reflected light beams from the disk, were fed into a digital counter with the timing gate set at one second. Since the tachometer disk was arranged with six alternating light and dark regions, six pulses from the photoelectric cell corresponded to one revolution of the turbine. The speed in rpm could, therefore, be obtained by multiplying the counter reading by ten.

Turbine speed was controlled by placing a pressure regulator between the laboratory air supply and the turbine. By regulating the air pressure to the turbine, it was found that the desired speeds could be obtained easily and quickly.

Radial and Axial Loading Techniques

The test bearing was loaded radially by the addition of lead weights to two stainless steel threaded rods located in the bearing housing 180° apart. These rods were oriented to be perpendicular to the bearing centerline and also to cause the radial load to act through the centerline of the balls within the bearing. The weights were secured to the rods by means of lock nuts. The total radial load acting on the bearing was, therefore, the sum of the weights of the housing, the rods, and the lead weights. The orientation of the radial load on the bearing is shown in Figure 4.

Axial loads were applied to the test bearing by means of a string and roller system using a weight as the loading force. A small bar, with an eye hook made of copper wire soldered to its center, was attached to the bearing housing cover plate. The bar was made symetrical so that this eye hook was also along the bearing centerline. A weight, corresponding to the desired axial load, was attached to one end of a thin dacron string with the other end being attached to the eye hook. The string and weight were then looped over the roller to apply the axial load. The roller, which was nothing more than a brass bushing, was cut with a V-groove for positioning the string. The roller frame was adjustable in three axes for positioning the roller in order that a true axial load could be applied to the bearing. Although the rotation of the bearing housing was small, due to the close distance between the nozzle and aluminum flapper, a small rotation of the string could cause errors in the torque reading. This problem was alleviated by calibrating the system with the axial load acting on the bearing.

Figure 6 shows the radial and axial loads acting on the test bearing along with the basic torque sensing elements.

Measurement of Pressure

The constant pressure, P_s, supplied to the flapper valve was determined by a manometer. A pressure regulator, located between the nitrogen bottle and the torque sensor, was equipped with a gauge plug which allowed connection to the manometer. This regulator was a pilot-operated instrument operating on a pneumatic-"null"-balance system which provided controlled output pressures which were practically independent of variations in flow.

A Statham pressure transducer was used to determine changes in the back pressure, P_b, between the orifice and nozzle. However, since these pressures could be very low due to the small torques acting on the flapper, a method for amplifying these back pressures was desirable.

This pressure amplification was carried out by means of a Moore Nullmatic Differential-Pressure Transmitter and Pilot Valve. The transmitter, which is a bellows-type instrument, measures differential pressures and transmits--pneumatically--an amplified value. The differential between the supply pressure,

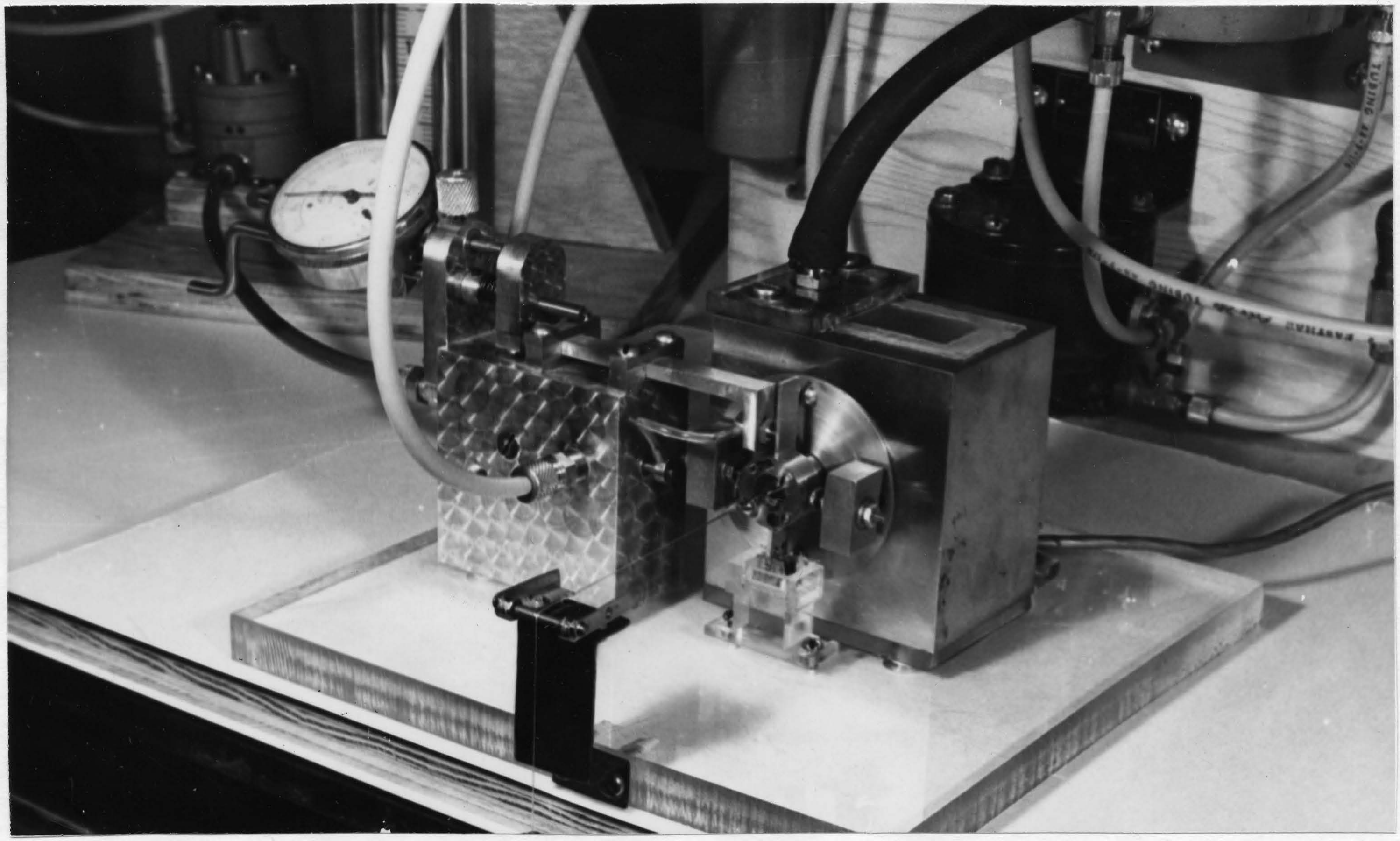

FIGURE 6. TORQUE SENSOR ASSEMBLY

P_s, and the back pressure, P_b, of the flapper valve was applied to the differential bellows in the transmitter. This difference in pressure results in a downward force to be applied to another bellows which is used to balance or null the differential bellows. By designing the effective area ratio of the balance bellows to the differential bellows to be less than one, a higher pressure is required to act in the balance bellows to null the differential bellows. This higher pressure required to balance the transmitter is proportional to the differential pressure and, therefore, serves as a pneumatic amplifier. The pilot valve supplies air at the necessary pressure to balance the transmitter by sensing the pressure drop across a nozzle contained within the balance bellows.

This transmitter-pilot valve arrangement was capable of sensing differential pressures ranging from 0 to 30 in. of water and amplifying to a range of 3 to 25 psi, respectively. The 3 psi output for zero differential pressure was obtained by preloading the differential bellows with an adjustable spring.

Since it was anticipated that the pneumatic amplifier would not be needed throughout the entire test program, a two-valve arrangement was used to bypass the amplifier. This arrangement is shown schematically in Figure 7.

The output strain from the pressure transducer was detected and amplified by means of a Bruel and Kjaer Type 1516 Strain Gauge Apparatus. The amplified strain voltage from this strain gauge apparatus then served as the input to a Moseley Autograph x-y plotter. By using sensitivity scales on both the strain gauge apparatus and the x-y plotter, a combination of the two was found to allow a relatively small pressure or torque range to be recorded over the entire 11-in. ordinate on the recorder. Due to this large ordinate, on which a small torque range could be plotted, very accurate values of torque were obtainable.

Calibration

The most important phase of each bearing test was the calibration of the pneumatic torque sensor. This calibration was carried out by hanging an accurately determined weight on the periphery of the bearing housing.

One end of a thin dacron string was secured to the bearing housing, near the aluminum flapper, with Epoxy. The string was placed such that its free end could wrap around the periphery of the bearing housing in the direction of bearing rotation. A small loop

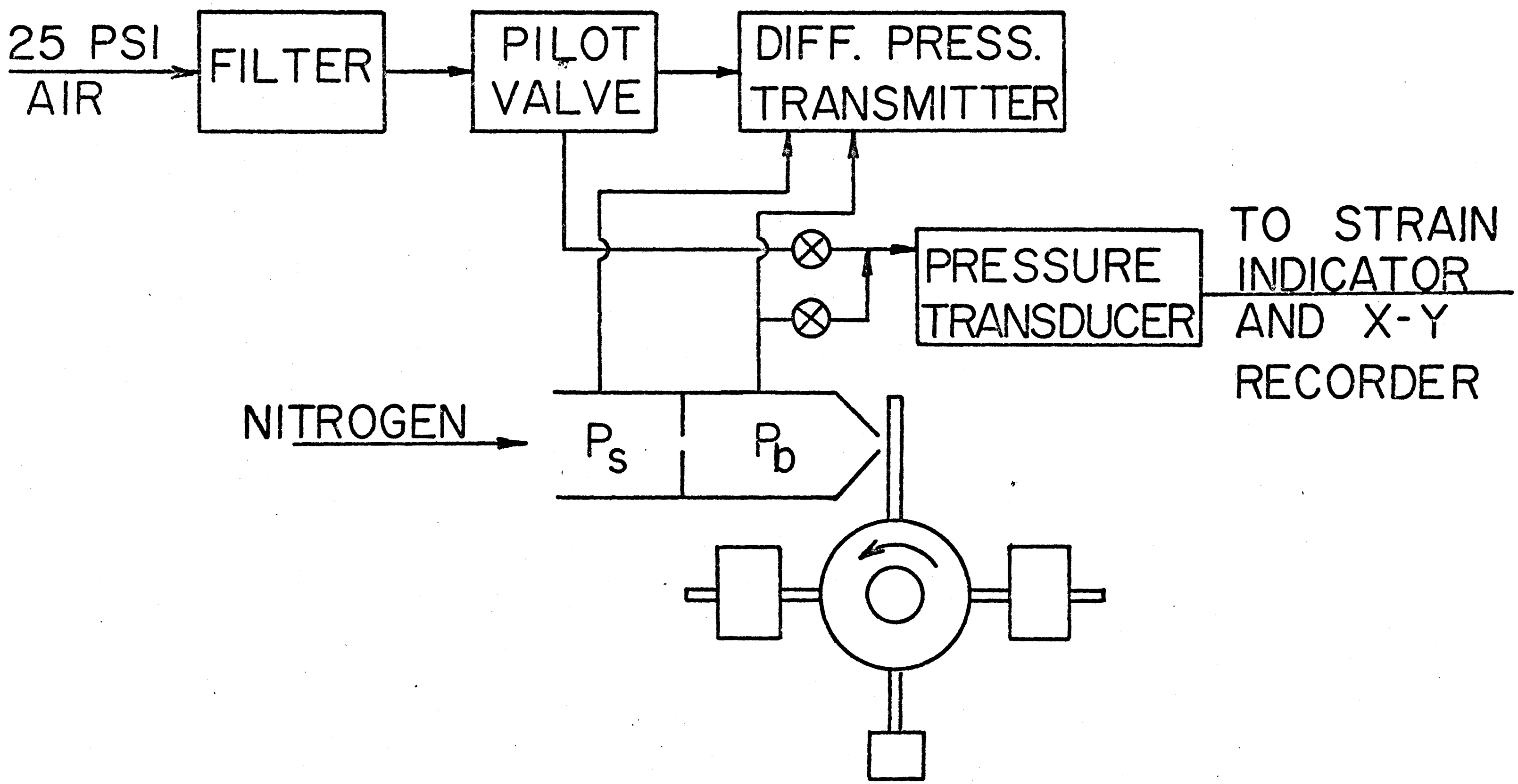

FIG. 7 PRESSURE MEASUREMENT

was then tied on the free end of the string to which a weight could be placed. Since the bearing and the housing were concentric, the torque arm could be determined from the housing diameter. The housing diameter was determined by a micrometer and upon dividing this value by two, the torque arm was found to be 0.4384 in.

The calibration weights were fabricated from lead and fitted with a small hook for attachment to the free end of the string. The exact weight was obtained by cutting away small bits of lead until the desired precalculated weights were obtained. Weights were made which would produce torques of 500, 1,000, 2,000, 5,000, 10,000, 20,000, 30,000, 50,000, and 100,000 mg-mm when hung from the bearing housing. The geometry of the calibration method is shown in Figure 8.

For the actual bearing calibration, the test bearing was mounted in the bearing housing and the assembly placed on the air turbine shaft. A supply pressure, P_s, was selected and weights were then hung from the bearing housing. The general procedure was to begin with a weight which corresponded to the maximum anticipated torque of the test bearing and work down. In this manner, it could be

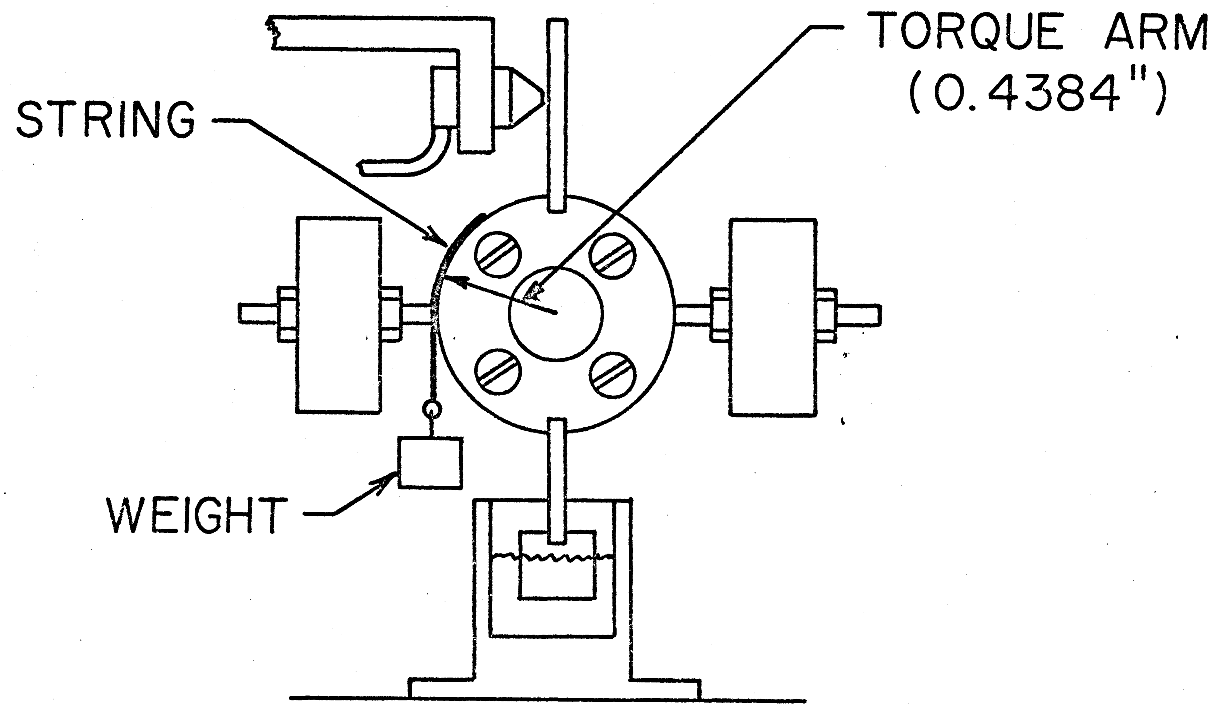

FIG. 8 CALIBRATION GEOMETRY

determined if the supply pressure was sufficient to counteract the bearing torque and also to set the x-y recorder so that full scale displacement could be used for the anticipated torque range. To be on the safe side, the calibration was conducted using two supply pressures. This was done in case the torque exceeded the anticipated maximum value. For the lower supply pressure, the recorder was set as mentioned above. For the higher supply pressure, the calibration was continued above the maximum anticipated value by switching the recorder to a less sensitive scale. In no instance was the zero adjustment of the recorder changed after the calibration at the lower supply pressure.

It was found that when weights were hung from the bearing housing, frictional effects within the bearing created some problem in obtaining a repeatable calibration. Although the deviations were small, on the order of two percent of full recorder deflection, it was determined that more consistent results could be obtained by oscillating the turbine shaft by hand after the weights had been applied to the housing. Calibration readings were not taken while oscillating the shaft, but immediately after this oscillation was discontinued. Also, when calibrating at low torques, such as 500 and 1,000 mg-mm, it was found that biasing the bearing housing toward the nozzle resulted in consistent calibration. This bias was achieved by creating a slight imbalance toward the nozzle with the lead weights used in applying the radial load. To insure against zero drift in the recording equipment, the calibration curve was spot-checked after each bearing test.

The calibration curves were found to be linear over most of the torque ranges. When operating very close to and relatively far from the nozzle, some non-linearities were observed. These non-linearities were not severe and could be eliminated by changing the supply pressure and recalibrating.

IV. INVESTIGATION

In order to determine the feasibility of the pneumatic torque sensor, an investigation of the torque characteristics of R-3 instrument ball bearings, under combined radial and axial loads, was undertaken. The R-3 ball bearings were selected for testing for several reasons. First, the torque characteristics of these bearings for constant radial load and zero axial load; and constant radial load, varying axial load were available from the investigations of Mabie[7] and Clarke,[8] respectively. This data could be used as a basis for comparison in evaluating the performance of the pneumatic sensor. Secondly, no investigations had been performed in determining the effect of variations in combined radial and axial loads on the running torque characteristics of instrument ball bearings. Before this investigation could be undertaken, however, a method for cleaning and lubricating the bearings had to be established.

Cleaning of Bearings

The R-3 instrument ball bearings to be used in the test program were the same as used by Mabie[7] and Clarke[8] in their investigations. Accordingly, the cleaning process closely followed the methods employed by those two investigators.

Since the bearings to be tested had been previously run, the first step was to clean them thoroughly. After the bearings were disassembled by removing the dust shields, each bearing was individually cleaned. The bearing was first agitated in benzene to loosen and remove the oil or grease from the previous lubrication. The bearing was then soaked in acetone to remove the remaining benzene and lubricant. After each bearing in the test group had been cleaned in the above manner, they were further cleaned ultrasonically in trichlorotrifluoromethane. Each bearing was individually held on a rack while suspended in the cleaning fluid. After several minutes, the bearings were removed from the cleaning tank while the fluid was still being agitated. This was done to minimize the possibility of foreign particles settling in the bearings. Immediately after removal from the cleaning tank, each bearing was placed in its own numbered glass container, which had been ultrasonically cleaned, to await weighing and lubrication.

<u>Weighing and Lubrication</u>

The weight of the cleaned bearings was determined to the nearest one ten-thousandth of a gram on an analytical balance. This dry weight was then used as a basis for determining the amount of lubricant to be added to each bearing.

Two types of lubricants were used in the test program. Univis P-38 oil, conforming to MIL-L-6085A, was added to the bearings by means of a hypodermic needle and syringe. By adding the oil to the balls near the inner and outer raceways, no oil fell through the bearing onto the weight pan in the analytical balance. The oil was added in small amounts so as not to exceed the desired amount of lubricant. Beacon 325 grease, conforming to MIL-G-3278A, was also used as a lubricant in the test program. The grease was added to the outer raceway between the balls by a hypodermic needle and syringe. The grease was then evenly distributed in the raceway by means of a jeweler's screwdriver. As this grease was added in small amounts, the bearings were rotated by holding the outer race with tweezers and turning the inner race by means of a cone inserted into the bearing bore. This was done to evenly distribute the grease. The lubricants were added to the bearings in the enclosure of the analytical balance so that the weight of lubricant added could be monitored at all times.

The weight of oil in bearings ordered from a manufacturer are generally not specified. It was previously determined by Mabie[7] that the quantity of oil added at the factory for R-3 size bearings was approximately 0.0076 gm. This was the weight of oil used in bringing the cleaned bearings up to factory oil specifications.

Grease lubricated bearings generally have the weight or degree of grease pack specified. This degree of pack is defined as the ratio of the weight of grease actually added to the maximum weight that can be packed into the bearing. Values of 1/16 and 1/8 pack were used in this test program which gave corresponding weights of 0.0087 and 0.0176 gm, respectively.

After each bearing had been weighed and lubricated, the dust shields were replaced on the bearings. Each bearing was then placed in a small envelope, which identified the bearing, lubricant, etc., to await testing. Figure 9 illustrates the equipment used for cleaning, weighing and lubricating the bearings.

Testing

Three sets of R-3 instrument ball bearings, consisting of six bearings per set, were chosen for testing. One set was lubricated with oil, one with 1/16 grease pack, and the other with 1/8 grease pack.

FIGURE 9. CLEANING, WEIGHING, AND LUBRICATION EQUIPMENT

Each set of bearings was tested under twelve combinations of radial and axial loads to speeds of 40,000 rpm.

For a given combination of radial and axial loads, the bearings in a set were tested to 40,000 rpm as originally lubricated--herein known as the original run. After the sequence of tests was completed, the same six bearings were then tested again without cleaning or relubricating. This sequence of tests will hereafter be referred to as the rerun. After the original and rerun tests were completed on the set of bearings, the bearings were cleaned and relubricated for testing at another combination of radial and axial load. The twelve combinations of load used in the test program were obtained from the following:

Radial Load-gm	Axial Load-gm
50	0
100	50
200	100
	200

The bearing torque was determined at eleven different speeds on each test. These speeds were 1,000 and 4,000 to 40,000 rpm in increments of 4,000 rpm. The torque data was obtained by measuring the displacement on the x-y recorder paper from a reference line. The torque reading in inches was then converted to milligram-millimeters from the displacement verse torque calibration curve previously determined.

With an original and rerun test conducted at each of twelve combinations of radial and axial load, a total of 144 tests were conducted on each set of bearings. The equipment used in obtaining the torque characteristics of R-3 instrument ball bearings is shown in Figure 10.

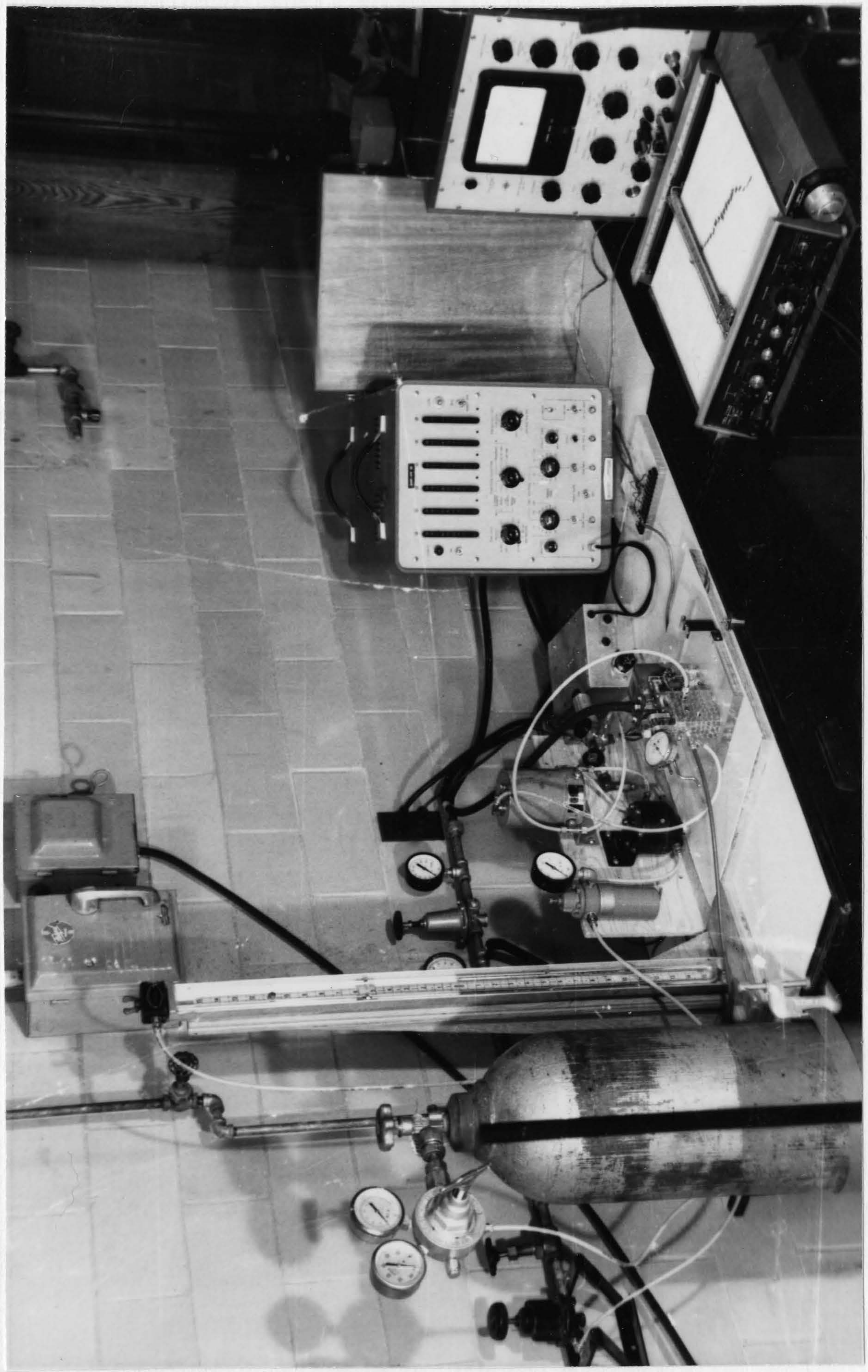

FIGURE 10. TEST APPARATUS

V. DATA

The torque characteristics of R-3 instrument ball bearings are presented in two forms. First, in Tables 1 through 36 the average torques and sample standard deviations for corresponding speeds are shown for various lubricant conditions and combinations of loads. Secondly, the data are shown graphically for rerun only in Figures 11 through 46. Vertical lines are also shown on these graphs which represent the plus and minus values of sample standard deviations at each speed.

The sample standard deviations, σ, were calculated by the following equation:

$$\sigma = \left[\frac{N(\Sigma X^2) - (\Sigma X)^2}{N^2}\right]^{1/2}$$

Where N = number of observations

X = individual torque values

TABLE 1

TORQUE CHARACTERISTICS FOR FACTORY OIL (0.0076 GM)

50 GM RADIAL LOAD

0 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	3150	627	1416	940
4000	6775	953	2233	1041
8000	7883	700	3200	1404
12000	7591	1068	3516	1041
16000	6875	1143	4966	980
20000	5858	704	5400	932
24000	5383	1330	5366	1367
28000	5841	1232	6050	892
32000	7658	636	6700	529
36000	8575	385	9733	286
40000	10066	293	12633	359

TABLE 2

TORQUE CHARACTERISTICS FOR FACTORY OIL (0.0076 GM)

50 GM RADIAL LOAD

50 GM AXIAL LOAD

SPEED (RPM)	ORIGINAL RUN AVERAGE TORQUE (MG-MM)	ORIGINAL RUN SAMPLE STANDARD DEVIATION (MG-MM)	RE-RUN AVERAGE TORQUE (MG-MM)	RE-RUN SAMPLE STANDARD DEVIATION (MG-MM)
1000	3791	652	2716	252
4000	11662	649	4325	647
8000	17770	1350	5450	1602
12000	14183	1371	5575	1077
16000	9058	1224	7250	1282
20000	9687	529	8991	1156
24000	10925	781	10241	806
28000	10700	2240	10958	1326
32000	13833	1381	12758	859
36000	16166	1795	15125	314
40000	17683	1228	17358	958

TABLE 3

TORQUE CHARACTERISTICS FOR FACTORY OIL (0.0076 GM)

50 GM RADIAL LOAD

100 GM AXIAL LOAD

SPEED (RPM)	ORIGINAL RUN AVERAGE TORQUE (MG-MM)	ORIGINAL RUN SAMPLE STANDARD DEVIATION (MG-MM)	RE-RUN AVERAGE TORQUE (MG-MM)	RE-RUN SAMPLE STANDARD DEVIATION (MG-MM)
1000	5366	687	2233	471
4000	13150	1500	5800	911
8000	16766	2536	7583	813
12000	11000	692	7608	1058
16000	10500	629	8375	1077
20000	10591	465	9008	1178
24000	12708	667	10833	1605
28000	14058	1032	11233	2365
32000	15483	1958	12366	2547
36000	18208	1954	14525	2550
40000	20791	1944	17816	2218

TABLE 4

TORQUE CHARACTERISTICS FOR FACTORY OIL (0.0076 GM)

50 GM RADIAL LOAD

200 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	6333	584	5066	359
4000	15166	604	9916	628
8000	19100	3428	12100	781
12000	16033	769	13983	267
16000	15000	1032	15933	935
20000	15566	1353	15933	928
24000	16766	1502	16533	1463
28000	17466	2083	17983	977
32000	18500	1668	18083	1535
36000	21800	2262	21400	1148
40000	23566	1448	23366	1174

TABLE 5

TORQUE CHARACTERISTICS FOR FACTORY OIL (0.0076 GM)

100 GM RADIAL LOAD

0 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	2733	280	900	387
4000	6816	343	2283	1301
8000	10333	692	3100	1131
12000	10650	1907	3333	851
16000	6150	1054	3666	628
20000	5883	841	4566	634
24000	7800	525	7366	555
28000	9550	1090	9366	1004
32000	11583	895	11150	848
36000	12816	474	12383	1026
40000	13733	743	13433	558

TABLE 6

TORQUE CHARACTERISTICS FOR FACTORY OIL (0.0076 GM)

100 GM RADIAL LOAD

50 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	4716	517	2700	957
4000	11433	1282	4950	512
8000	16883	1005	6916	338
12000	11983	1442	7516	429
16000	9416	1952	8700	1029
20000	9133	438	9566	926
24000	10733	1054	11933	1411
28000	13600	1394	14400	1290
32000	15766	1341	16200	1024
36000	16483	1435	16916	773
40000	17450	1229	17183	821

TABLE 7

TORQUE CHARACTERISTICS FOR FACTORY OIL (0.0076 GM)

100 GM RADIAL LOAD

100 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	5316	302	2666	414
4000	11000	1238	4766	767
8000	16400	2402	6533	729
12000	12133	1299	7450	793
16000	9566	760	8050	828
20000	10016	589	9083	633
24000	12566	1277	11833	1133
28000	15716	2391	15616	1337
32000	18283	2519	17800	781
36000	21700	2235	20733	1081
40000	26833	1478	25933	1205

TABLE 8

TORQUE CHARACTERISTICS FOR FACTORY OIL (0.0076 GM)

100 GM RADIAL LOAD

200 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	6200	472	2433	508
4000	13983	895	6016	548
8000	19425	2842	8225	463
12000	16816	1257	10808	808
16000	17100	1589	13416	587
20000	16558	1280	14050	869
24000	16954	1182	14316	982
28000	18300	1754	16408	626
32000	19766	1348	17433	590
36000	22408	1258	19283	498
40000	23016	893	20858	591

TABLE 9

TORQUE CHARACTERISTICS FOR FACTORY OIL (0.0076 GM)

200 GM RADIAL LOAD

0 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	3663	719	2225	695
4000	8866	349	3930	731
8000	12416	712	5025	622
12000	10400	1083	5905	613
16000	7875	516	7178	785
20000	8441	394	8561	845
24000	9483	233	9933	861
28000	10566	228	10708	864
32000	10791	458	10716	755
36000	10700	378	10983	483
40000	10533	659	10758	589

TABLE 10

TORQUE CHARACTERISTICS FOR FACTORY OIL (0.0076 GM)

200 GM RADIAL LOAD

50 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	4300	391	3016	285
4000	8566	531	4866	521
8000	12566	377	6016	477
12000	13450	1157	7166	696
16000	9550	381	8150	694
20000	10166	679	9916	1086
24000	11700	571	11700	1112
28000	12283	380	12450	937
32000	12750	593	12633	760
36000	13600	685	13250	818
40000	14000	723	14183	539

TABLE 11

TORQUE CHARACTERISTICS FOR FACTORY OIL (0.0076 GM)

200 GM RADIAL LOAD

100 GM AXIAL LOAD

SPEED (RPM)	ORIGINAL RUN AVERAGE TORQUE (MG-MM)	ORIGINAL RUN SAMPLE STANDARD DEVIATION (MG-MM)	RE-RUN AVERAGE TORQUE (MG-MM)	RE-RUN SAMPLE STANDARD DEVIATION (MG-MM)
1000	4883	323	3366	749
4000	11833	874	6333	1263
8000	14450	859	8183	1612
12000	13600	1665	10183	1330
16000	11450	1341	11383	1288
20000	12316	367	12433	1930
24000	13916	511	14366	1293
28000	14883	669	15433	825
32000	16883	942	16200	704
36000	18100	1093	17066	1030
40000	19183	1169	18300	991

TABLE 12

TORQUE CHARACTERISTICS FOR FACTORY OIL (0.0076 GM)

200 GM RADIAL LOAD

200 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	4933	997	3616	601
4000	13666	829	9000	1562
8000	16083	2002	12116	1693
12000	16666	1267	14316	1508
16000	14833	495	15150	1390
20000	15716	1010	15916	1688
24000	17100	1445	17816	1849
28000	18200	1653	18750	1945
32000	19883	2172	19650	2269
36000	21233	2752	21183	2041
40000	22000	2773	21966	2189

$IBSYS

TIME	1450	TOTAL TIME	19683		
PROG	360	DATA STORAGE	41	AVAILABLE CORE	12385

TABLE 13

TORQUE CHARACTERISTICS FOR 1/16 GREASE PACK (0.0087 GM)

50 GM RADIAL LOAD

0 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	6900	774	6116	296
4000	10850	1498	11200	824
8000	12950	711	13600	1069
12000	13566	1499	13200	1093
16000	13733	2124	10700	516
20000	13400	816	10466	1422
24000	11366	1104	8450	1669
28000	15800	2215	11366	2021
32000	14033	835	10433	865
36000	16116	1208	13766	1277
40000	18016	1077	15600	972

TABLE 14

TORQUE CHARACTERISTICS FOR 1/16 GREASE PACK (0.0087 GM)

50 GM RADIAL LOAD

50 GM AXIAL LOAD

SPEED (RPM)	ORIGINAL RUN AVERAGE TORQUE (MG-MM)	ORIGINAL RUN SAMPLE STANDARD DEVIATION (MG-MM)	RE-RUN AVERAGE TORQUE (MG-MM)	RE-RUN SAMPLE STANDARD DEVIATION (MG-MM)
1000	8750	957	7916	964
4000	15000	1925	15625	1390
8000	18541	1318	18333	1133
12000	21108	1431	18366	1816
16000	20950	1286	15791	2522
20000	21875	1982	17125	1908
24000	23491	2561	19750	1802
28000	26291	2952	22608	3193
32000	26708	3254	24083	2443
36000	28791	3664	25750	2358
40000	30416	3801	27083	1532

TABLE 15

TORQUE CHARACTERISTICS FOR 1/16 GREASE PACK (0.0087 GM)

50 GM RADIAL LOAD

100 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	9641	1479	8958	1734
4000	16075	1479	19466	2754
8000	25233	2725	23650	3216
12000	27991	1248	23908	1741
16000	28625	2805	21500	1493
20000	28458	2895	20833	731
24000	30950	2551	23850	1372
28000	33708	3882	25533	2125
32000	32833	2640	26000	629
36000	32708	3498	28683	1148
40000	35125	3633	31666	2243

TABLE 16

TORQUE CHARACTERISTICS FOR 1/16 GREASE PACK (0.0087 GM)

50 GM RADIAL LOAD

200 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	11300	1263	14333	1017
4000	20800	2103	27883	1547
8000	32500	3032	29833	2640
12000	36300	4464	29583	3167
16000	37816	2503	32416	3951
20000	41866	2281	34783	3356
24000	42416	1902	36283	1378
28000	43866	3780	40616	1717
32000	47583	3481	40916	5013
36000	49333	3582	45033	3194
40000	51916	2225	49916	4086

TABLE 17

TORQUE CHARACTERISTICS FOR 1/16 GREASE PACK (0.0087 GM)

100 GM RADIAL LOAD

0 GM AXIAL LOAD

SPEED (RPM)	ORIGINAL RUN AVERAGE TORQUE (MG-MM)	ORIGINAL RUN SAMPLE STANDARD DEVIATION (MG-MM)	RE-RUN AVERAGE TORQUE (MG-MM)	RE-RUN SAMPLE STANDARD DEVIATION (MG-MM)
1000	7750	1155	6633	1808
4000	13083	527	11866	1944
8000	16416	1436	14033	2163
12000	18166	1806	15583	2032
16000	18116	1168	13883	2019
20000	19716	2932	12483	1476
24000	22183	2743	15433	1385
28000	25583	3993	18450	1669
32000	28416	3334	20833	1322
36000	30266	2949	23233	960
40000	28900	2033	25650	1471

TABLE 18

TORQUE CHARACTERISTICS FOR 1/16 GREASE PACK (0.0087 GM)

100 GM RADIAL LOAD

50 GM AXIAL LOAD

SPEED (RPM)	ORIGINAL RUN AVERAGE TORQUE (MG-MM)	ORIGINAL RUN SAMPLE STANDARD DEVIATION (MG-MM)	RE-RUN AVERAGE TORQUE (MG-MM)	RE-RUN SAMPLE STANDARD DEVIATION (MG-MM)
1000	9816	915	7733	890
4000	15700	2390	16850	1260
8000	21266	891	21566	1938
12000	23983	924	21866	1894
16000	26733	1286	18316	3479
20000	26400	1813	18316	3507
24000	27850	2099	21316	2699
28000	30100	1605	24733	3123
32000	27116	11052	27033	1056
36000	32866	3640	28866	1049
40000	29283	1833	29650	1154

TABLE 19

TORQUE CHARACTERISTICS FOR 1/16 GREASE PACK (0.0087 GM)

100 GM RADIAL LOAD

100 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	9633	784	6916	975
4000	15966	1719	16816	1529
8000	22883	1547	22583	3432
12000	26966	2401	20200	2344
16000	27633	1388	16666	2211
20000	28050	2433	17850	1050
24000	31500	2608	23033	2157
28000	32966	2255	26350	1817
32000	36266	1604	30000	1772
36000	34700	2146	31400	1738
40000	37250	1749	35566	1657

TABLE 20

TORQUE CHARACTERISTICS FOR 1/16 GREASE PACK (0.0087 GM)

100 GM RADIAL LOAD

200 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	8250	908	9966	1146
4000	17200	1522	21025	1264
8000	26716	2001	25550	1021
12000	33133	3590	25866	2347
16000	35658	3537	28866	1650
20000	38333	3073	27266	2445
24000	41308	2589	28150	2192
28000	44833	2418	31316	3695
32000	45333	3916	32900	3124
36000	44866	2152	38200	2190
40000	45691	1135	40333	3182

TABLE 21

TORQUE CHARACTERISTICS FOR 1/16 GREASE PACK (0.0087 GM)

200 GM RADIAL LOAD

0 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	8500	611	4833	274
4000	14750	647	11616	318
8000	18383	2065	15700	1181
12000	19616	2681	17683	1562
16000	21900	1787	17066	1117
20000	23383	2008	16633	669
24000	25316	1105	17650	1499
28000	26950	1470	19633	1677
32000	28166	1782	20300	1514
36000	26250	1861	22016	944
40000	25550	2141	22383	1303

TABLE 22

TORQUE CHARACTERISTICS FOR 1/16 GREASE PACK (0.0087 GM)

200 GM RADIAL LOAD

50 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	8550	747	6266	861
4000	14350	518	14450	398
8000	19666	1760	17633	492
12000	19783	966	18900	374
16000	22833	3325	19383	389
20000	25816	2596	20566	385
24000	26616	2425	22533	799
28000	28300	885	24100	1170
32000	28900	1531	25666	1916
36000	29850	1648	25966	1562
40000	28116	664	25416	1925

TABLE 23

TORQUE CHARACTERISTICS FOR 1/16 GREASE PACK (0.0087 GM)

200 GM RADIAL LOAD

100 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	9700	532	9100	789
4000	15950	1400	17416	2069
8000	21733	1179	20700	1616
12000	26083	3232	21700	1494
16000	30433	4303	21166	1983
20000	34450	5117	21516	2256
24000	37366	3229	23950	1768
28000	38583	1815	27816	1994
32000	40266	3083	30216	1148
36000	39883	1606	32116	1626
40000	38850	2039	33166	1771

TABLE 24

TORQUE CHARACTERISTICS FOR 1/16 GREASE PACK (0.0087 GM)

200 GM RADIAL LOAD

200 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	10116	747	9633	471
4000	15950	793	19283	1245
8000	27516	1225	26650	1372
12000	30933	1885	29516	1858
16000	36100	725	29200	2539
20000	39266	2500	33100	1297
24000	40950	2410	34516	1264
28000	43550	2893	38583	1394
32000	42116	3346	38766	2256
36000	43183	3410	41116	1197
40000	43533	2787	41733	631

TABLE 25

TORQUE CHARACTERISTICS FOR 1/8 GREASE PACK (0.0174 GM)

