THE MEASUREMENT OF THE RUNNING TORQUE OF OIL AND GREASE LUBRICATED INSTRUMENT BALL BEARINGS UNDER COMBINED RADIAL AND AXIAL LOADS

by

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I. INTRODUCTION

In recent years, the trends toward miniaturization and the needs for increased efficiency and reliability particularly in space applications have led to a marked increase in the use of miniature and instrument ball bearings. This necessarily has pointed up a need for more extensive knowledge of the operational characteristics of ball bearings.

From previous bearing experience, the starting torque of bearings was considered to be the main performance parameter of ball bearings. Many devices were constructed to measure the starting torque of ball bearings. Smoothness of operation of the ball bearings also generated considerable interest. By necessity, this smoothness determination was made at speeds generally less than two revolutions per minute. Few investigators have considered the running torque of ball bearings at very high speeds. In all of the above areas of investigation, there were few cases of the ball bearings being loaded both radially and axially. In the case of high-speed running torque, no report has been made of a device to determine accurately the running torque of a bearing loaded both axially and radially.

Accordingly, the object of this investigation was to design, develop, and construct a device which would

permit the measurement of the running torque of instrument ball bearings operating at high speeds under combined radial and axial loads.

II. REVIEW OF LITERATURE

A study of methods of measuring the running torque of instrument ball bearings revealed that most work has centered around the design of a machine to test ball bearings according to MIL-STD-206 specifications.[1] Briefly, these specifications are concerned with running bearings one revolution clockwise then one revolution counterclockwise at a speed of 0.5 revolutions per minute maximum. The bearings are loaded with either 75 grams or 400 grams axial thrust for the duration of the test. All tests are conducted at ambient room temperature.

Nearly all of the major bearing manufacturers have constructed machines to measure the running torque of bearings being tested according to the above specifications.[2] Fafnir Corporation has a machine with a sensitivity range of 0.18 to 1.80 gm-cm. Bendix Corporation and MPB Corporation have dual range machines with ranges of 50 to 20,000 mg-mm and 2,500 to 50,000 mg-mm. However, the Bendix and MPB machines also have a capability of speeds up to two revolutions per minute and can vary the axial load between 15 grams and 400 grams. The most popular of the commercial testers is the Sunshine Torque Tester which is essentially the same as the Bendix and MPB machines. In none of the above machines are there provisions for operation at high speed or under combined

bearing load.

In an effort to produce a more flexible machine, the Sadamel Electro-microdynamometer[3] was designed. This machine has sensitivity ranges of 0.0003 to 3 gm-cm and 0.03 to 300 gm-cm with a 0.5% accuracy. Although the bearings can only be loaded axially, they may be run between 1 and 2800 rpm. Although the exact details of construction were not available, it is known that the operation is based on a wire resistance-type gage set in a null balance bridge with a galvanometer being the direct torque reading instrument. To enhance readout stability, the system was arranged with a spiral spring between the test bearing and the torque sensing device in order to smooth out the usual torque fluctuations. The shortcomings of this device are the low speeds available and the inability to subject the bearing to combined loads.

In studying the operation of grease-lubricated ball bearings under actual service conditions, Mr. P.R. McCarthy of Gulf Research[4] built a device to test ball bearings at high speeds, high temperatures, and under combined loading. The bearings used in the grease tests were of relatively large size; on the order of 0.75 to 1.00 inches bore. To supply the driving power for the bearings, a radial inward-flow air turbine was used. The useful speed range was between 20,000 and 45,000 rpm. The bearings were radially loaded by means of a weighted steel strap looped

over the stationary outer race of the bearing. A beam was attached to the loop to prevent the loop from spinning over the bearing. The cantilever beam was instrumented with strain gages and served as a torque sensor. Measurement of the deflection of the beam due to its restraining action served as an indirect measurement of the running torque of the ball bearings. The axial load was applied by means of a weight and lever system. The weights caused a dowel to make point contact with a cup which was positioned against the outer race of the bearing. Thus, a force on the pin caused the cup to apply an axial force on the bearing. Yet, the outer race of the bearing was still free to rotate since the point contact of the rounded dowel on the cup created very little friction to angular motion when the contact took place at the center of the cup. It will be noted that in the axial and radial loading mechanisms of this machine some small amount of inherrent friction was present although the frictional forces were comparatively small and did not appreciably affect the results of tests on such large bearings. Also, the speed range of the machine was limited to the mid-to high-speed range due to the design of the type of air turbine used.

Considerable investigation of the running torque of instrument ball bearings has been carried out by H.H. Mabie.[5] This work was done on bearings with bores ranging from

1/8 inch to 1/4 inch and at speeds of 1,000 to 40,000 rpm. However, in the entire series of tests, the bearings were loaded radially only.

The power source for Mabie's tester was again an air turbine. However, the design chosen was similar to a Pelton wheel with a single nozzle. The turbine wheel physically resembled a thick ratchet wheel with the nozzle, a copper tube, directing the air stream tangentially to the turbine wheel. The design was selected primarily for the ease and speed of construction since the efficiency of the unit was of little consequence if the desired speed range could be attained. The outer ends of the turbine shaft served as the shafts on which the test bearings were mounted. One test bearing was mounted on each end so that the machine could test two bearings simultaneously. The inner races of the test bearings were driven at turbine speed while the outer races were restrained from rotation by a pendulous disk fitted over the outer race of each bearing. In addition to restraining rotation, the pendulous disks also served as the torque measuring mechanism. Although the disks did not rotate, they would undergo a small angular displacement proportional to the drag on the outer race of each bearing. Thus, a measure of the angular displacement could easily be converted into a measure of the running torque of the bearing. To measure the angular displacement of the disks, the disks had

marks machined on them at one degree increments around the periphery. A pointer was fixed to the top of a stationary part of the turbine frame and rotation of the disks was observed relative to the fixed pointers. The torque disks weighed between 40 and 60 grams depending upon the size of the test bearings. The pendulous mass of the disks was approximately 5 grams per disk. This was found to be sufficient to permit torque readings between 1,000 and 30,000 mg-mm. Speed of the turbine was measured electronically by means of a light and photocell. The light beam was incident upon a cylindrical disk attached to the turbine shaft. The disk was arranged with light and dark regions around the circumference of the cylindrical portion so the light would reflect into a properly located photocell when incident upon a light spot. Through appropriate circuitry, the frequency of the pulses was displayed on an electronic digital counter. The frequency displayed was the average rpm over a one second period. Initially, there was some problem with windage drag between the tachometer disk and the pendulous torque disk on one side of the turbine. This problem was easily relieved by the placement of a baffle between the two disks.

As is characteristic of ball bearings, the torque readout system showed distinct torque fluctuations or spikes which tended to make the pendulous disk oscillate

slightly , thereby complicating the task of obtaining accurate average torque readings. To smooth out the fluctuations a damper was added to both disks. The dampers were small rectangular troughs filled with a damping fluid through which the lower part of the pendulous disk would move. The drag of the disk moving through the fluid greatly retarded any sudden movement of the disk. The first damping fluid used was mercury which proved to have very satisfactory damping characteristics. However, it was feared that there was a possibility of some flotation of the disk in the mercury causing erroneous data. To eliminate this possible source of error, the damping fluid was changed to SAE 50 oil. The oil proved to be a satisfactory substitute in the damping troughs.

While probably not of primary importance in the initial design stages, one of the safety features of the Mabie machine was its fail-safe operation in the event of a bearing seizure. Since the only restraints on the torque disks were the pendulous masses at the bottom, a seizure would merely cause the entire disk to rotate with the inner race at the test speed. Indication of the failure would be given immediately in the form of oil slinging caused by the disk rotating through the damper at high speed. As a further precaution against failures at high speed, the entire test apparatus was enclosed in a wood-lined steel case. Provisions for observation were

made by installing lucite ports in the case in such a way that the centers of the ports coincided with the axis of rotation of the turbine shaft.

Some work was also undertaken at SKF Bearing Company by J.C. Lawrence.[6] His work was concerned with the running torque at very low speeds and at mid-range speeds while under very high thrust loads. The power source for this tester was again an air turbine which was capable of speeds in the region of 20,000 rpm. The axis of the driving shaft was vertical so that any weight resting on the test bearing would create an axial load on the bearing. The actual loading mechanism was a steel cup which fitted over the outer race of the bearing while the inner race was fitted on the vertical turbine shaft. The loading cup resting on the outer race of the test bearing could be filled with varying amounts of mercury to provide selected axial loads. The inner race of the test bearing was driven at turbine speed while the angular motion of the outer race was restrained by a spiral spring. The resisting torque of the spring was calibrated according to torque watch readings. Further, the torque watch readings were used to calibrate angular displacement of the loading cup as measured by a pointer fixed to the cup and moving relative to the fixed frame of the turbine case. Damping was taken care of by the high inertia of the loading cup on the outer race. Very little testing

was conducted at the upper speed ranges of the machine used.

Torque Measurement--Strain Gages

In addition to the obvious strain measurements which may be made with strain gages, these gages may be applied indirectly to measure such things as weight, pressure, torque, and other items of particular interest. Torque measurements are most often made by the application of strain gages at specific angles on the shaft being subjected to the torque.[7] However, this approach is not directly applicable to torque measurement with instrument ball bearings mainly due to the very small torques involved. When applied to a very thin beam, however, the strain gages are very capable of detecting small forces applied to the free end of a cantilever beam. The force on the end of the beam causes a small amount of strain to occur all along the beam; the strain being greatest at the base of the beam. By attaching the strain gages at the base of the beam, the greatest sensitivity is achieved. The strain which causes some resistance changes in the strain gages is measured by means of a Wheatstone bridge. For dynamic uses such as in fluctuating torque measurement, the bridge is not balanced for each reading, but rather a deflection of the galvanometer placed across the bridge is the strain indicator. This arrangement is particularly convenient for making rapid

readings since a meter reading only need be taken with no adjustments necessary for each reading.[8]

When using strain gages, there is always a problem of temperature compensation. Temperature compensation is necessary to prevent even small changes of ambient or local temperature being indicated as changes of strain.[7] The most popular method is to use two similar gages, each in separate legs of the Wheatstone bridge. By proper selection of the legs used, additional benefits may be obtained. In the case of a cantilever beam, the gages may be connected in adjacent legs of the bridge. This arrangement has several favorable points. First, the installation is fully temperature compensated. Secondly, the gages connected this way and mounted on opposite sides of the beam will double the output or needle deflection for a given amount of strain. Thirdly, the gages mounted in this way are insensitive to axial loading of the beam. Fourth, the gages are insensitive to twisting of the beam such as may be caused by asymmetrical loading.[8] Thus, the cantilever beam installation is suited very well to the detection and measurement of small forces applied at the end of the beam. The contemplated forces are due to a torque and because of construction tolerances and geometry cannot be expected to be perfectly symmetrical and with axial components.

III. INVESTIGATION

A. BEARING HANDLING

In considering the investigation of the effects of combined axial and radial load on the running torque of ball bearings, it was considered best to conform in as many respects as possible to procedures followed by other investigators so that meaningful comparative data might be obtained. Due to the ready availability of similar equipment and data, this investigation follows closely the work of H.H. Mabie[5] with radially loaded bearings. His work covered three sizes of instrument bearings, the basic bearing dimensions of which are shown in the following table.

Table 1 Bearing Sizes

Designation	R-2	R-3	R-4
Outside Diameter (inches)	3/8	1/2	5/8
Bore (inches)	1/8	3/16	1/4

The R-3 size bearing was selected for this investigation because of its previous history of consistent behavior. Only one bearing size was chosen so that very comprehensive and statistically accurate data could be obtained for the intended conditions within a reasonable test period.

It was fortunate that the same bearings used in Mabie's investigation were available for further testing. However, this did necessitate the establishment of well

controlled handling techniques.

A well-planned bearing handling program was required to assure thorough cleaning of bearings between test segments while minimizing the possibility of contamination which could produce highly erratic results.

Since the bearings to be used in the tests had been run previously, the first step was to clean them thoroughly. However, removal of the dust shields proved to be a difficult task. It was extremely hard to hold the bearing and pry out the retaining ring holding the dust shield in. Even if the retaining ring would pry loose, it would generally spring out and become lost.