50 GM RADIAL LOAD

0 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	6600	3189	5100	2365
4000	9950	4548	10100	4590
8000	12083	5489	13816	6254
12000	17600	8217	13816	6383
16000	21233	11690	10400	5351
20000	23966	12196	10200	5057
24000	34766	18245	7866	3944
28000	47183	22178	9016	4451
32000	47433	22399	9766	4436
36000	46533	22168	11550	5181
40000	40033	19904	14033	6285

TABLE 26

TORQUE CHARACTERISTICS FOR 1/8 GREASE PACK (0.0174 GM)

50 GM RADIAL LOAD

50 GM AXIAL LOAD

SPEED (RPM)	ORIGINAL RUN AVERAGE TORQUE (MG-MM)	ORIGINAL RUN SAMPLE STANDARD DEVIATION (MG-MM)	RE-RUN AVERAGE TORQUE (MG-MM)	RE-RUN SAMPLE STANDARD DEVIATION (MG-MM)
1000	8650	3973	6958	3123
4000	13316	6111	14916	6746
8000	18766	8525	18558	8449
12000	23000	10536	18333	8502
16000	25166	11549	17000	8036
20000	28166	12723	18500	8500
24000	35450	16141	20416	9239
28000	31916	14595	21666	9809
32000	35500	16196	23083	10453
36000	37416	17229	24125	10887
40000	38333	17534	24833	11156

TABLE 27

TORQUE CHARACTERISTICS FOR 1/8 GREASE PACK (0.0174 GM)

50 GM RADIAL LOAD

100 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	8791	4065	8883	4118
4000	15616	7034	18608	8601
8000	22633	10282	22966	10360
12000	26250	12300	22966	10335
16000	31350	15048	22583	10184
20000	30683	14391	20833	9366
24000	34875	16012	23000	10344
28000	37708	17058	25750	11536
32000	42458	19150	25191	11289
36000	40625	18288	26708	11967
40000	33433	16072	29333	13215

TABLE 28

TORQUE CHARACTERISTICS FOR 1/8 GREASE PACK (0.0174 GM)

50 GM RADIAL LOAD

200 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	9250	4210	9416	4285
4000	17083	7791	20416	9311
8000	26083	11702	26750	12058
12000	37000	17447	29316	13213
16000	48250	22492	29816	13608
20000	49866	23164	28116	12626
24000	49333	22414	31000	14071
28000	52283	23628	35833	16097
32000	56583	25336	39416	18326
36000	54283	24442	40916	18462
40000	57250	25634	47083	21221

TABLE 29

TORQUE CHARACTERISTICS FOR 1/8 GREASE PACK (0.0174 GM)

100 GM RADIAL LOAD

0 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	7633	3537	5433	2449
4000	11766	5314	10900	4892
8000	15150	6930	14016	6347
12000	21700	10267	14116	7138
16000	23866	11783	11500	6159
20000	24800	12471	10500	5465
24000	38950	18484	13466	6378
28000	59283	27118	15616	7163
32000	72433	32716	17633	7972
36000	80333	36376	19183	8703
40000	73783	36711	20750	9419

TABLE 30

TORQUE CHARACTERISTICS FOR 1/8 GREASE PACK (0.0174 GM)

100 GM RADIAL LOAD

50 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	6433	2974	6666	3391
4000	10650	5004	14700	7105
8000	16400	7587	19383	8877
12000	23666	10849	19200	8815
16000	26666	12474	14883	7019
20000	30283	13840	15350	6979
24000	44666	21194	18250	8163
28000	61600	28921	22133	9923
32000	75783	34118	25183	11321
36000	82500	37424	26916	12096
40000	62533	28355	31533	14197

TABLE 31

TORQUE CHARACTERISTICS FOR 1/8 GREASE PACK (0.0174 GM)

100 GM RADIAL LOAD

100 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	10650	4911	8333	4307
4000	14083	6438	18616	9076
8000	21633	9929	23083	10470
12000	32450	14594	22066	10231
16000	36000	16276	19550	9500
20000	41100	18990	20183	9381
24000	52466	24323	25333	11591
28000	63950	30203	29650	13961
32000	73250	34234	32083	15318
36000	81033	37100	35766	17094
40000	87583	39894	36283	16445

TABLE 32

TORQUE CHARACTERISTICS FOR 1/8 GREASE PACK (0.0174 GM)

100 GM RADIAL LOAD

200 GM AXIAL LOAD

SPEED (RPM)	ORIGINAL RUN AVERAGE TORQUE (MG-MM)	ORIGINAL RUN SAMPLE STANDARD DEVIATION (MG-MM)	RE-RUN AVERAGE TORQUE (MG-MM)	RE-RUN SAMPLE STANDARD DEVIATION (MG-MM)
1000	8525	4021	8833	3965
4000	15466	7446	19216	8970
8000	25183	11286	23916	11264
12000	32750	15061	22850	10266
16000	38383	17260	26516	11898
20000	45033	20403	26183	11882
24000	52133	24281	27041	12263
28000	58750	28528	30516	13728
32000	64408	32133	31766	14298
36000	71683	35337	35266	15893
40000	79533	38459	39350	17700

TABLE 33

TORQUE CHARACTERISTICS FOR 1/8 GREASE PACK (0.0174 GM)

200 GM RADIAL LOAD

0 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	6400	3271	4750	2268
4000	11266	5145	12033	5577
8000	17116	8042	18083	8263
12000	24166	11319	18166	8229
16000	33733	15221	15783	7670
20000	35083	16233	17366	8354
24000	47916	23626	21000	10230
28000	62750	32870	23666	11585
32000	78833	36361	24666	11685
36000	91283	41324	28333	13585
40000	89816	45134	30750	14270

TABLE 34

TORQUE CHARACTERISTICS FOR 1/8 GREASE PACK (0.0174 GM)

200 GM RADIAL LOAD

50 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	6616	3932	4783	2265
4000	11616	5779	10083	4624
8000	18200	10114	13600	6401
12000	23683	13259	16050	7251
16000	31750	15566	17583	8429
20000	31166	14587	20616	9978
24000	36416	16902	30100	17528
28000	45333	22793	36933	20504
32000	56416	30578	41133	20709
36000	71083	37939	43133	21158
40000	84250	44764	50000	22449

TABLE 35

TORQUE CHARACTERISTICS FOR 1/8 GREASE PACK (0.0174 GM)

200 GM RADIAL LOAD

100 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	6616	3022	8483	3884
4000	12316	5623	18166	8209
8000	21000	9568	24666	11171
12000	28450	12897	26850	12169
16000	42466	19314	28116	12593
20000	44366	20355	27250	12348
24000	51250	23826	30633	13924
28000	62016	30369	32333	14610
32000	70866	35205	37033	16834
36000	78533	38161	39933	18456
40000	78600	39160	43250	19773

TABLE 36

TORQUE CHARACTERISTICS FOR 1/8 GREASE PACK (0.0174 GM)

200 GM RADIAL LOAD

200 GM AXIAL LOAD

	ORIGINAL RUN		RE-RUN	
SPEED (RPM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)	AVERAGE TORQUE (MG-MM)	SAMPLE STANDARD DEVIATION (MG-MM)
1000	10833	5217	8166	3771
4000	17166	7695	18083	8126
8000	26816	12056	24166	10849
12000	32500	14679	28083	12657
16000	48933	22286	28083	12716
20000	46833	21234	31333	14117
24000	46333	21296	36416	16341
28000	51166	24628	42666	19110
32000	57666	28369	50166	22610
36000	66166	32371	53666	24613
40000	72000	37939	59483	27029

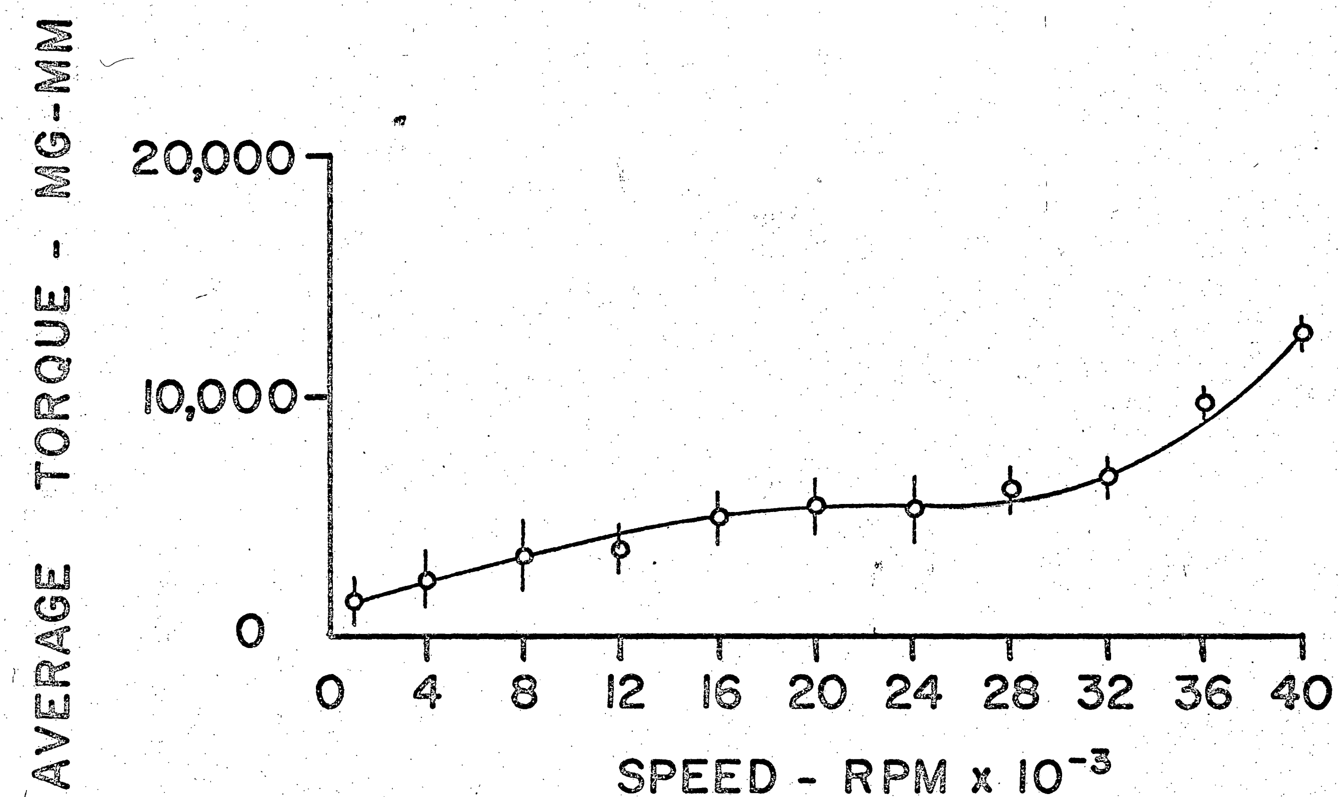

FIG. 11 OIL, 50 GM. RADIAL, 0 AXIAL

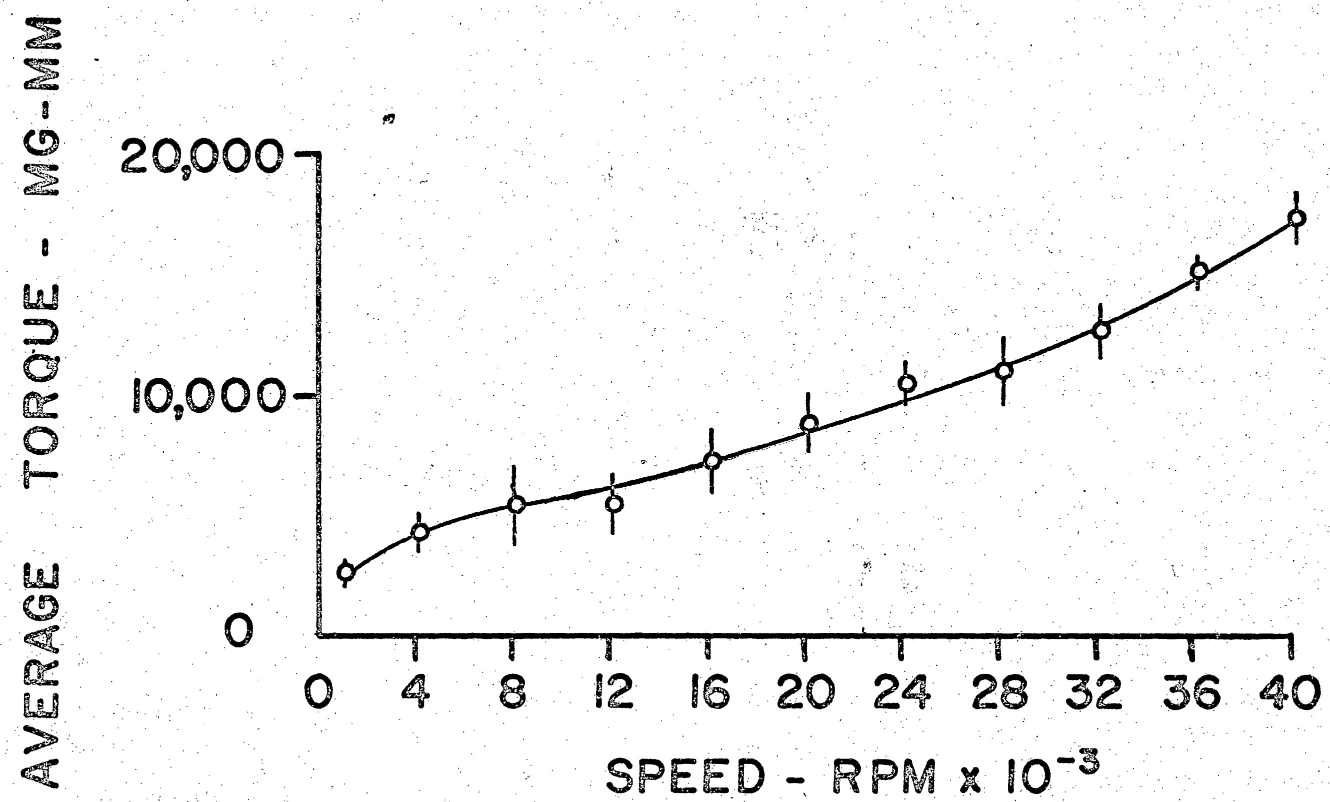

FIG. 12 OIL, 50 GM. RADIAL, 50 GM. AXIAL

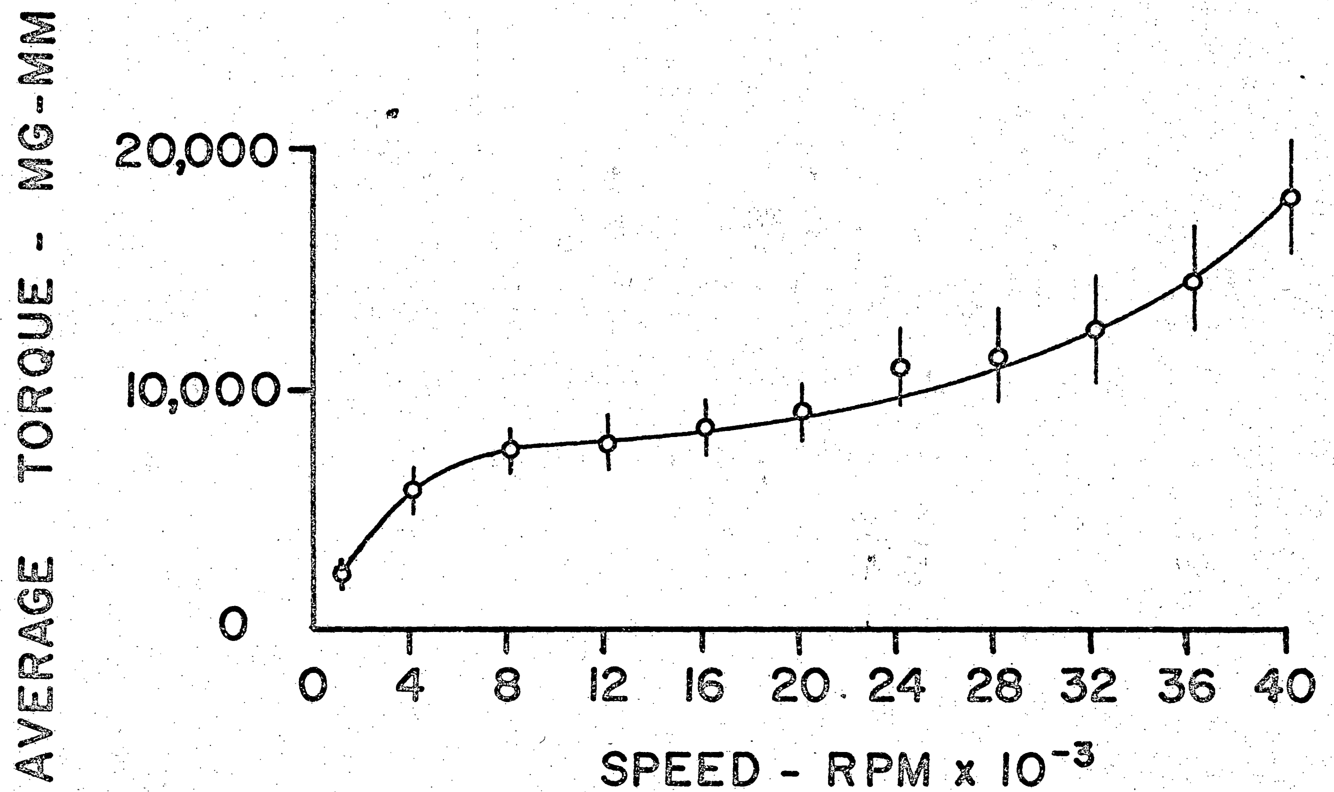

FIG. 13 OIL, 50 GM. RADIAL, 100 GM. AXIAL

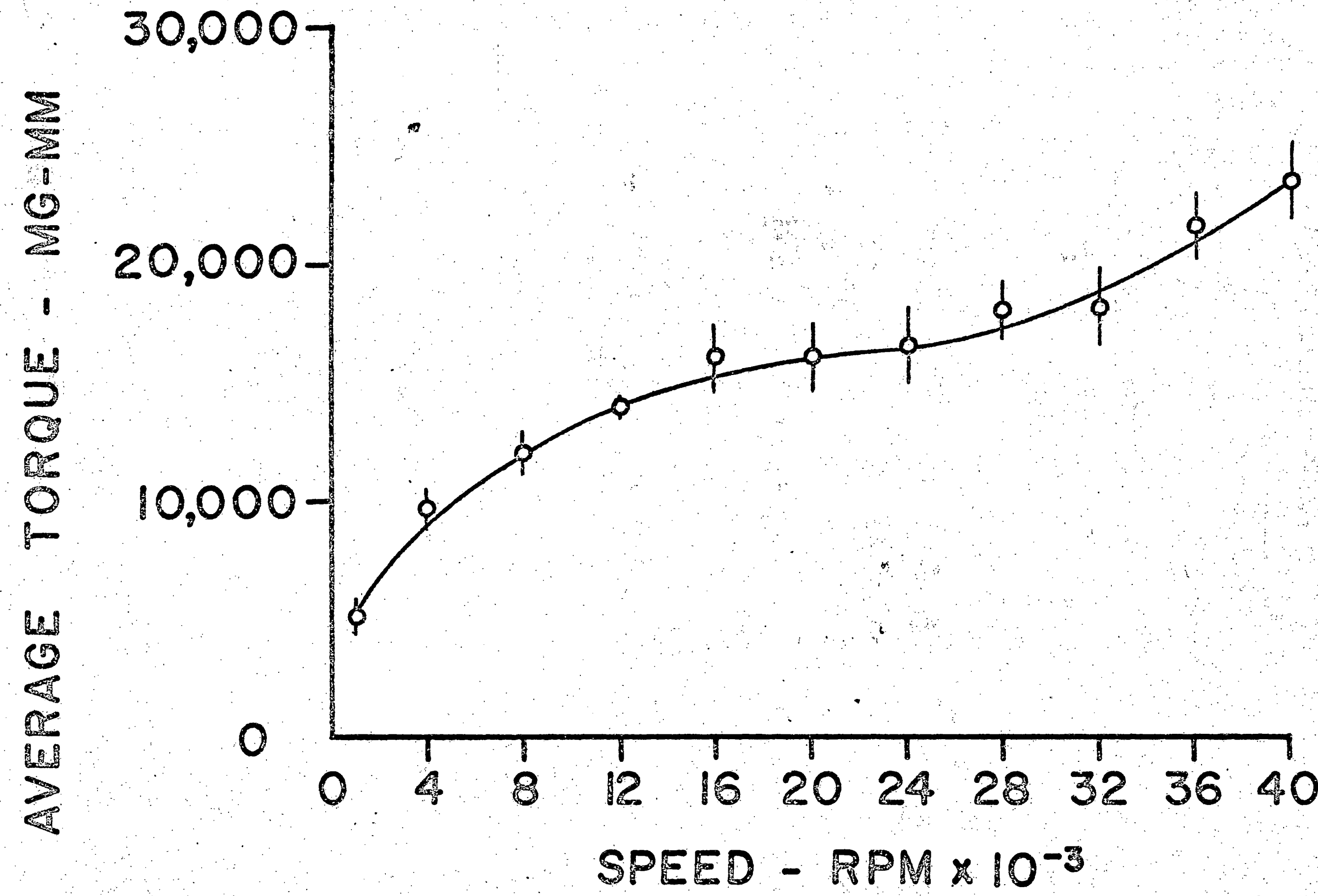

FIG. 14 OIL, 50 GM. RADIAL, 200 GM. AXIAL

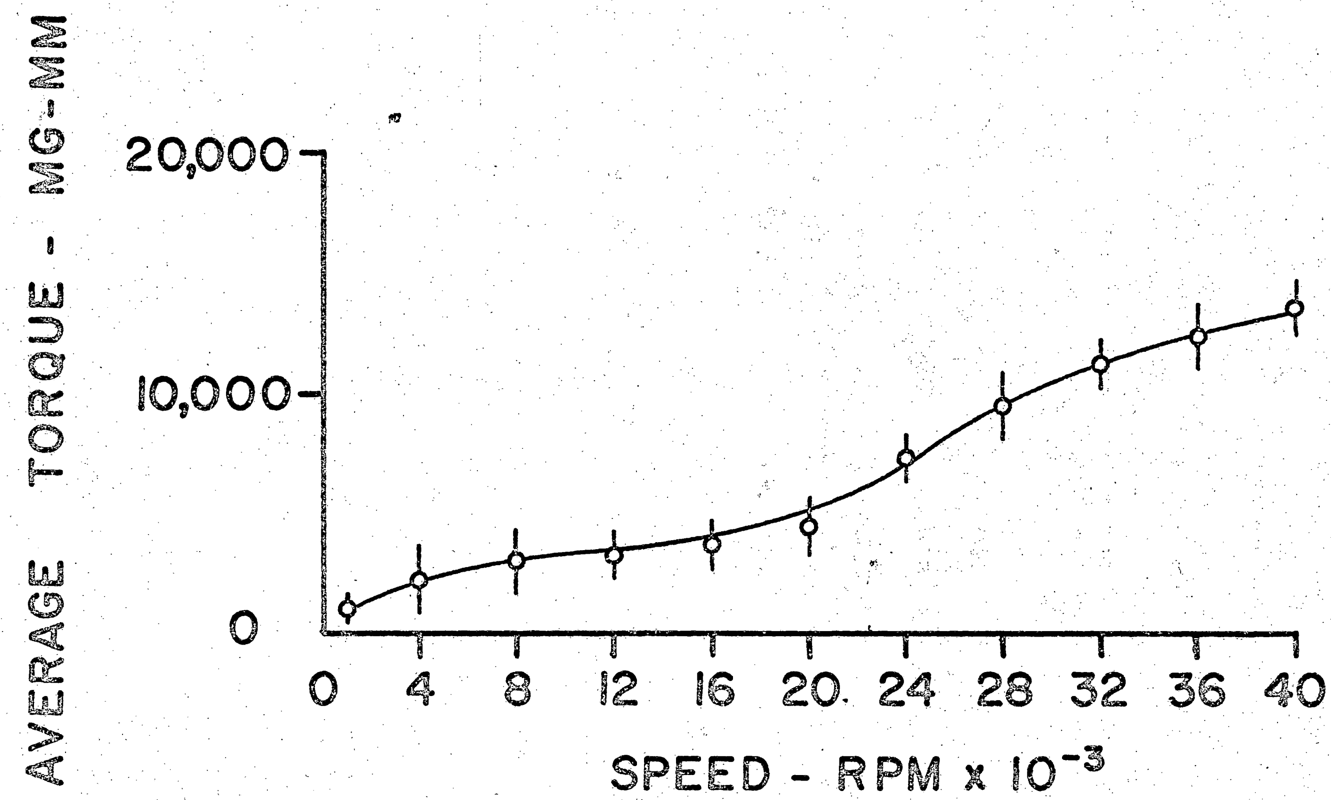

FIG. 15 OIL, 100 GM. RADIAL, 0 AXIAL

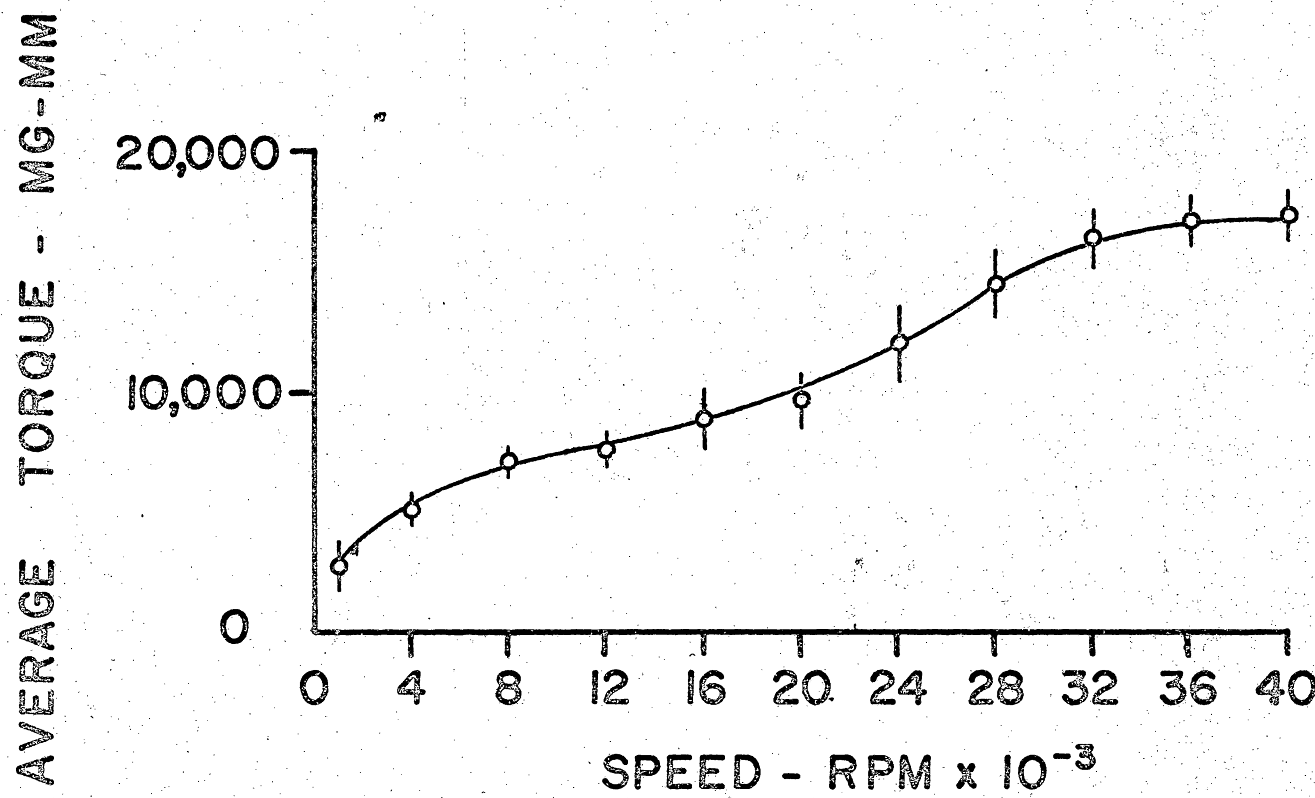

FIG. 16 OIL, 100 GM. RADIAL, 50 GM. AXIAL

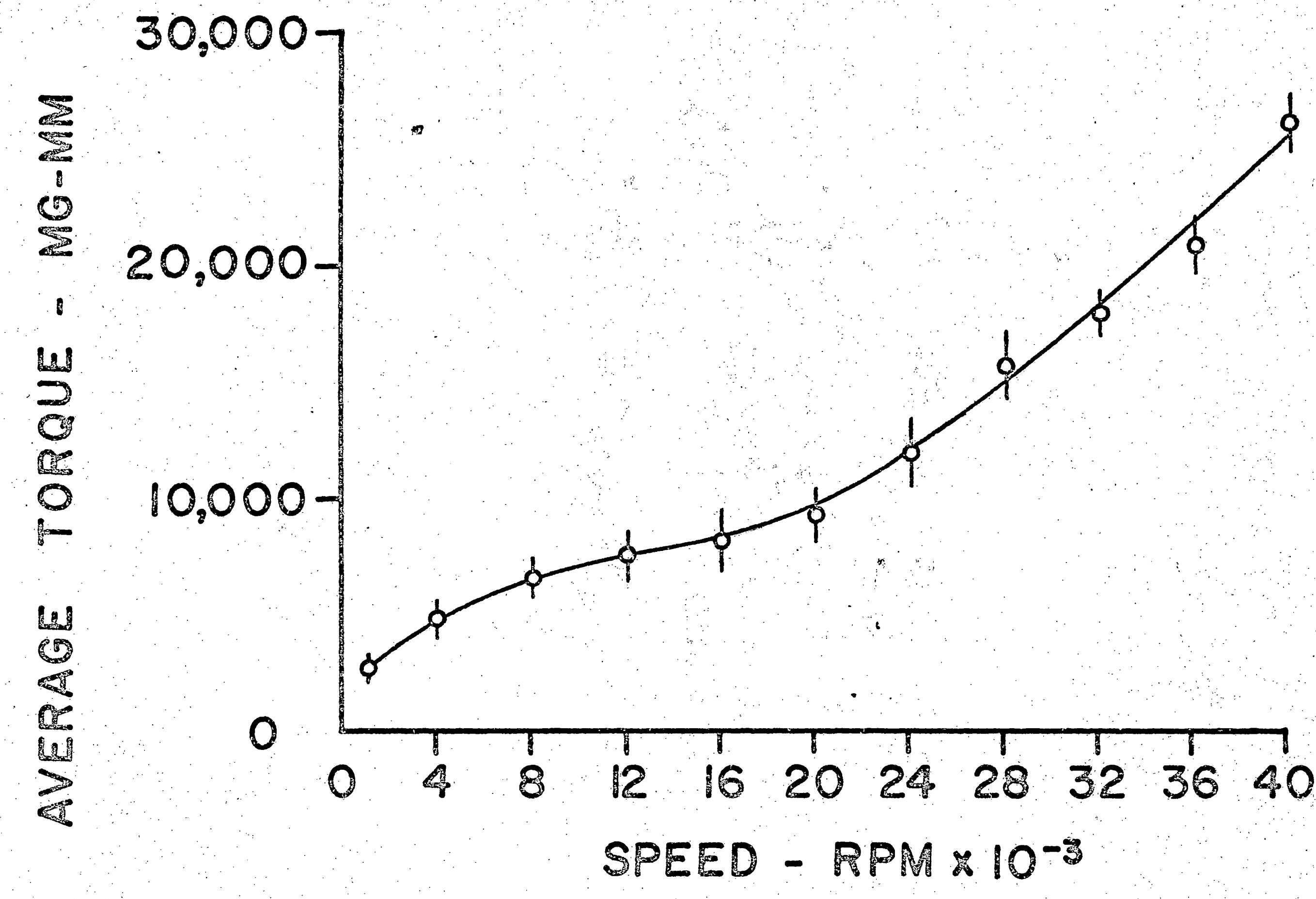

FIG. 17 OIL, 100 GM. RADIAL, 100 GM. AXIAL

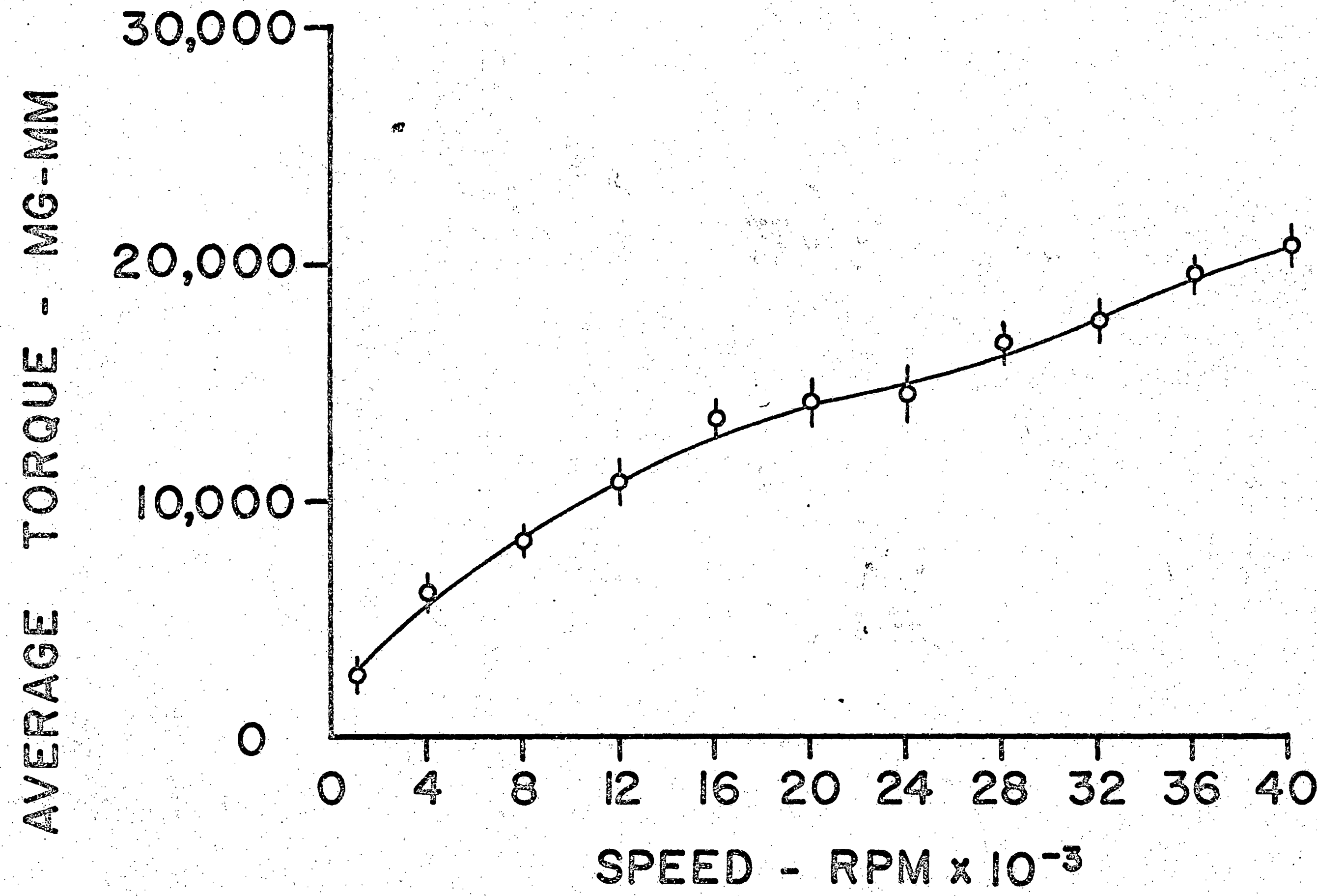

FIG. 18 OIL, 100 GM. RADIAL, 200 GM. AXIAL

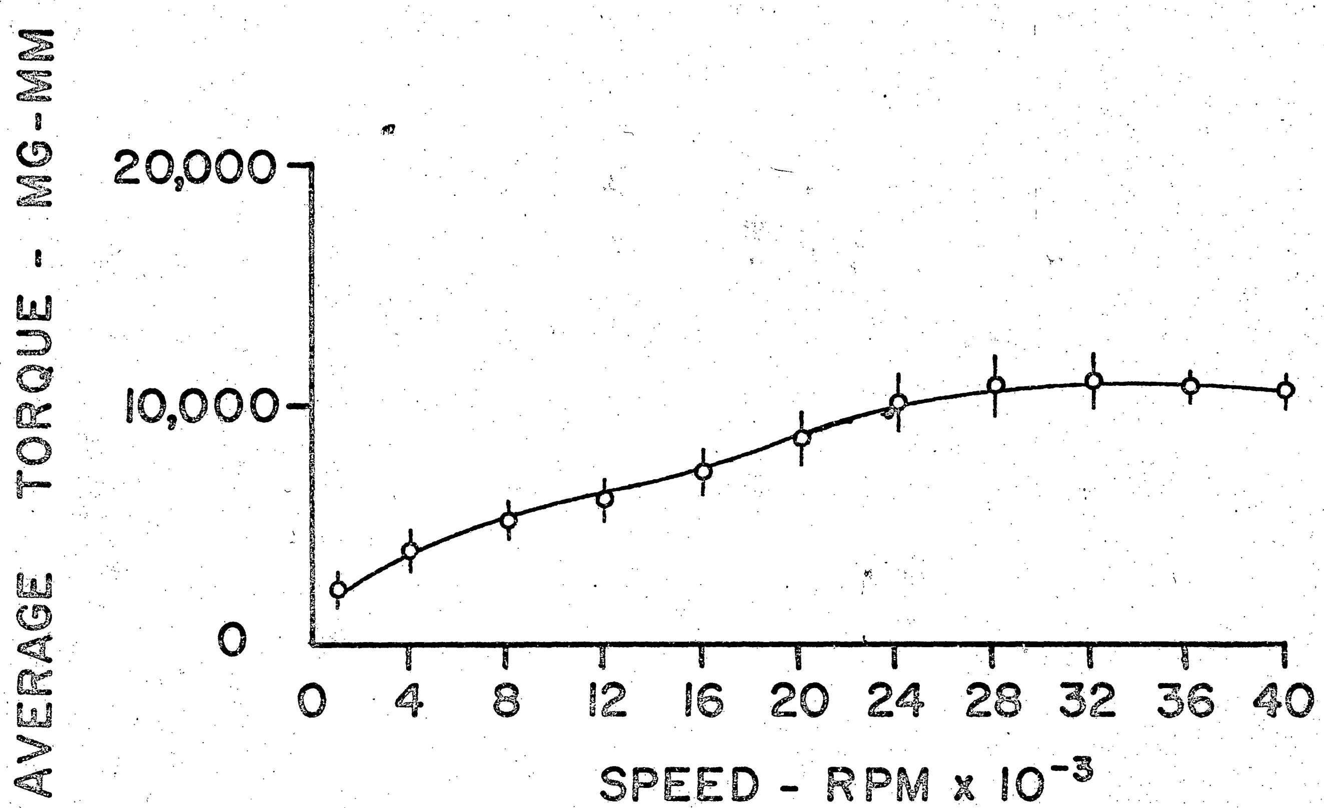

FIG. 19 OIL, 200 GM. RADIAL, 0 AXIAL

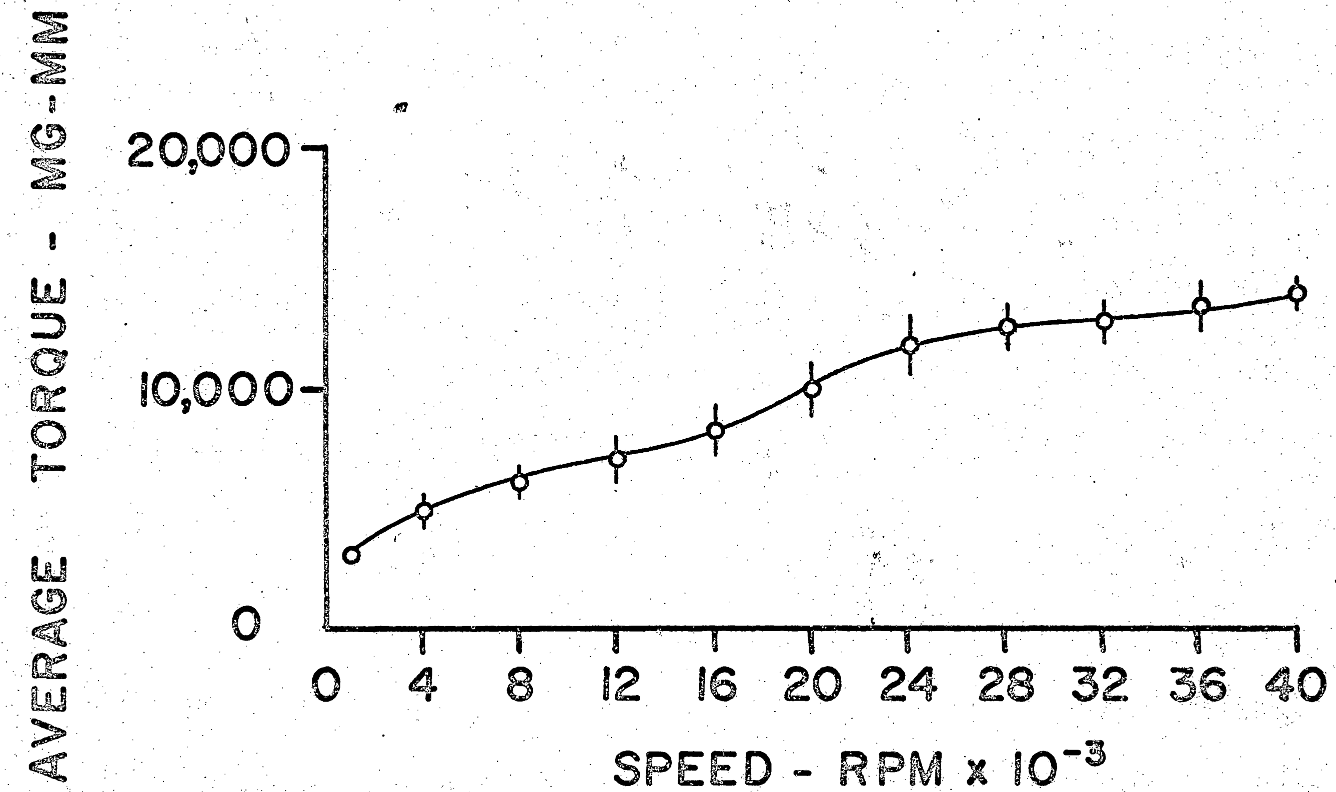

FIG. 20 OIL, 200 GM. RADIAL, 50 GM. AXIAL

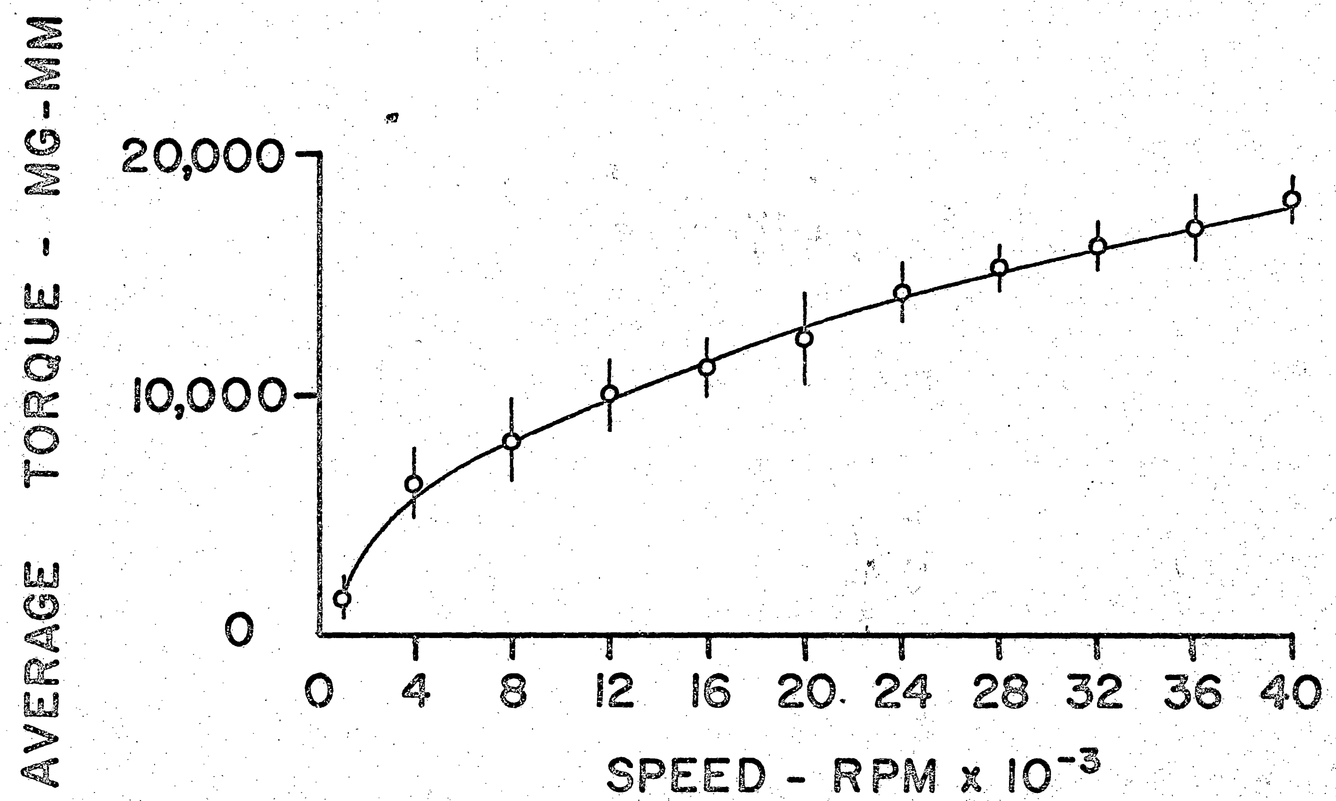

FIG. 21 OIL, 200 GM. RADIAL, 100 GM. AXIAL

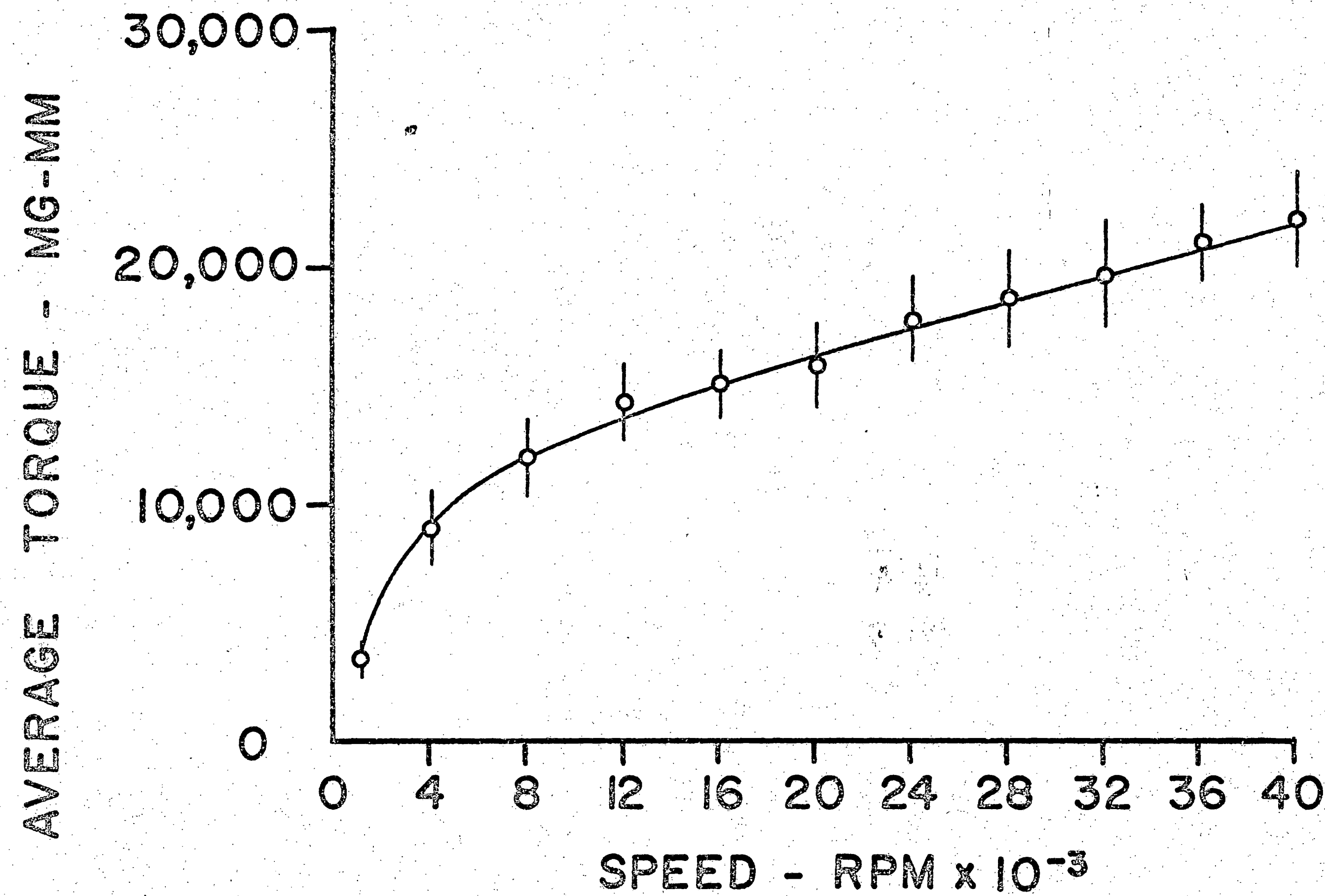

FIG. 22 OIL, 200 GM. RADIAL, 200 GM. AXIAL

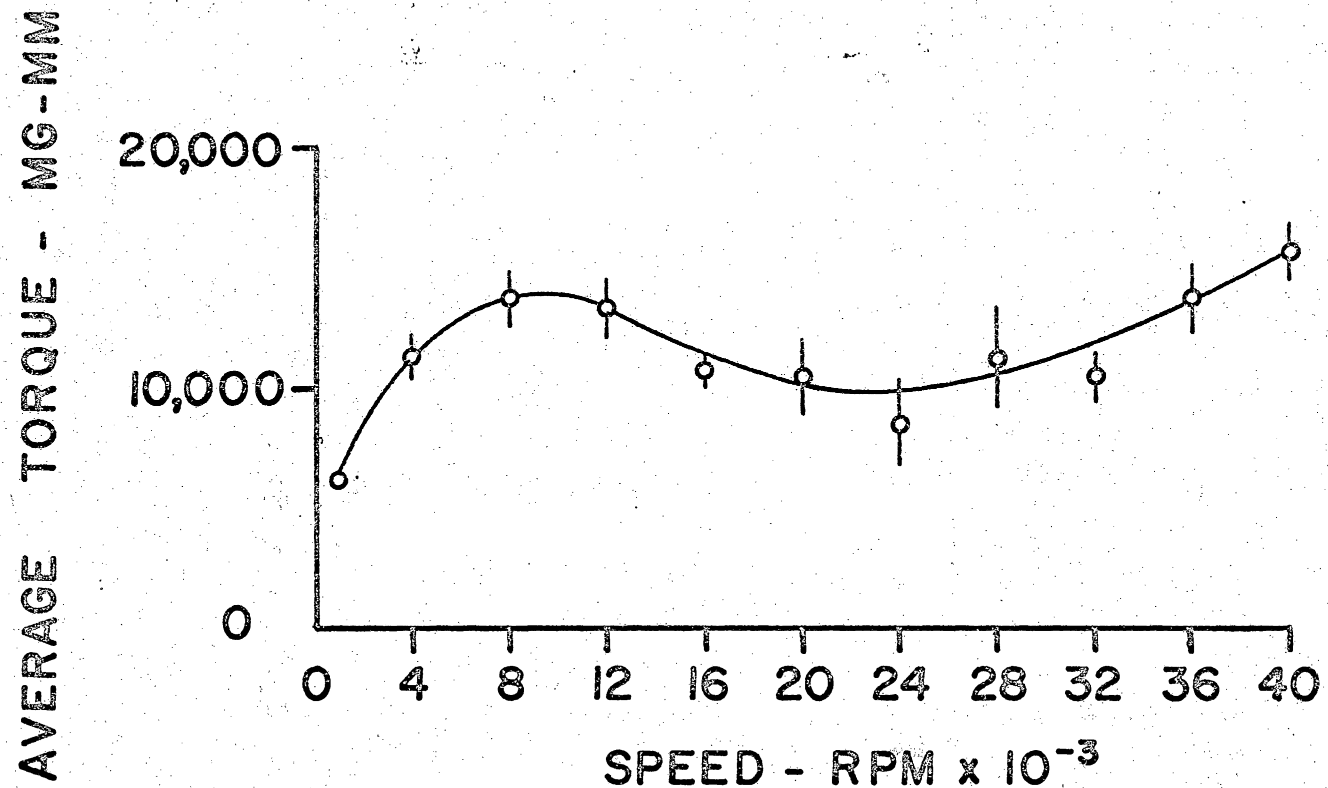

FIG. 23 1/16 GREASE PACK, 50 GM. RADIAL, 0 AXIAL

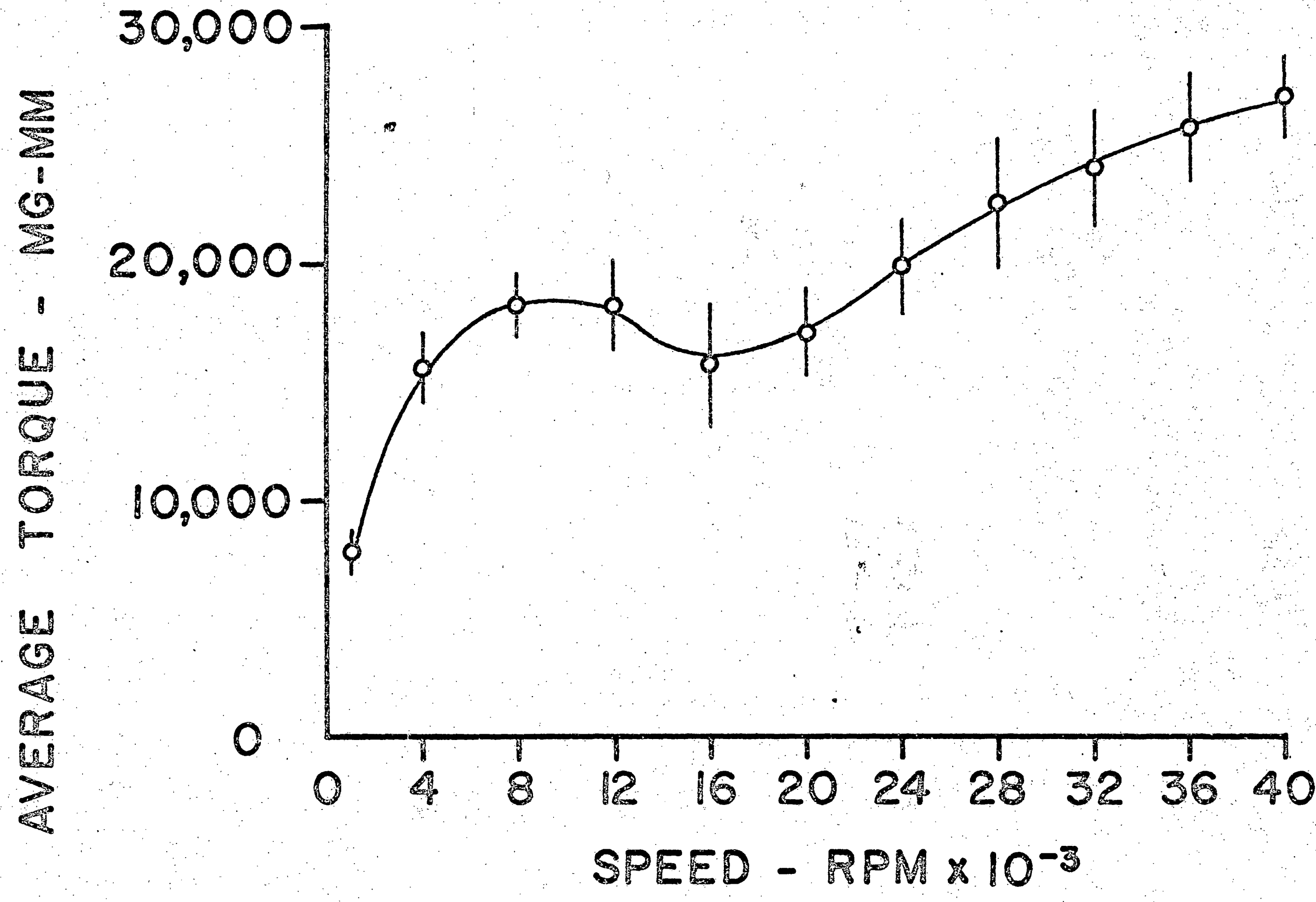

FIG. 24 1/16 GREASE PACK, 50 GM. RADIAL, 50 GM. AXIAL

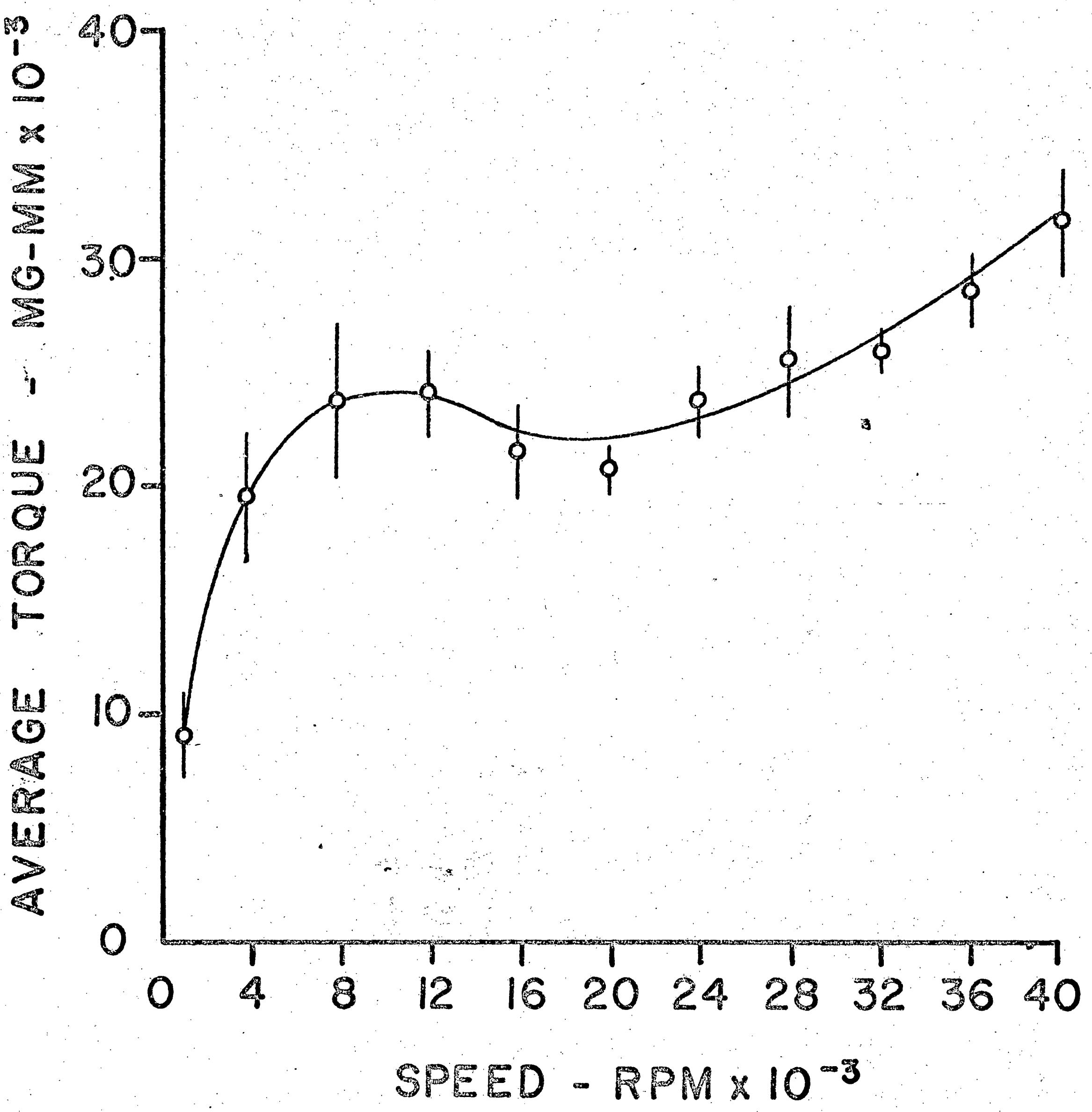

FIG. 25 1/16 GREASE PACK
50 GM. RADIAL, 100 GM. AXIAL

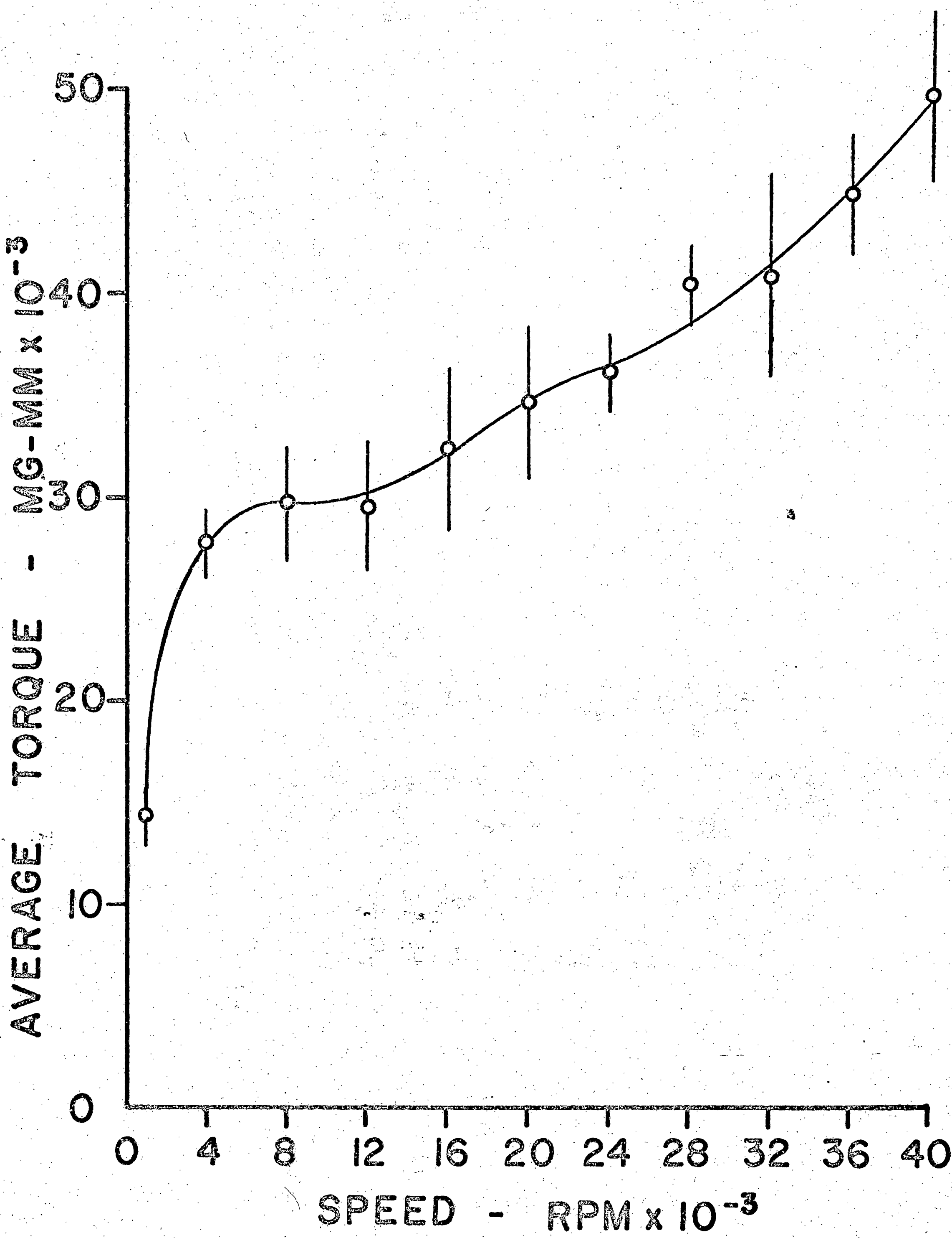

FIG. 26 1/16 GREASE PACK
50 GM. RADIAL, 200 GM. AXIAL

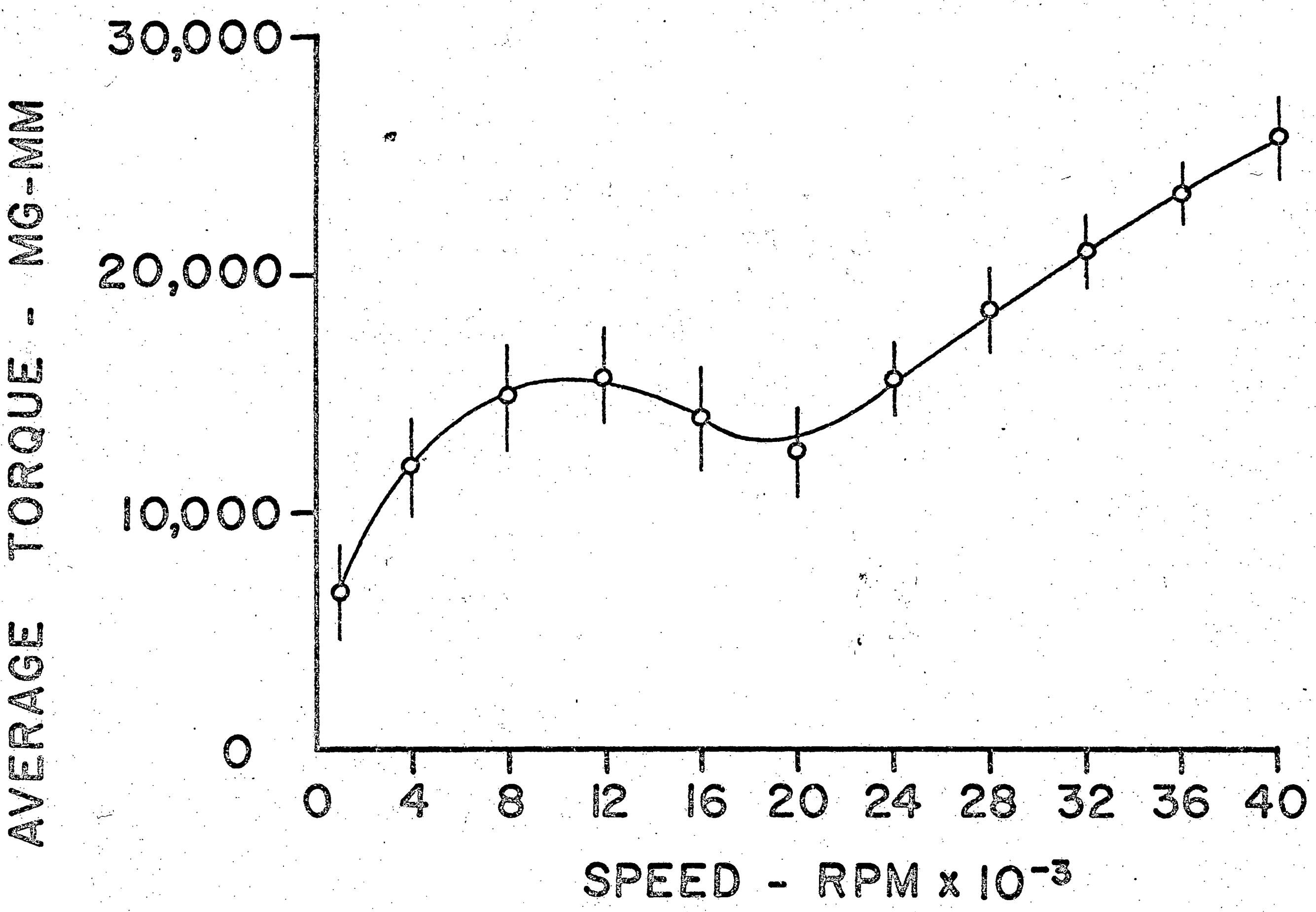

FIG. 27 1/16 GREASE PACK, 100 GM. RADIAL, 0 AXIAL