To assist in this disassembly, a special jig was fabricated which held the bearing securely without causing any damage and which prevented the loss of the retaining rings. The jig which is shown in Fig. 1 was based on a 4 x 5 inch steel plate. The plate was drilled and tapped to take a screw which fitted through the 3/16 inch bore of the bearing. The screw thus held the inner race firmly to the plate. A swinging arm was positioned so that one end of the arm would contact the outer race of the bearing. In the contact region of the arm, was placed a small piece of thick foam rubber. The arm was tensioned against the outer race by means of rubber bands attached to the other end of the pivoted arm. Thus, the inner race was held stationary by the hold down screw. The outer race was

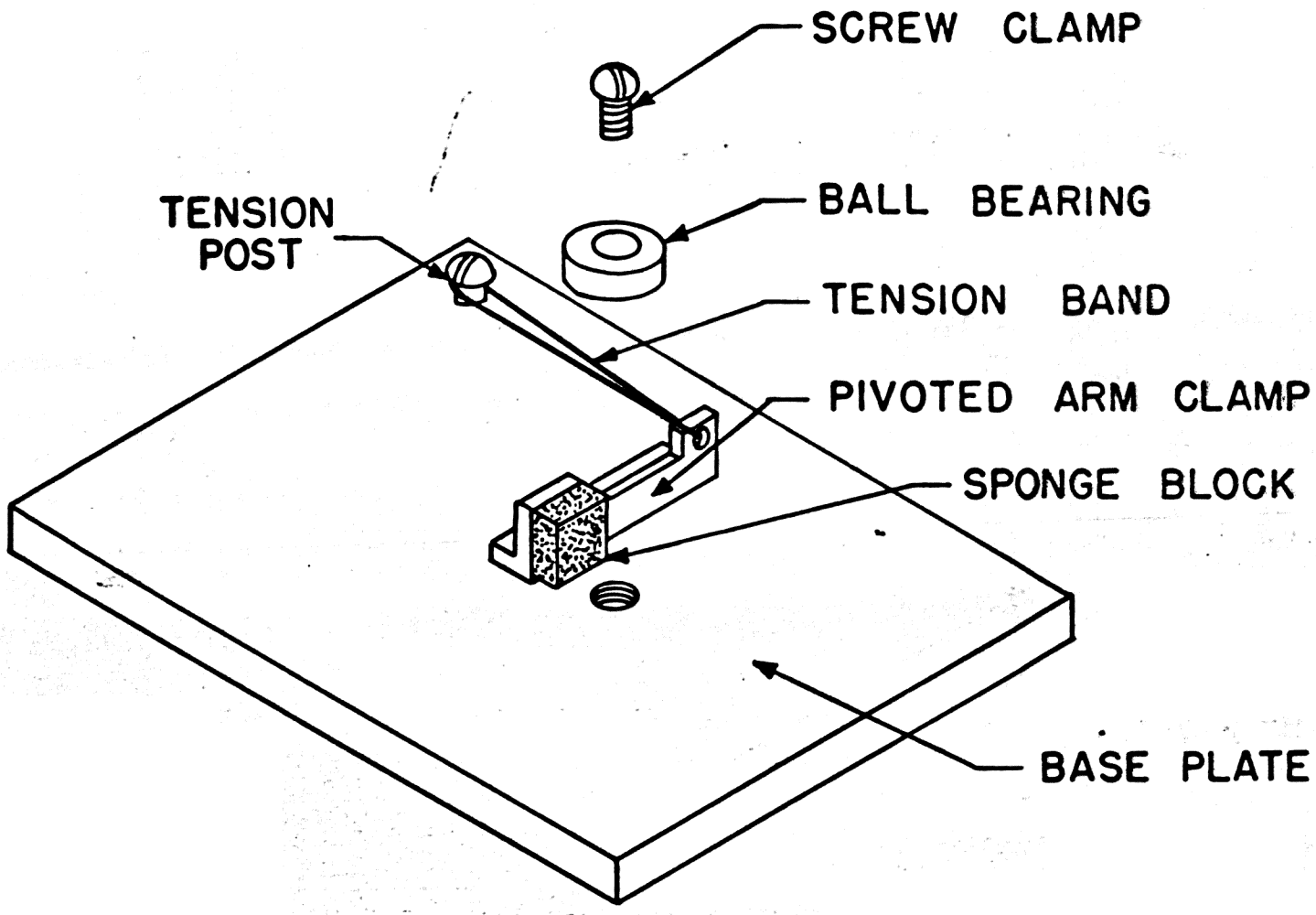

Fig. 1 Bearing Holding Jig

prevented from rotating by means of the pivoted arm. The foam rubber prevented the arm from damaging the outer race,and,in addition,covered part of the top of the bearing preventing the retaining ring from springing completely free. The actual removal of the retaining rings was accomplished by means of two sheet metal scribes. One scribe was used to prevent rotation of the ring in its groove while the other scribe picked the end of the ring from the groove. This system proved very efficient and allowed a test sample of six bearings to be disassembled in less than ten minutes.

The identity of all bearings was maintained through all test procedures. This was done by working with only one bearing at a time and always keeping each bearing in its own numbered container.

After removal of the bearing shields, each bearing was individually cleaned. The bearing was first soaked in benzene to loosen any remaining grease or oil. Then the bearing was removed from the benzene and soaked in acetone to remove the remaining benzene and to permit low residue drying upon removal from the acetone. When the bearing had dried, it was placed in its own glass container which had been ultra-sonically cleaned and sealed closed. This operation was repeated for the rest of the bearings in the test group. The chemically washed bearings were then further cleaned ultra-sonically in an inert cleaning

fluid. Each bearing was individually held on a plastic coated rack while suspended in the cleaning fluid. The cleaning was continued for five minutes and the entire rack was removed from the cleaning tank while the fluid was still being agitated to minimize the possibility of any foreign particles settling in the bearings. The bearings were again placed in their individual glass containers for transportation to the weighing and lubrication area.

The cleaned bearings were weighed on an analytical balance to obtain the dry weight to one ten-thousandth of a gram. The dry weight was very important since all subsequent weighing for lubrication would be dependent upon the dry weight. During the weighing operation, the selected lubricant would also be added to the bearing. In the case of oil lubricated bearings, Univis P-38 conforming to MIL-L-6085A was used. It was added by means of a syringe and a #27 hypodermic needle. It was found that by adding the oil to the balls in the area of the ball retainer, little or no oil fell through the bearing and onto the balance pan. In the case of grease lubricated bearings, Beacon 325 conforming to MIL-G-3278A was used. The grease was placed in the bearing in small quantities by means of a jewelers screwdriver. Here, it was found that grease could best be added to the outer race between the balls. The grease was added in several

small quantities so as to control better the weight added as well as to assure that the grease was actually placed in the ball races and not deposited in the dust-shield retainer-ring grooves where it would have no effect on the actual lubrication. Fig. 2 shows much of the handling equipment used including the bearing holding jig with a bearing fastened in working position.

It was previously determined that the quantity of oil added at the factory was approximately 0.0076 grams for this size bearing and this was the weight of oil used in bringing the dry bearings up to factory oil specifications. The weight of oil in bearings ordered from a manufacturer is generally not specified,and the bearings are used with the as-received lubrication. However, grease lubricated bearings usually have the weight or degree of grease pack specified. The degree of pack is the ratio of the weight of grease actually added to the weight of grease that it is possible to pack into a bearing. The values of 1/16 and 1/8 pack were chosen since they are fairly common specified values, and previous torque data was available for ready comparison. From attempts to pack completely a ball bearing with grease, it was found that R-3 bearings held an average of 0.1408 grams of grease. This gave a corresponding 1/8 pack of 0.0176 grams and a 1/16 pack of 0.0087 grams. These were the weights of grease used in the 1/8 pack

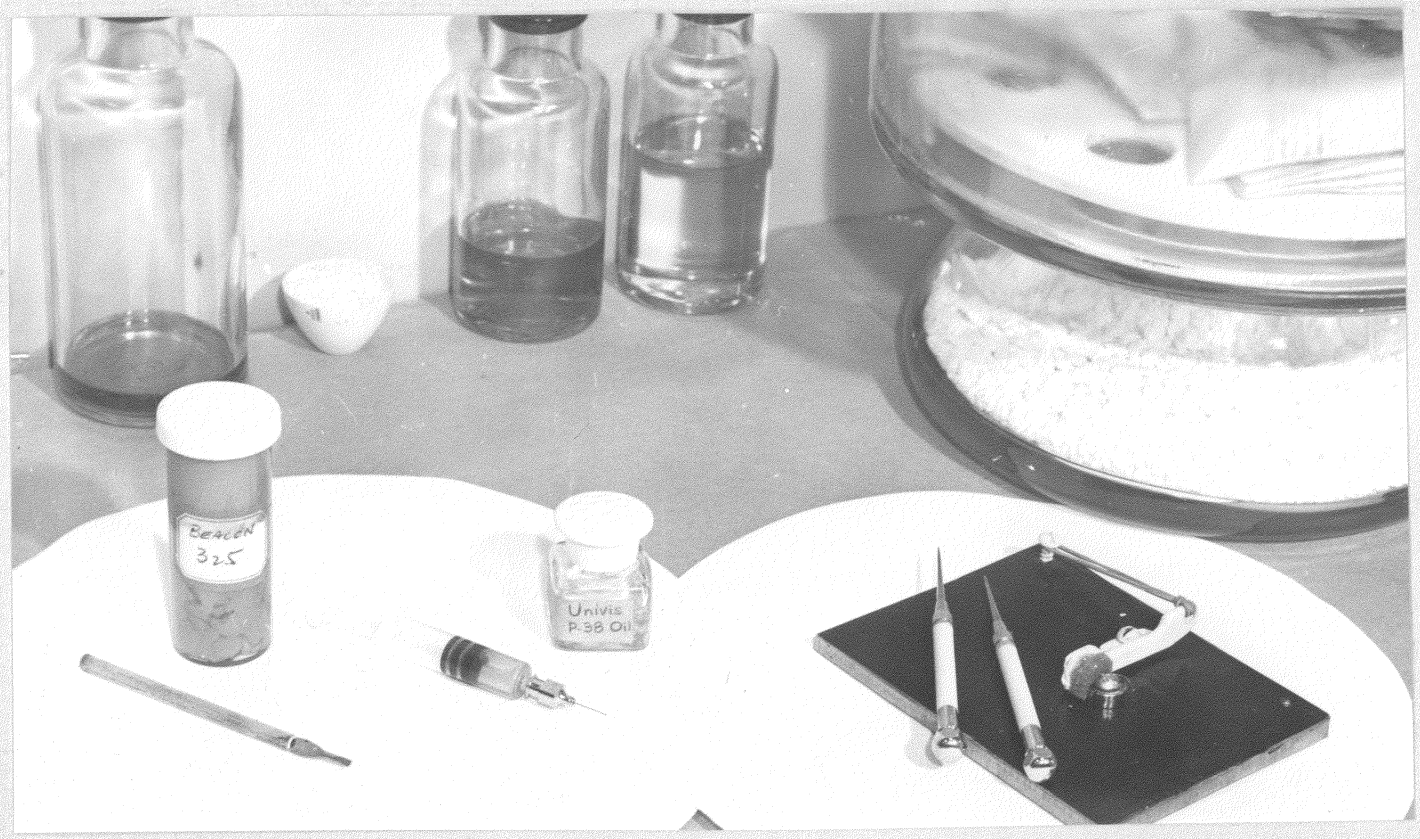

Fig. 2 Bearing Handling Equipment

and 1/16 pack grease tests.

After each bearing was weighed and properly lubricated, the shields were replaced and the bearings were placed in small individual envelopes for subsequent handling.

In any test sequence which called for re-oiling, the bearings were first weighed to determine if any lubricant loss had occurred. If there was any deficiency, the dust shield was removed from one side of a bearing and the weight of lubricant would be brought up to the original specification. The dust cover was immediately replaced and the bearing placed back in its envelope. To as great an extent as possible, the bearings were not handled by hand, but by tweezers and were always kept in their containers. In an effort to reduce atmospheric variables as much as possible, the bearings were always stored in a chemical dessicator at a controlled temperature except when being weighed and actually tested.

B. APPARATUS

Power Source

Several methods were considered for driving the test bearings at speeds that ranged between 1,000 and 40,000 rpm. Electric motors with direct drive, geared drive, and belt drive were initially considered. The direct drive electric motors were discarded because of the questionable reliability of the motor bearings.

Geared and belt drives used with electric motors were not acceptable because of smoothness considerations. Of the several types of air drives used by other investigators, the most suitable type seemed to that type used by H.H. Mabie. This Pelton wheel type turbine had proved that it worked well over the entire speed range to be covered: 1,000 to 40,000 rpm. In addition, it was smooth running and had proved to be reliable and safe at the high speeds expected. Thus a duplicate of the Mabie was constructed. A detailed drawing of the Mabie turbine is shown in Fig. 3. The turbine is also shown with the accompanying torque testing equipment in Fig. 4 which presents an overall view of the apparatus.

Torque Sensor

During preliminary investigations, optical systems seemed to offer a method of very accurately and easily determining the running torque of the instrument ball bearings. The first method considered made use of the pendulous disk in which was mounted the test bearing. However, it was thought that by optical means, the measuring arm could be increased by 10 to 20 times over measuring the torque directly by angular rotation of the disk. A light-weight celluloid cylinder was constructed, and graduated markings placed on the outer walls of the cylinder in a direction parallel to the axis of the