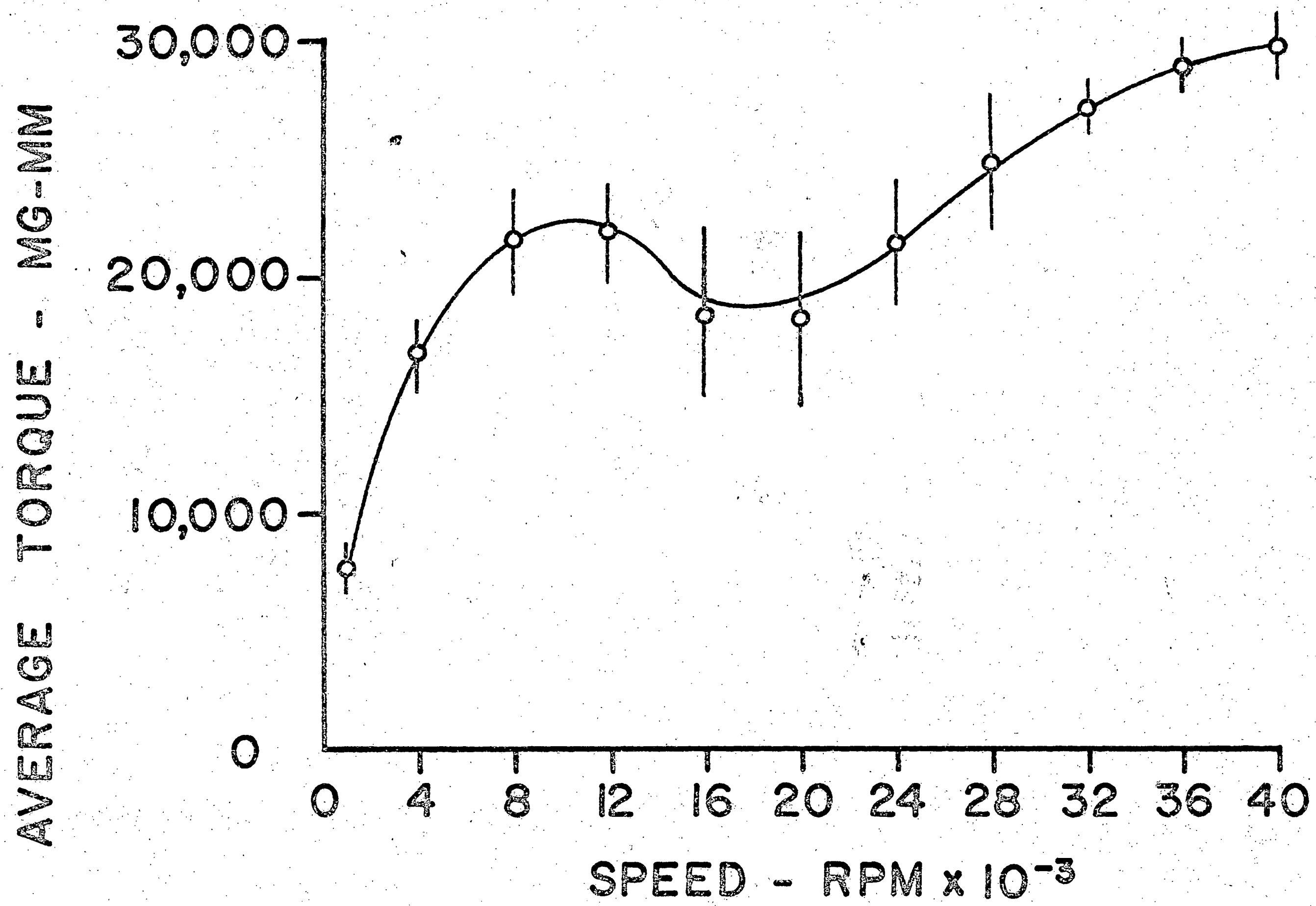

FIG. 28 1/16 GREASE PACK, 100 GM. RADIAL, 50 GM. AXIAL

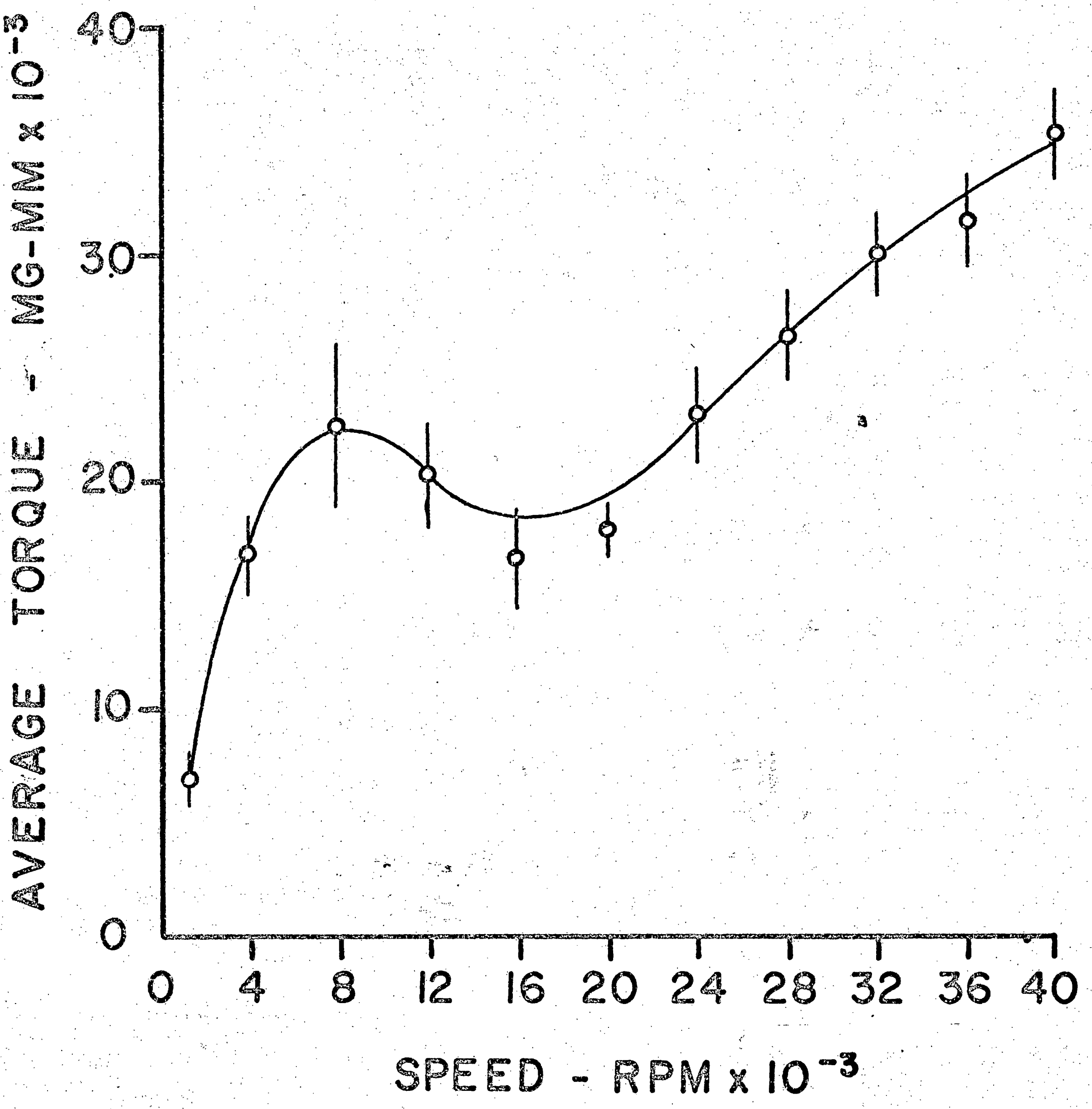

FIG. 29 1/16 GREASE PACK
100 GM. RADIAL, 100 GM. AXIAL

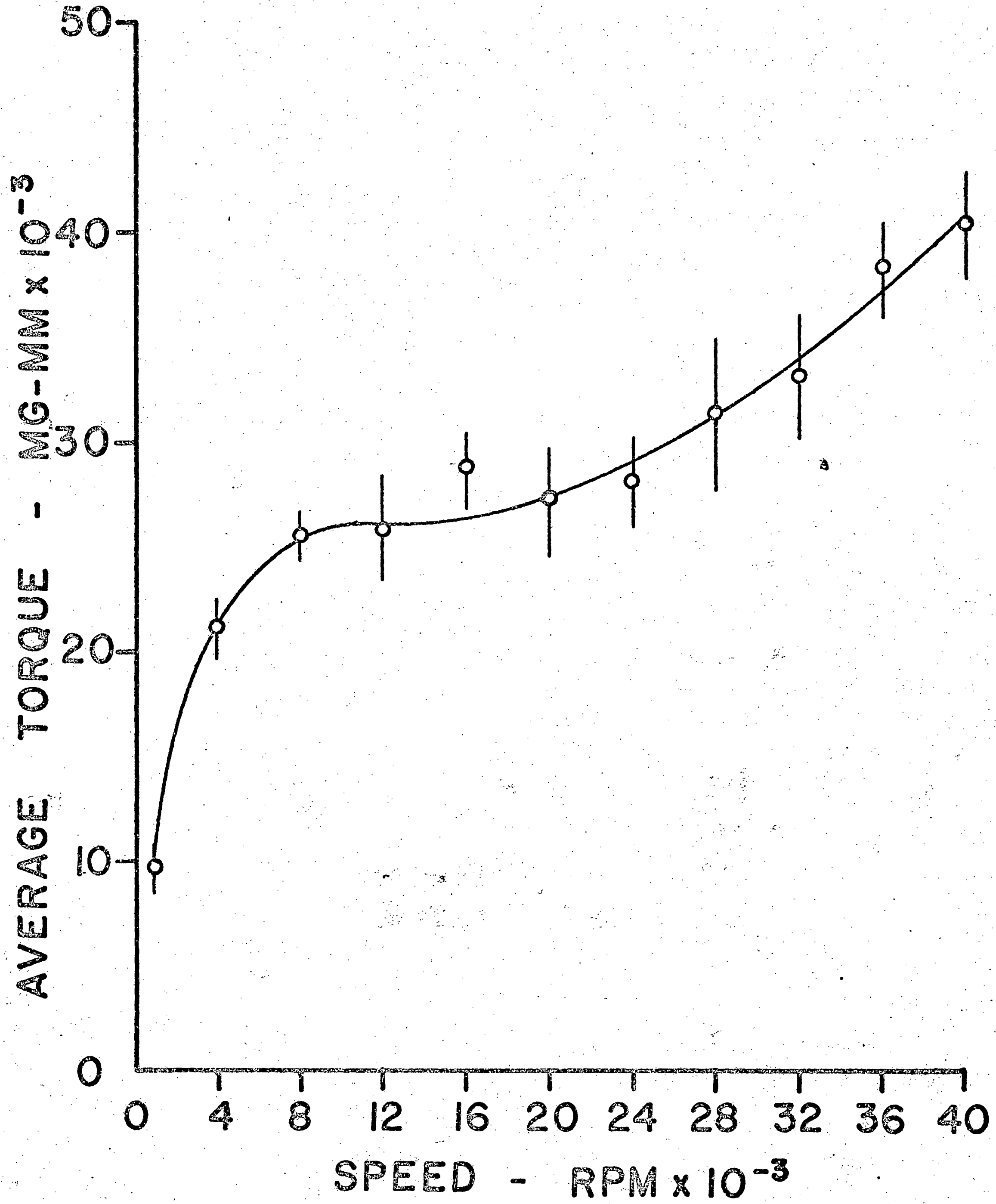

FIG. 30 1/16 GREASE PACK
100 GM. RADIAL, 200 GM. AXIAL

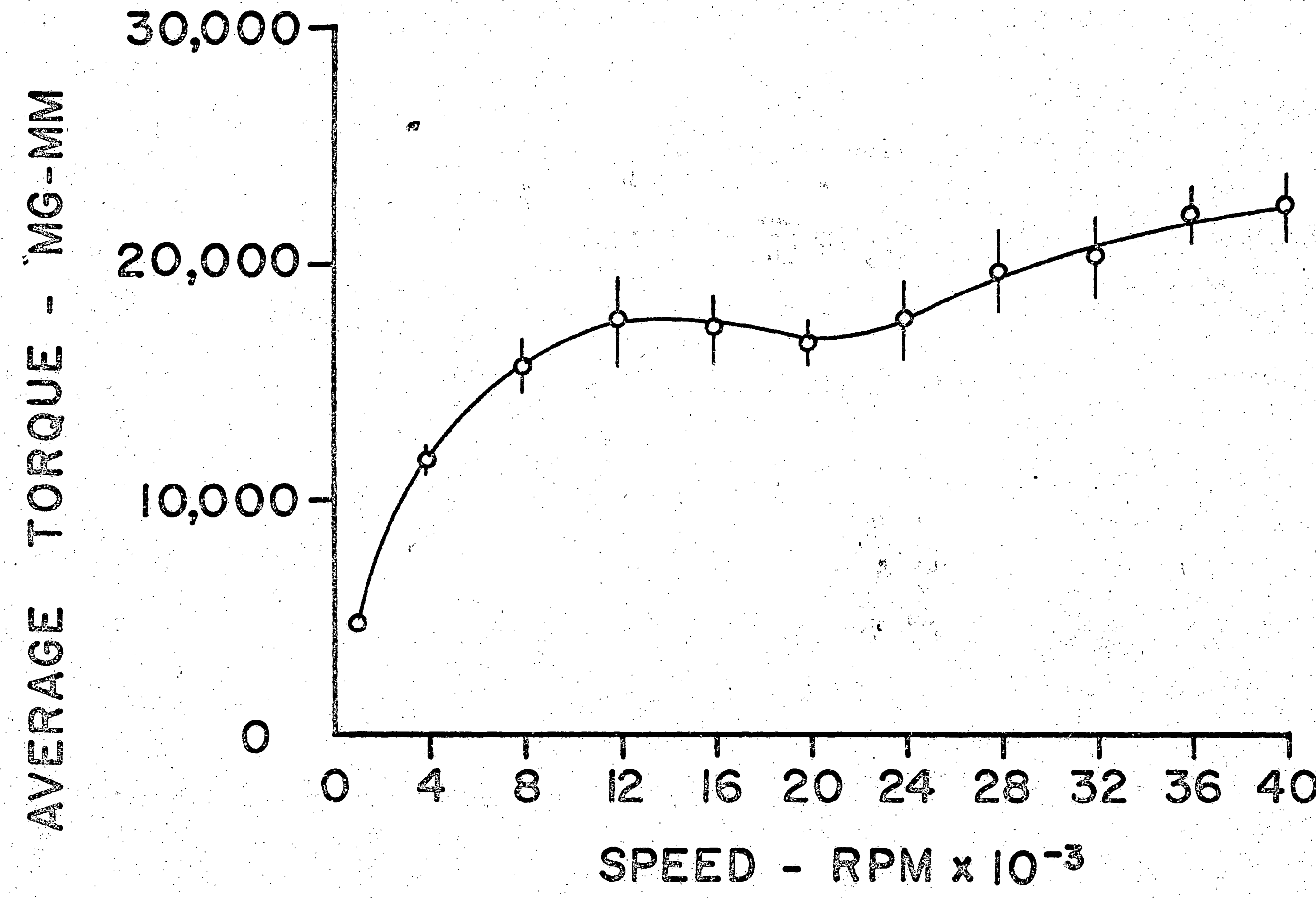

FIG. 31 1/16 GREASE PACK, 200 GM. RADIAL, 0 AXIAL

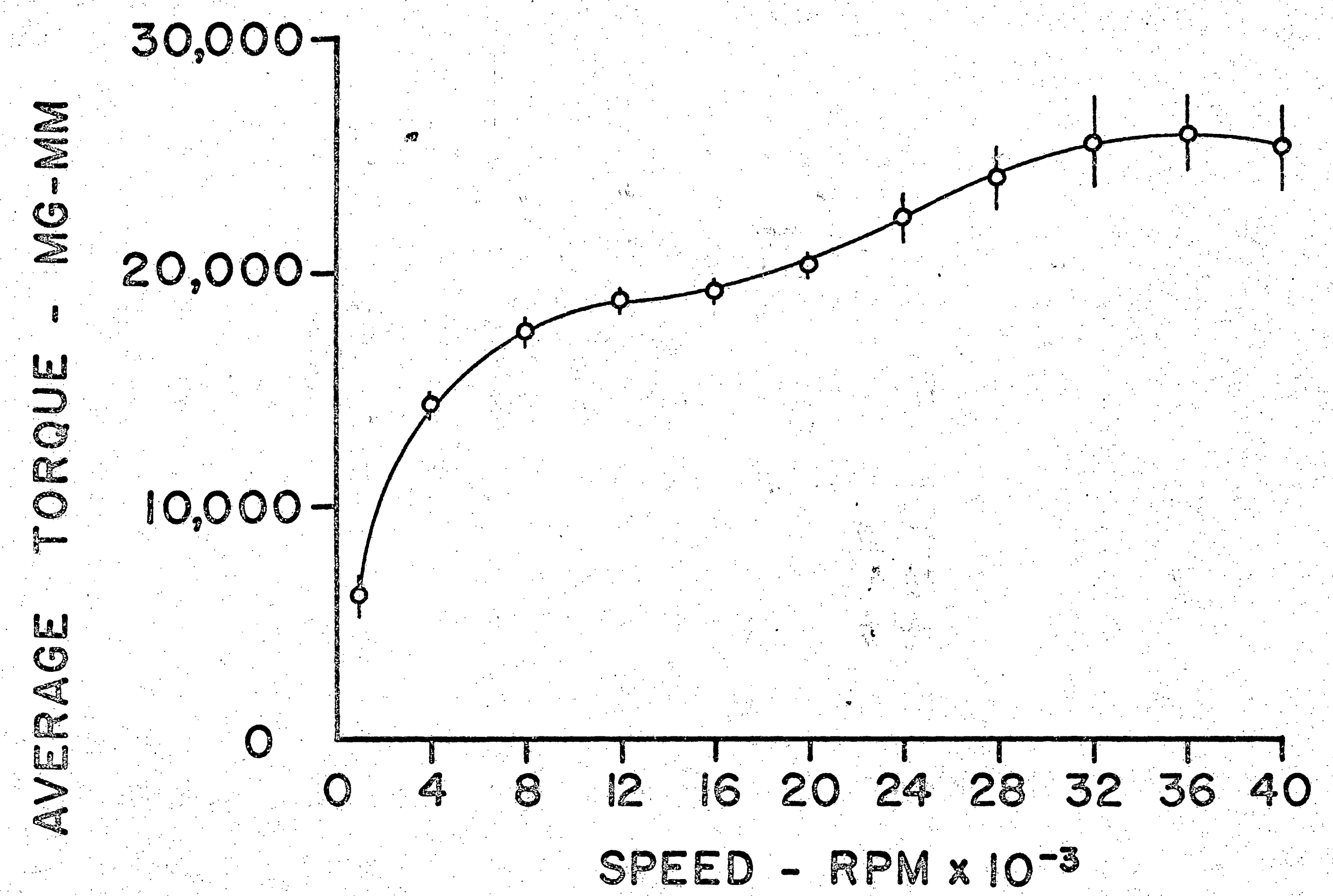

FIG. 32 1/16 GREASE PACK, 200 GM. RADIAL, 50 GM. AXIAL

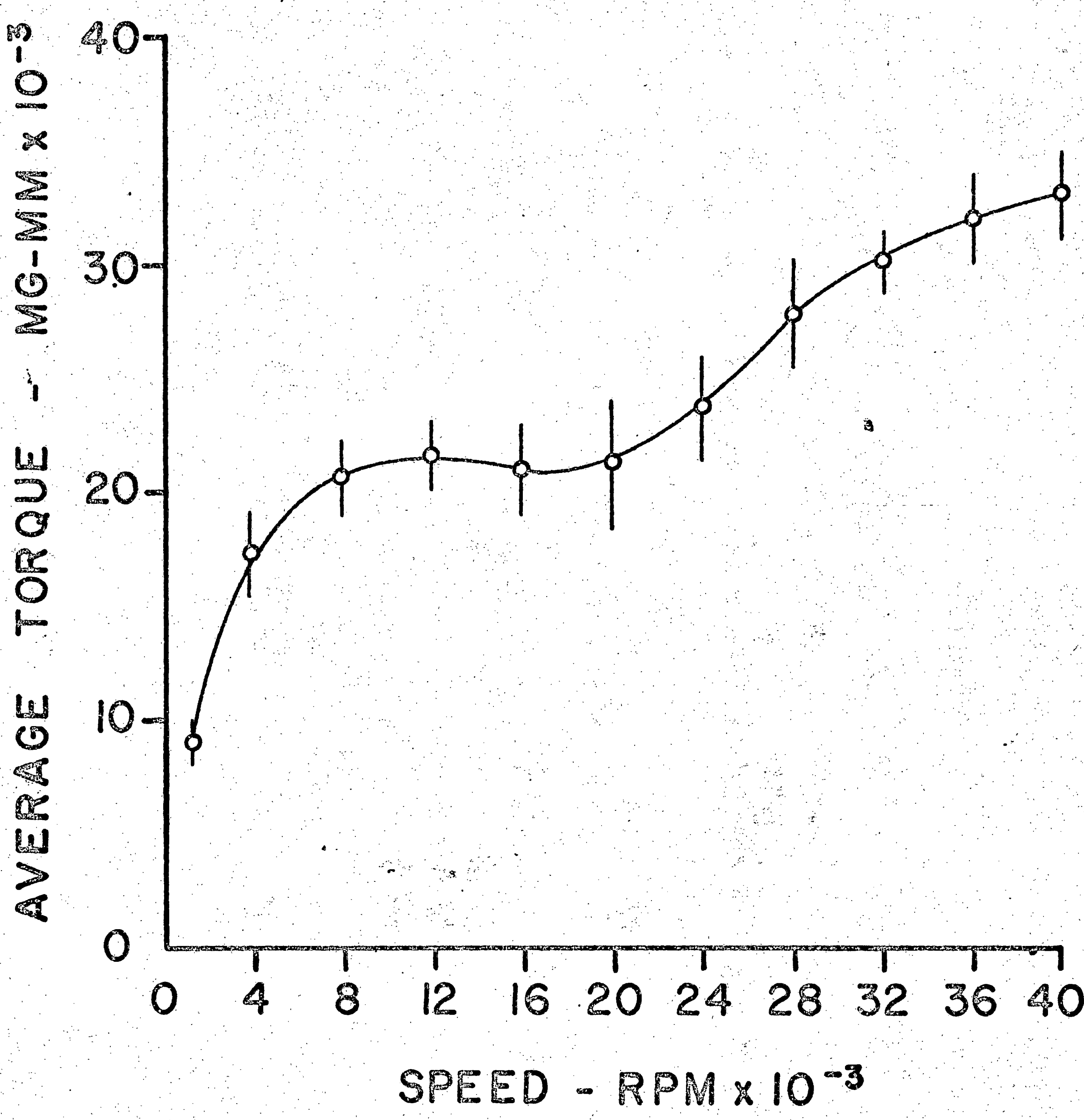

FIG. 33 1/16 GREASE PACK
200 GM. RADIAL, 100 GM. AXIAL

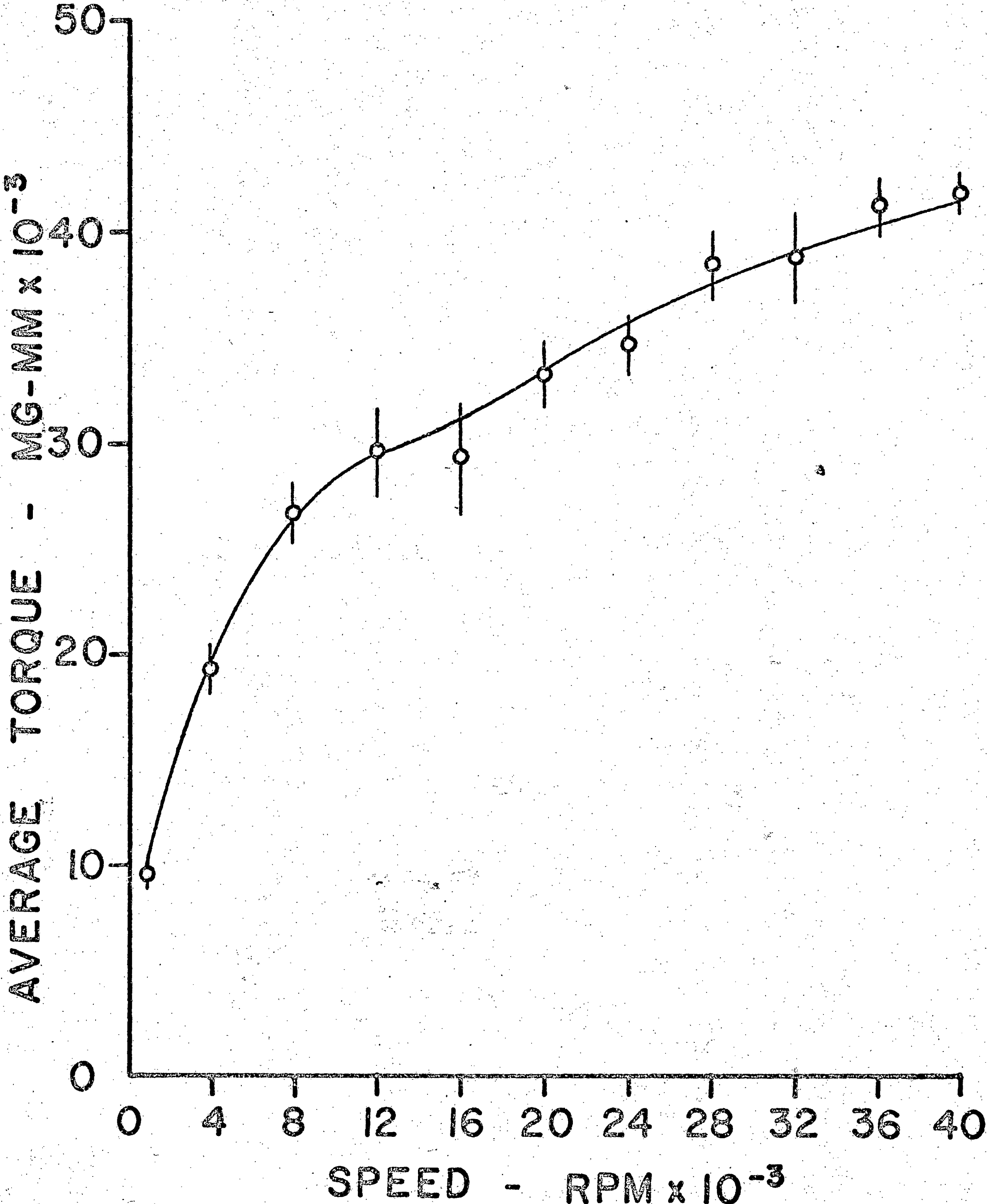

FIG. 34 1/16 GREASE PACK
200 GM. RADIAL, 200 GM. AXIAL

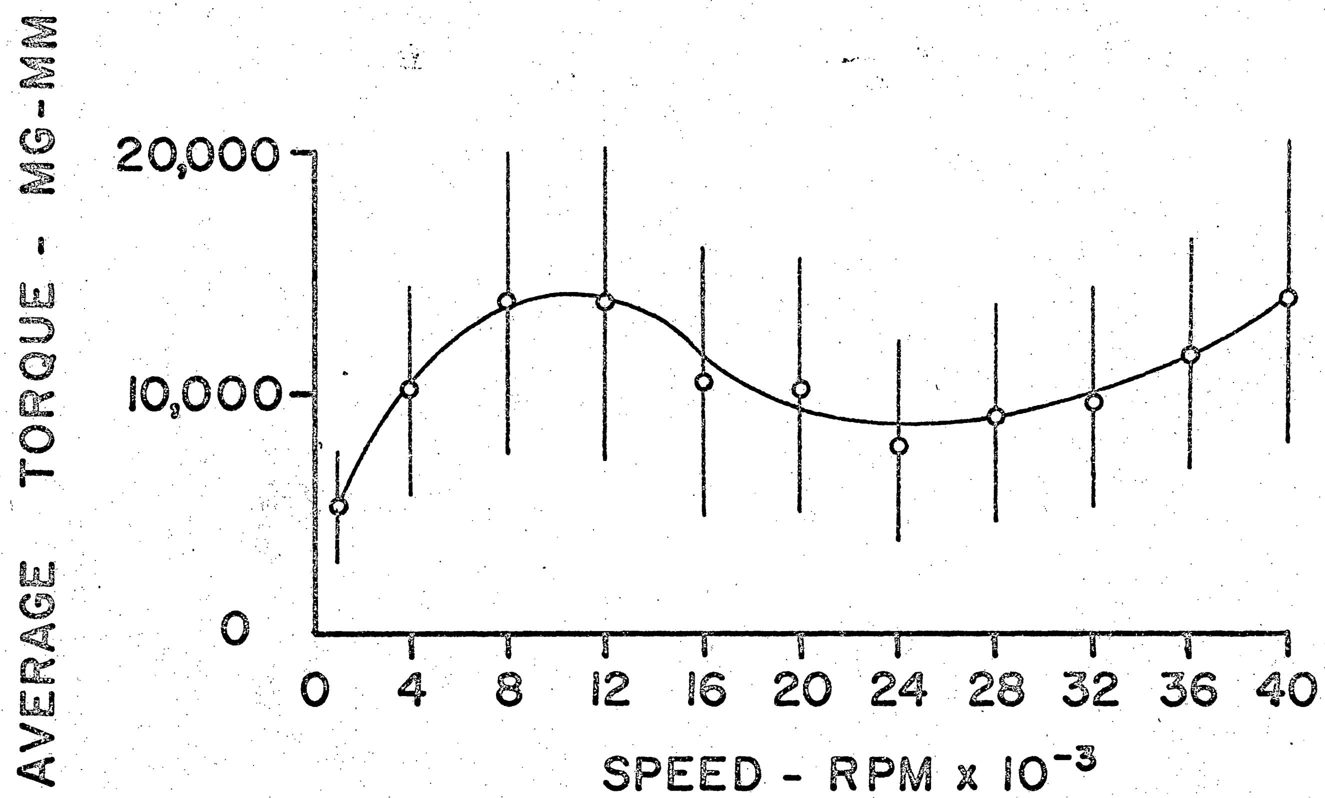

FIG. 35 1/8 GREASE PACK, 50 GM. RADIAL, 0 AXIAL

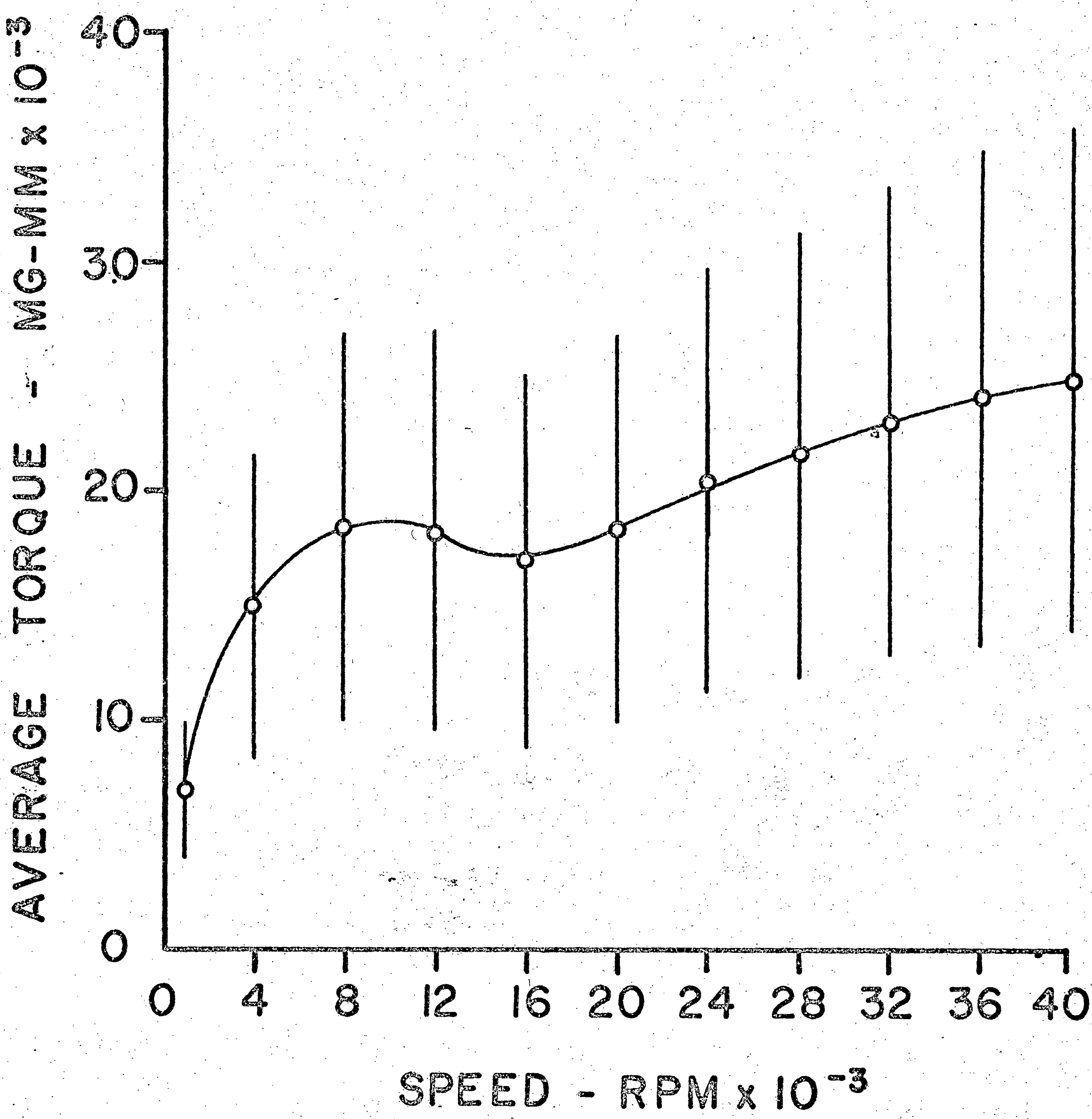

FIG. 36 1/8 GREASE PACK
50 GM. RADIAL, 50 GM. AXIAL

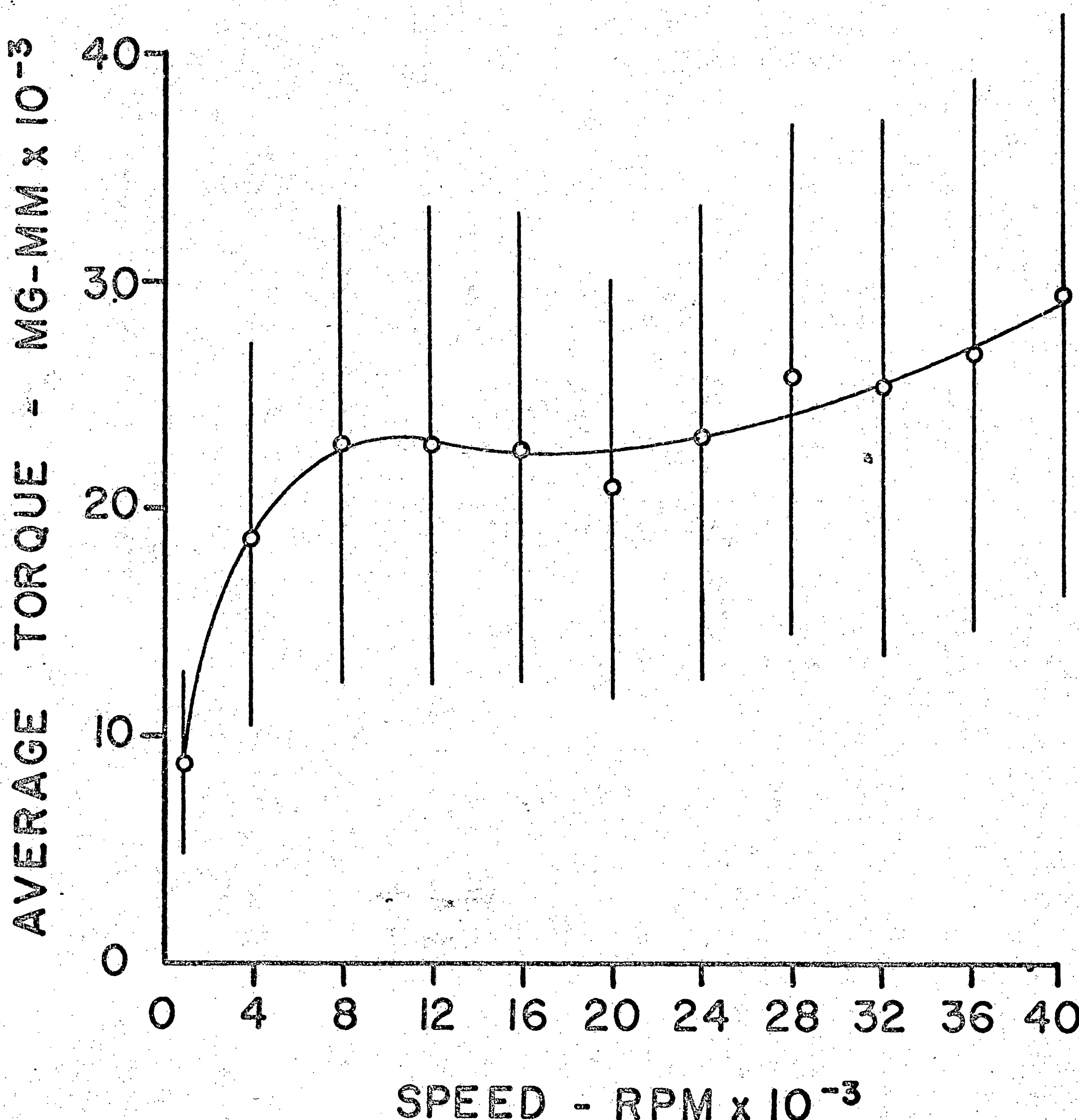

FIG. 37 1/8 GREASE PACK
50 GM. RADIAL, 100 GM. AXIAL

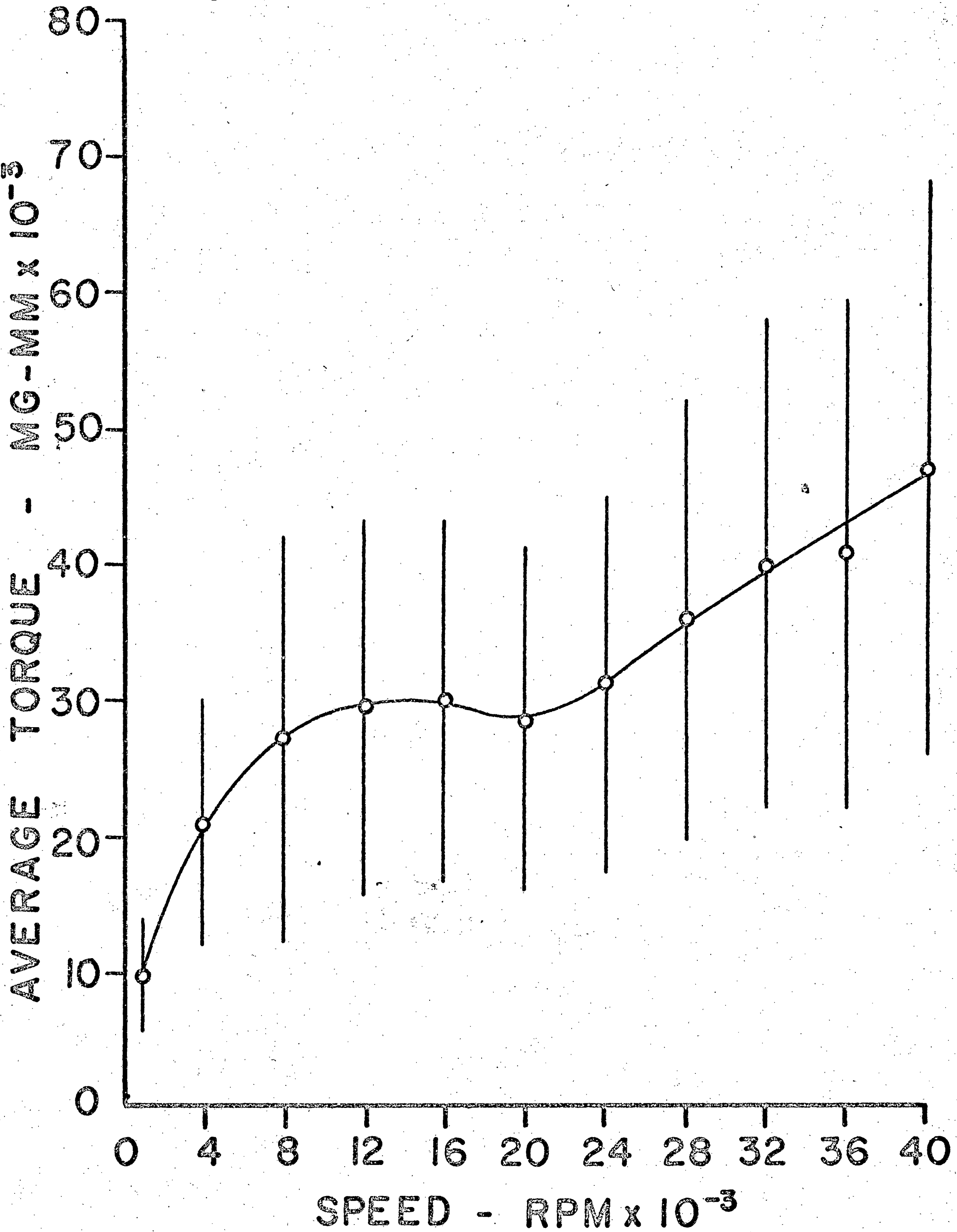

FIG. 38 1/8 GREASE PACK
50 GM. RADIAL, 200 GM. AXIAL

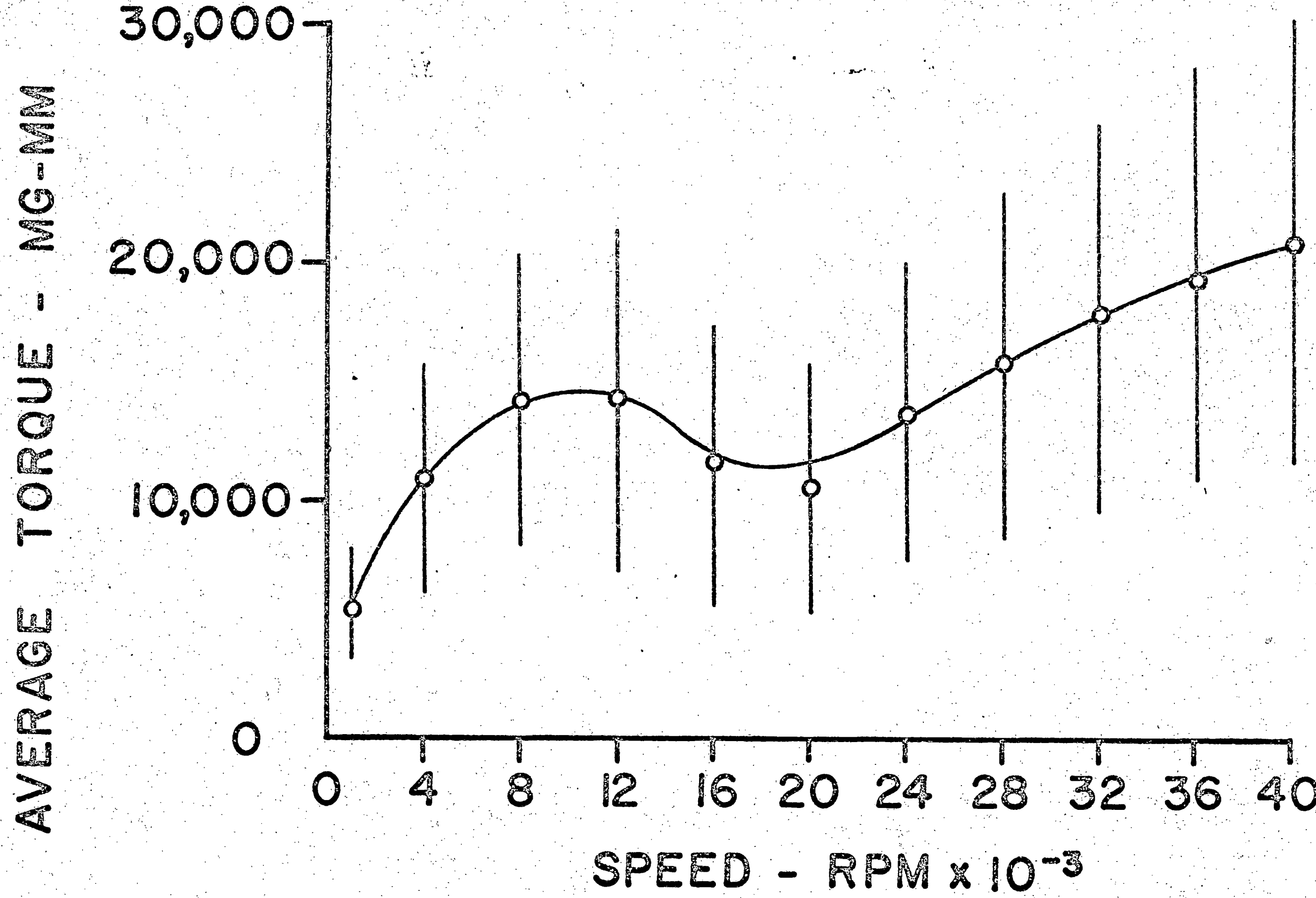

FIG. 39 1/8 GREASE PACK, 100 GM. RADIAL, 0 AXIAL

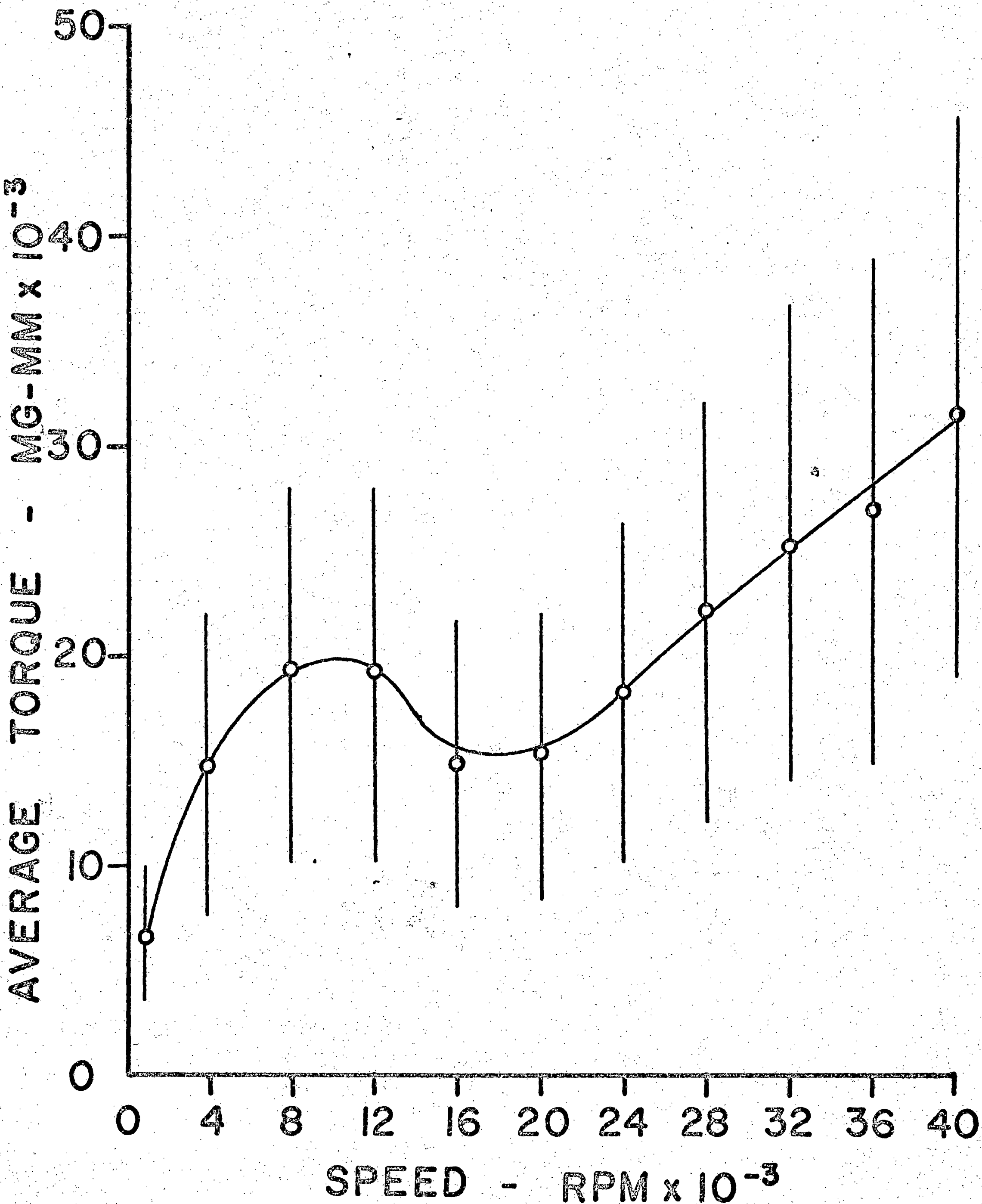

FIG. 40 1/8 GREASE PACK
100 GM. RADIAL, 50 GM. AXIAL

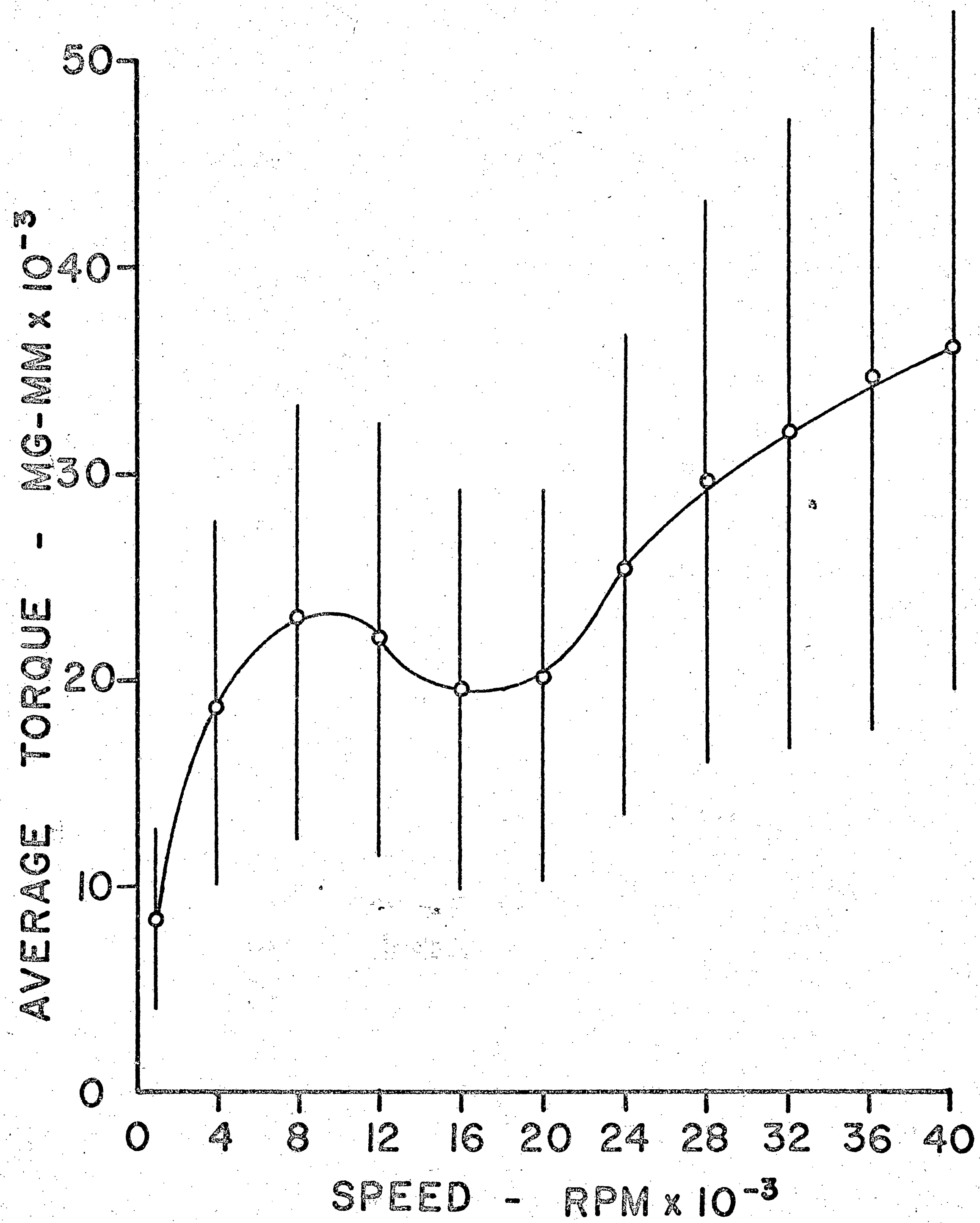

FIG. 41 1/8 GREASE PACK
100 GM. RADIAL, 100 GM. AXIAL

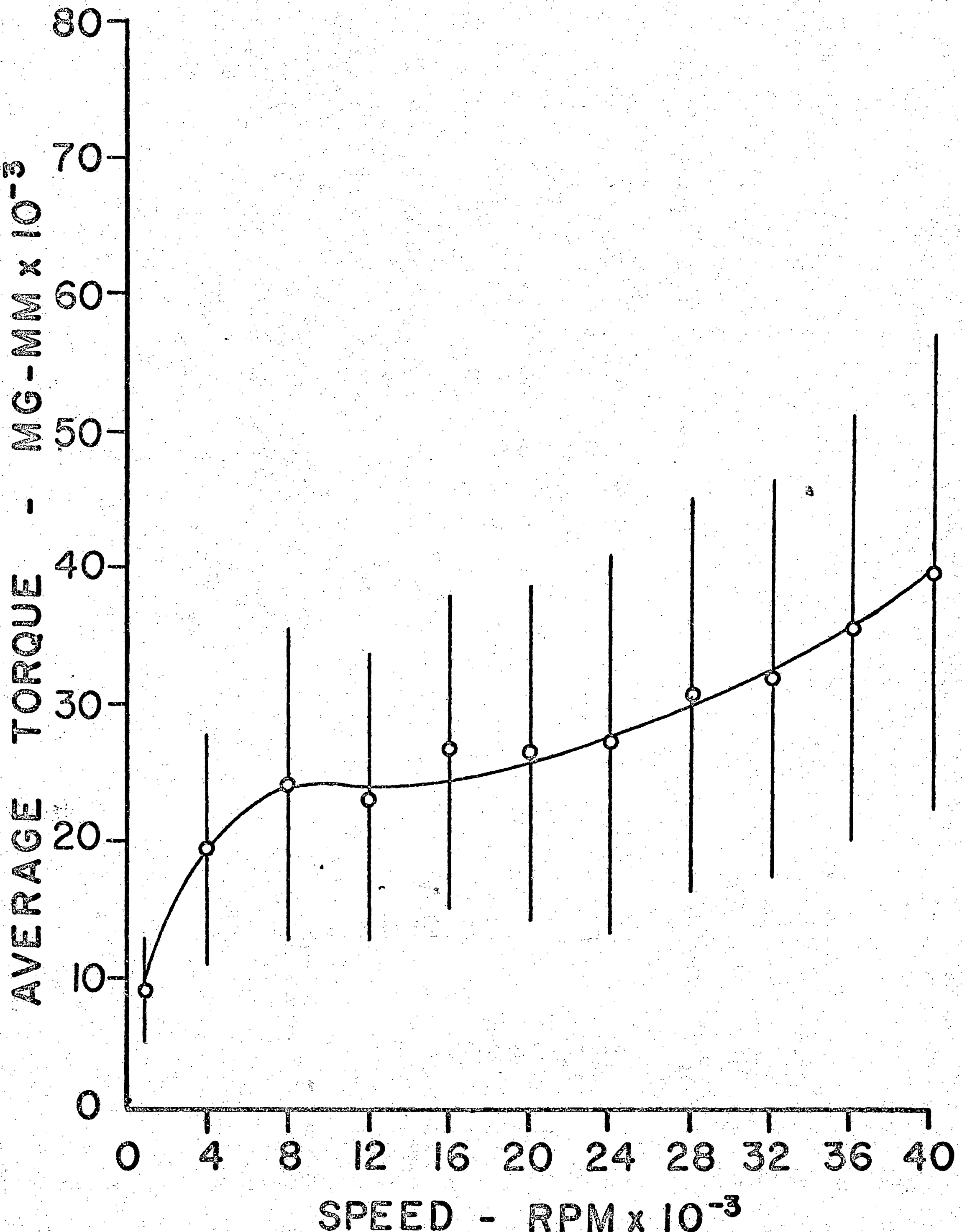

FIG. 42 1/8 GREASE PACK
100 GM. RADIAL, 200 GM. AXIAL

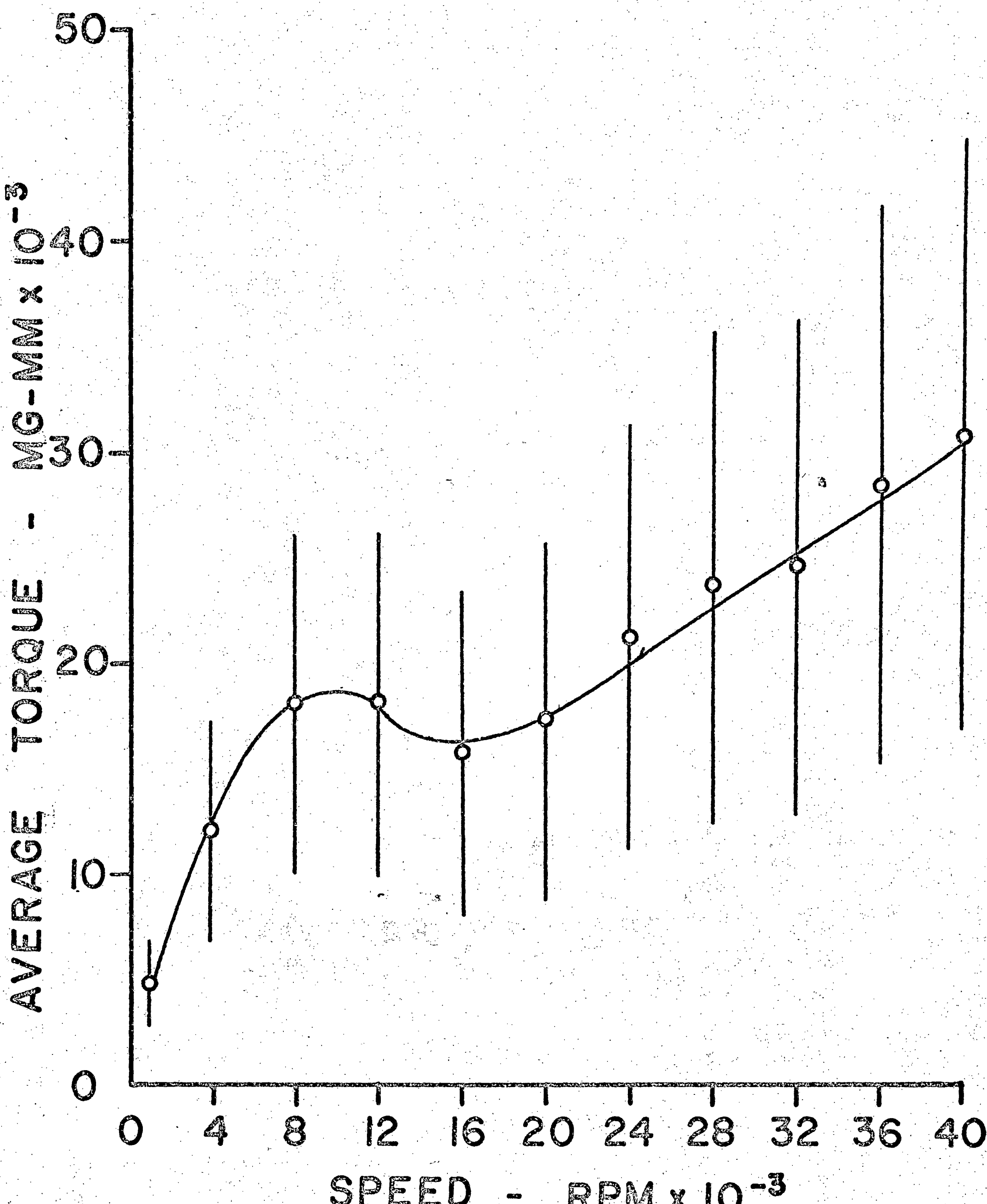

FIG. 43 1/8 GREASE PACK