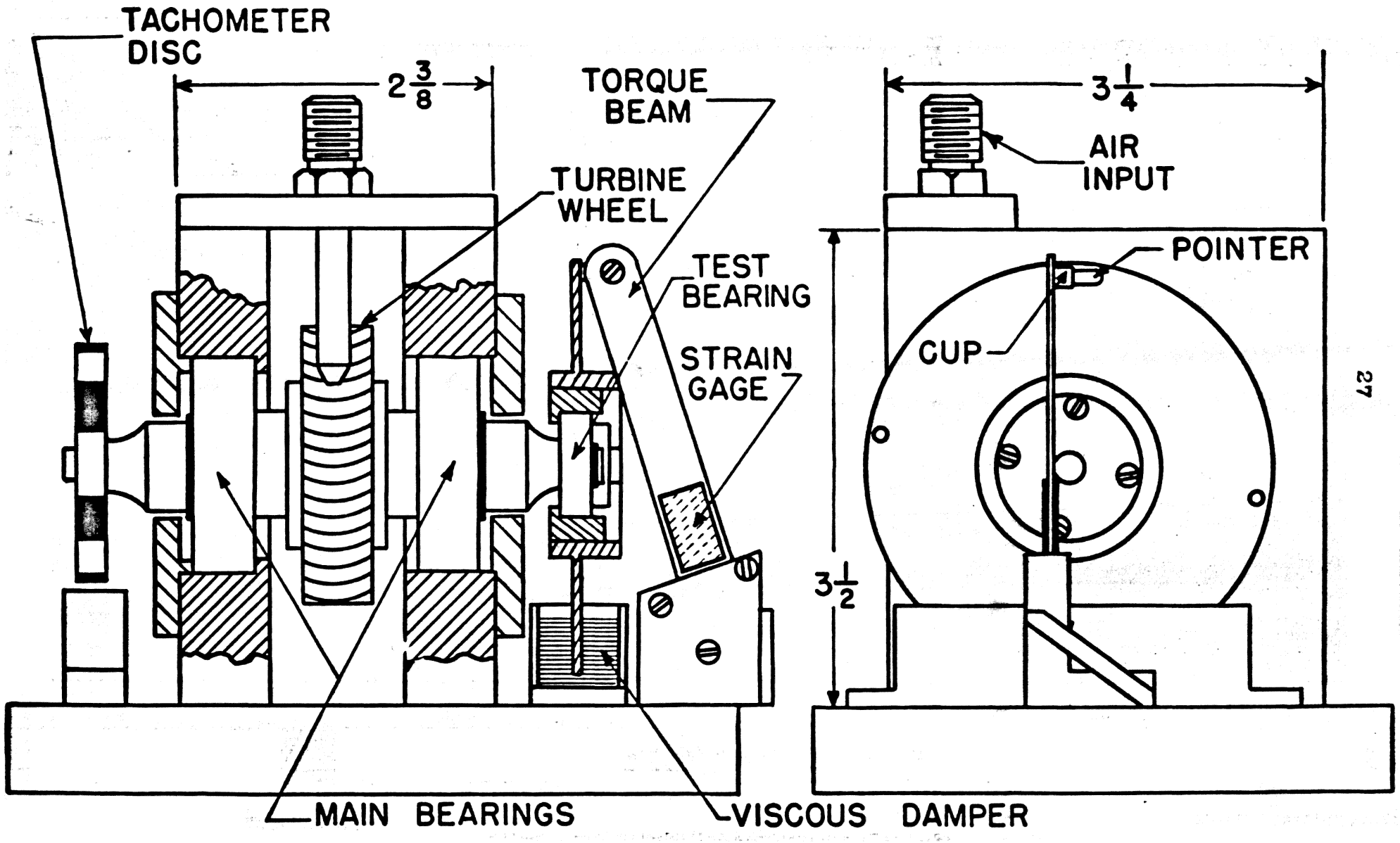

Fig. 3 High Speed Torque Tester

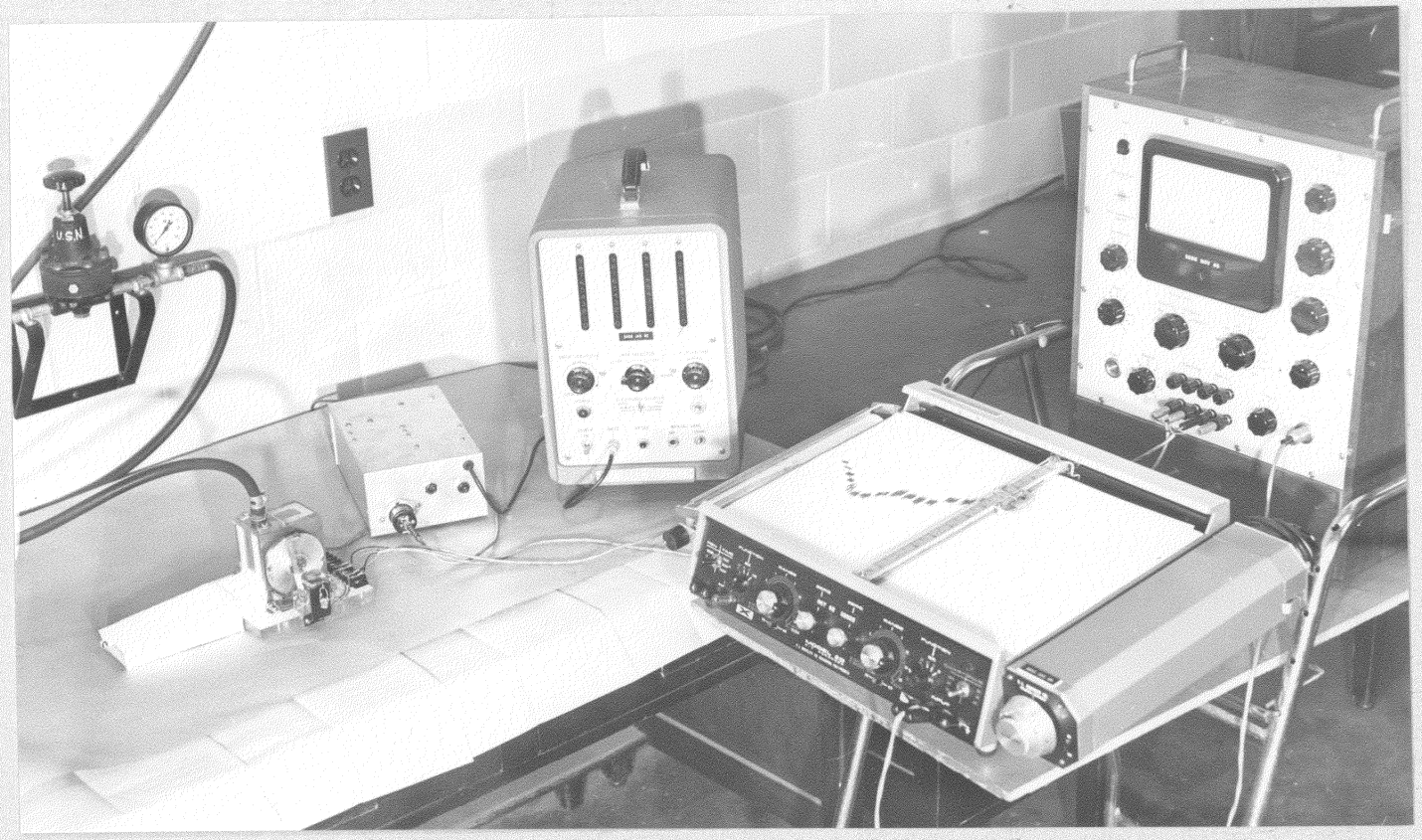

Fig. 4 Test Apparatus

cylinder. This cylinder was placed on the pendulous disk such that the disk formed a bottom to the cylinder. The open end of the cylinder faced outward, away from the turbine. A small electric light source was mounted so that the light was within the cylinder and approximately on the axis of the cylinder. The theory is that the light will shine out through the celluloid cylinder and through appropriate lenses onto a viewing screen. Angular displacements could be read by noting the graduated markings on the cylinder as they were projected on the screen. Several problems were encountered, though, and the plan was eventually abandoned. Some such problems were the weight of the pendulous disk and cylinder assembly, as well as its balance, difficulty in producing the desired light source characteristics, and indications of less accuracy than expected.

The second method of torque sensing tried was an approach using strain gages. With this approach, the test bearing was mounted in a disk which was statically balanced so that it would have no preferred orientation. Thus, rotation of the inner race of the bearing could eventually cause rotation of the outer race and the attached disk. To restrain this rotation, the disk was equipped with an L-shaped conical-nosed pointer mounted near its periphery and pointing in the direction of intended rotation. Fixed to the base of the test

stand was a long, thin beam. Attached to the free end of this cantilever beam was a small nylon cup, shaped to accept the nose of the pointer on the disk. The pointer and cup in operating position are shown in Fig. 3. Thus, the cantilever beam prevented the rotation of the disk while allowing some angular displacement to take place. It was decided that the maximum angular displacement that could be permitted was 5°. This is the generally accepted upper limit for which the sine and tangent are considered equal. It was further assumed that the maximum anticipated torque would be 40,000 mg-mm. It was known from the anticipated disk diameter of three inches that the 5° limit would correspond to a linear displacement of about 1/8 inch at the circumference of the disk. From these considerations, a suitable beam was designed. The final design was a stainless steel beam with the dimensions 0.004 x 0.375 x 2.0 inches. A beam of these dimensions, subjected to the anticipated loads, would experience a maximum of 3800 psi at the base of the beam. This was considered a very moderate loading. The final dimensions were selected partly in consideration of the difficulty of fabricating straight, flat beams of small thickness. The desired dimensions and material requirements were met in the form of a single blade from a commercial thickness gage.

Strain gages were mounted near the base of the beam

with one gage on each flat side of the beam. The gages used were Baldwin SR-4 type C-7 gages. These gages had a short gage length which was considered desirable in order to take advantage of the relatively high strains which occur at or near the base of the beam. In addition, C-7 gages have a gage factor of approximately 3.59 which is relatively high. The high gage factor means that the gage will undergo a relatively high resistance change for a given amount of strain. Moreover, the gages had a resistivity of approximately 499 ohms which,again, is relatively large. This large resistivity meant that for the 3 volt maximum bridge voltage available, there would be very little Joulean heating of the gages.

The gages were mounted on the beam with Eastman 910 cement. This cement was selected because of its extremely fast setting characterisitcs and need for little or no pressure during its curing time. After the gages were mounted, and the cement had cured, the integrity of the bond was tested. The testing was accomplished by means of a very accurate bridge circuit used to measure the resistance of the gages. A resistance measurement was made on each gage initially. Then the beam was subjected to severe bending, but not so severe as to cause any permanent deformation or damage to the cement bond. The resistances of the gages were again checked and matched the original resistivities exactly. The sensitivity of

of the measuring bridge was 1 part in 50,000 which was quite sufficient for this check. The gage to ground resistance was also checked and found to be essentially infinite for both gages. A large discrepancy in the resistance readings before and after bending could have indicated a poor bond between the gage and the beam. Low gage to ground resistance would also have indicated an improper gage installation.

The torque beam was mounted rigidly and isolated electrically from the turbine by its adjustable lucite clamp. Rigid mounting of the strain gage leads was found to be important in preventing zero drift during operation. Thus, the leads were permanently epoxied to the beam clamp. The beam installation is shown in Fig. 5.

Disk Stabilizer

The running torque of ball bearings is not a constant value but rather a fluctuating torque superimposed on some average torque. This characteristic makes it difficult to measure the average running torque, particularly if the measuring device tries to follow the oscillations. In attempting to damp the oscillations of the torque disk caused by the torque fluctuations, the lower edge of the disk was immersed in a viscous fluid. This damping method may easily be seen in Fig. 3. Initially glycerine was tried as a damping fluid because of its high viscosity. Also, being clear, it gave the appearance of a "clean"

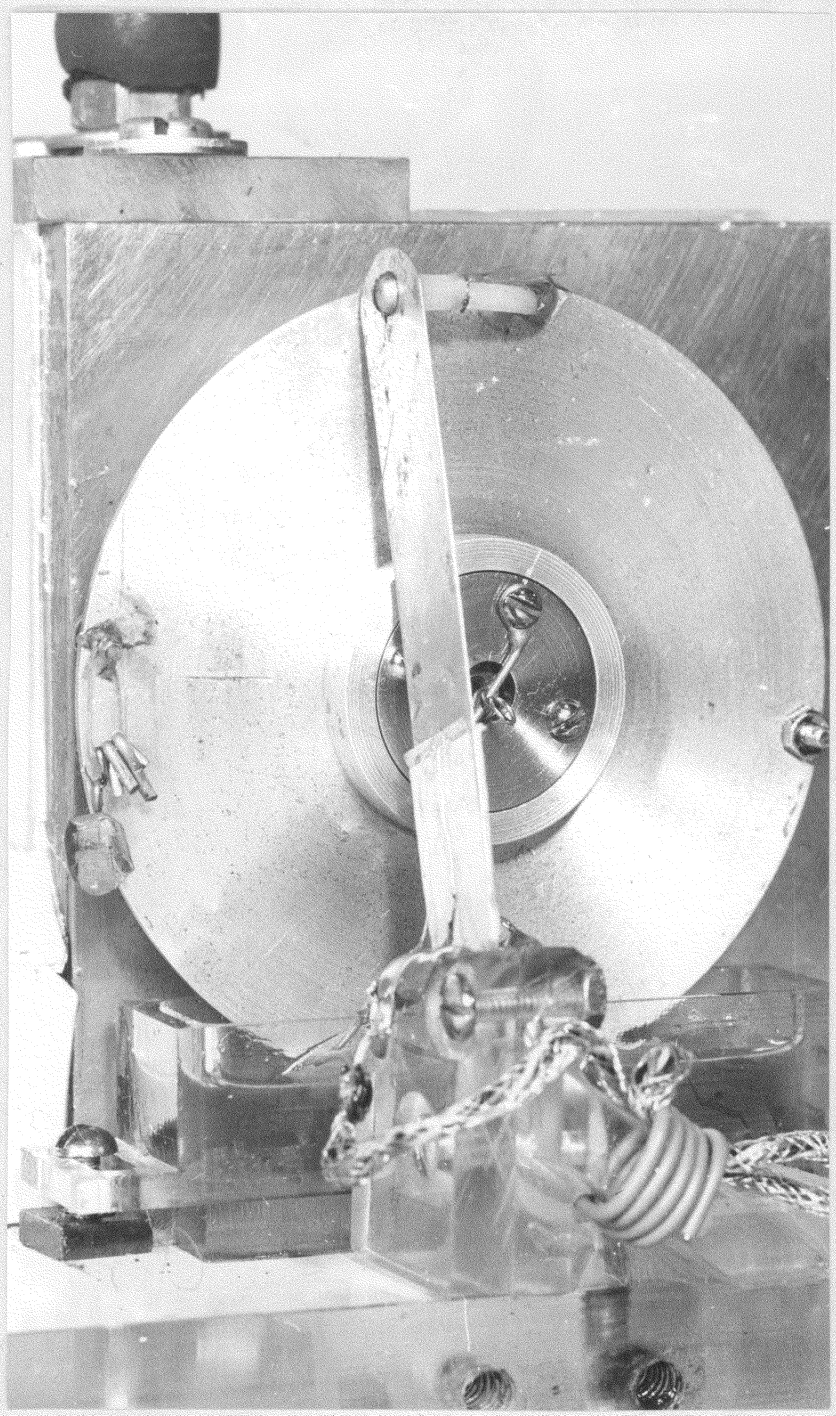

Fig. 5 Basic Torque Sensor Assembly

fluid. Preliminary tests,however, proved that it was not capable of damping out the torque fluctuations, particularly at higher rotational speeds. Although other investigators had little success with mercury as a damping fluid, it was, nevertheless, tried. When the mercury was sufficienty deep in the damper trough, it definitely caused the torque disk to rotate away from the measuring beam. This was due to the tendency of one of the balancing weights on the bottom of the disk to float to the surface of the mercury,thus pulling the bottom of the disk with it. Mercury was eliminated as being highly unsuitable. Finally, SAE 50 oil was tried in the damper. It proved capable of reducing the oscillations to an acceptable level without causing any measurable flotation of the torque disk. The oil damper is shown in Fig. 5.