200 GM. RADIAL, 0 AXIAL

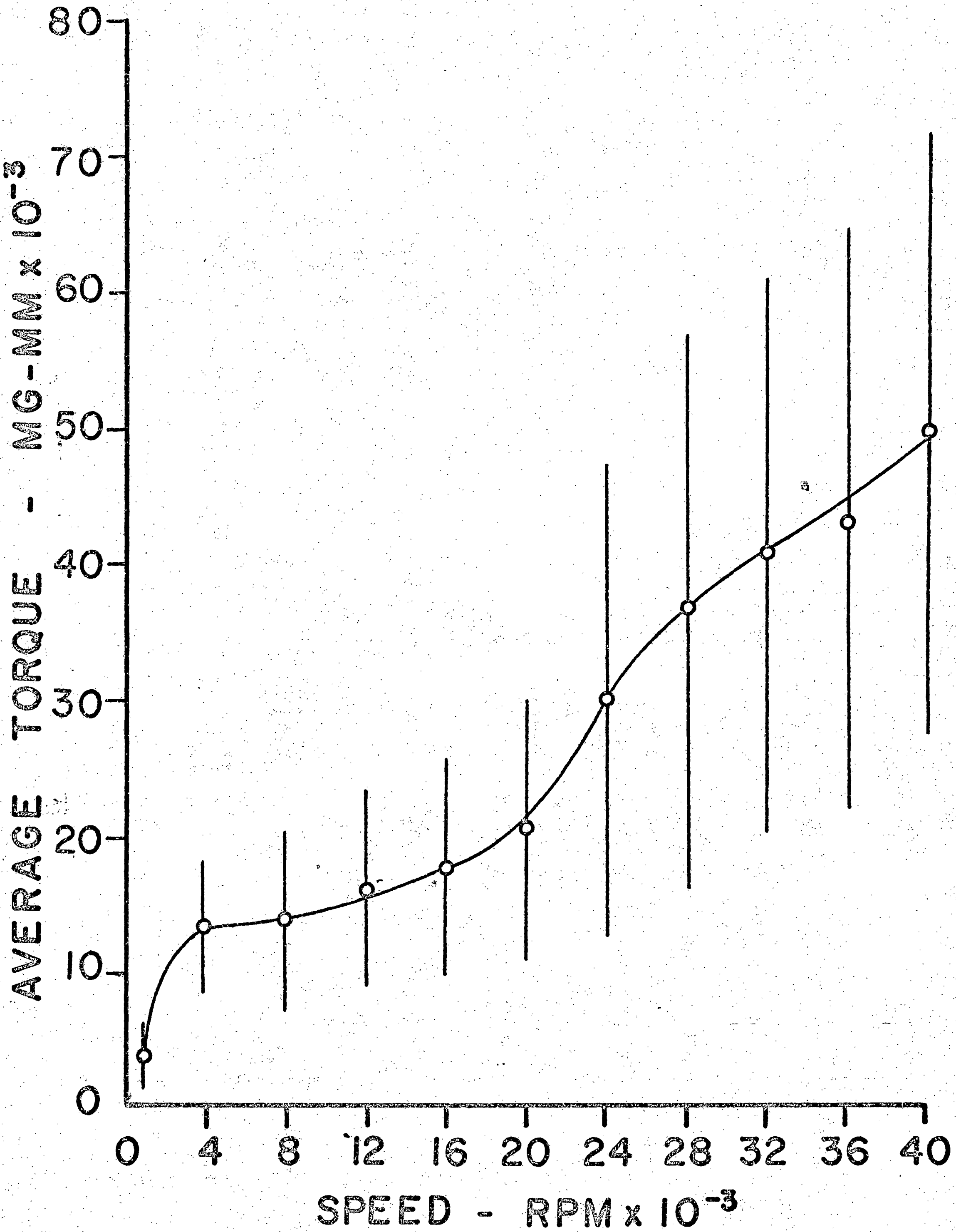

FIG. 44 1/8 GREASE PACK
200 GM. RADIAL, 50 GM. AXIAL

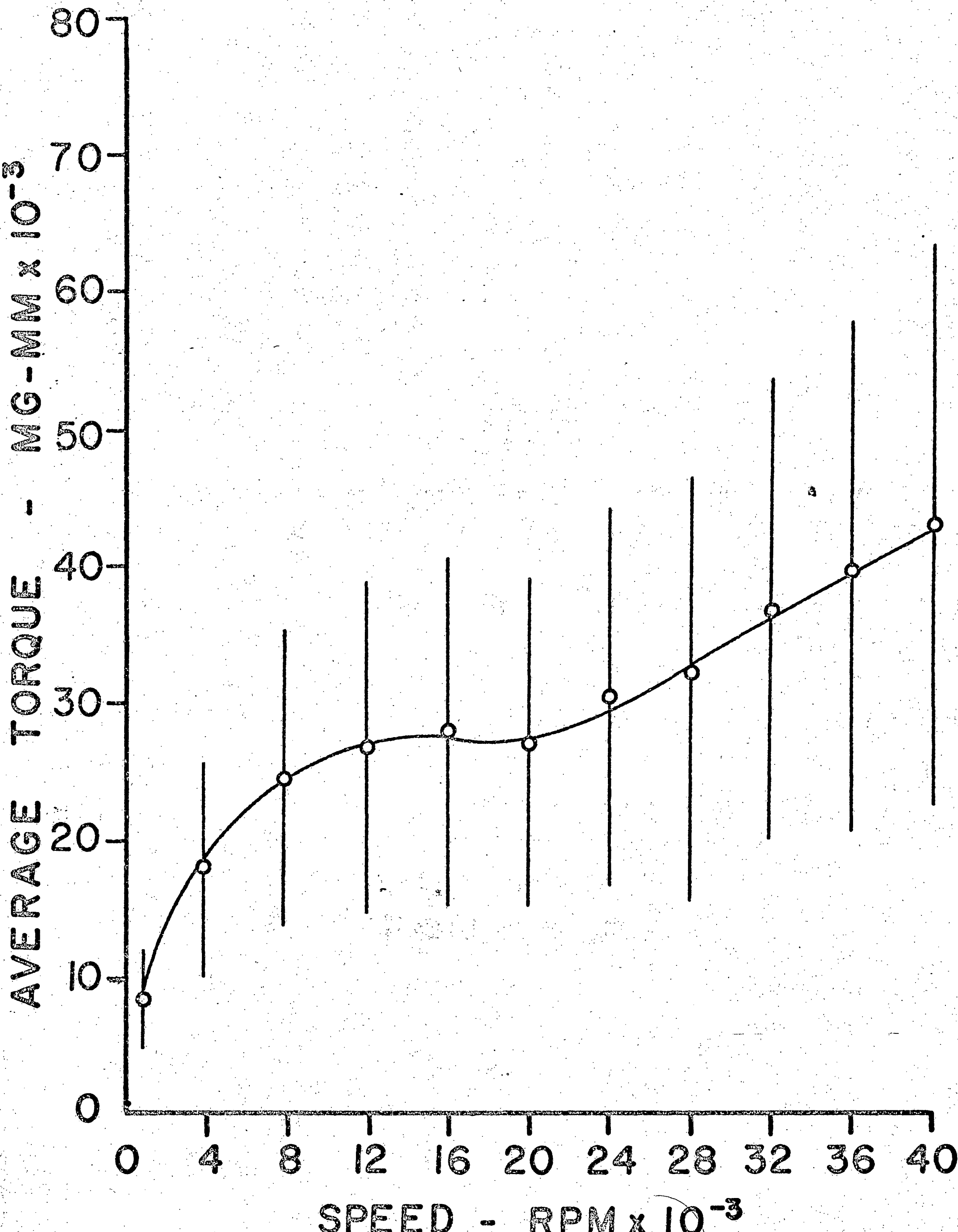

FIG. 45 1/8 GREASE PACK
200 GM. RADIAL, 100 GM. AXIAL

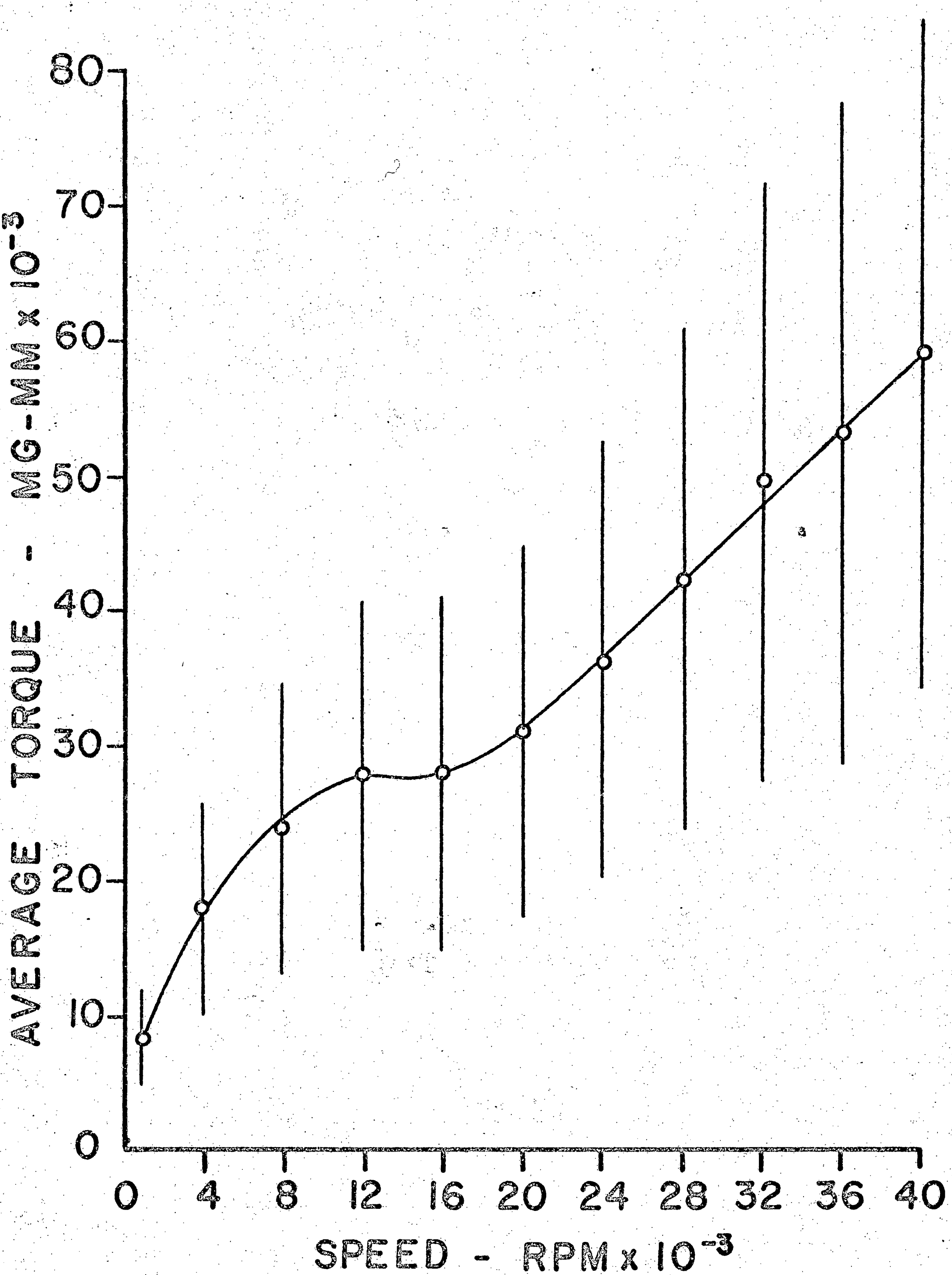

FIG. 46 1/8 GREASE PACK
200 GM. RADIAL, 200 GM. AXIAL

VI. DISCUSSION OF RESULTS

The data shown in the last section has been replotted in order to observe the effects of combined loading on the torque characteristics. For each of the three lubricant conditions used in the test program, seven graphs are presented. Four of these graphs illustrate the effect on average torque when varying the radial load while maintaining constant axial load. The remaining three graphs are plotted for constant radial load with varying axial load.

Figures 47 through 53 illustrate the average torque characteristics for oil lubricated bearings. In Figures 47 through 50, it can be observed that the torque is not significantly affected by varying the radial load while maintaining a constant axial load. However, for varying axial load, constant radial load, it can be observed in Figures 51 through 53 that the average torque was significantly affected by the value of axial load.

Figures 54 through 60 and 61 through 67 illustrate the effects of combined loading on the average torque for the 1/16 and 1/8 grease pack, respectively. In general, the torque of the bearings lubricated with 1/16 grease pack behaved quite similar to that