Speed Measurement

The rotational speed of the turbine was measured through electronic means and displayed on a digital counter. The measuring system depended on the small disk mounted on the non-testing end of the turbine shaft. This tachometer disk is shown in Fig. 3. The circumference of the disk was marked with six light areas alternating with six dark areas. A small light was directed at these light and dark areas, and a photoelectric cell was positioned so as to detect the reflected beam from the light areas. The signal pulses from the photoelectric

cell were the input to a Hewlett-Packard digital counter. The timing gate of the counter was set at one second. Thus with the counter registering six pulses per revolution for a period of one second, the speed in rpm was obtained by multiplying by 60 seconds per minute and dividing by 6 pulses per revolution. The net result being that the readings had to simply be multiplied by ten to obtain the rotational speed in rpm. The tachometer circuitry is shown in Fig. 6. This provided a quick and easy method for accurately measuring the rotational speed of the turbine without interfering with its operation.

Speed Regulation

One of the major advantages of the air turbine was that its speed could be easily regulated over a wide speed range. The regulation was accomplished by installing an air pressure regulator between the air line and the turbine. The speed of the turbine could be regulated to within plus or minus 10 rpm over the entire speed range used. The regulator and pressure gage are seen mounted to the wall in Fig. 4. In order to streamline the testing procedure the tolerance used was 10% of the difference between speed intervals. Since in all but the initial case, the intervals were 2,000 rpm, the speed was held to within 200 rpm of the nominal speed while torque readings were taken. As some indication of the ease and speed of this regulation method,

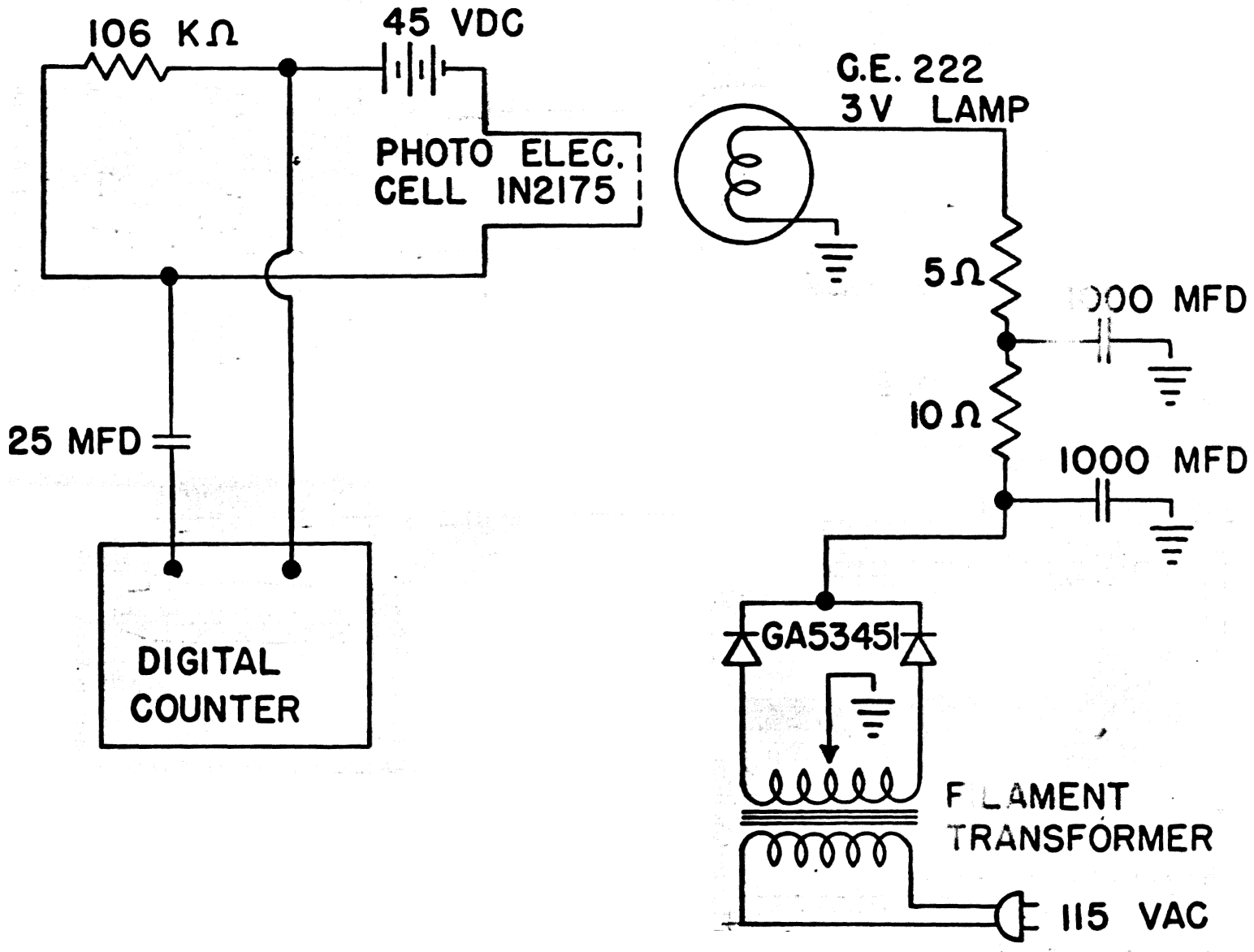

Fig. 6 Tachometer Electrical Schematic

a series of 21 data points ranging from 1,000 to 40,000 rpm could te taken in a period of 10 minutes. This means that the speed could be changed and stabilized in a period of less than thirty seconds.

Strain Measurement

The strain measurement and subsequent torque measurement was accomplished by means of a Bruel and Kjaer Model 1516 strain indicator. As mentioned previously, a four-arm bridge was used with the active strain gages constituting two adjacent arms of the bridge. The common point between the two active gages was also connected to the shielding used on the strain gage leads. It was imperative that all leads be shielded and the beam be grounded since measurements were being made near the limit of sensitivity of the instrument. Preliminary investigation had shown that a Baldwin-Lima-Hamilton SR-4 Strain Indicator used with an oscilloscope was not capable of measuring the small resistance change of the gages. The Bruel and Kjaer instrument, however, proved capable of producing a good indication of strain. This instrument is shown with the other apparatus in Fig. 4.

Strain Recording

After efforts to use an oscilloscope with a Polaroid camera to record strain readings amplified from the Baldwin Lima-Hamilton SR-4 bridge proved unsuccesful, a Moseley

Autograph x-y plotter was tried with the Bruel and Kjaer instrument. This combination proved very compatable and provided satisfactory low-cost torque recordings. The x-y plotter accepted 11 x 15 inch sheets of paper. A complete torque test series from 1,000 to 40,000 rpm for a single ball bearing was put on each sheet. The torque values were recorded as a deflection on the 11 inch ordinate. The pen was manually moved along the 15 inch abscissa to a new position for each speed test point. Because the ordinate was relatively large, torque readings could reasonably be expected to have an accuracy of ± 1/2% of the full scale deflection.

Axial Loading Mechanism

The problem of axially loading the test bearing was originally believed to be complex because of unknown variables which might be introduced in the final torque measurement. The calibration system used, however, eliminated this problem and cleared the way for several possible loading methods.

In preliminary investigations, compressed air was used to load the torque disk axially. A small tube was fixed to the torque disk such that the axis of the tube was coincident with the axis of the disk. The disk formed a bottom to the cylindrical tube. A smaller tube was positioned so that it fitted in the open end

of the tube fixed to the disk, but neither tube was permitted to contact the other. It was thought that compressed air directed through the smaller tube would impinge on the bottom of the cylinder formed by the larger tube and the disk. This impinging air stream would thus load the disk and bearing axially. The air was permitted to escape through the annular area between the two tubes. The loading system did work and appeared not to induce any rotational motion of the disk. There was a problem, though, in that the air escaping from the annular area causing turbulence in the proximity of the torque beam. This air flow caused very erratic readings of the torque sensor. To combat this turbulence, extensive shielding and duct work was fabricated and installed. However, the problem still persisted and forced the abandonment of the compressed air loading system.

Because of this indication of the extreme sensitivity of the torque beam, a much simplified method of axial loading was sought. The method which eventually proved successful was a string and roller system using a dead weight as the loading force. The loading mechanism is shown schematically in Fig. 7. A small bar was made which fitted across the bearing cap using the existing cap screws for attachment. To this bar was attached a tiny loop which was located along the center-line of the bearing cap. The loading weight was hung on a small

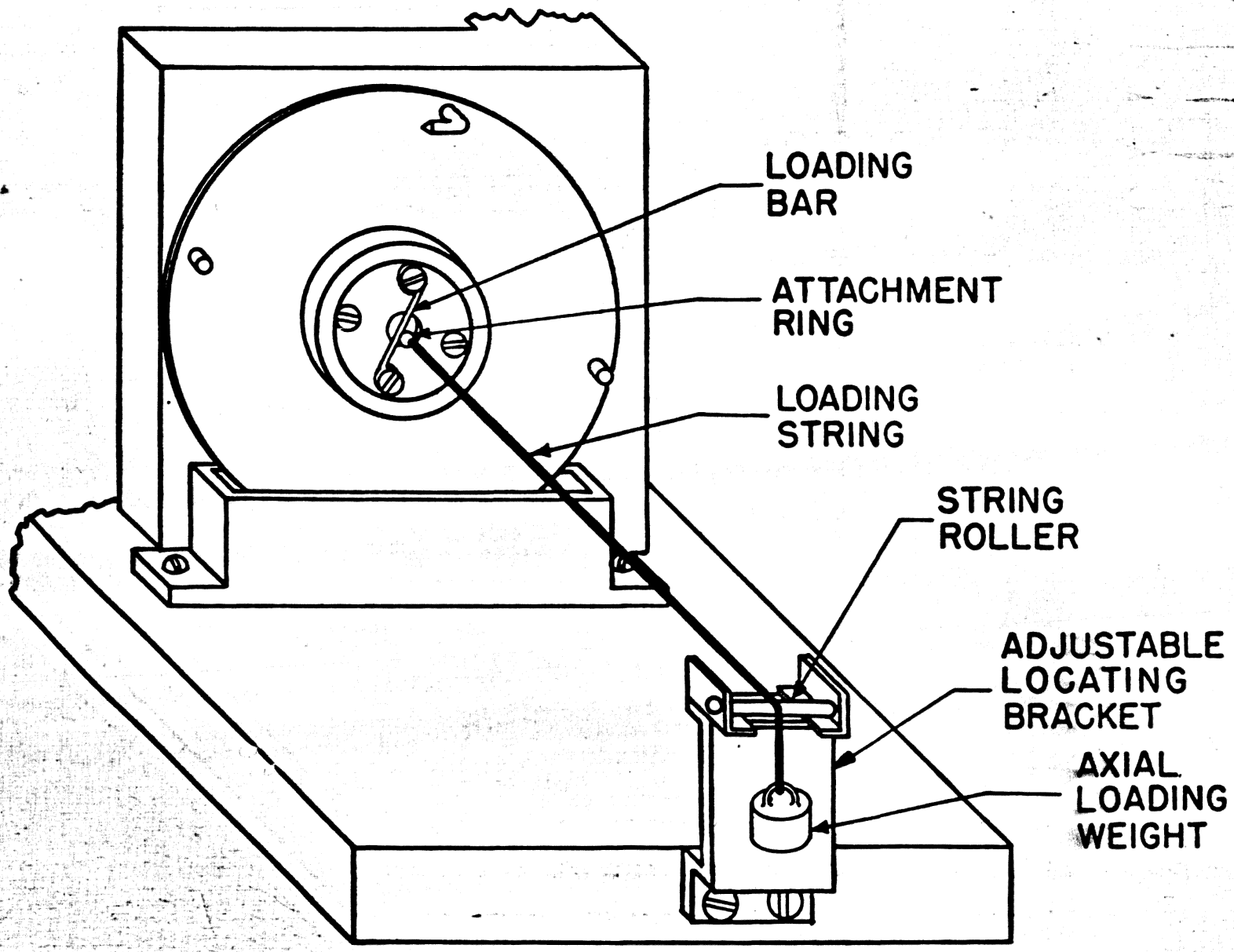

Fig. 7 Axial Loading Mechanism

thread which was looped over a roller, and the opposite end of the string was attached to the loop on the bearing cap. The string selected was wound clockwise so that the counter-clockwise motion of the disk during testing would not tend to tighten the string. The roller was copper alloy and fitted over a brass journal bearing. The roller was cut with a V-groove to position the string. The frame holding the roller was adjustable in all three directions. In this way, it was assured that the string exerted a true axial load without skew radial components. The axial loading mechanism is shown mounted to the turbine base in Fig. 8.

The weights used for axial loading were fabricated from lead cylinders and plates, and fitted with solder loops to attach them to the tensioning string. The weights were adjusted in size by shaving away small bits of metal until exactly the desired weight was attained. The weights used are listed in the following table.

Table 2 Axial Load Weights

Axial Load Ratio*	1/8	1/4	1/2	1	2
Weight (grams)	5.84	11.69	23.39	46.78	93.56

*Note: Axial Load Ratio is defined as Axial Load/Radial Load

Additional Air Shielding

The high sensitivity of the torque beam to random air currents pointed out the need for more isolation of

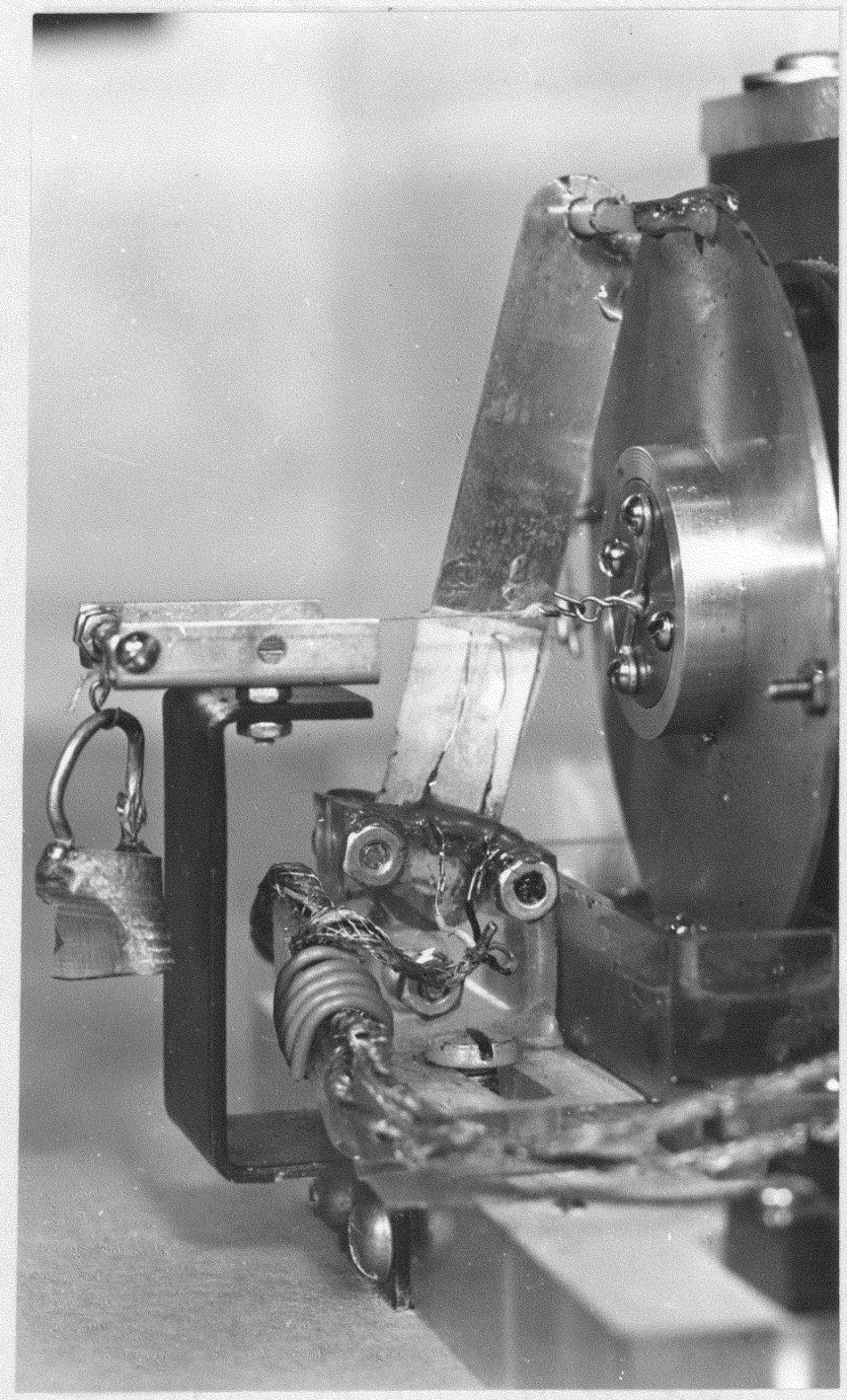

Fig. 8 Torque Sensor with Axial Loading Mechanism

the beam from the rest of the test equipment that might cause disturbances. To this end, the turbine wheel was completely enclosed and fitted with an air duct which would carry the exhaust air outside of the protective steel hood. It was found that this enclosure and ducting also considerably reduced the noise level during operation.

To guard further against stray air currents, a small cardboard enclosure was constructed to fit snugly around the volume enclosing the torque beam, torque disk, axial loading mechansim, and the viscous damper. It was found that accurate calibration of the instrument could not be made without this enclosure, even in still air. This box was also equipped with celluloid observation ports positioned so that the enclosed apparatus could be observed with the steel hood in place.

Safety Measures

The primary hazard considered was the possibility of a failure of the rotating parts at high rotational speeds. To best protect against human injury, it was decided to cover the entire turbine and torque sensing gear with a strong and absorbant hood. The hood case was constructed from 1/4 inch steel plates on both ends and the top. The sides were 1/8 inch steel plates. The steel gave the hood sufficient strength. Impact absorption was built in by lining the entire hood with 1/2 inch plywood. The sides

of the protective hood were fitted with 4-inch lucite ports to permit observation of the test apparatus during operation. See Fig. 9 for the integrated system schematic.

C. PROCEDURE

Calibration

One of the most important phases of each test series was the calibration of the torque sensing unit. This was particularly necessary since full-scale recorder deflection was used as much as possible to increase the accuracy of the final readings.

Calibration was carried out by the use of dead weights hung on the periphery of the torque disk. A calibration weight may be seen hung on the left side of the torque disk in Fig. 5. The weights were specially made so that when hung on the post provided on the torque disk, they would produce even equivalent torque forces. The torques chosen were 1/2 g-cm, 1 g-cm, 2 g-cm, and 3 g-cm. The weights themselves were fabricated from lead and fitted with a small hook to fit on the calibration post on the disk. The exact weight was obtained by shaving away tiny bits of lead from the weights until the desired pre-calculated weight was obtained. The geometry of the calibration method is shown in Fig. 10.

For the actual calibration, the test bearing was mounted in the torque disk and the assembly fitted on

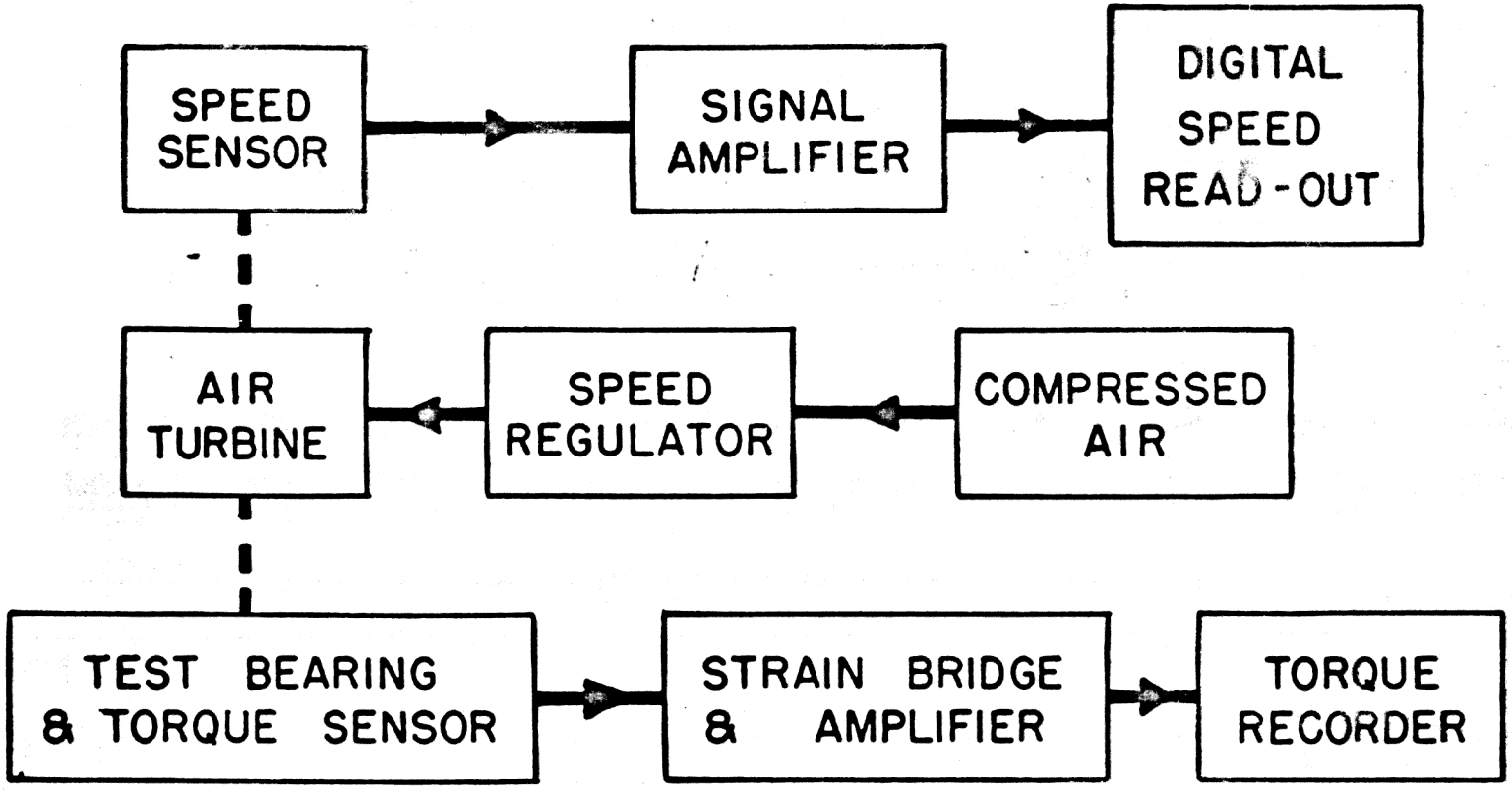

Fig. 9 System Flow Chart

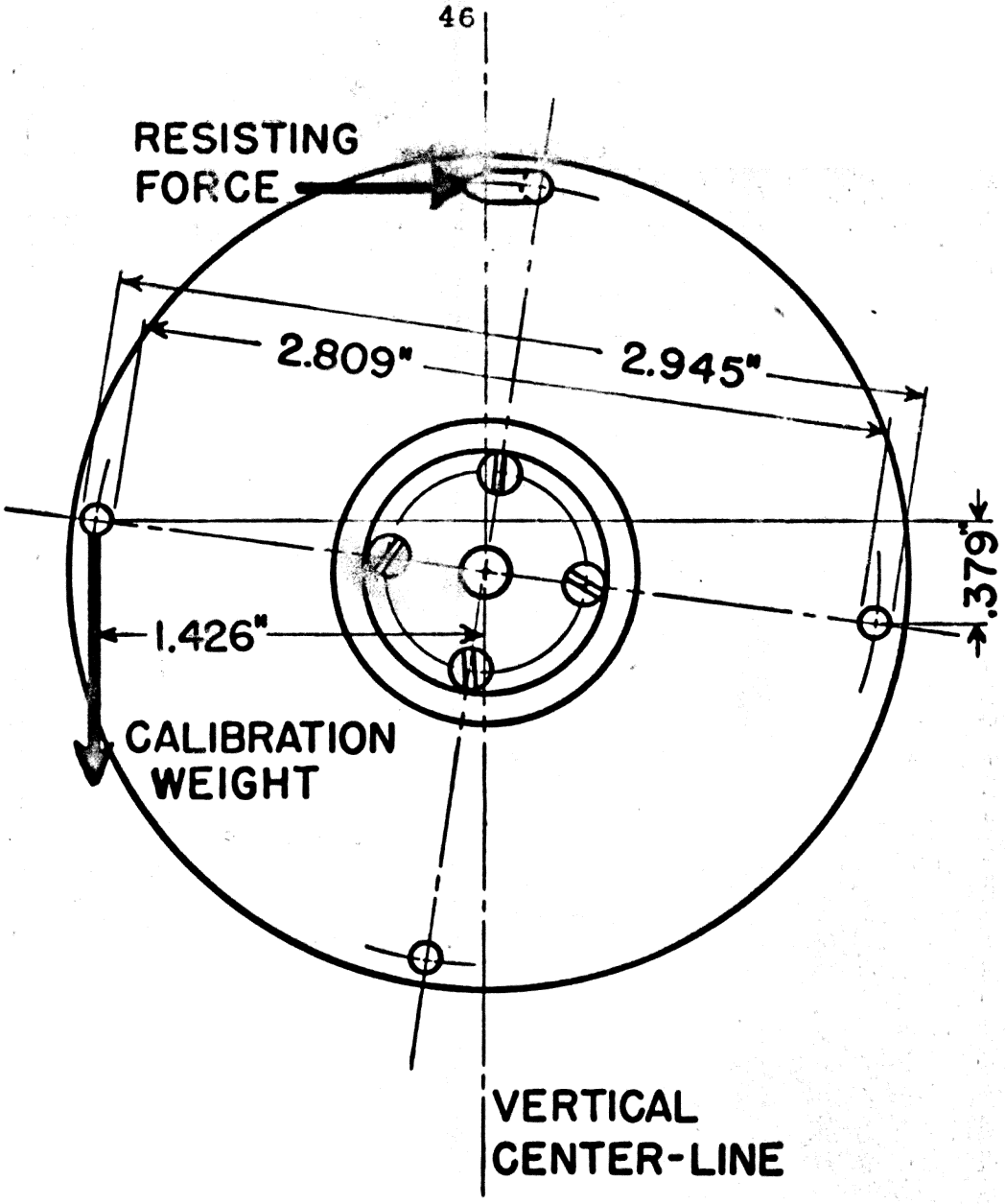

Fig. 10, Torque Calibration Geometry

the turbine shaft. The torque sensing beam was carefully aligned with the pointer on the torque disk. A zero reading was taken with the pointer and cup just out of contact to assure a zero strain condition. A trace of this zero strain condition was made by the recorder and may be seen in Fig. 11. Next the 1/2 g-cm calibration