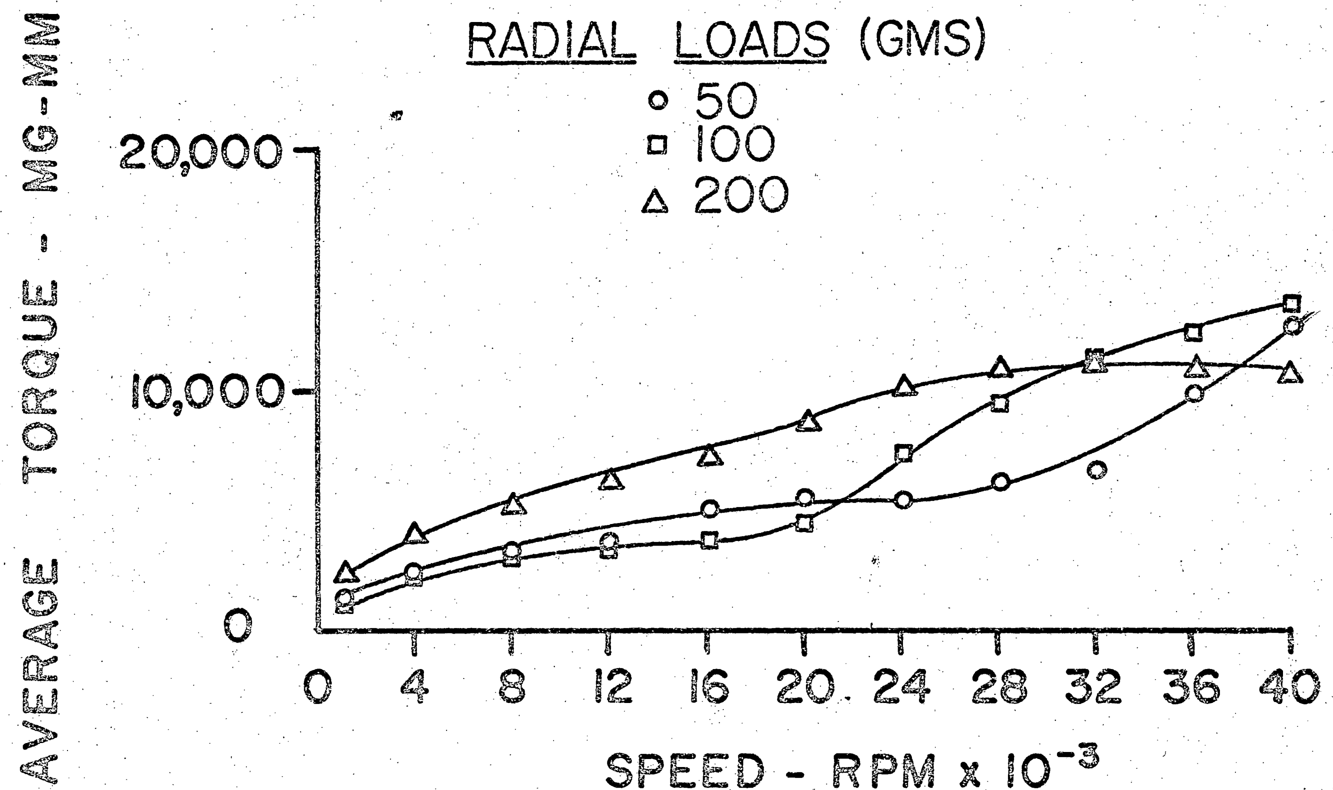

FIG. 47 OIL, 0 GM. AXIAL LOAD

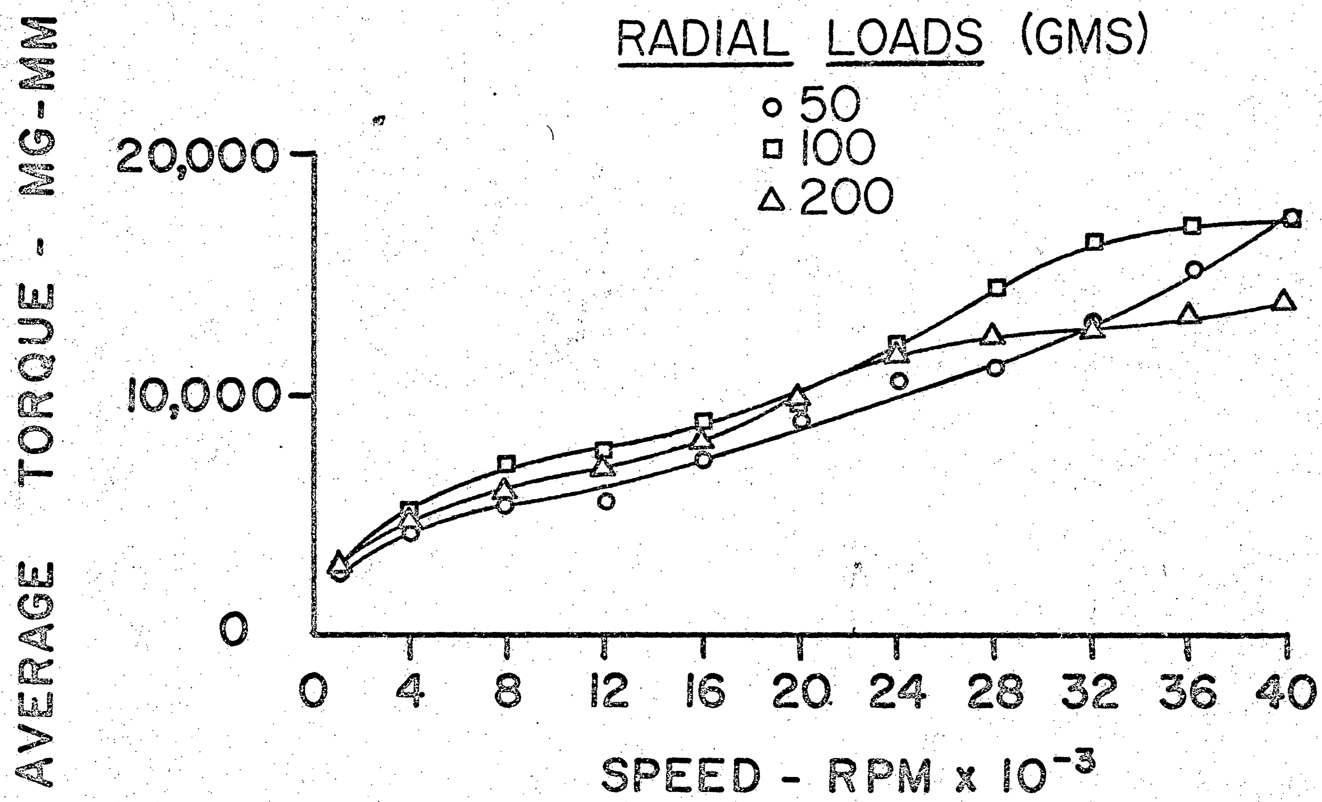

FIG. 48 OIL, 50 GM. AXIAL LOAD

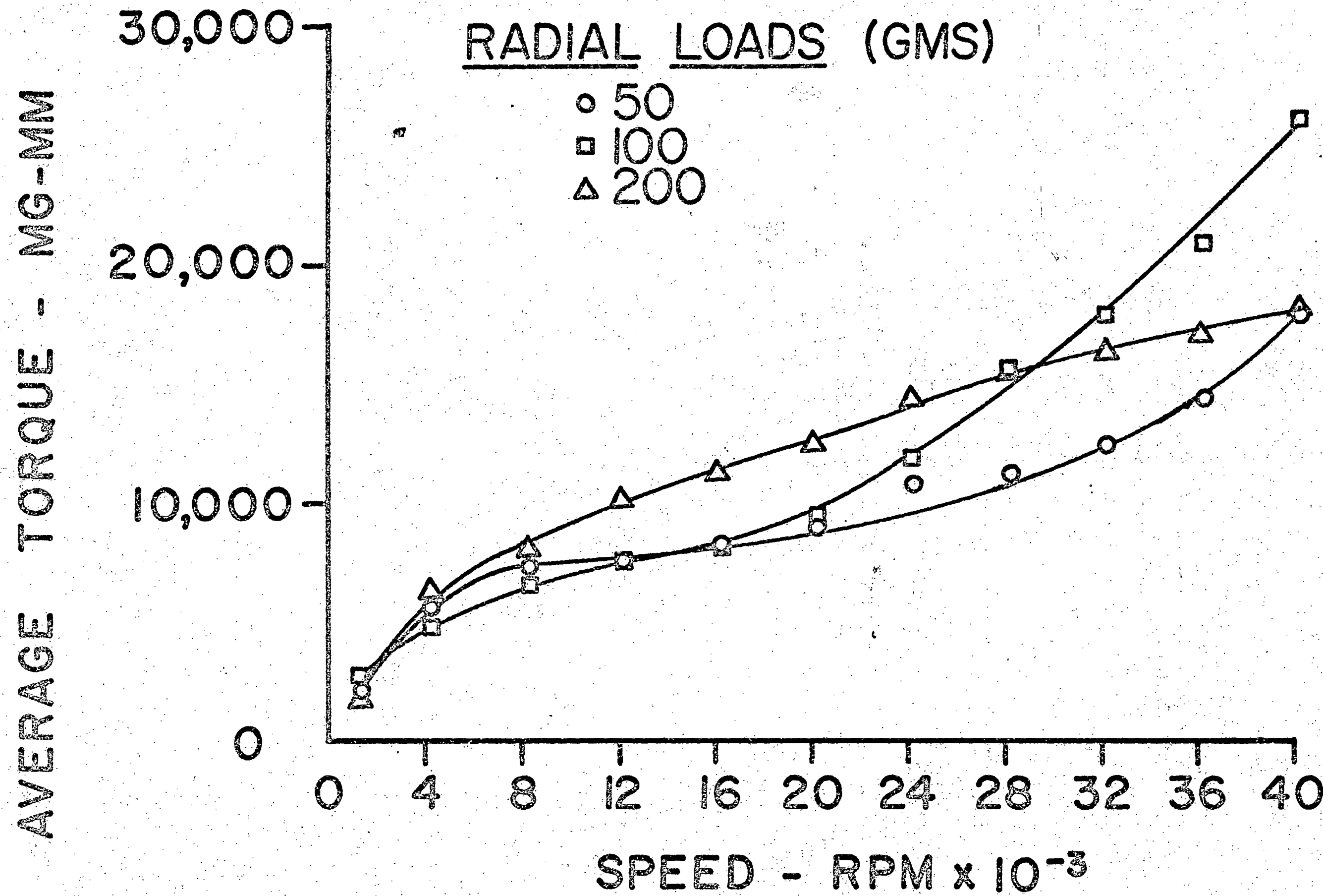

FIG. 49 OIL, 100 GM. AXIAL LOAD

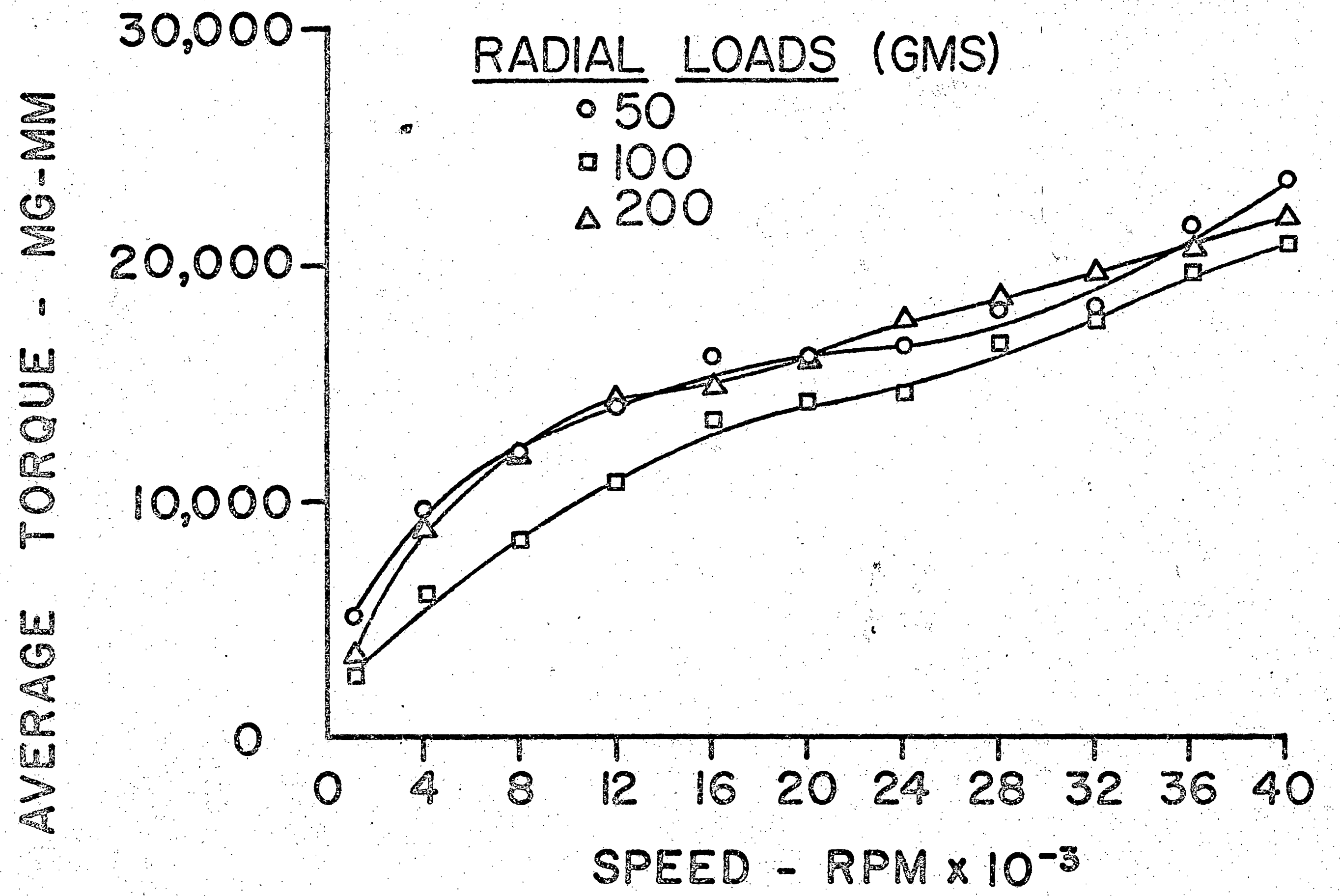

FIG. 50 OIL, 200 GM. AXIAL LOAD

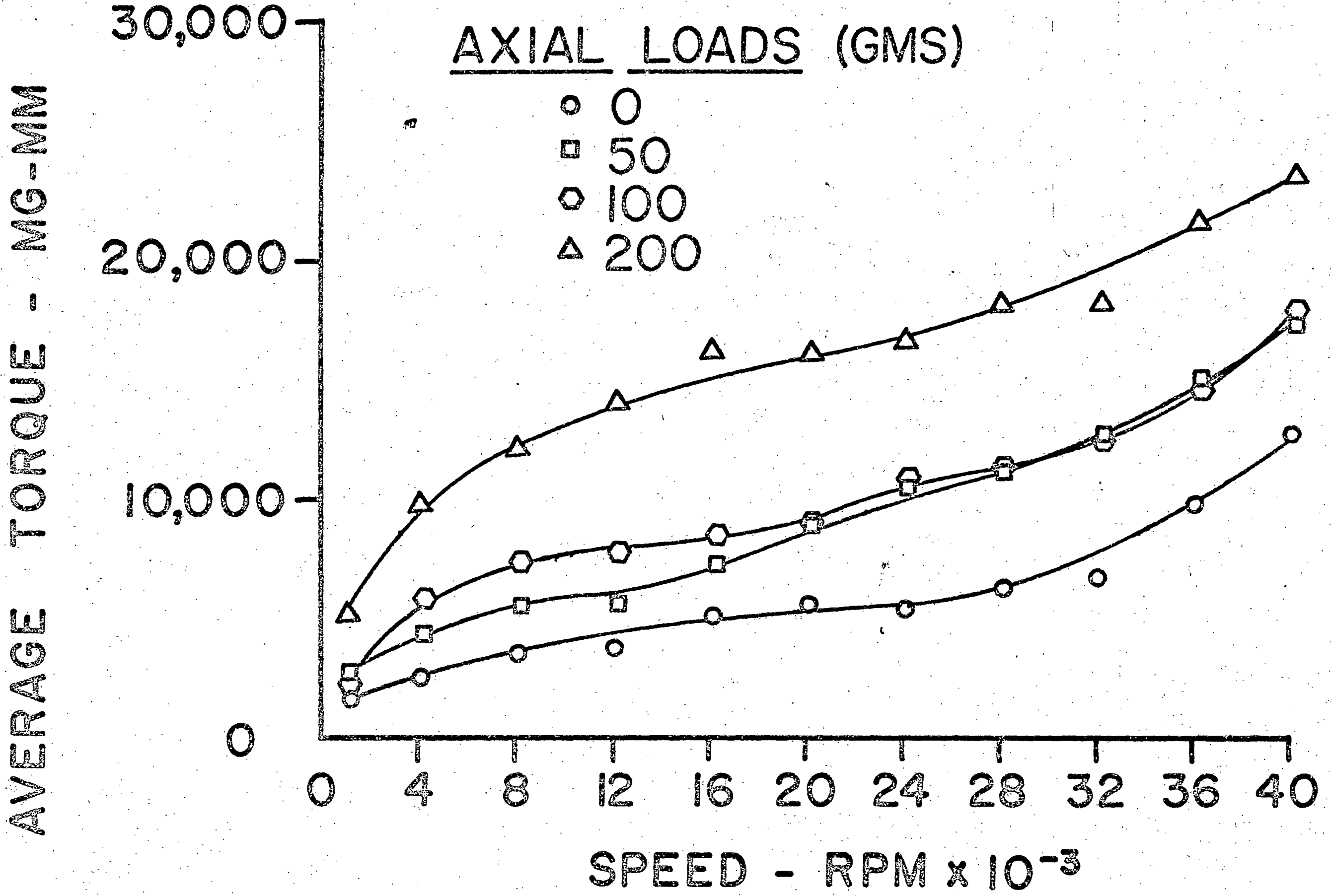

FIG. 51 OIL, 50 GM. RADIAL LOAD

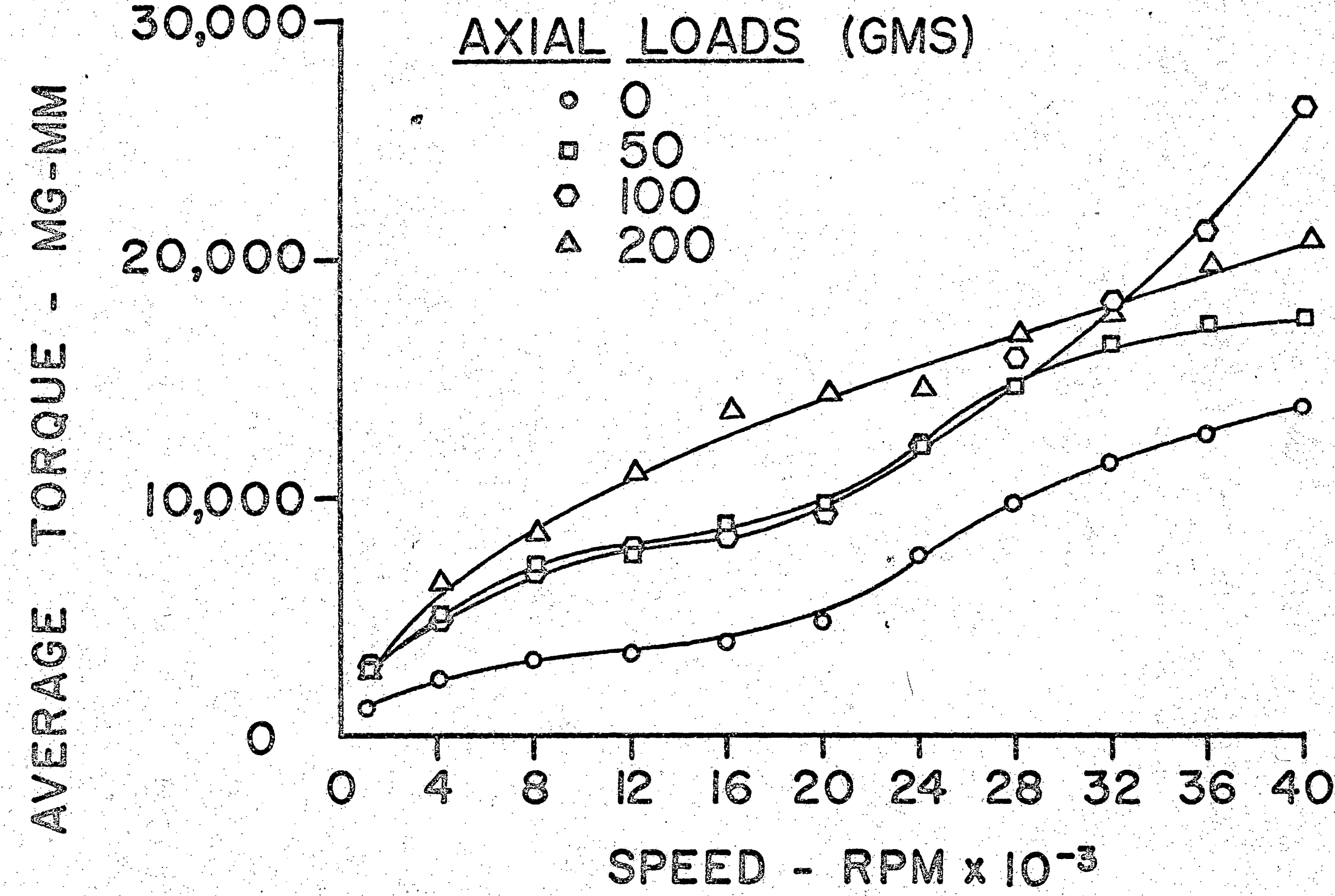

FIG. 52 OIL, 100 GM. RADIAL LOAD

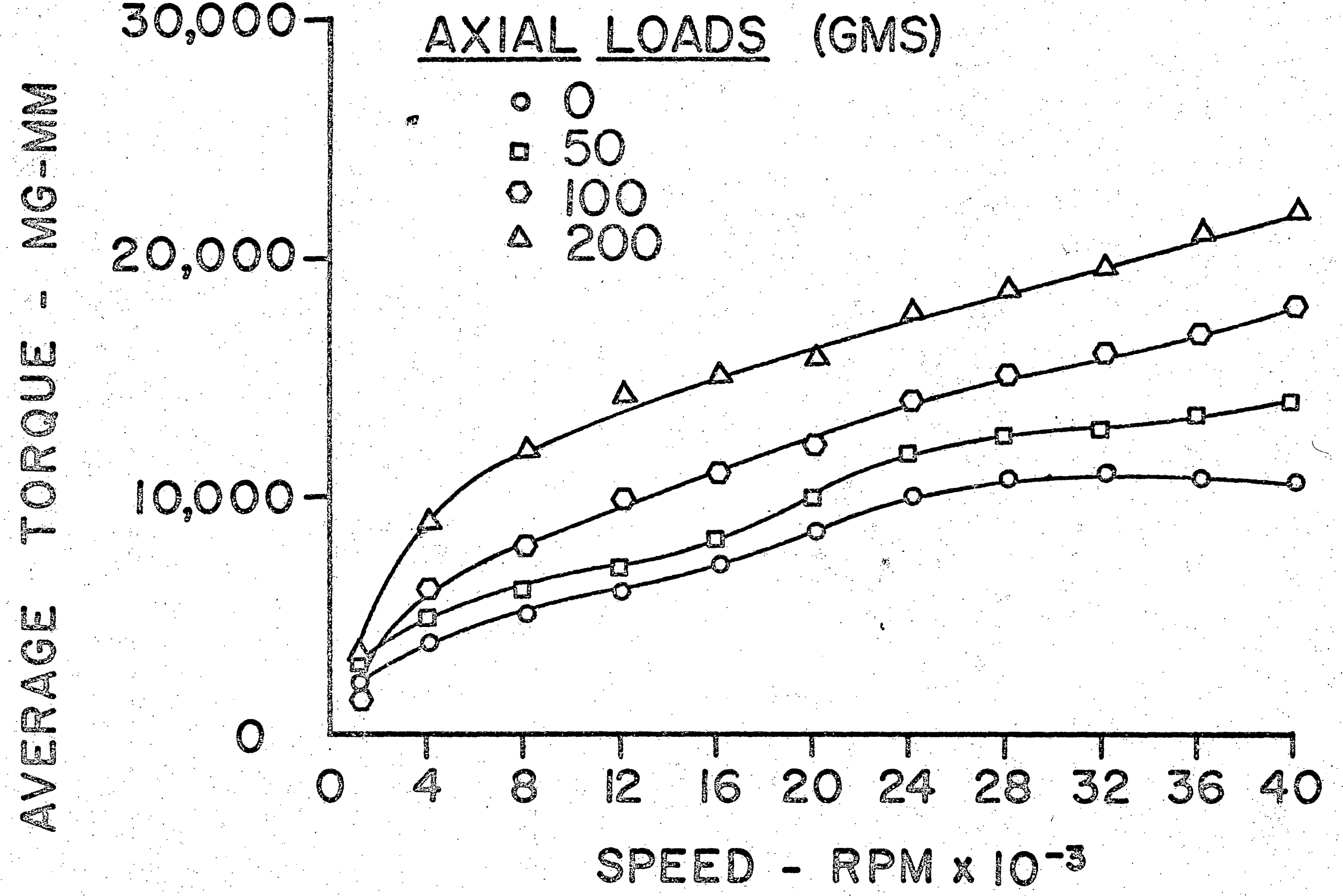

FIG. 53 OIL, 200 GM. RADIAL LOAD

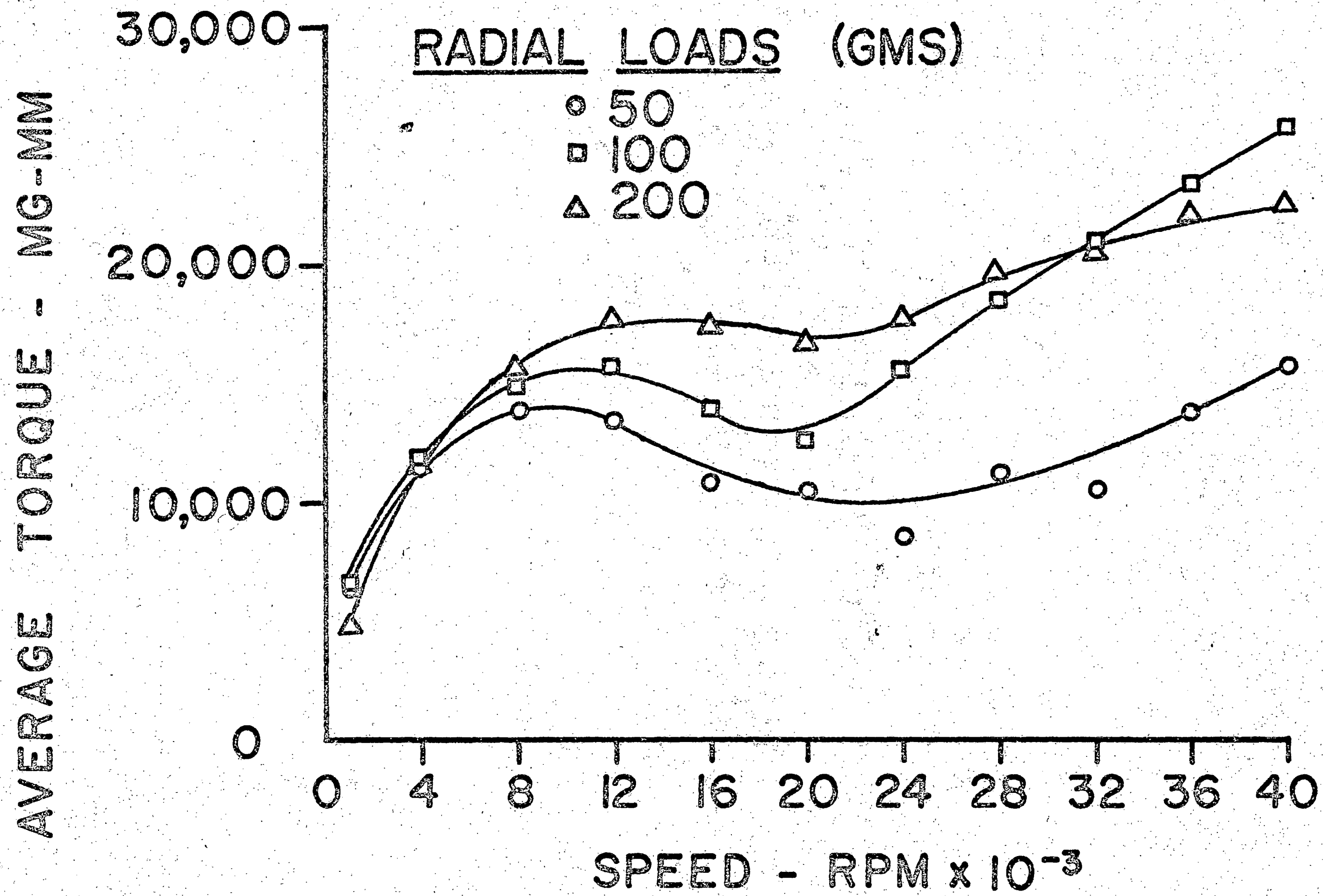

FIG. 54 1/16 GREASE PACK, 0 GM. AXIAL LOAD

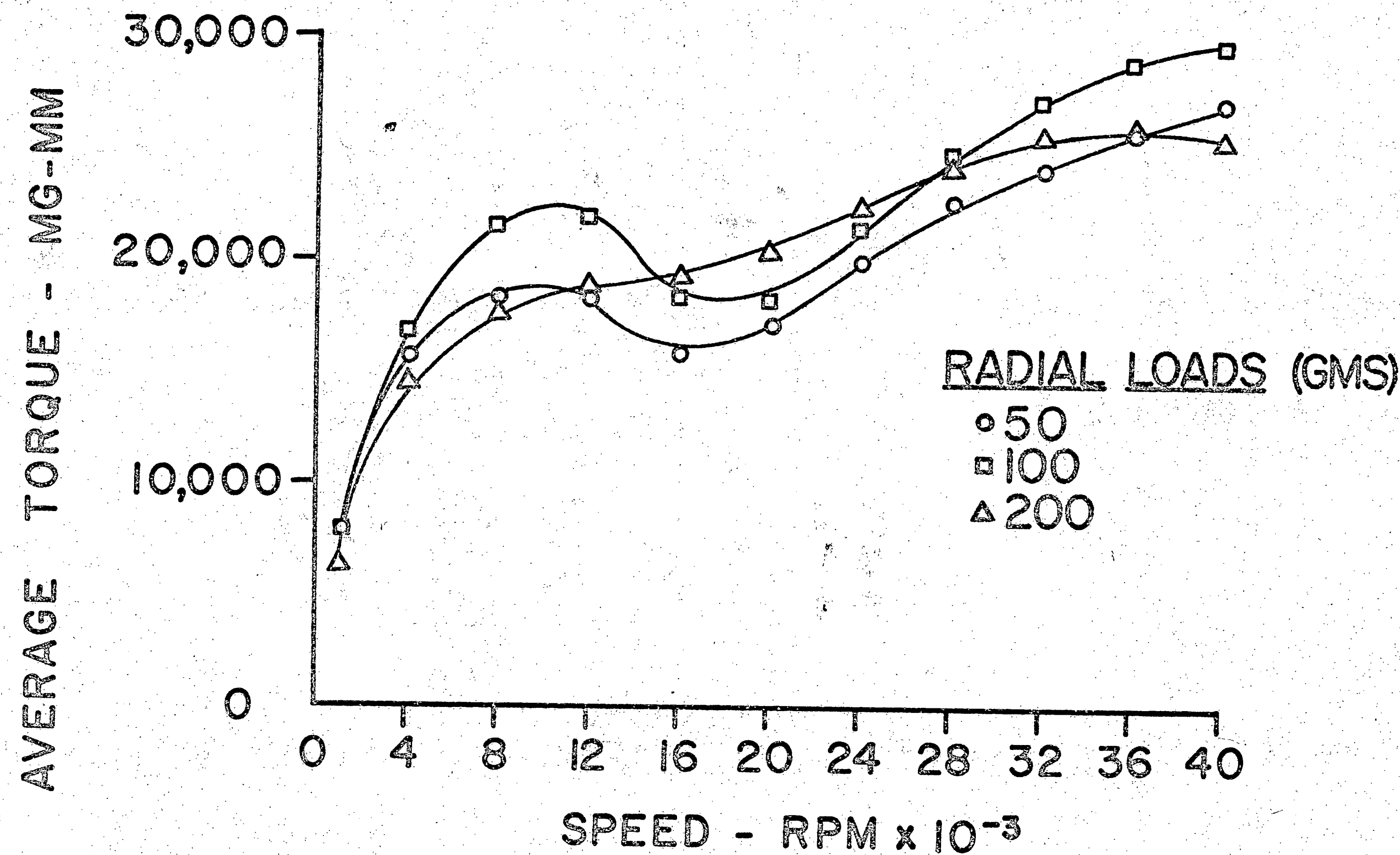

FIG. 55 1/16 GREASE PACK, 50 GM. AXIAL LOAD

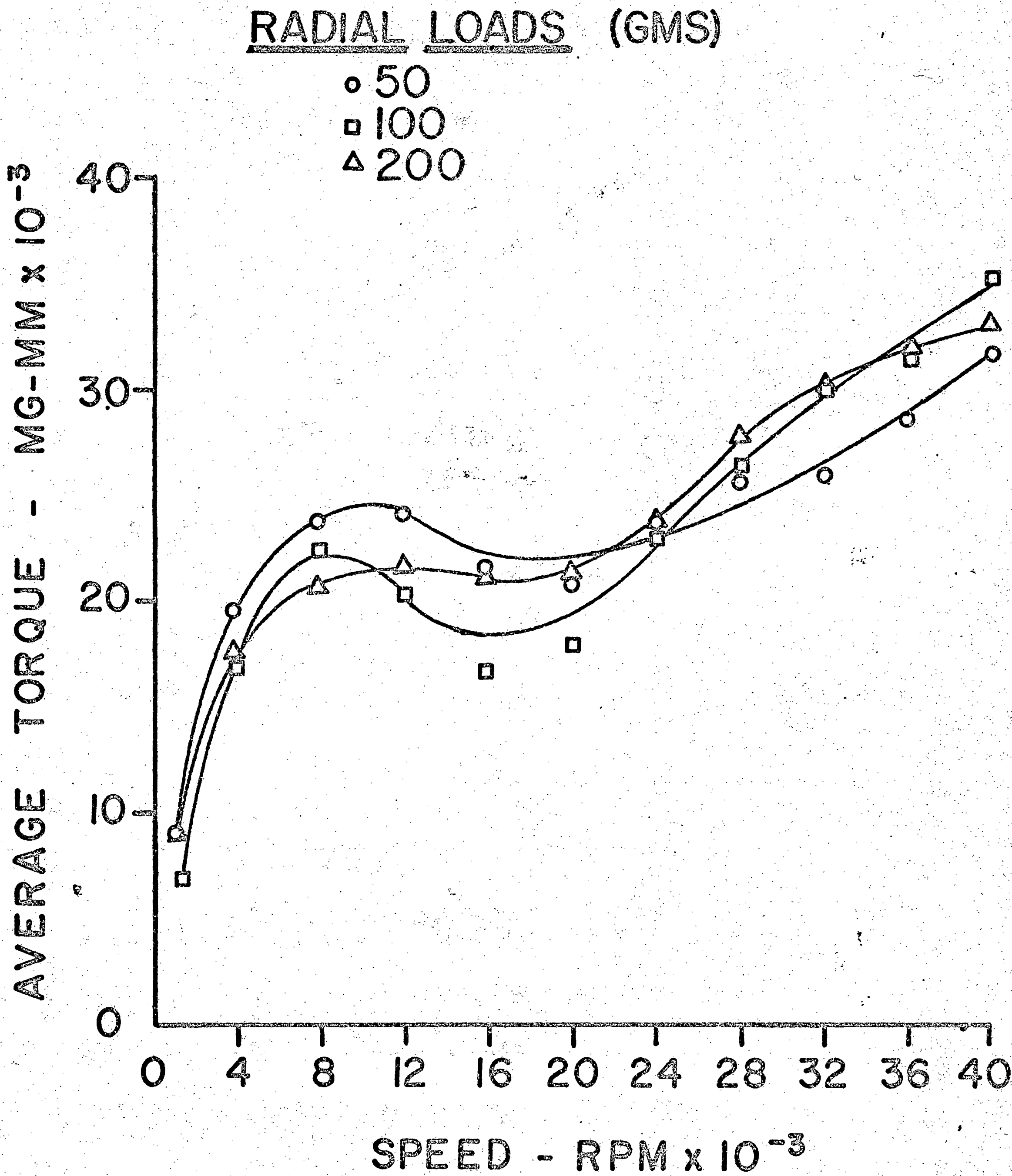

FIG. 56 1/16 GREASE PACK

100 GM. AXIAL LOAD

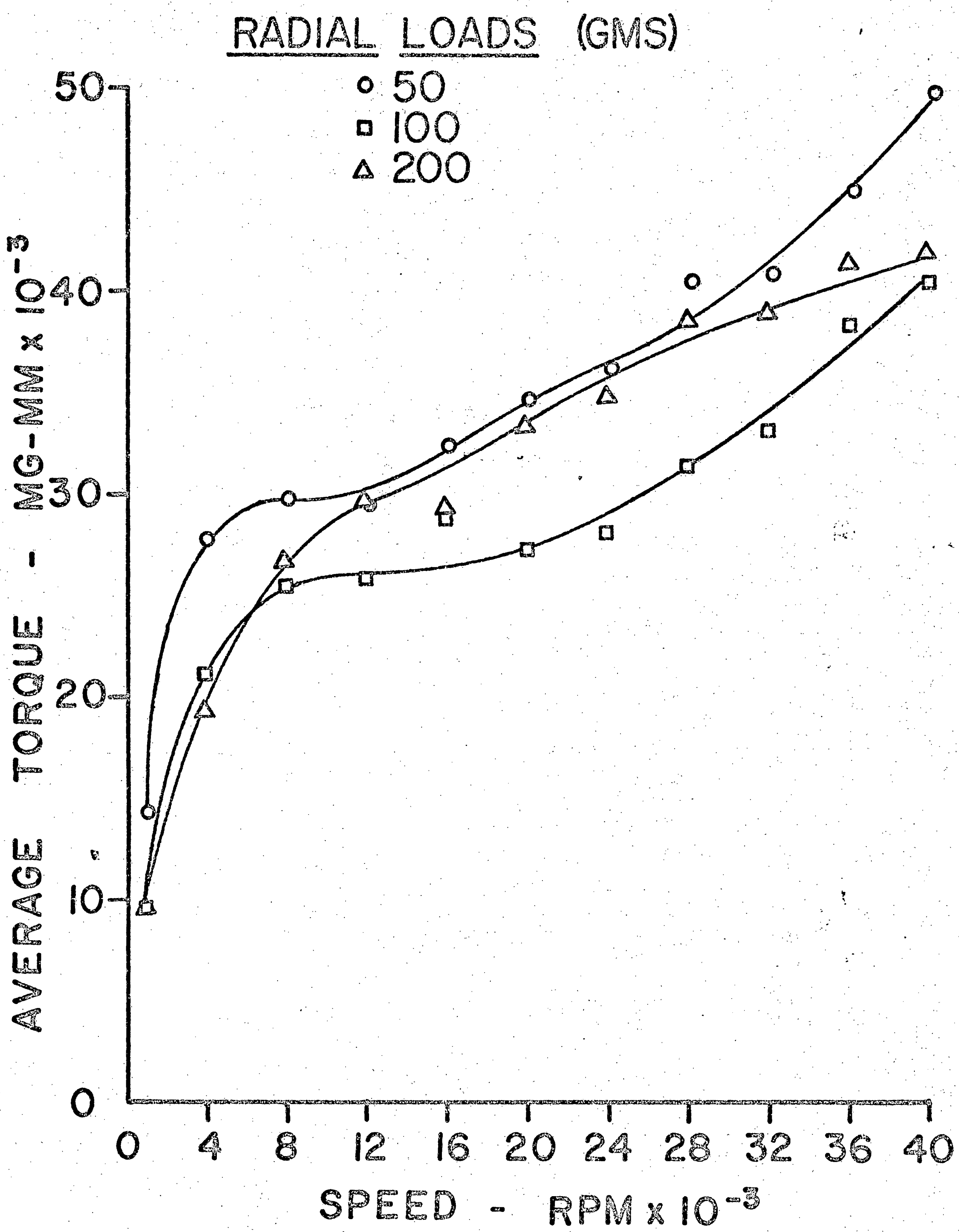

FIG. 57 1/16 GREASE PACK
200 GM. AXIAL LOAD

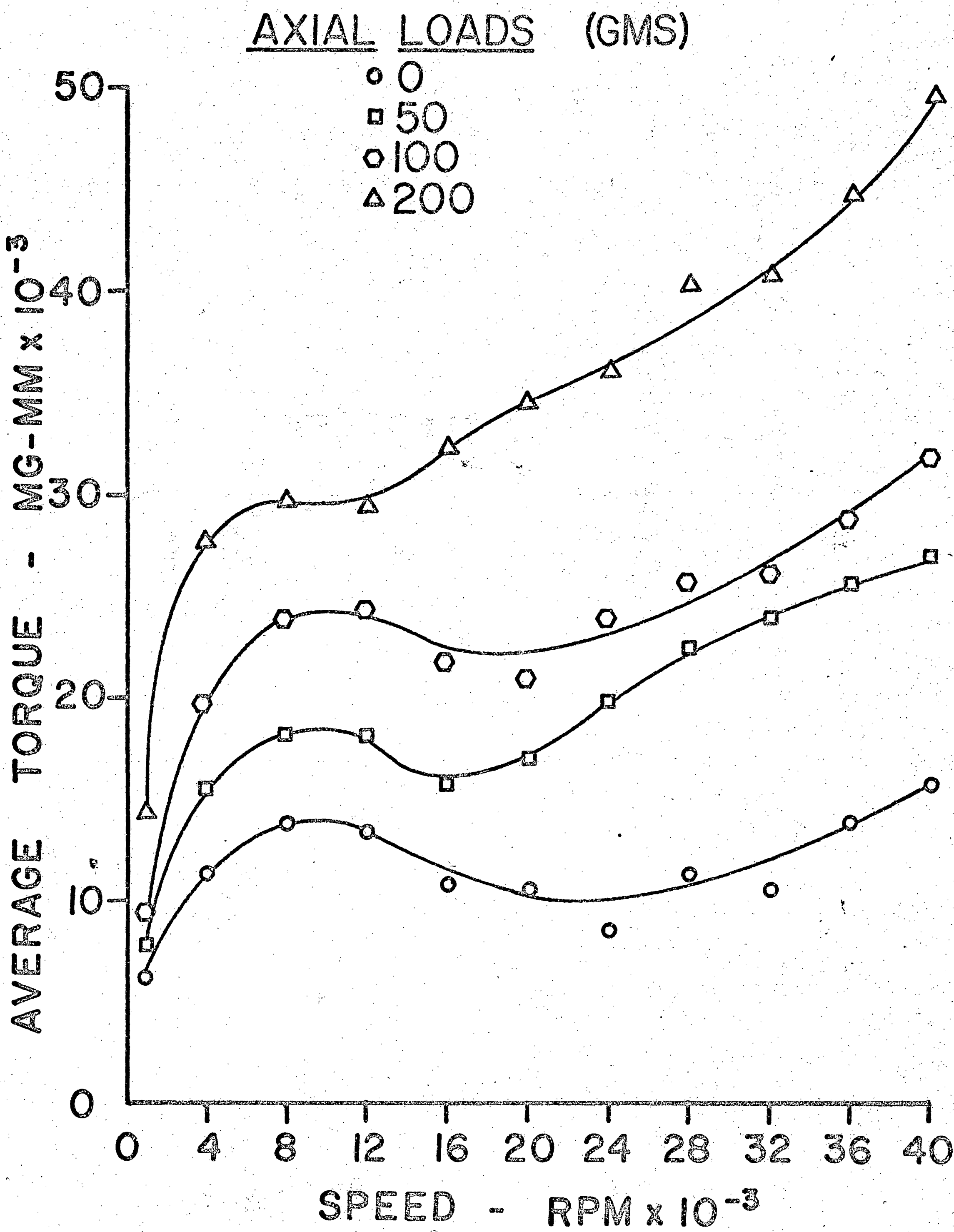

FIG. 58 1/16 GREASE PACK
50 GM. RADIAL LOAD

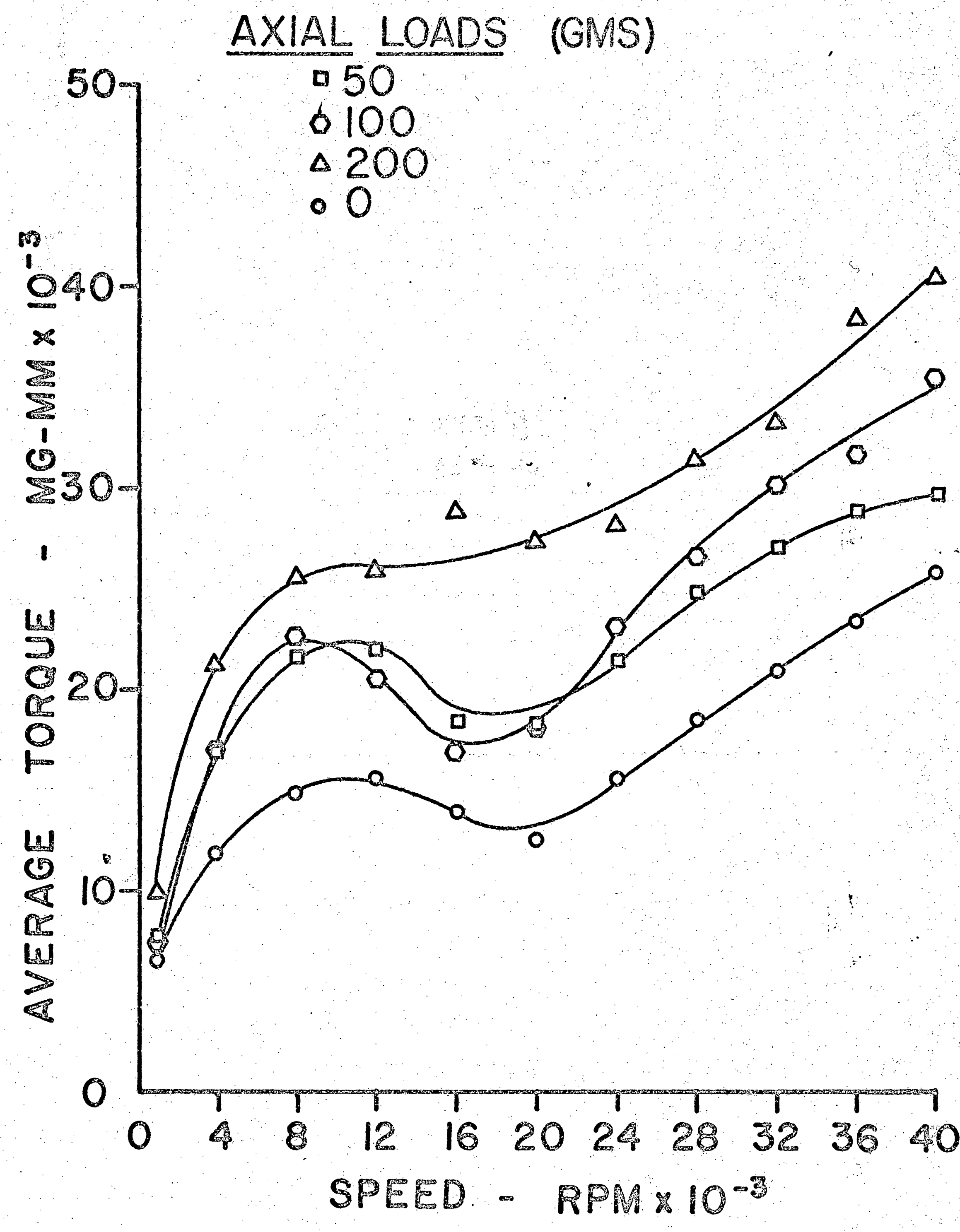

FIG. 59 1/16 GREASE PACK
100 GM. RADIAL LOAD

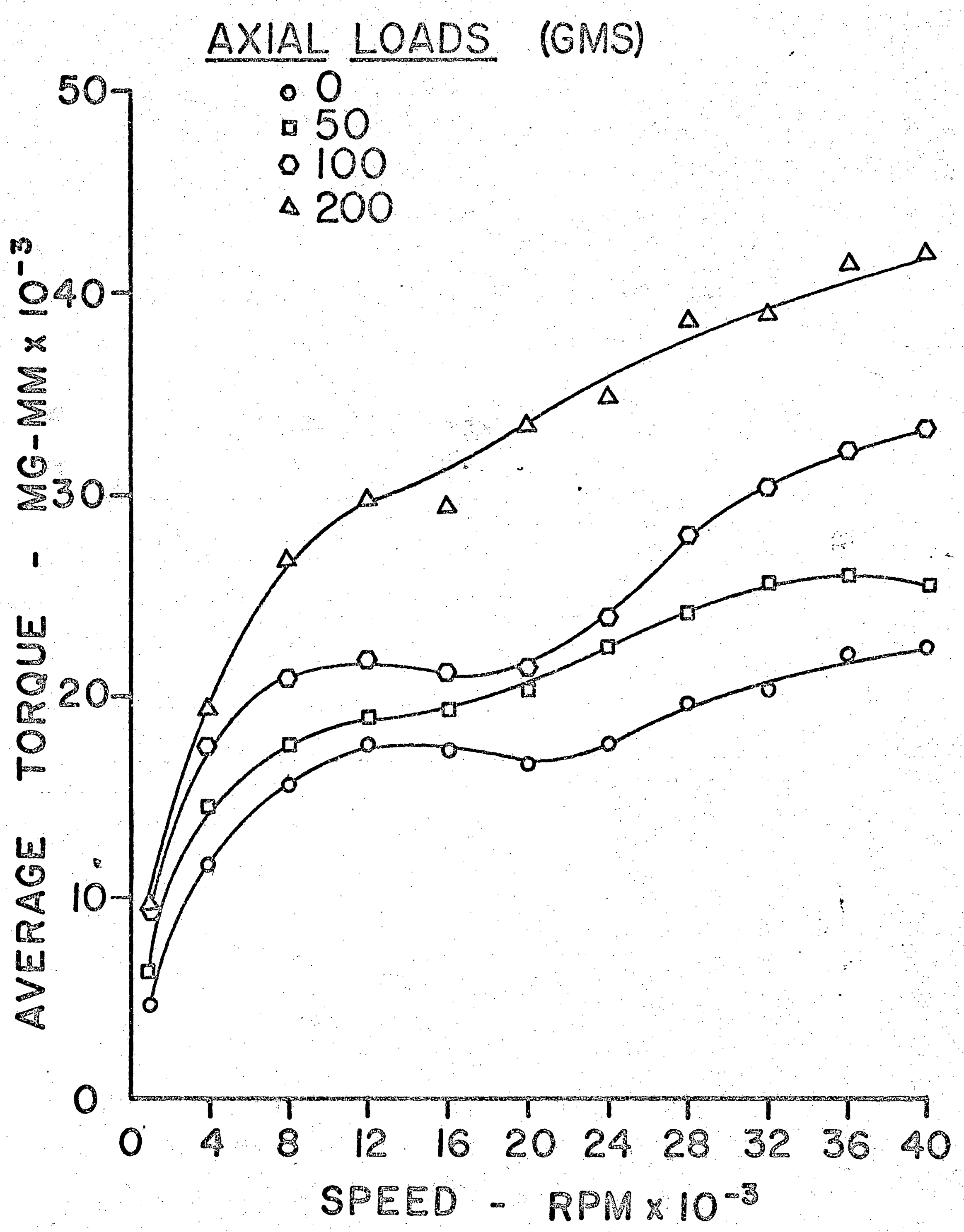

FIG. 60 1/16 GREASE PACK
200 GM. RADIAL LOAD

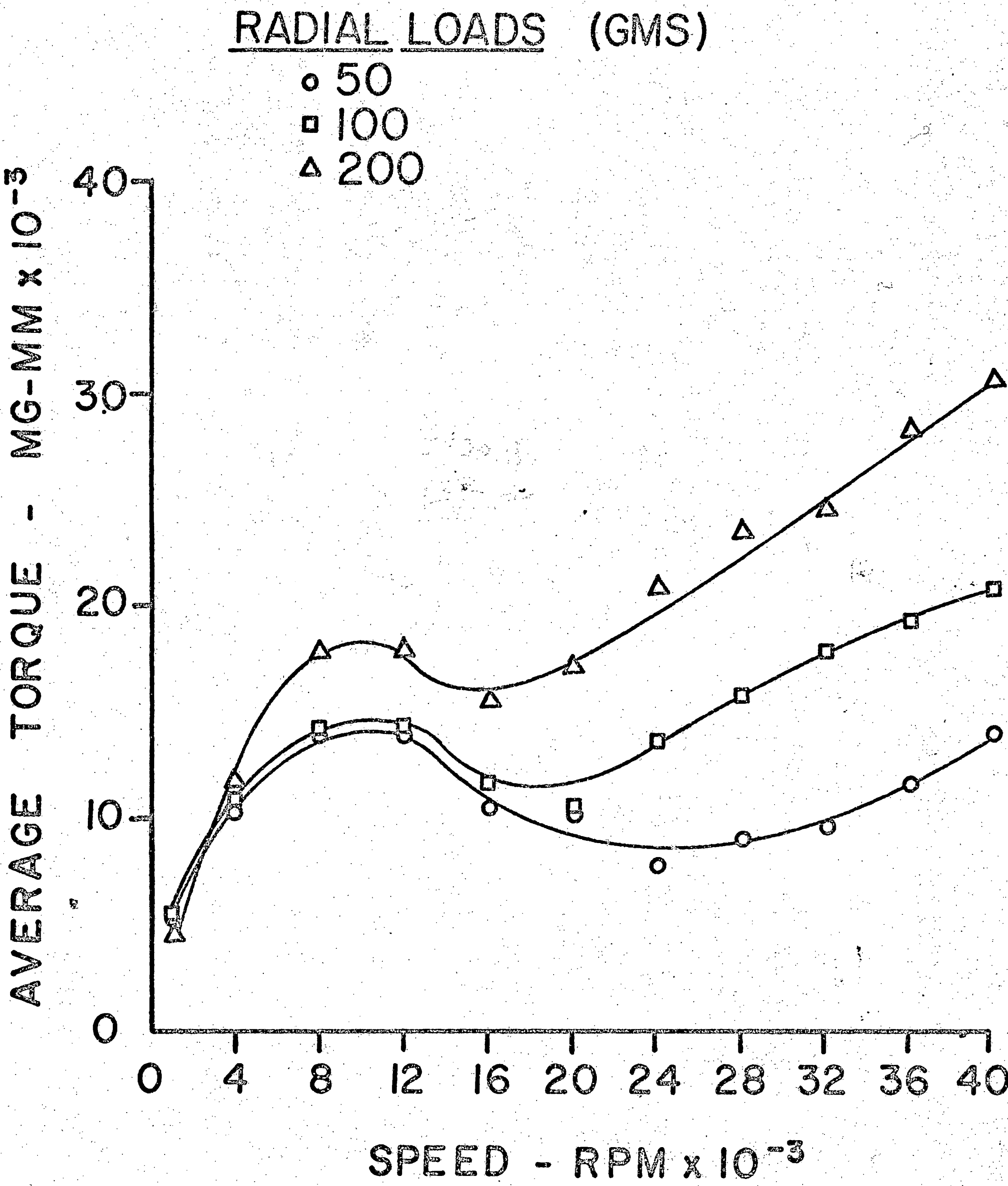

FIG. 61 1/8 GREASE PACK
0 GM. AXIAL LOAD

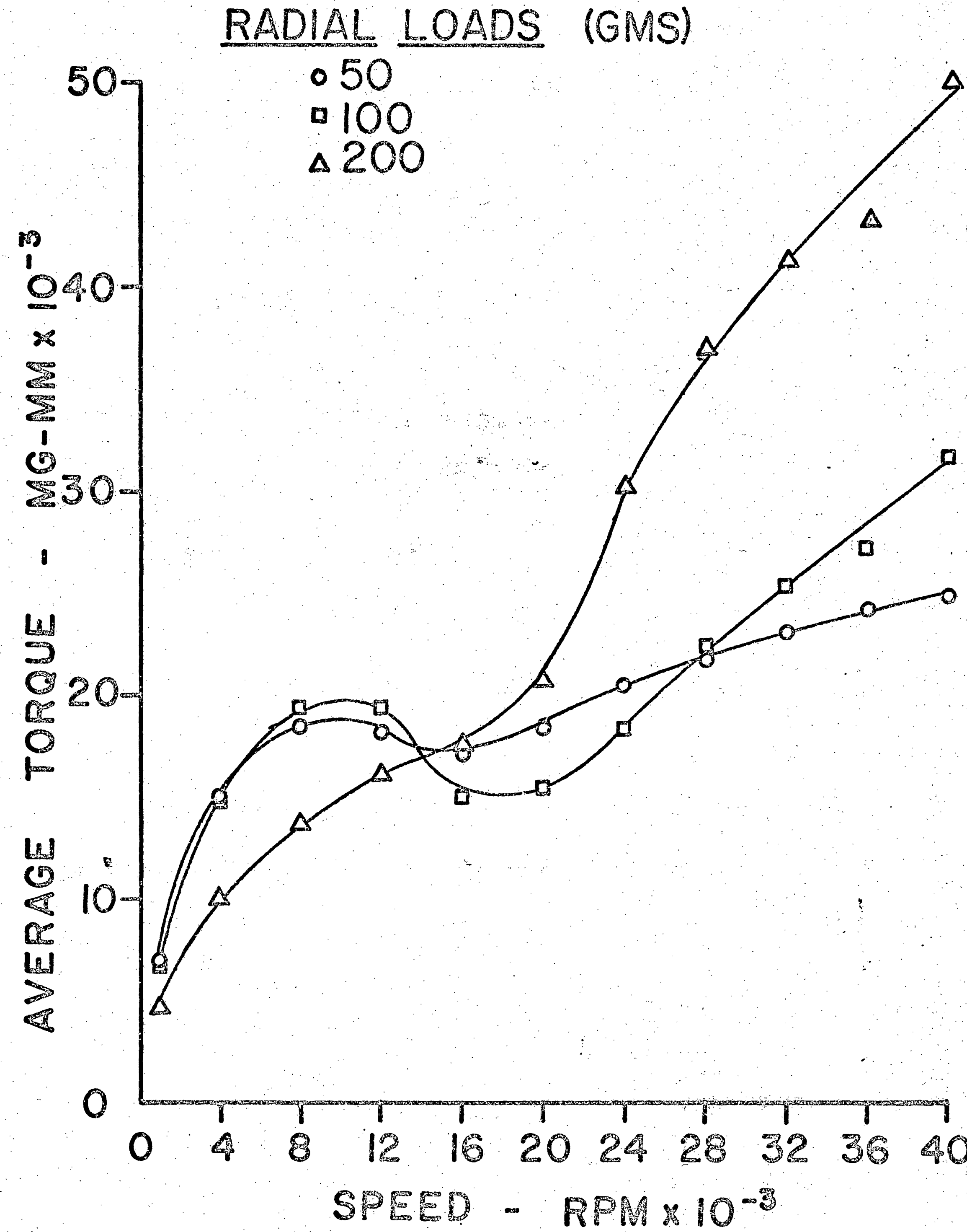

FIG. 62 1/8 GREASE PACK
50 GM. AXIAL LOAD

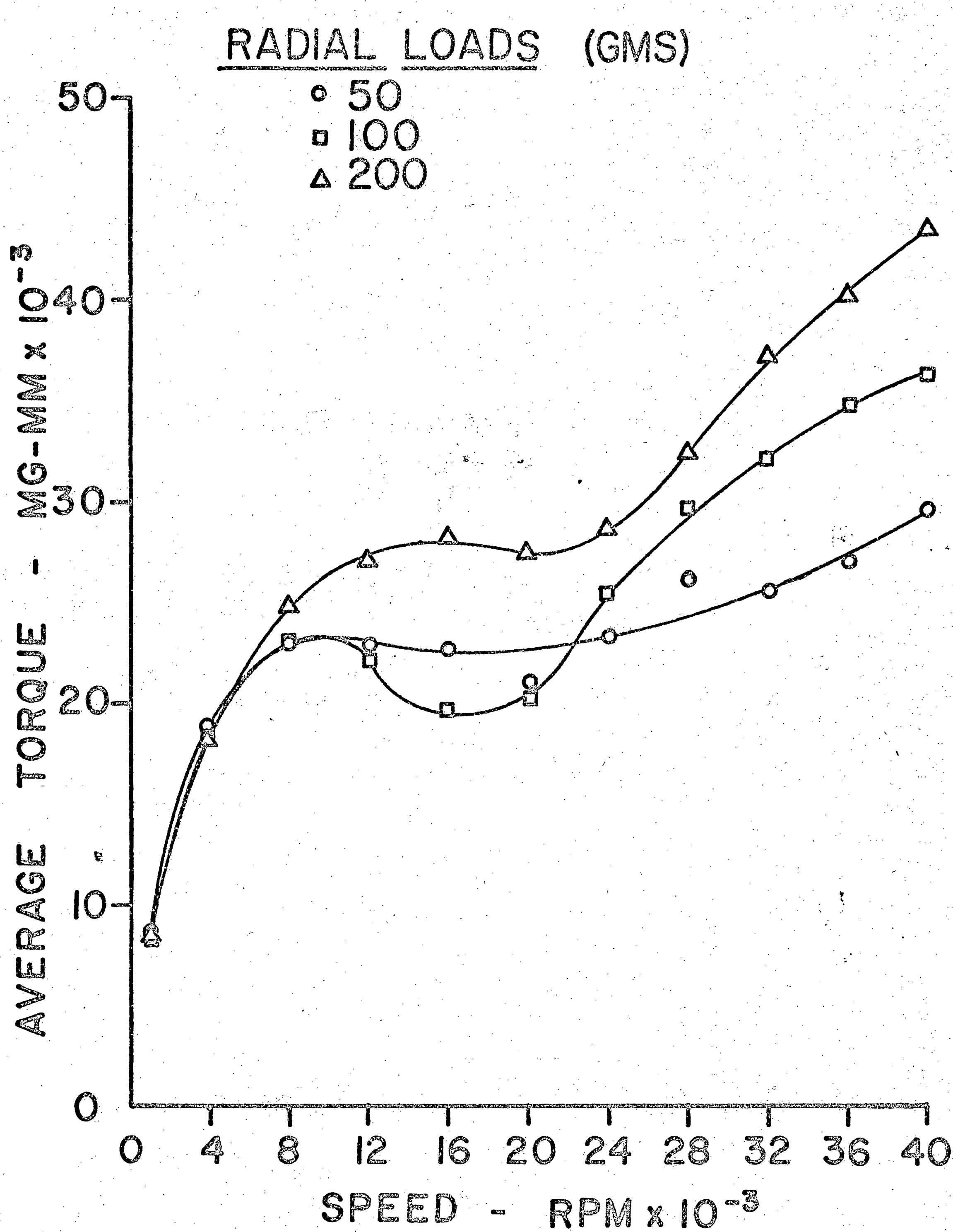

FIG. 63 1/8 GREASE PACK
100 GM. AXIAL LOAD

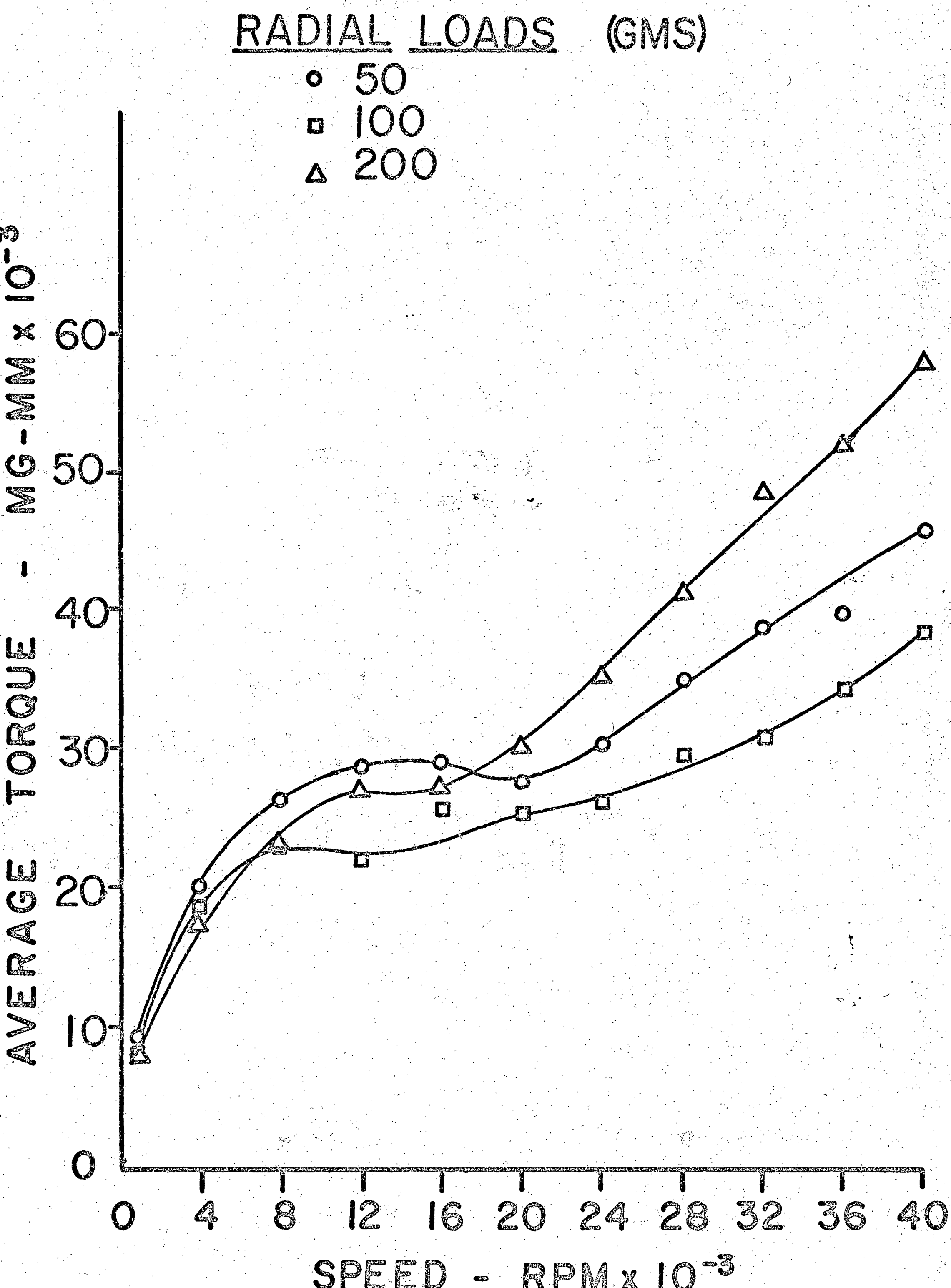

FIG. 64 1/8 GREASE PACK
200 GM. AXIAL LOAD

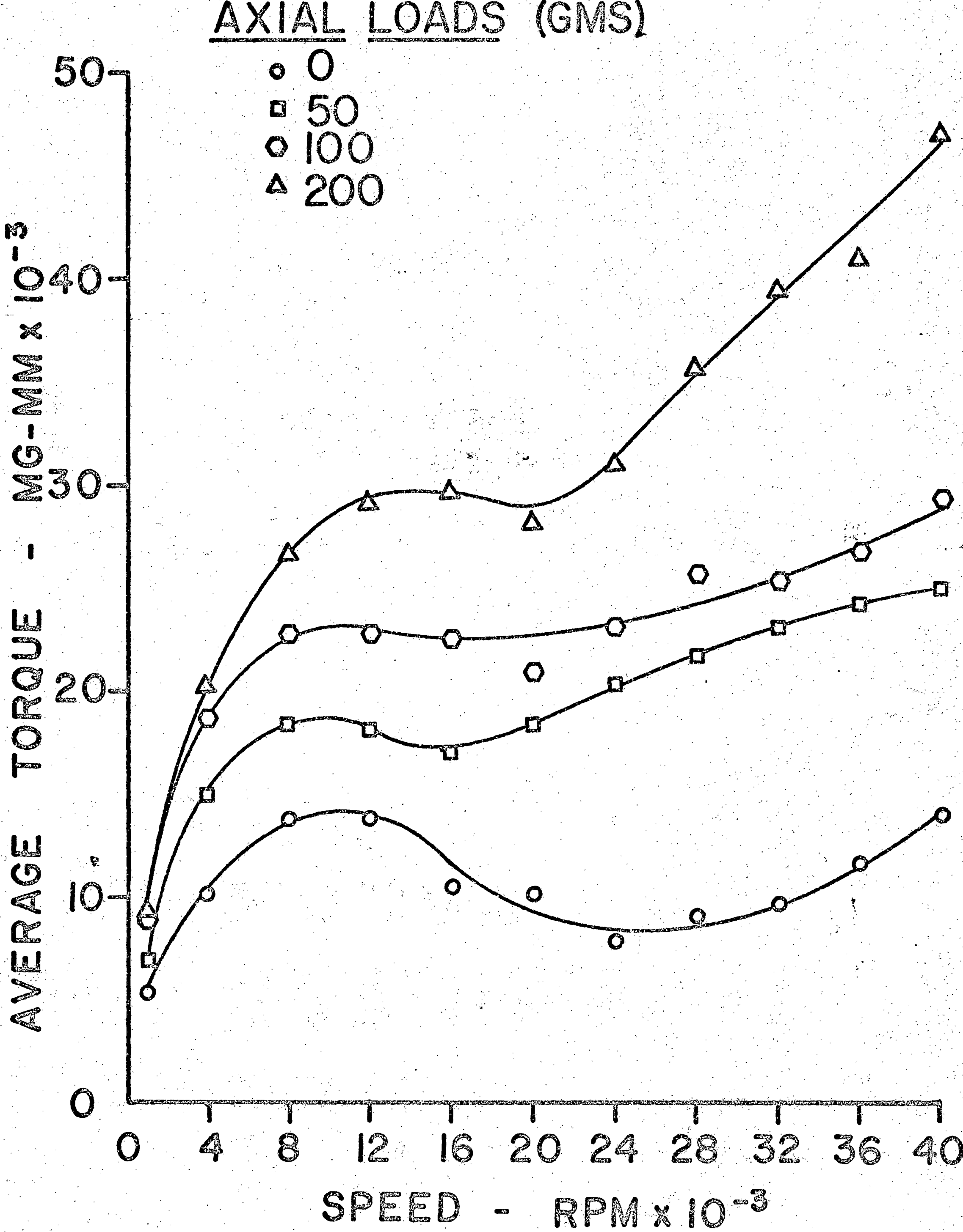

FIG. 65 1/8 GREASE PACK
50 GM. RADIAL LOAD

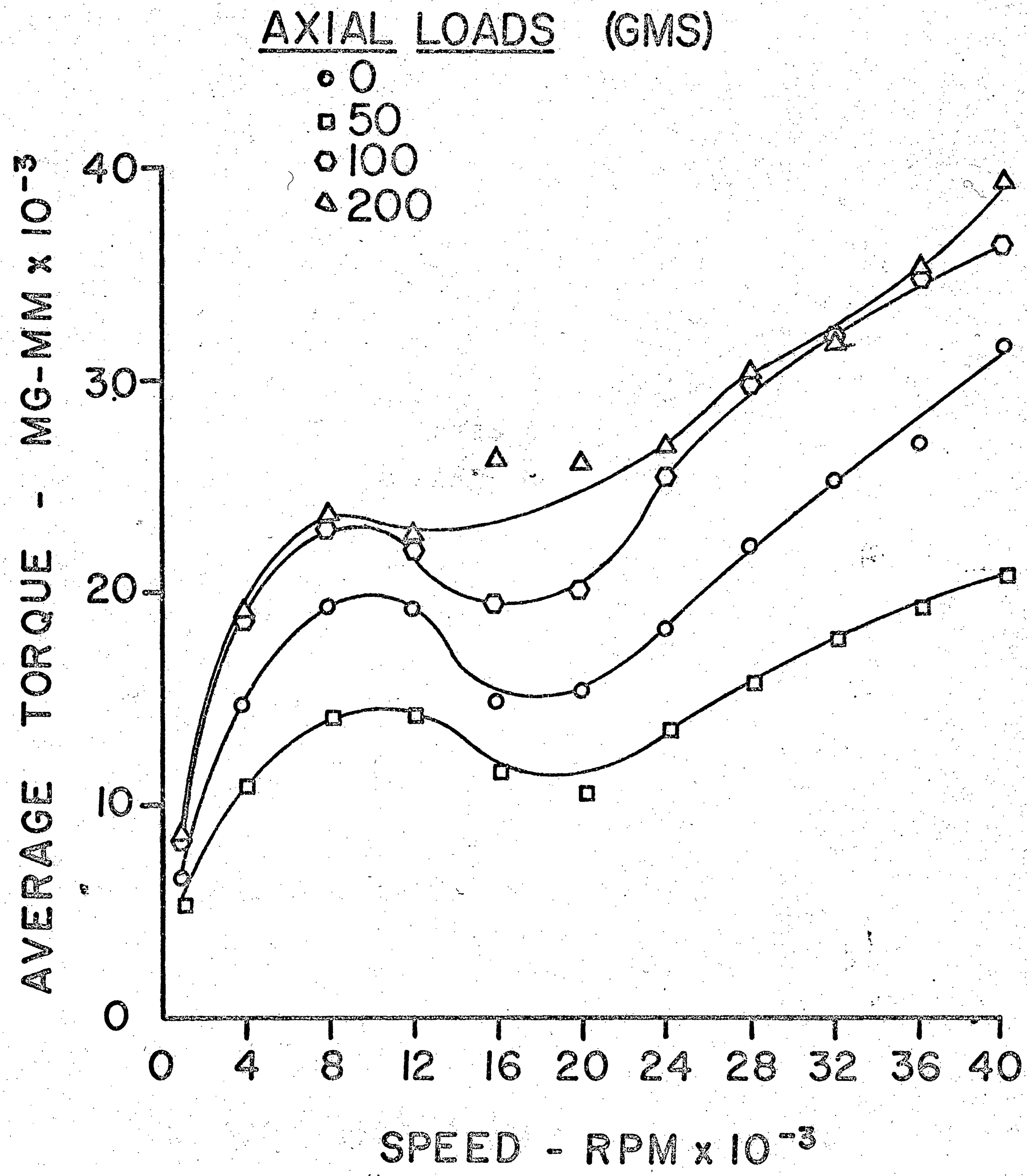

FIG. 66 1/8 GREASE PACK
100 GM. RADIAL LOAD

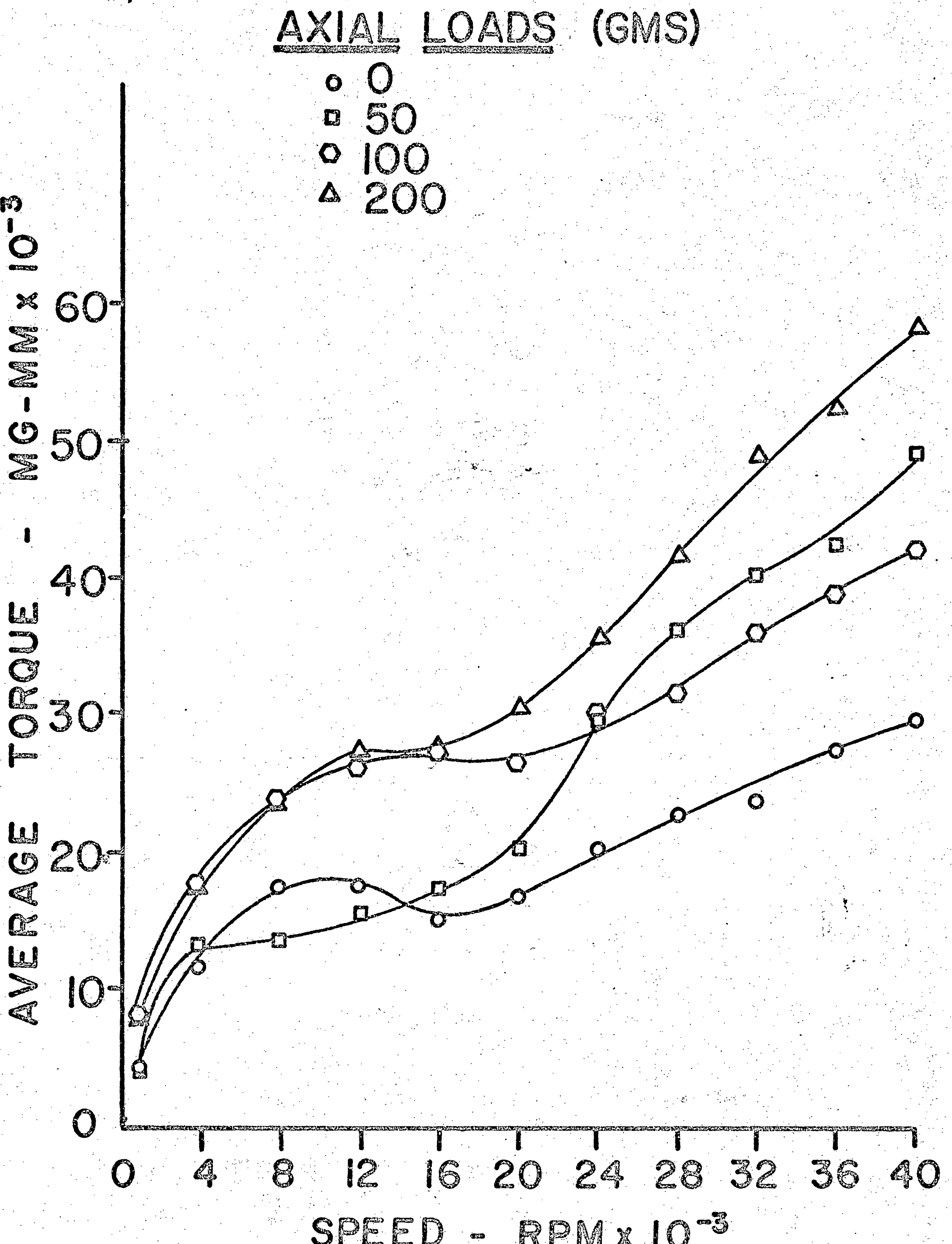

FIG. 67 1/8 GREASE PACK
200 GM. RADIAL LOAD

which was observed for oil; i.e., apparent insensitivity to changing radial load, sensitive to changing axial load. For the 1/8 grease pack, the torque changed considerably with varying axial loads which was also observed for oil and 1/16 grease pack. However, the torque may be seen to also vary significantly with changing radial load which was not observed for the other two lubricant conditions. The validity of the 1/8 grease pack torque data, however, is somewhat questionable due to poor reproducibility as evidenced by the large sample standard deviations.

Since the oil and 1/16 grease pack bearings were lubricated with approximately the same weight of lubricant, it was interesting to note that the torque for 1/16 grease pack was about 2.5 times the torque for oil for the same loading. This observation applies only to rerun tests since original runs were not plotted.

In comparing the average torques between the 1/16 and 1/8 grease pack, it was found that no significant differences existed between the two for constant radial load with varying axial loads. However, some considerable variation was observed at high speeds when the radial loads were varied.

This correlation between 1/16 and 1/8 grease pack torque data could have been expected since these comparisons were made for the rerun tests. In the original runs, the excess grease was probably pushed out of the effective lubricating areas, thereby leaving approximately the same amount of grease in the raceways for both the 1/16 and 1/8 grease pack rerun tests.

Although the objectives of this work did not include an analysis of the friction torque sources in instrument ball bearings, a limited study was initiated to determine if a statement could be made concerning the apparent insensitivity of torque to varying radial loads.

The first approach was to consider the equivalent radial loads acting on the bearings. Manufacturers of ball bearings recommend an equation of the following form for determining the equivalent load:

$$P_e = XR + YA \qquad (1)$$

Where P_e = equivalent radial load

R = applied radial load

A = applied axial load

X = radial load factor

Y = axial load factor

The values of X and Y depend upon the clearance and bearing design. These values were not specified by the manufacturer of the ball bearings used in the test program. However, a survey of another bearing manufacturer's catalog revealed that the radial load factor, X, was approximately 0.56 for instrument ball bearings.[13]

The manufacturer of the ball bearings used in the test program recommended an equation of the following form to calculate the equivalent load.

$$P_e = RF \qquad (2)$$

$$\text{Where} \quad F = X+Y(A/R)$$

Values of F were given for various ratios of A/R. By assuming X equal to 0.56, values of Y were determined from the given values of F. It was found that Y varied from approximately 1.0 to values greater than 2.0 from high to low ratios of A/R. Although these values for X and Y may not have been exactly correct, they did show that an increase in axial load had a much higher effect on the equivalent load than did a corresponding increase in radial load.

The equivalent radial loads, as determined from equation (2), are shown in the following table.

TABLE 37 EQUIVALENT RADIAL LOAD

	Equivalent Radial Load, P_e (lbs.)			
Axial / Radial	0 gm.	50 gm.	100 gm.	200 gm.
50 gm.	0.11023	0.1951	0.3053	0.5247
100 gm.	0.22046	0.2822	0.3902	0.6107
200 gm.	0.44092	0.4806	0.5644	0.7804

A comparison of torque and equivalent radial load indicated that the torque increased in proportion to the equivalent load when axial loads were varied. However, this trend was not observed for varying radial loads. As noted previously, the torque changed very little with increased radial load. Due to the closeness of the curves and the random nature in which the curves varied, no correlation between the equivalent radial load and torque was found.