Table 3 Torque Calibration Weights

Torque (g-cm)	1/2	1	2	3
Calibration Wt. (grams)	.1382	.2764	.5528	.8292

weight was hung on the calibration post. This produced a strain which was recorded as a trace across the abscissa of the calibration sheet. The weight was removed and the zero strain condition again checked to assure no zero drift error in the calibration. The procedure was repeated with each increasing size weight until the anticipated maximum value had been reached. In no case were less than three points used for calibration and in several cases as many as five points were used. In all calibration checks, the recorder showed the indicated torque to be a linear function of the actual torque. The question of the effective lever arm of the calibration weight changing during small angular displacements proved to be no problem. At the maximum designed deflection, the change of the effective lever arm increased by less than 0.0015 inches. This corresponded to a change of 0.01%

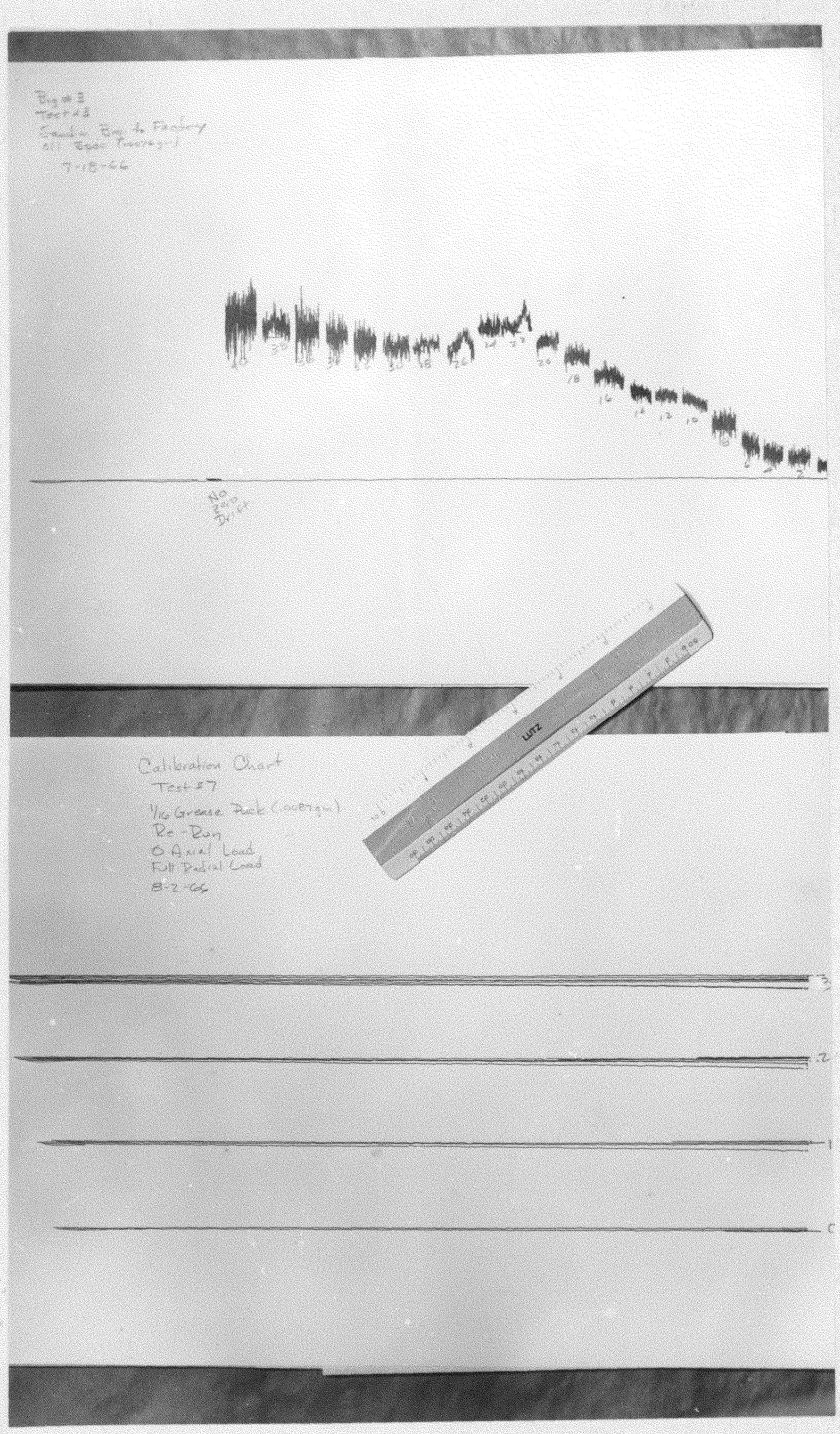

Fig. 11 Sample Torque and Calibration Charts

which was at least one order of magnitude smaller than the recorder error alone. An additional advantage of this type of calibration was that variables which might affect the torque disk deflection could be added to the system before calibration, and the plotter deflection would automatically compensate for this variable. More simply, the system was calibrated with the variable already included. This assured accurate results whether or not the axial loading system was used.

Testing

The speeds chosen for the final investigation were 1,000 rpm and then ranged from 2,000 rpm to 40,000 rpm in increments of 2,000 rpm. Lubrication specifications required that tests be run with factory oil, re-run without re-oiling, and re-oiled to factory specifications and re-run. Also, bearings were run with 1/16 and 1/8 grease pack and re-run without re-greasing. This constituted three oil test series and 4 grease test series for 0, 1/3, and 1/4 axial load ratios. After this group of tests was completed, it was necessary to run additional tests for axial load ratios of 1/2, 1, and 2. From the results of the first battery of tests, it was thought necessary to conduct tests for only the factory oil, 1/16 pack, and 1/8 pack conditions. This constituted a program of 30 test series. Each series

consisted of tests on six individual ball bearings. One group of six bearings was used for all twelve oil tests. Another group of six bearings was used for the nine 1/16 grease pack tests. A third group of six bearings was used for the nine 1/8 grease pack tests.

The test series were conducted such that all six bearings in a group would be run as originally lubricated before going on to the re-run tests. This generally allowed one to two hours between the original run and the re-run.

Data was taken from the original torque recordings by measuring the displacement from the zero base by means of a scale. The measurement was made to the nearest 0.05 inches. The torque reading in inches was converted to milligram-millimeters by multiplying by the appropriate scale factor as determined from the calibration charts for each test series.

D. LIST OF EQUIPMENT

1. Strain Gage Apparatus
   Model 1516
   Serial No. 132912
   Bruel and Kjaer, Naerum, Denmark
   Use: To measure and amplify the strain signal from the torque sensor.

2. Moseley Autograph
   Model 2 D X-Y Plotter
   Serial No. 1217
   F.L. Moseley Company, Pasadena, California
   Use: To record and amplify the torque signal from the strain indicator.

3. Electronic Counter
   Model 521 A
   Serial No. 120-04376
   Hewlett-Packard, Palo Alto, California
   Use: To count and display the speed of the air turbine.

4. Nullmatic Pressure Regulator
   Model 40 E 100
   Serial No. 3827M29661
   Moore Products Company, Philadelphia, Pennsylvania
   Use: To regulate the pressure of the inlet air to the air turbine.

5. Gramatic Balance
   Model H 5
   Serial No. 113176
   Mettler Instrument Corp., Hightstown, New Jersey
   Use: To weigh the test bearings and the amount of lubricant added to the bearings.

IV. DATA AND RESULTS

The data and results are presented in two forms. First, in Tables 4 through 13, the average torques and sample standard deviations for corresponding speeds are shown for various lubricant conditions and axial loads. Similar lubricant conditions are grouped together in order of increasing axial load. The second method of presentation is graphical. Figures 12 through 41 show average torques as a function of running speed. In addition, plus and minus values of sample standard deviation are shown as vertical lines at each of the data points. The average torques were calculated as the arithmetic mean of the torque values taken at each speed point.

$$\text{Average Torque} = \bar{X} = \frac{\Sigma X_i}{N}$$

The sample standard deviations were calculated according to normal statistical methods.[9]

$$\text{Sample Standard Deviation} = \sqrt{\frac{\Sigma X_i^2}{N} - \bar{X}^2}$$

Where: X_i are the individual torque values

N is the number of torque values summed

$\bar{X}$ is the average torque value

TABLE 4. FACTORY OIL (0.0076 gm), ORIGINAL RUN

	0 AXIAL LOAD		1/8 AXIAL LOAD		1/4 AXIAL LOAD	
SPEED (rpm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)
1000	975	1690	1052	2090	265	3593
2000	1690	3850	2130	3278	403	4718
4000	2450	5400	1710	4070	594	6214
6000	2520	6325	1880	4225	1210	6932
8000	2920	7575	1455	4070	1805	7612
10000	2850	8275	1276	5104	2400	9047
12000	2520	7750	1393	6160	2030	9301
14000	2390	7075	1200	6534	1660	8544
16000	2420	6675	1410	7194	1202	8640
18000	2130	6600	1428	8294	1040	9127
20000	2230	6650	1175	9702	1108	9320
22000	1990	6850	1422	10890	1443	10000
24000	1470	6700	730	11088	1202	10271
26000	1690	6950	823	10516	1780	8894
28000	1420	6800	1285	10120	1740	8369
30000	1195	6900	1640	10098	1348	8136
32000	1110	7350	1381	10010	1365	8039
34000	815	7825	380	9812	1365	8232
36000	637	8350	590	10230	1202	8330
38000	755	9025	1619	11154	1140	8563
40000	1048	10525	2100	12210	1165	9534

TABLE 5. FACTORY OIL (0.0076 gm), ORIGINAL RUN

	1/2 AXIAL LOAD		1 AXIAL LOAD		2 AXIAL LOAD	
SPEED (rpm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)
1000	1170	3453	637	3446	2370	5338
2000	1040	5892	1600	6052	3030	7963
4000	1600	7644	1445	9555	3960	12027
6000	1880	7963	1770	10234	3200	13459
8000	1910	7326	1380	10669	4320	14575
10000	1880	8322	1065	11785	3940	14575
12000	1495	9186	1490	10937	4070	14575
14000	1273	8509	1465	9874	4300	13613
16000	1992	9447	1160	9874	4350	13778
18000	1705	10561	1100	10879	3690	13696
20000	1395	11835	935	11893	3390	15128
22000	2060	12689	1060	12313	3630	15205
24000	2670	12262	1940	12154	3570	15607
26000	2915	11835	2190	12313	3260	16664
28000	3345	11039	2230	12581	3660	16664
30000	3160	10459	2240	12211	4100	17123
32000	3380	10511	2410	13058	3120	17995
34000	3000	10937	2080	14385	3690	18792
36000	2290	12371	2140	14696	1425	19352
38000	2230	12313	2110	13854	3570	18155
40000	2570	12529	2560	14014	2650	16326

TABLE 6. FACTORY OIL (0.0076 gm), RE-RUN

	0 AXIAL LOAD		1/8 AXIAL LOAD		1/4 AXIAL LOAD	
SPEED (rpm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)
1000	582	1331	516	2231	768	1708
2000	826	1784	650	2808	860	2048
4000	1286	1928	690	3096	842	2772
6000	1004	2387	705	3327	906	3284
8000	1300	3060	515	3731	830	3904
10000	1090	3228	577	4654	592	4500
12000	1558	3672	635	4750	1115	5056
14000	1765	3311	456	5231	748	5568
16000	1490	3470	674	5923	995	6484
18000	1645	3510	1155	6913	1080	7040
20000	1680	3849	860	8019	1302	7444
22000	1790	4207	1292	8288	1955	7596
24000	1955	4703	1292	8211	1955	7596
26000	1990	4691	1122	7673	2050	6699
28000	1890	4691	920	7557	2180	6441
30000	1830	5098	860	7596	1850	6252
32000	1825	6096	1155	7615	1653	6764
34000	1445	6628	1525	7480	1770	7636
36000	1270	8109	1537	7461	1808	8448
38000	2125	9434	1462	7923	1840	9516
40000	2695	11068	1415	8634	1885	10752

TABLE 7. FACTORY OIL (0.0076 gm), RE-OILED & RE-RUN

	0 AXIAL LOAD		1/8 AXIAL LOAD		1/4 AXIAL LOAD	
SPEED (rpm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)
1000	1175	3333	1240	3477	1530	5651
2000	1725	4571	849	4587	1850	6872
4000	2075	5924	1432	6191	2010	7825
6000	3560	6959	1700	6159	2385	8479
8000	2265	6065	1600	6001	2720	8361
10000	2090	7562	1625	7001	3060	9282
12000	1832	7704	1280	7795	3390	10146
14000	1650	7590	953	7795	2650	10085
16000	1470	7821	1335	7207	3200	10828
18000	1925	8221	1115	7477	2210	11156
20000	1640	8566	672	8731	1850	12227
22000	1365	8856	490	9382	375	12702
24000	1580	8597	575	9302	2110	12917
26000	1850	8684	427	8433	2110	12674
28000	1850	8339	695	7477	1490	12227
30000	2910	7704	850	6938	1755	10917
32000	1545	7849	346	6858	2160	10917
34000	3220	8021	968	6968	2505	12345
36000	2610	8797	105	7049	1265	11842
38000	2440	8970	347	7906	1215	12049
40000	2150	9660	1251	8921	800	12941

TABLE 8. 1/16 GREASE PACK (0.0087 gm), ORIGINAL RUN

	0 AXIAL LOAD		1/8 AXIAL LOAD		1/4 AXIAL LOAD	
SPEED (rpm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)
1000	697	8057	1870	6777	2825	8004
2000	1020	10058	2170	8390	2690	10278
4000	210	12286	2200	10397	3230	13613
6000	1253	13006	2680	12173	3080	15394
8000	1540	14280	2150	13613	2450	16448
10000	2490	15561	2815	13894	2320	18116
12000	3510	17342	3140	14674	2480	18896
14000	2150	18116	2720	15281	2815	19843
16000	2250	19843	2320	16281	3170	18676
18000	2930	20117	2530	16281	2065	18396
20000	1640	20677	2980	17896	2290	19176
22000	1915	20950	2380	18729	1920	19509
24000	1915	20730	3460	19116	2195	18783
26000	1840	21284	2770	19176	2060	19563
28000	2800	21117	3840	20450	2285	18616
30000	2445	20730	3710	18949	2440	18478
32000	1720	20844	3180	19009	2520	18843
34000	1220	22011	4260	20510	2360	19616
36000	1650	19563	5740	20450	1890	18449
38000	1775	19676	5630	21011	1835	18343
40000	3000	21177	7800	23452	2710	21451

TABLE 9. 1/16 GREASE PACK (0.0087 gm), ORIGINAL RUN

	1/2 AXIAL LOAD		1 AXIAL LOAD		2 AXIAL LOAD	
SPEED (rpm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)
1000	1745	4948	2630	7084	3030	9528
2000	2355	7708	2880	10340	3650	14160
4000	2585	10844	3680	15792	3460	20116
6000	2630	11528	5140	18296	4400	23816
8000	3190	12408	3860	19424	3500	25004
10000	3575	13475	3880	20439	3915	27072
12000	3680	14852	3410	20928	3220	27320
14000	2520	16859	4220	21146	3565	26192
16000	2195	18236	3430	21432	4520	24876
18000	2740	18739	2940	21304	4080	24004
20000	3155	20432	3320	23252	2830	25132
22000	2565	20552	2280	23816	3320	24124
24000	3470	20055	2165	23748	976	24944
26000	3610	19364	2500	23936	2400	25440
28000	4995	17672	2990	23748	1680	26259
30000	3995	16108	3420	23124	1680	26696
32000	3300	16168	3810	22808	1810	28012
34000	3530	16792	3710	21680	1680	30456
36000	4620	16920	3700	21808	2460	32396
38000	4420	16168	3100	20552	2660	31396
40000	3190	16980	3270	21620	3700	28892

TABLE 10. 1/16 GREASE PACK (0.0087 gm), RE-RUN

	0 AXIAL LOAD		1/8 AXIAL LOAD		1/4 AXIAL LOAD	
SPEED (rpm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)
1000	1340	6276	2170	7837	1515	7891
2000	1540	8891	2370	10392	1640	9611
4000	1930	11612	3110	13560	1915	11779
6000	1870	13894	2890	15448	1350	13447
8000	2480	16115	3230	16728	1420	14894
10000	3260	17115	3200	16948	2650	16115
12000	2850	17782	3180	16281	2370	17562
14000	2310	17562	2270	14781	2480	17229
16000	2460	18009	2270	14561	1710	17115
18000	1680	18896	2150	15114	1895	17842
20000	1570	20784	2270	16342	1955	19116
22000	2770	17675	2770	17675	1180	19509
24000	3650	21844	3000	17949	2020	19677
26000	4100	22398	2810	18616	1720	19116
28000	2400	20950	3560	18116	2120	19283
30000	3290	20063	3560	18116	1585	18282
32000	3010	19616	3720	18175	1990	18949
34000	1930	20230	2040	18449	2665	20230
36000	1615	17675	2150	16948	1640	18116
38000	1420	17895	845	18282	1500	18676
40000	2075	19509	1060	19283	1305	21617

TABLE 11. 1/8 GREASE PACK (0.0175 gm), ORIGINAL RUN

	0 AXIAL LOAD		1/8 AXIAL LOAD		1/4 AXIAL LOAD	
SPEED (rpm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)
1000	5100	14903	2795	14742	3130	11662
2000	4510	16346	3740	18869	5110	14679
4000	5770	19348	5250	22903	4900	18564
6000	5070	20672	5720	25883	5660	21106
8000	4300	20999	4610	28034	5610	23486
10000	4350	22115	3580	29305	5880	25780
12000	5700	23229	2910	31243	1450	30302
14000	6080	24845	3040	30716	1730	31892
16000	6400	27557	4320	30358	7150	29436
18000	6260	29326	3250	30358	3130	30149
20000	5800	30614	4840	32643	2130	31816
22000	5060	30922	4820	32643	3580	32444
24000	5370	32056	4360	33349	4750	33796
26000	4830	30768	5150	32200	4420	33482
28000	5020	31095	4390	30537	5550	32368
30000	4300	30287	4810	29042	3630	30388
32000	5440	29806	2999	28252	4290	29436
34000	3320	33498	4255	29305	3130	32444
36000	5440	32056	2425	28084	4020	27294
38000	9400	31979	3670	28778	3640	24752
40000	11700	33653	5160	30189	1505	25942

TABLE 12. 1/8 GREASE PACK (0.0175 gm), ORIGINAL RUN

	1/2 AXIAL LOAD		1 AXIAL LOAD		2 AXIAL LOAD	
SPEED (rpm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)
1000	2020	6900	4080	14800	5250	20000
2000	3050	10700	3160	20200	5250	25670
4000	2980	14400	2900	28100	4500	32580
6000	3760	17400	4430	32000	2880	40050
8000	4430	20800	2450	34200	2590	41750
10000	3900	23600	4700	34800	3420	46330
12000	4030	25500	3000	34900	4830	47420
14000	4590	27900	2450	35200	6260	46920
16000	2640	28200	5480	35800	6700	44330
18000	3320	28500	5480	34800	5170	43750
20000	3130	29700	6000	36500	4380	44920
22000	3440	32700	5390	37600	4090	44830
24000	4130	33500	4360	38400	4830	44830
26000	4340	31900	4700	39700	4280	48000
28000	4990	30700	5660	39400	4820	50500
30000	6010	29800	5830	40000	5550	50670
32000	6900	29400	7280	41300	5480	53250
34000	6800	30800	5920	40400	5630	56670
36000	5080	32400	5100	40500	5320	56000
38000	5570	29400	4700	39300	7410	55500
40000	4670	28400	6000	34100	4090	51750

TABLE 13. 1/8 GREASE PACK (0.0175 gm), RE-RUN

	0 AXIAL LOAD		1/8 AXIAL LOAD		1/4 AXIAL LOAD	
SPEED (rpm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)	S. S. DEVIA. (mg-mm)	AVG. TORQ. (mg-mm)
1000	5030	13317	2510	12080	5500	10330
2000	6140	18062	2970	15920	6450	13250
4000	8460	24645	5180	19850	6050	17000
6000	8960	28402	6310	23000	6590	20250
8000	10120	30799	9050	25670	6050	23580
10000	10330	31139	9560	26830	4930	26080
12000	9960	30799	9310	27420	4370	27920
14000	8900	29301	8350	27080	4810	28580
16000	7540	27972	7390	26000	5410	27330
18000	6970	28551	6790	26830	6230	27640
20000	6989	29471	7520	28170	6740	28080
22000	9060	29800	7650	29250	6850	27830
24000	6940	30389	8400	30580	6900	29330
26000	7440	30719	7420	30670	7600	30080
28000	7150	30549	7790	29580	6910	29920
30000	6450	30219	7550	28410	6990	29000
32000	6040	29221	5400	28080	5380	29000
34000	6320	29970	6120	28670	4910	34170
36000	6320	29970	3820	27240	4910	30000
38000	6440	29800	4490	26970	4200	30500
40000	6340	31638	4960	26670	3940	32000

V. DISCUSSION OF RESULTS

The raw data, as shown in Fig. 11, was physically measured with a scale and converted to a torque value through multiplication by the appropriate constant, as determined from the calibration chart for the particular test. By observation of many test series, it was found that the shape of the average torque curve could be varied slightly by the speed with which the test was conducted. By conducting the test slowly, which gave the lubricant ample time to be pushed into a low resistance configuration, the average torques tended to be slightly lower. This was particularly true in the initial low speed phases of the test. The prospect of setting up a test criteria for the rate at which the speed steps were traversed seemed artificial and arbitrary. It was thought that in general practice, ball bearings are brought up to their operating speeds as rapidly as the driving source will permit, which in most cases is more rapidly than the speed ranges were traversed during the tests. For this reason, it was considered most realistic to traverse the speed ranges as rapidly as speed stabilization within 200 rpm at the data points would permit. It was found that the response lag due to the viscous damper was less than two-seconds, so that the only criteria limiting the speed at which the tests were conducted was the speed

stabilization. Thus, a typical speed traverse would require 10 to 12 minutes.

The accumulated data from the author's tests could be compared in two basic ways. First, the effect of re-runs and re-oiling could be compared for fixed values of axial loading. Secondly, the effect of axial load for similar lubricant conditions could be compared. It was the intention of this study to consider primarily the effect of axial loading on the running torque. It should be remembered during comparisons that the radial load was held constant at 46.78 grams through all the tests. Because the radial load was constant, it is not mentioned in the discussion, or shown in the data tables or on the graphical presentations. The axial load, which did vary, is referred to as some proportion of the radial load.

Graphs for comparing the factory oiled bearings are shown in Figs. 12 through 17. The plots of the average running torque show that an increase of axial loading tended to create slightly higher running torques in the mid-speed range for axial loads of 1/4 or less. The starting and high-speed torques seem to take on approximately the same values for low-axial loads. However, as the axial load was increased to 1/2 and above, the entire average torque curve was raised. The 1/2 axial load caused approximately a 20% increase of the average running torque. The 1 axial load increased the average