If the film of lubricant between the balls and raceways was not of sufficient thickness to prevent any asperities on the balls from contacting asperities on the raceways, then an increase in either the axial or radial load should cause even more asperities to contact one another, thus increasing the torque. For these conditions, known as boundary lubrication, changes in axial loads corresponded well with the above hypothesis whereas changes in radial loads did not.

As another possible means of explaining the insensitivity of torque to increases in radial load, the mode of lubrication was investigated. It has been known for quite some time that hydrodynamic lubrication was possible in ball bearings. In ball bearing literature, this type of lubrication is known as "elastohydrodynamic" lubrication. Under the high lubricant pressures present between contacting rolling bodies, the two surfaces are known to deform elastically; hense, the term "elasto." Theoretical and experimental results have verified that the film thickness required for elastohydrodynamic was almost insensitive to load.[14]

In order to determine if the bearings tested were operated under conditions of elastohydrodynamic lubrication, empirical methods developed by Harris[14] were used. This analysis was conducted only for oil lubricated bearings due to the uncertainty of viscosity for grease to be used in the equations. This analysis indicated that at approximately 1,000 rpm elastohydrodynamic lubrication should have been obtained. This corresponded to the lowest speed at which data were taken. Therefore, for factory oil lubricated bearings, elastohydrodynamic lubrication should have existed for the entire speed range at which data were taken.

Palmgren[15] had conducted studies to determine the friction torque when elastohydrodynamic lubrication exists in ball bearings. Based on these studies, Palmgren determined empirically that the friction torque was a function only of bearing size, lubricant, and speed; i.e., a viscous torque. These empirical equations for friction torque were determined for large bearings operating at relatively slow speeds. Although no load term was present in these equations, the value of torque, as determined by these equations, are at best approximate. Because these empirical methods are not sufficiently developed at this time, considerable investigation is warranted to determine the dependence of load on the torque.

Although the discussion thus far has dealt with data observed during rerun tests, several interesting observations of torque characteristics were noted on the original runs. In the factory oil original runs, the torque was observed to increase rather rapidly and then decreased in the 1,000 to 16,000 rpm speed range. This characteristic "hump" is shown by the A to C region of the torque curve in Figure 68. One possible explanation for this trend was thought to be the uneven distribution of oil in the bearing after lubrication. With the oil not being distributed evenly throughout the bearing, some unlubricated surfaces are able to contact one another as the bearing is rotated resulting in higher torques. As the bearing continues to rotate at increasing speeds, the oil was redistributed by the rolling action and centrifugal effects resulting in the development of a thin film of lubricant between the contacting surfaces. With less metal to metal contact, the torque decreased as shown by the portion of the curve from B to C.

For grease lubricated bearings, the original run torque curves either leveled out or decreased considerably at high speeds when the speed range was traversed from 0 to 40,000 rpm. In general, the bearings lubricated with 1/16 grease pack leveled out, whereas the 1/8 grease packed bearings decreased sharply at high speeds. It was found, however, that when the bearings were run up to speeds of 40,000 rpm very rapidly and torque readings then taken at decreasing

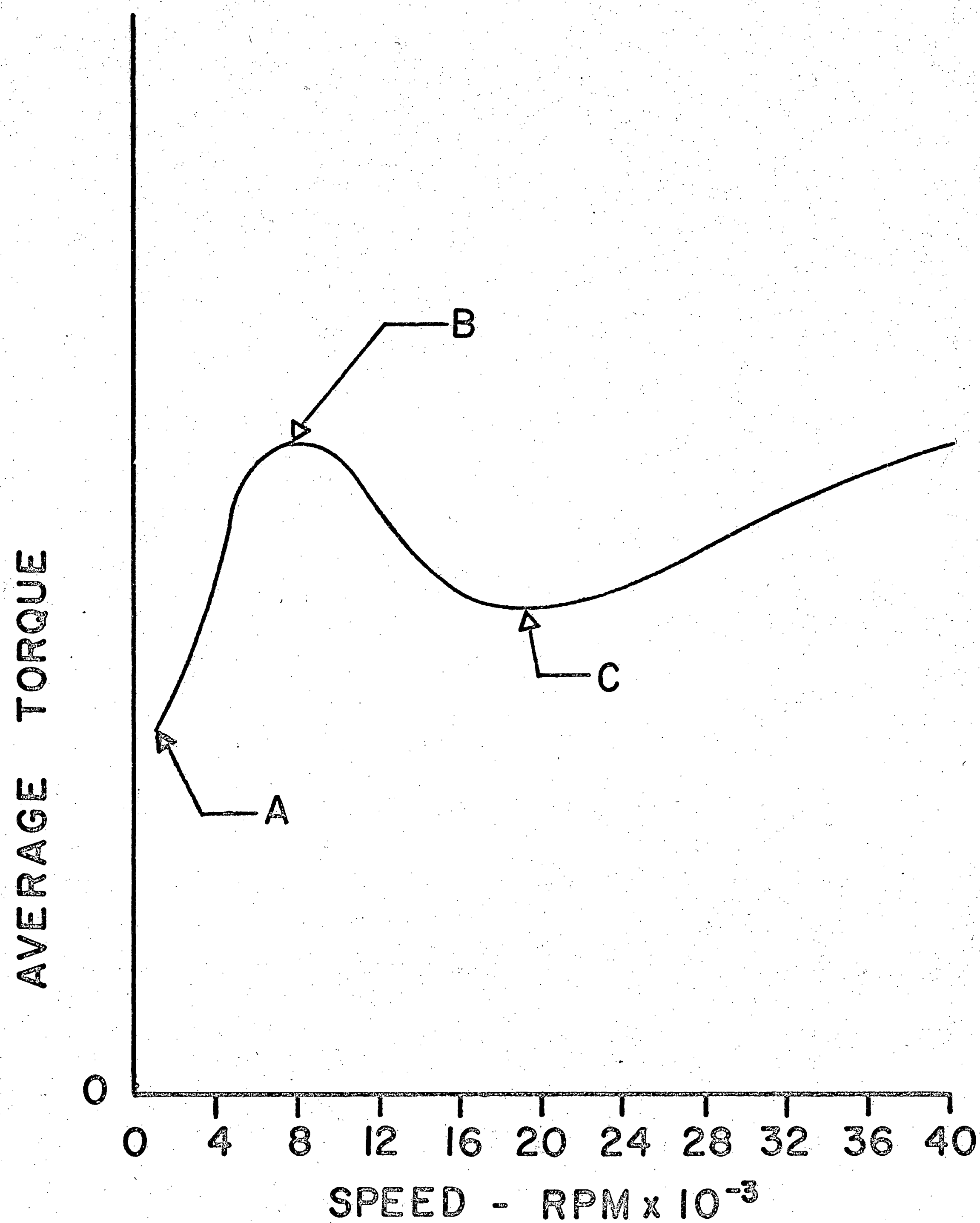

FIG. 68 TYPICAL ORIGINAL RUN TORQUE CURVE FOR OIL

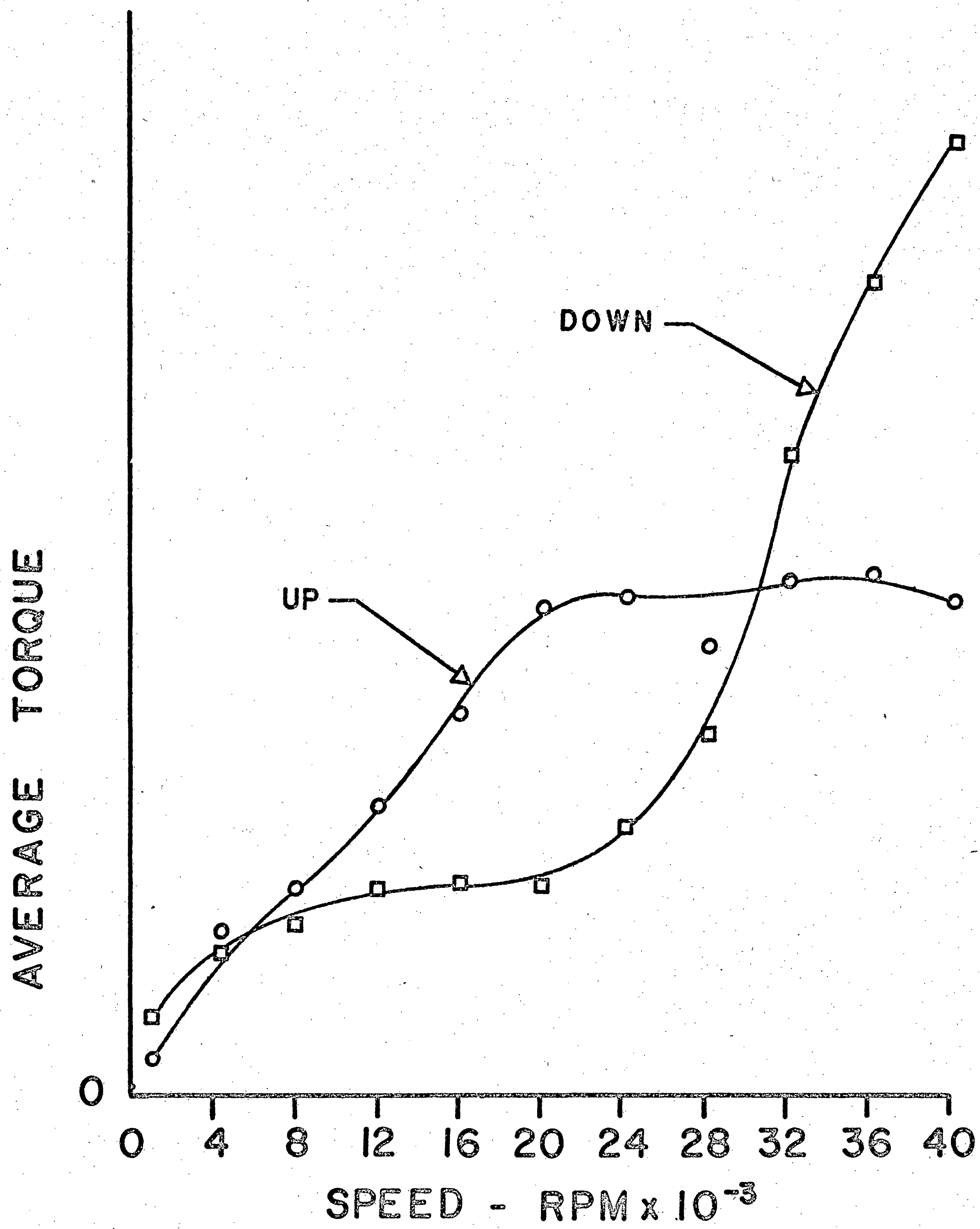

FIG. 69 EFFECT OF INCREASING AND DECREASING SPEED ON ORIGINAL RUNS - GREASE, 1/16 PACK

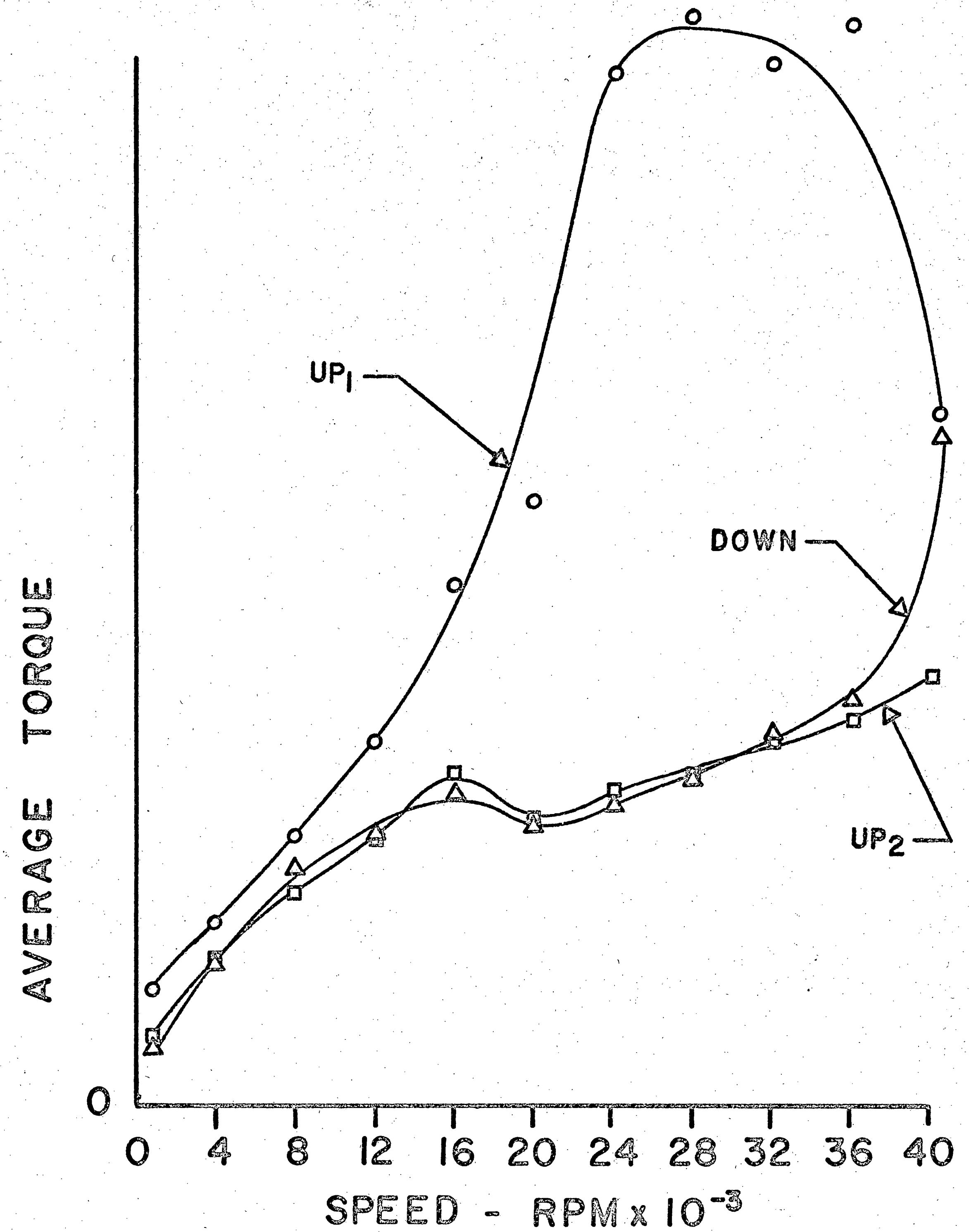

FIG. 70 EFFECT ON TORQUE OF INCREASING AND DECREASING SPEEDS AT CONTINUOUS OPERATION-GREASE(1/8)

speeds, the torques were found to vary considerably from torques taken at increasing speeds. Figure 69 shows the typical torque variations for 1/16 grease pack when the bearing was rotated from 0 to 40,000 rpm (up) and 40,000 to 0 rpm (down). Bearings lubricated with 1/8 grease pack followed the same trend except at high speeds the torque decreased sharply. Both the up and down curves were determined for original runs of the same bearing. Figure 70 illustrates the effect of running to 40,000 rpm (up_1) back to 0 rpm and back to 40,000 rpm (up_2) at continuous operation for 1/8 grease pack. It was suspected that the decrease in torque at high speeds for grease lubricated bearings was simply due to moving the excess grease from the raceways. However, a heating effect, resulting in a lower viscosity of the grease, could have also caused a reduction in torque.

It was possible to make direct comparisons with the work of Mabie[7] and Clarke[8] on the same bearings under several conditions of load and lubricant. These comparative plots are shown in Figures 71 through 82.

For factory oil, shown in Figures 71 through 74, it may be seen that the data agreed quite well. Only when axial loads were applied to the test bearings do

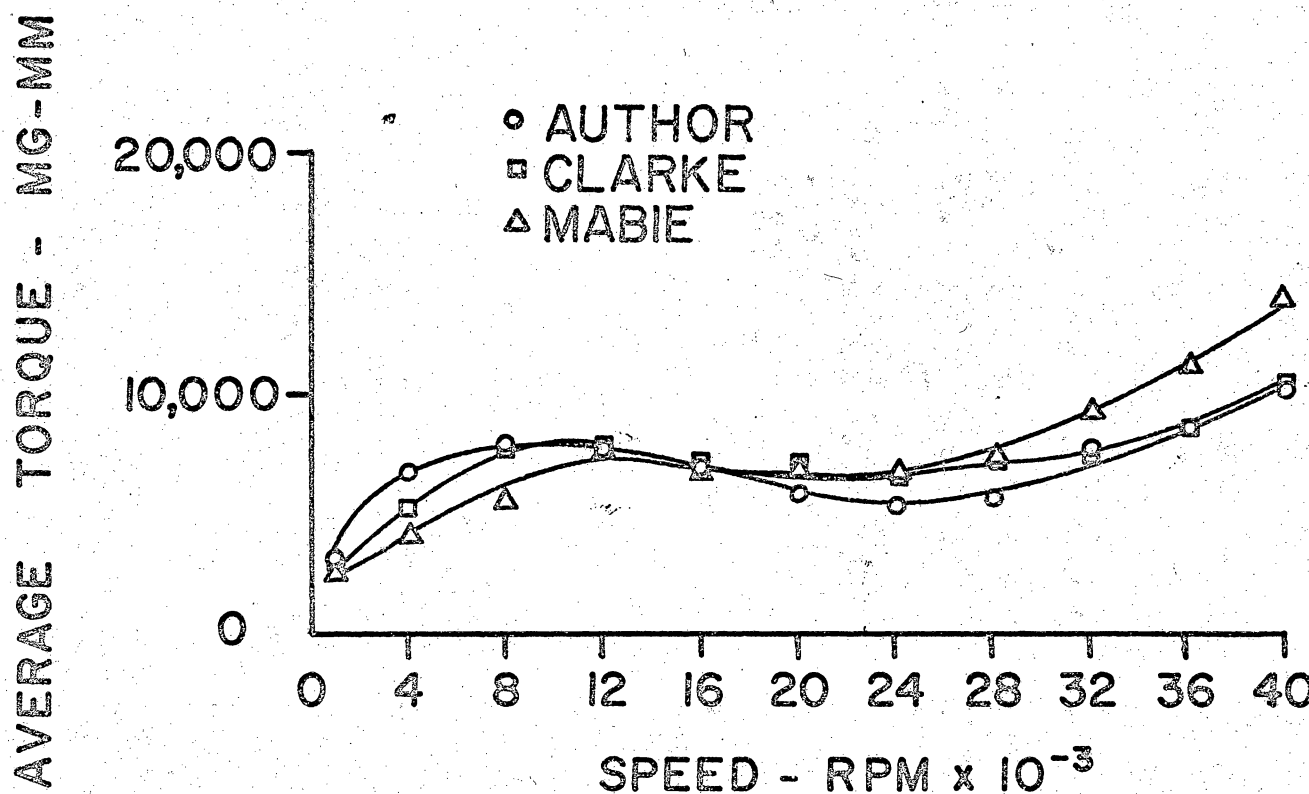

FIG. 71 COMPARISON OF OIL, ORIGINAL RUN, 50 GM. RADIAL, 0 GM. AXIAL

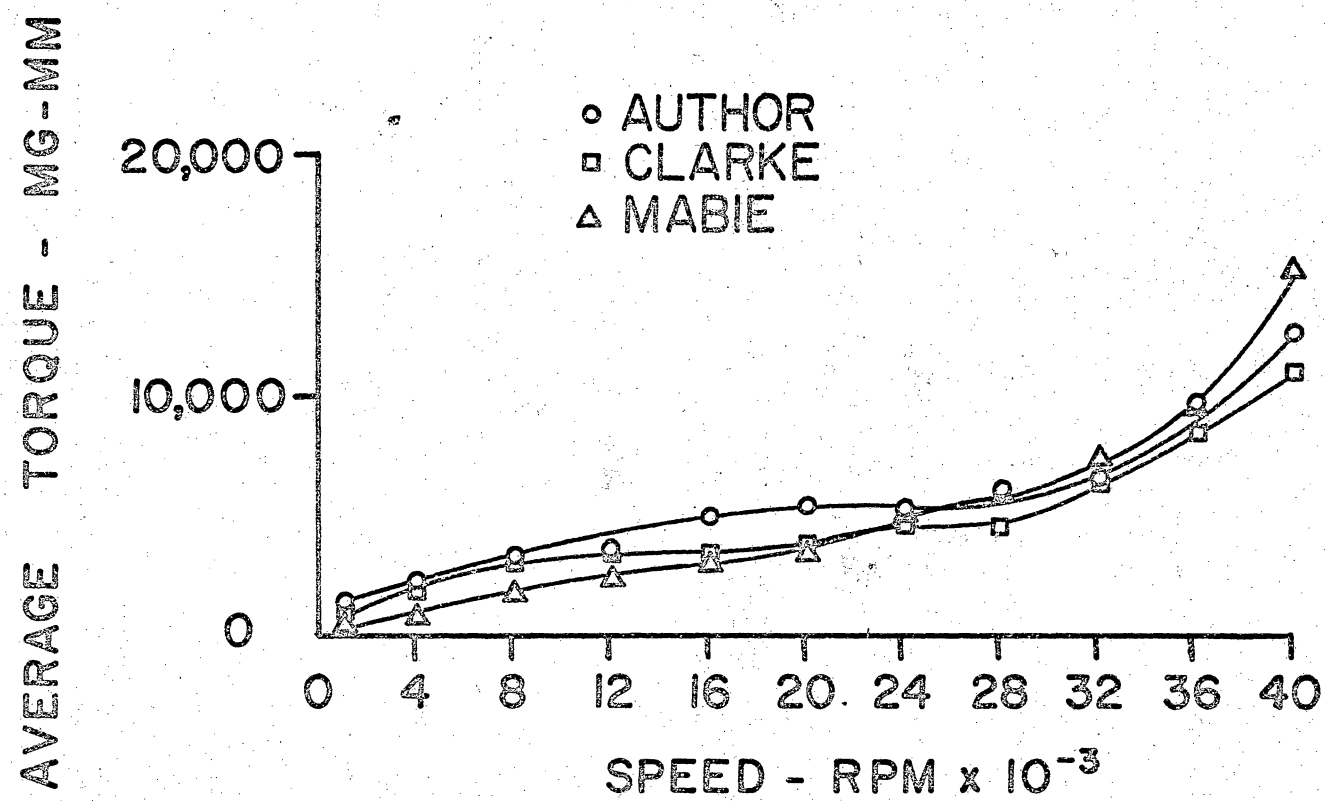

FIG. 72 COMPARISON OF OIL, RERUN, 50 GM. RADIAL, 0 GM. AXIAL

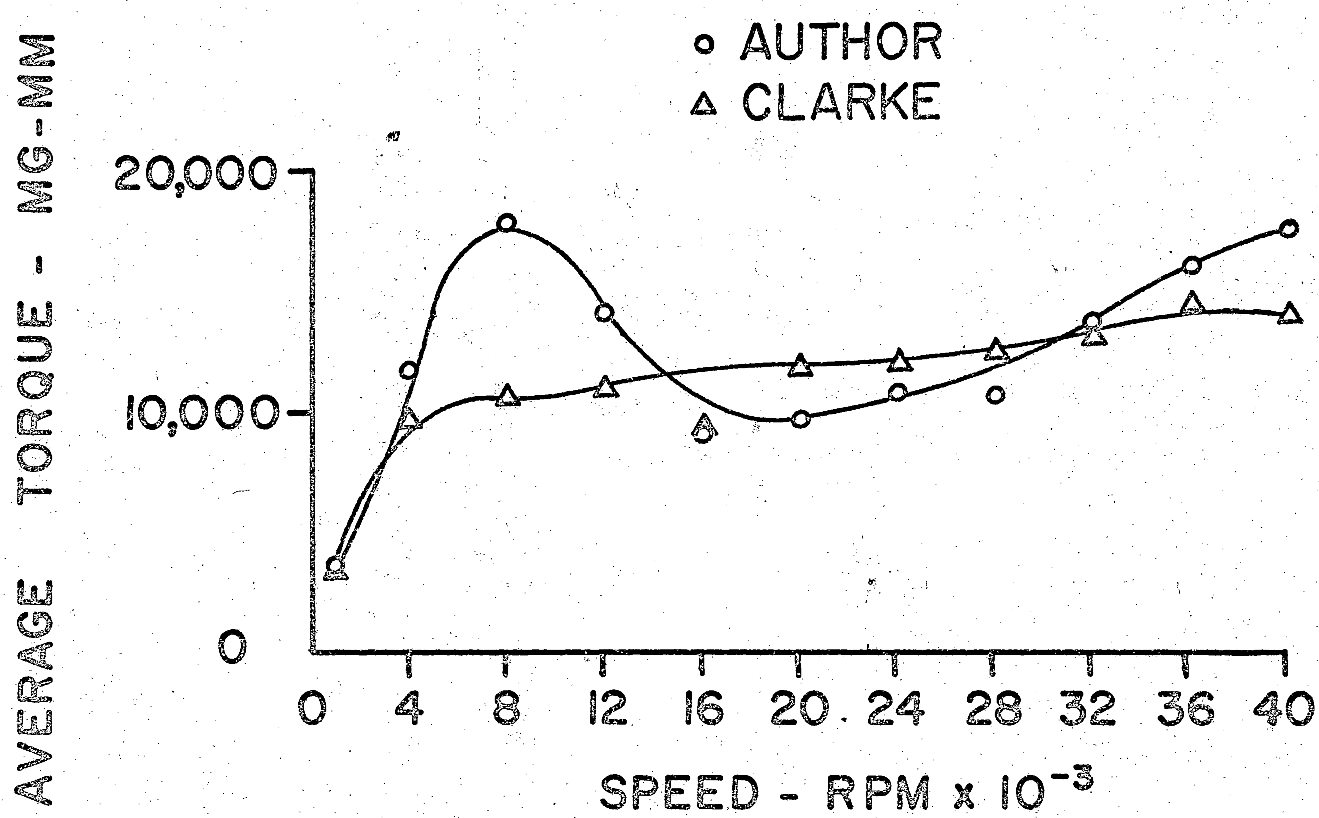

FIG. 73 COMPARISON OF OIL, ORIGINAL RUN, 50 GM. RADIAL, 50 GM. AXIAL

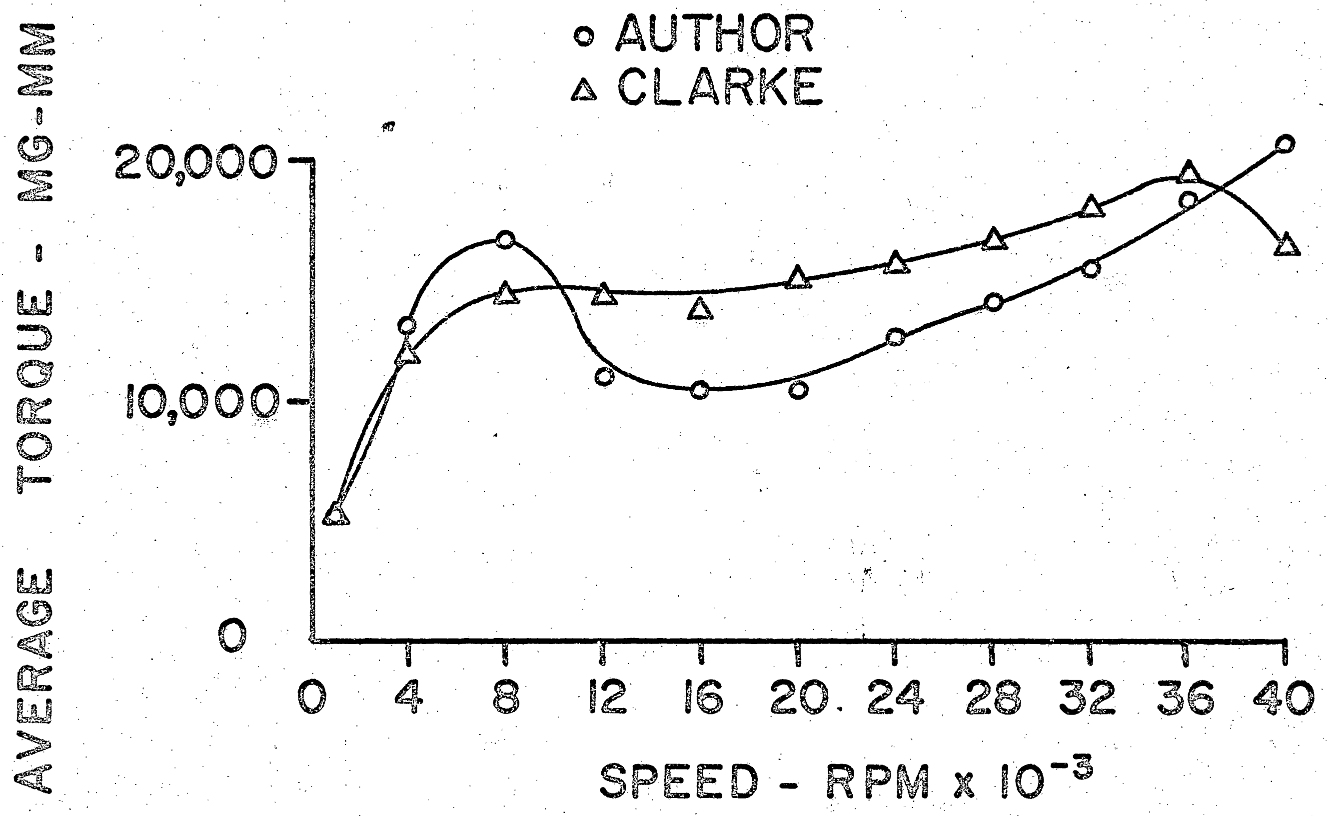

FIG. 74 COMPARISON OF OIL, ORIGINAL RUN, 50 GM. RADIAL, 100 GM. AXIAL

some deviations appear in the author's and Clarke's data. Since these deviations in torque occurred in the original runs, it was assumed that the major contributor to these deviations was due to how well the oil was distributed in the bearing at the beginning of the tests.

The torque for bearings lubricated with grease showed considerably more deviation then did bearings lubricated with oil. For 1/16 grease pack, Figures 75 through 78, the author's data were in agreement with Mabie's data at low speeds and with Clarke's at high speeds. For bearings lubricated with 1/8 grease pack, Figures 79 through 82, no general agreement was reached among the three investigators. The author observed during testing, however, that the torque particularly for 1/8 grease pack, was highly dependent upon the method of packing the grease in the bearing. Thus, the application of grease in the effective lubricating areas of the bearing could vary with individuals. This could account for the discrepancy in the torque data for the 1/8 grease packed bearings.

Although there are several points of disagreement in the data of the author, Mabie, and Clarke,

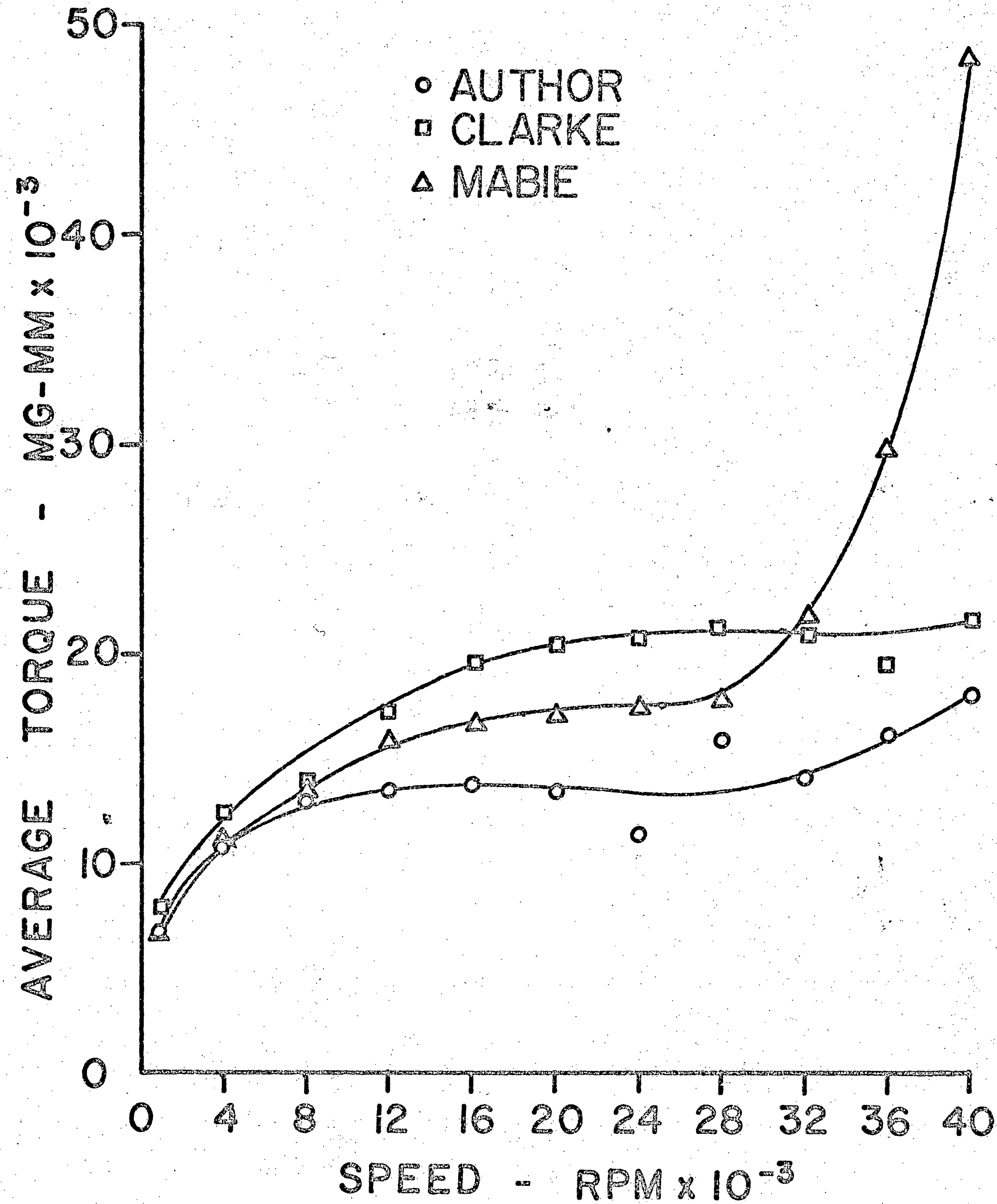

FIG. 75 COMPARISON OF 1/16 PACK, ORIGINAL RUN, 50 GM. RADIAL, 0 GM. AXIAL

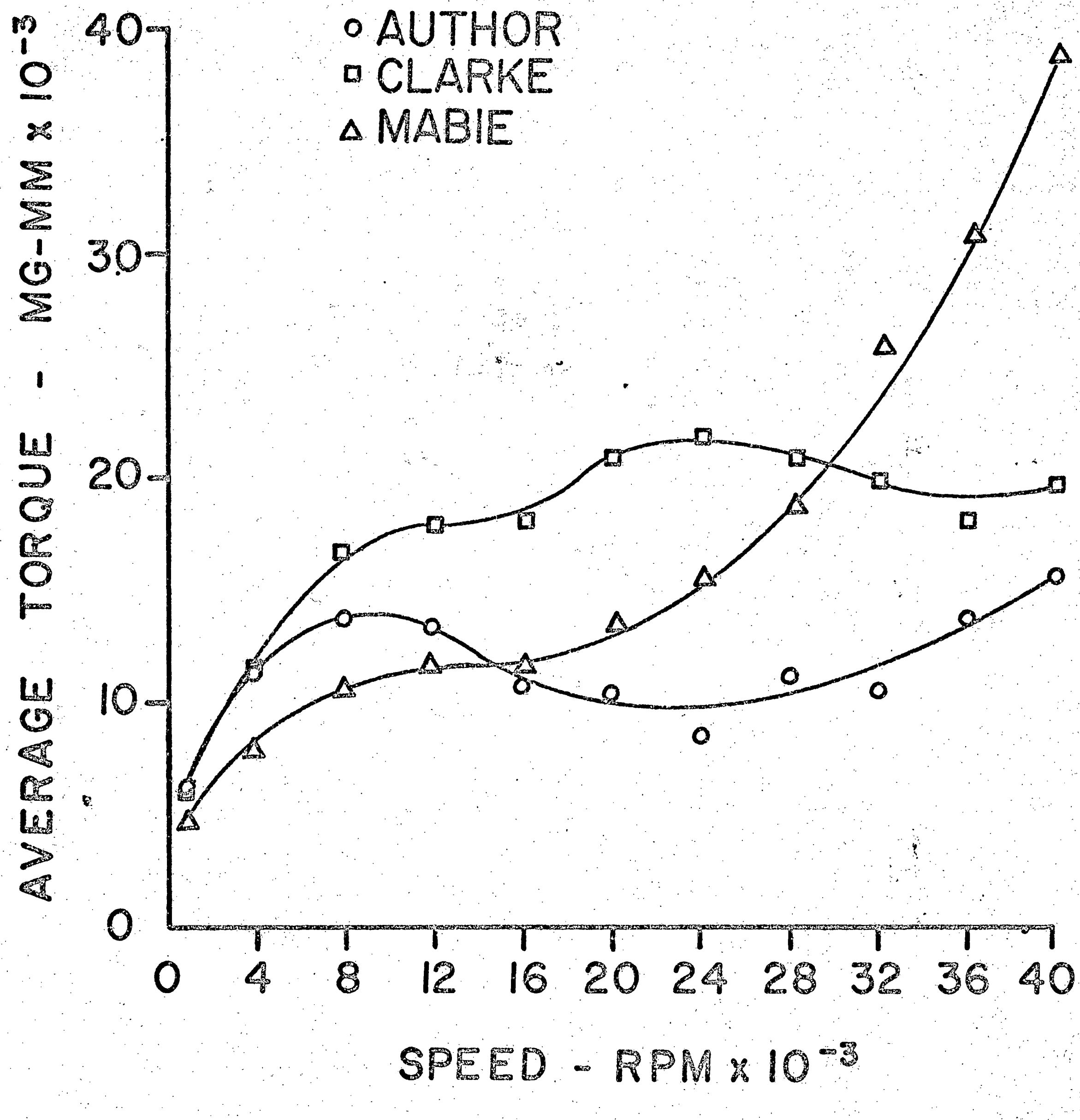

FIG. 76 COMPARISON OF 1/16 PACK, RERUN, 50 GM. RADIAL, 0 GM. AXIAL

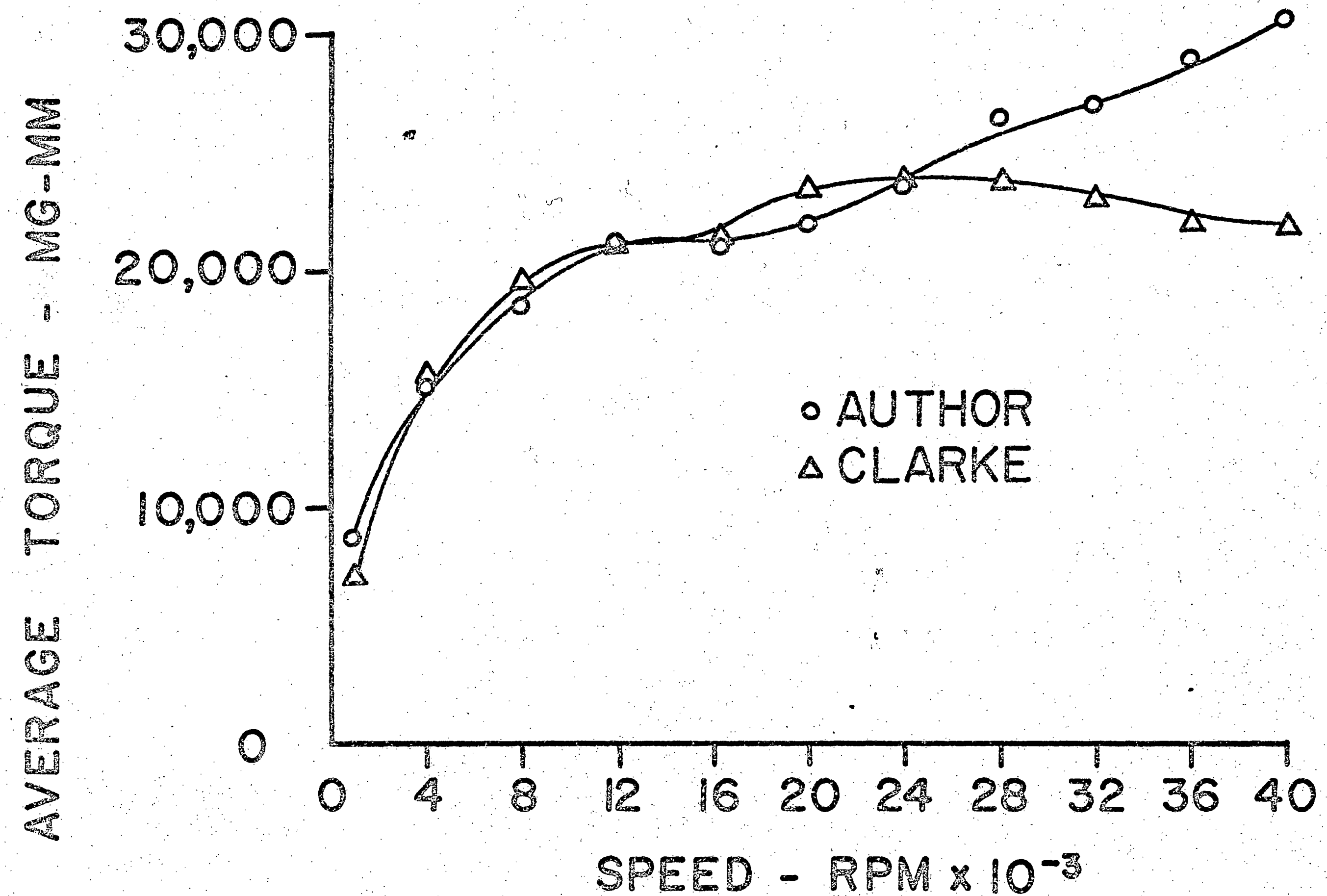

FIG. 77 COMPARISON OF 1/16 PACK, ORIGINAL RUN, 50 GM. RADIAL, 50 GM. AXIAL

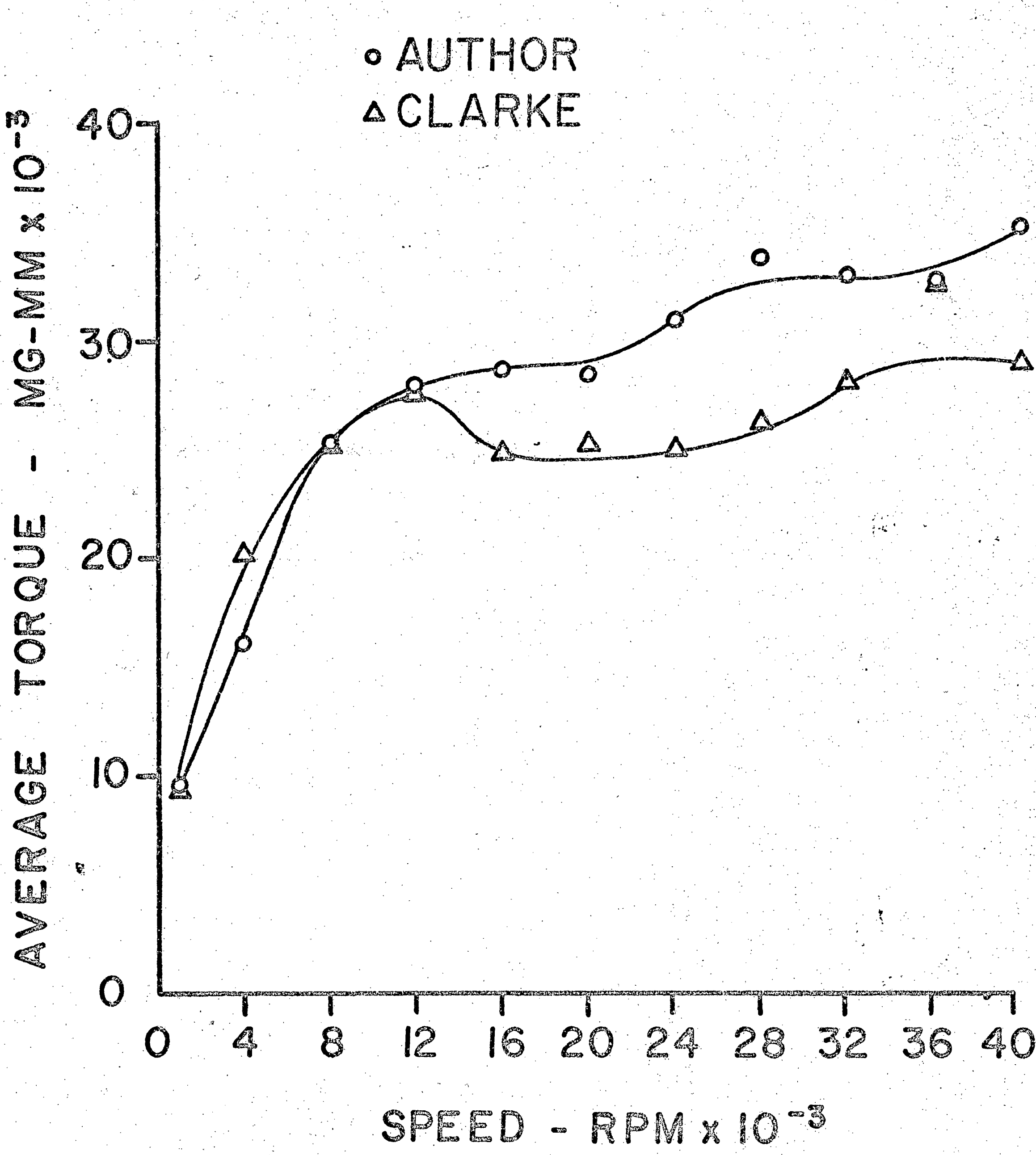

FIG. 78 COMPARISON OF 1/16 PACK ORIGINAL RUN, 50 GM. RADIAL, 50 GM. AXIAL

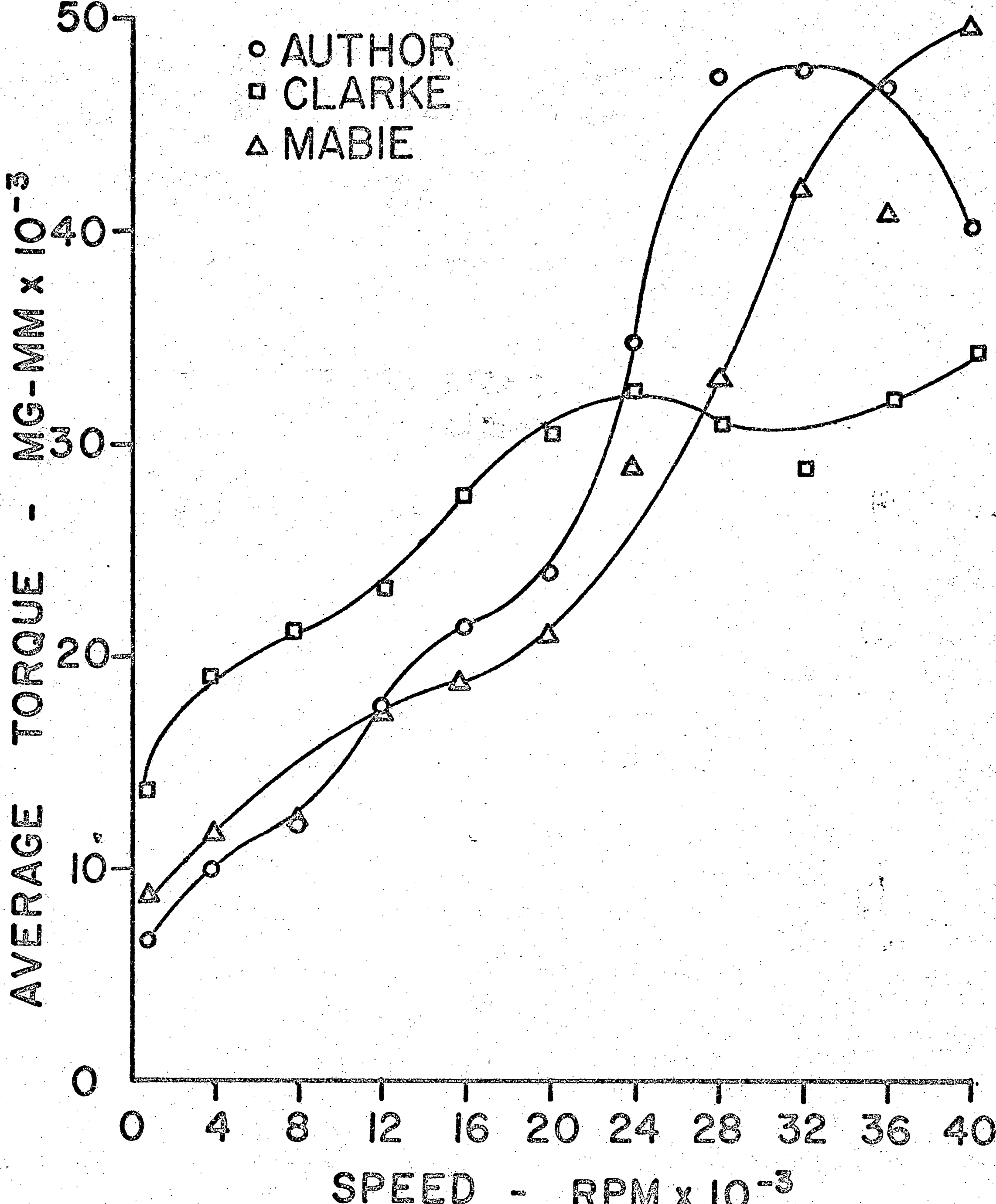

FIG. 79 COMPARISON OF 1/8 PACK, ORIGINAL RUN, 50 GM. RADIAL, 0 GM. AXIAL

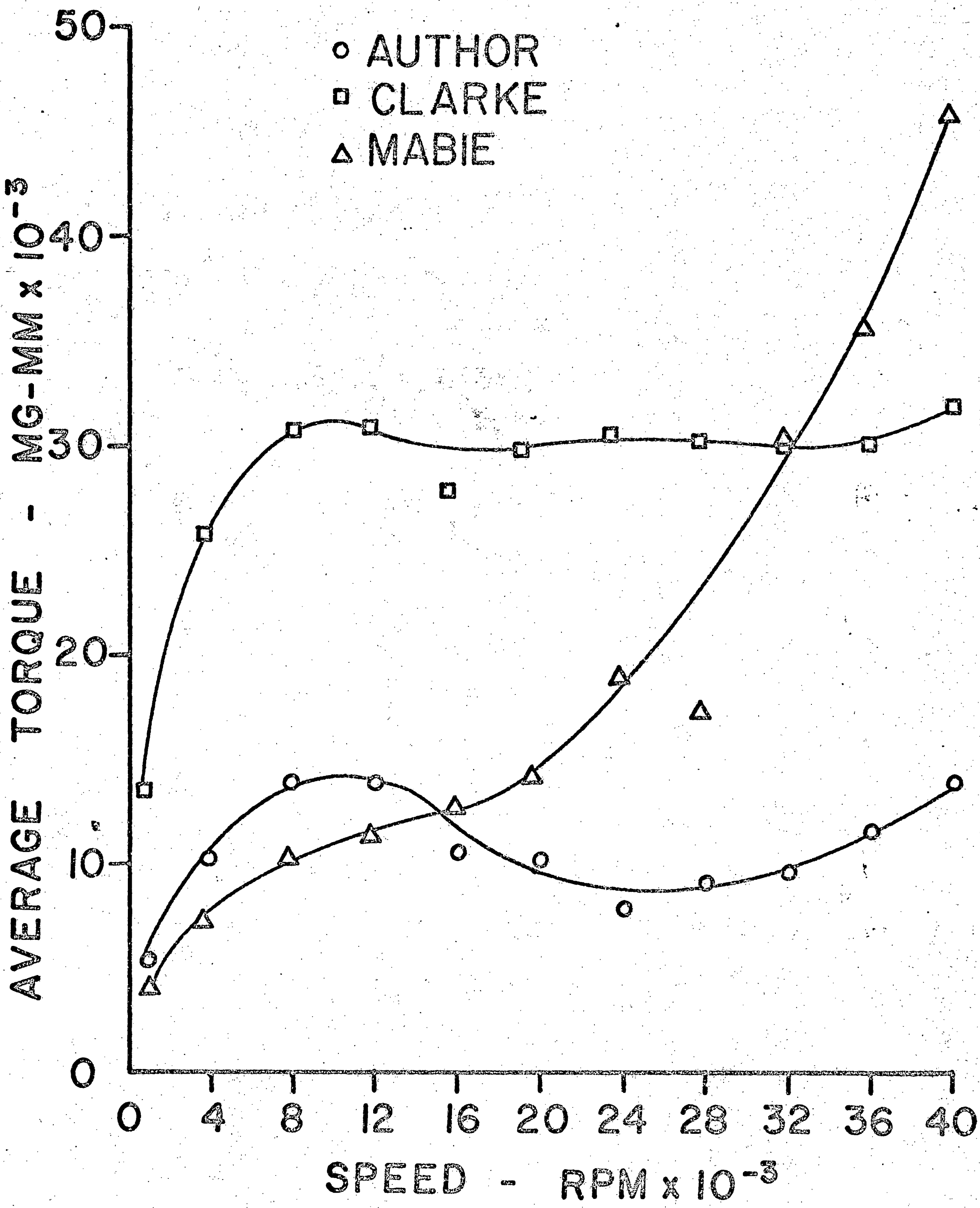

FIG. 80 COMPARISON OF 1/8 PACK, RERUN, 50 GM. RADIAL, 0 GM. AXIAL

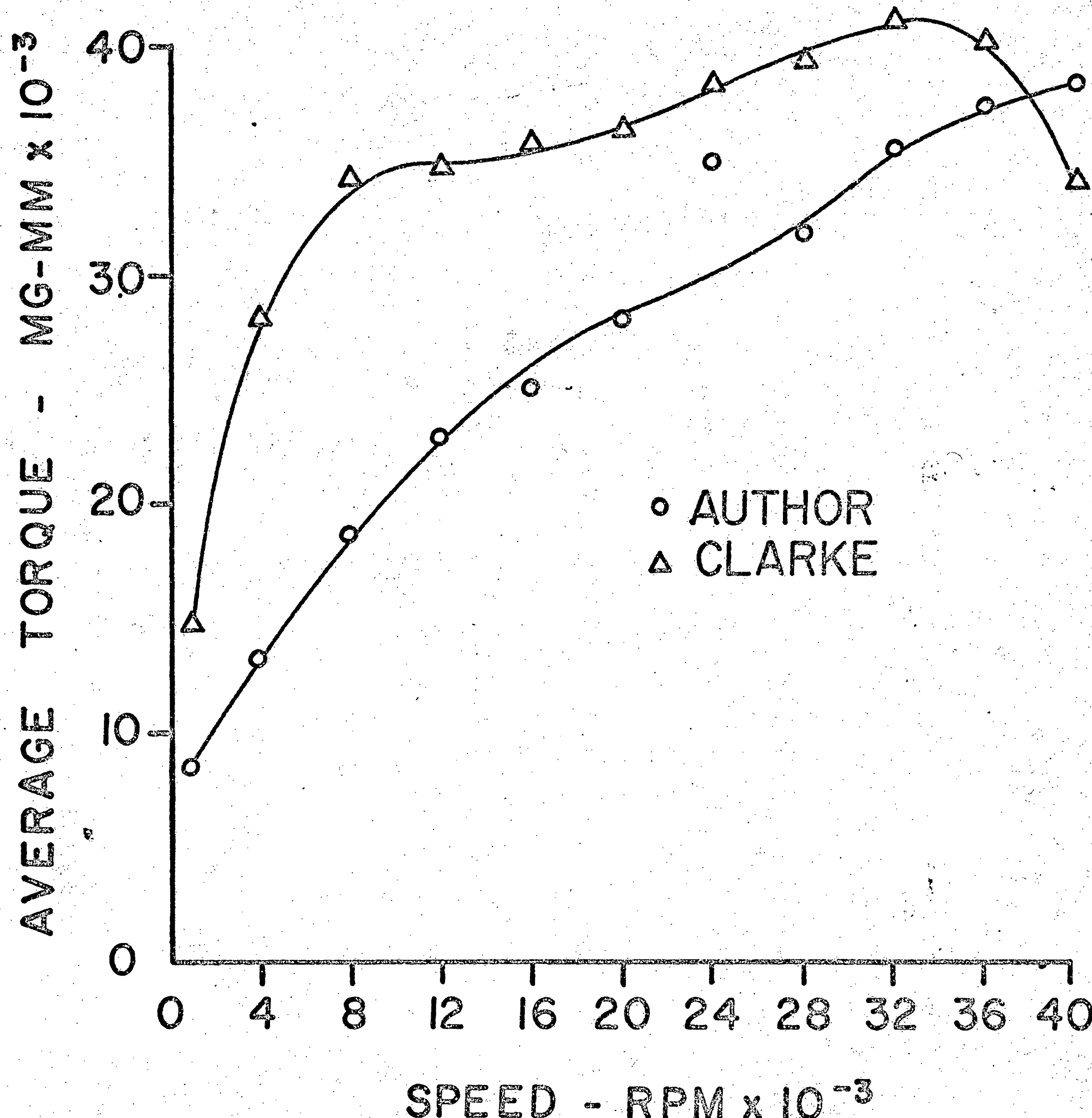

FIG. 81 COMPARISON OF 1/8 PACK, ORIGINAL RUN, 50 GM. RADIAL, 50 GM. AXIAL

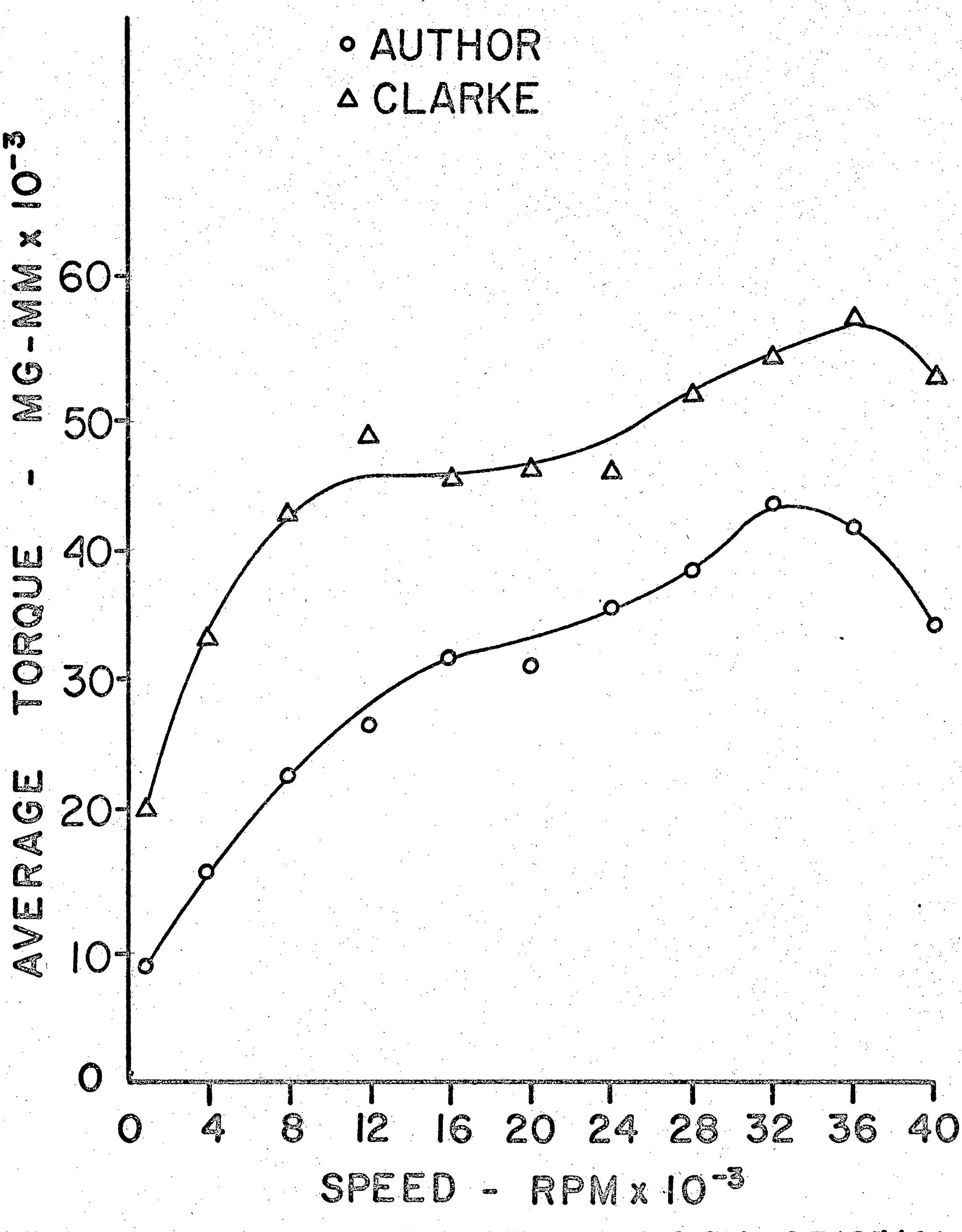

FIG. 82 COMPARISON OF 1/8 PACK, ORIGINAL RUN, 50 GM. RADIAL, 100 GM. AXIAL

the data did compare reasonably well. This was particularly true for oil lubricated bearings where reproducibility was known to be quite good as evidenced by the sample standard deviations.

This agreement with data from other investigators did justify confidence in the pneumatic technique employed in measuring the torque of R-3 instrument ball bearings.

VII. CONCLUSIONS

Based upon the results of this investigation, the following conclusions were made:

1. A pneumatic technique employing the flapper-nozzle valve will accurately measure the running torque of R-3 instrument ball bearings.
2. The torque data agreed well with the findings of other investigations working with the same load, lubricant, and speed range.
3. The effect of radial loading in the 50 to 200 gm range had no apparent effect on the torque for bearings lubricated with oil and 1/16 grease pack.
4. For axial loads from 0 to 200 gm, the torque for all lubricants increased approximately in proportion to the equivalent static load acting on the bearing.
5. For rerun tests, grease lubricated bearings exhibited a torque two to three times as great as bearings lubricated with an equal weight of oil.

6. The torque for oil and grease lubricated bearings on original runs depended largely on how the oil or grease was located in the bearings.

VIII. RECOMMENDATIONS

1. It is recommended that a study of elastohydrodynamic lubrication in instrument ball bearings be initiated. This study should be concerned primarily with the effect of radial and axial loads on torque.
2. It is recommended that studies be initiated to determine consistent methods of lubricating ball bearings with grease.

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DEVELOPMENT OF A PNEUMATIC SENSOR FOR MEASURING THE TORQUE OF INSTRUMENT BALL BEARINGS

Earl Garland Edwards

Abstract

Of the studies that have been conducted on the operational characteristics of instrument ball bearings, a great majority have been in accordance with MIL-STD-206. Since tests in compliance with this specification determine bearing quality or rate bearings comparatively, nothing was known of the operational characteristics of the bearings in their final application. A few investigators have developed sensors to study torque characteristics of instrument ball bearings. However, in no case has a report been made of the effect on torque when both radial and axial loads were varied.

In seeking to obtain improvements in methods of measuring small torques, a pneumatic sensor was developed for testing R-3 instrument ball bearings under varying radial and axial loads. This sensor was based upon the principle of the flapper-nozzle valve. The flapper valve consisted of two orifices in series, one of constant area, the other of variable area, which was determined by flapper position. Since the pressure

between the two orifices was dependent upon flapper position, indirect measurements of torque acting on the flapper were obtainable by measuring this pressure.

As a result of this study, it was concluded that the pneumatic sensor accurately measured the running torque of R-3 instrument bearings. This statement was based upon good agreement with data from other investigators working under identical conditions.

It was also concluded that for a range of 50 to 200 gm. radial loading, no significant effect on torque was observed. For axial loads in the same range, the torque was found to vary in proportion to the equivalent load acting on the bearing.