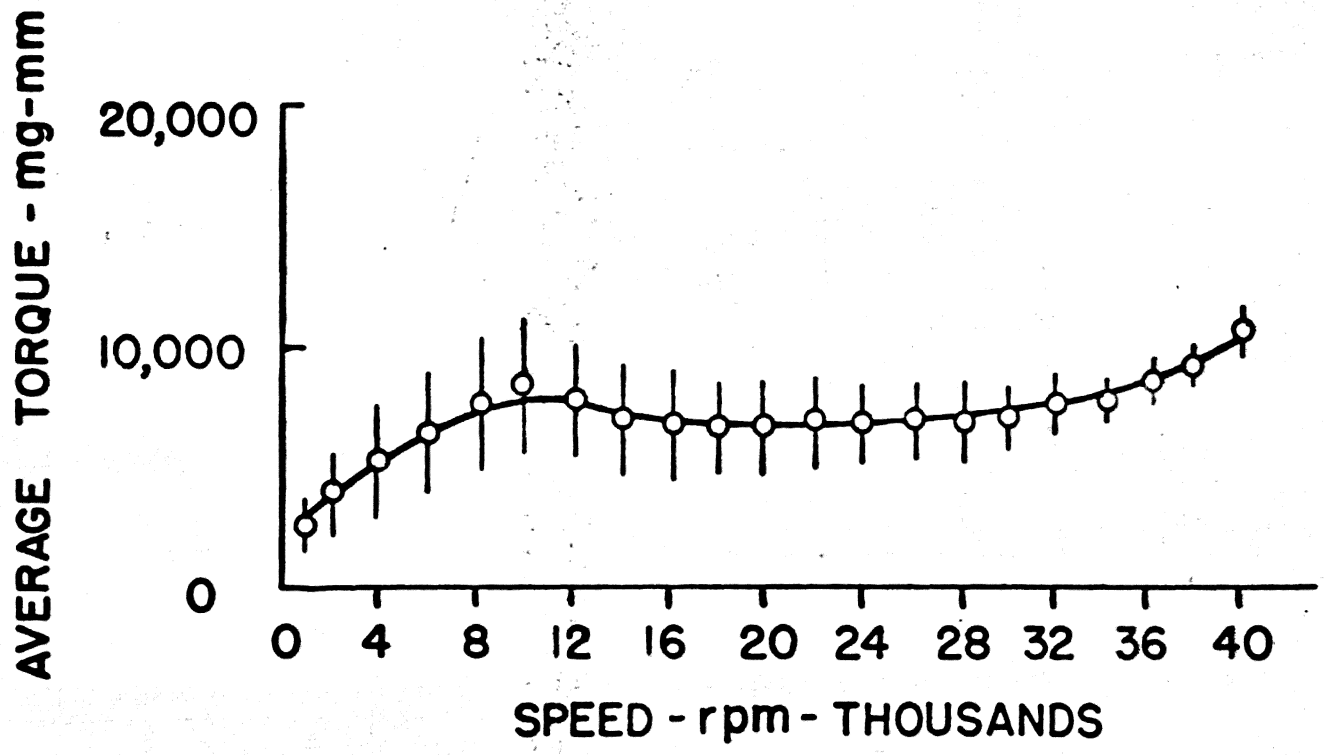

Fig. 12 Factory Oil, Original Run, 0 Axial Load

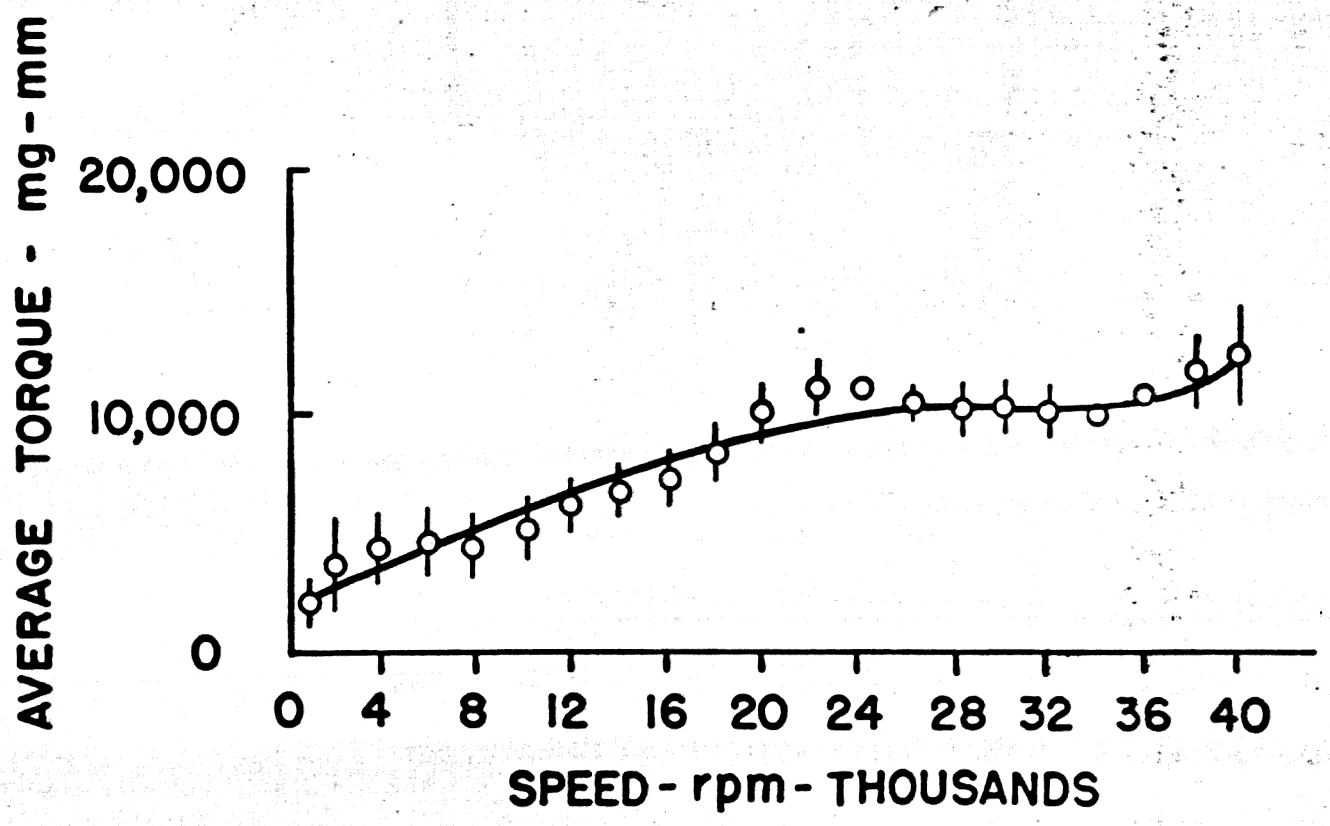

Fig. 13 Factory Oil, Original Run, 1/8 Axial Load

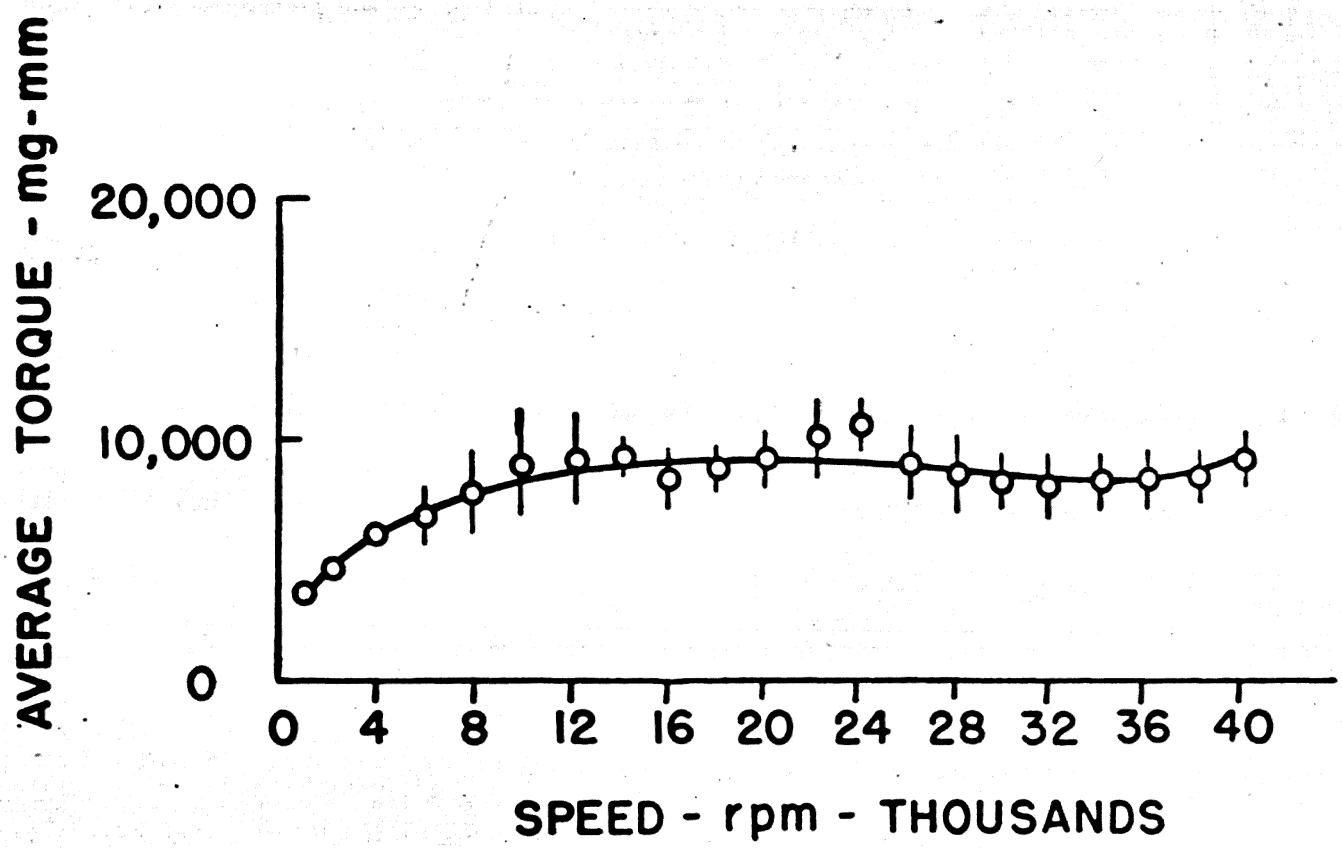

Fig. 14 Factory Oil, Original Run, 1/4 Axial Load

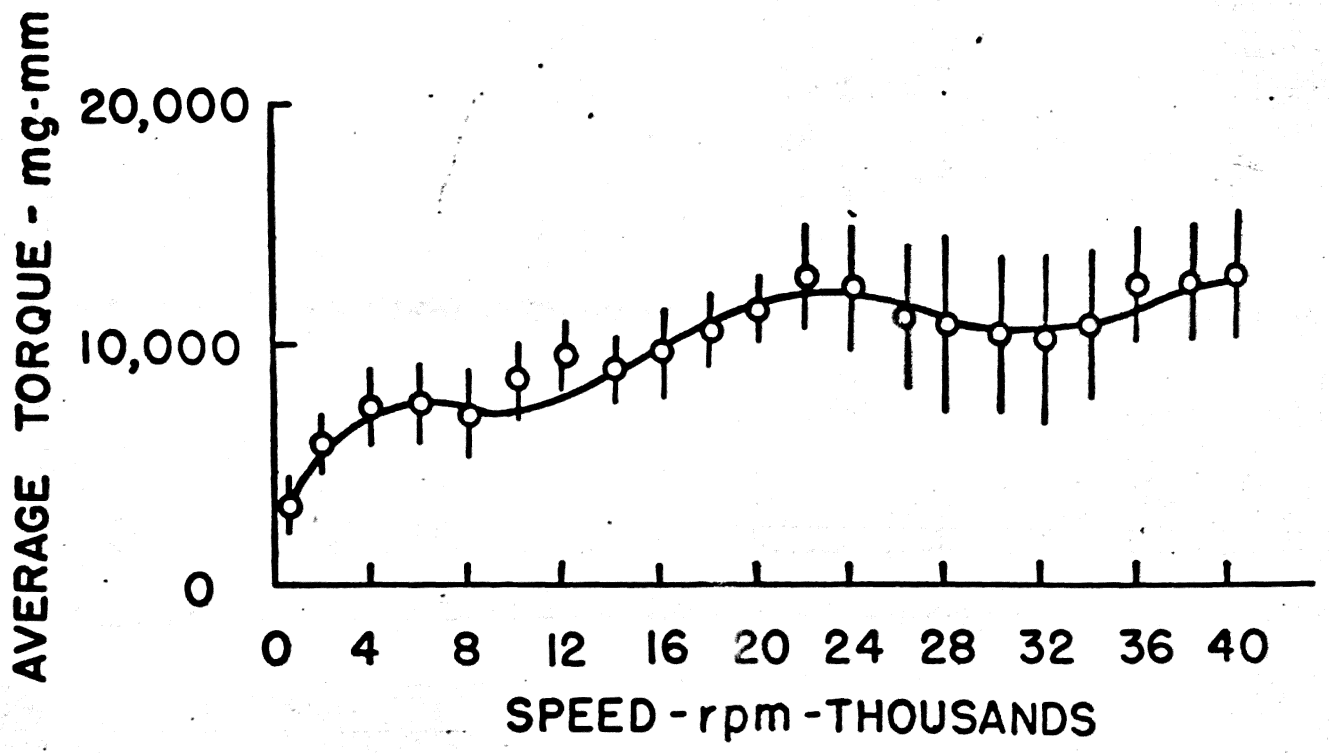

Fig. 15 Factory Oil, Original Run, 1/2 Axial Load

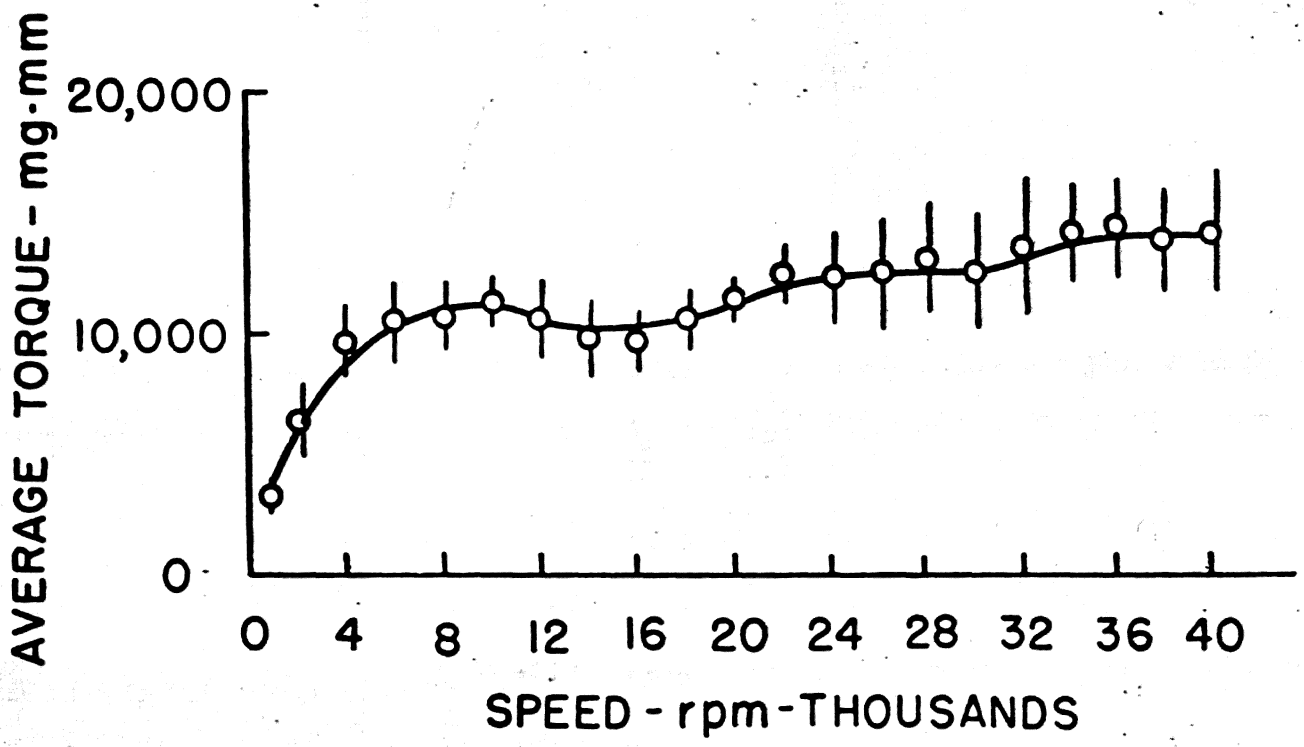

Fig. 16 Factory Oil, Original Run, 1 Axial Load

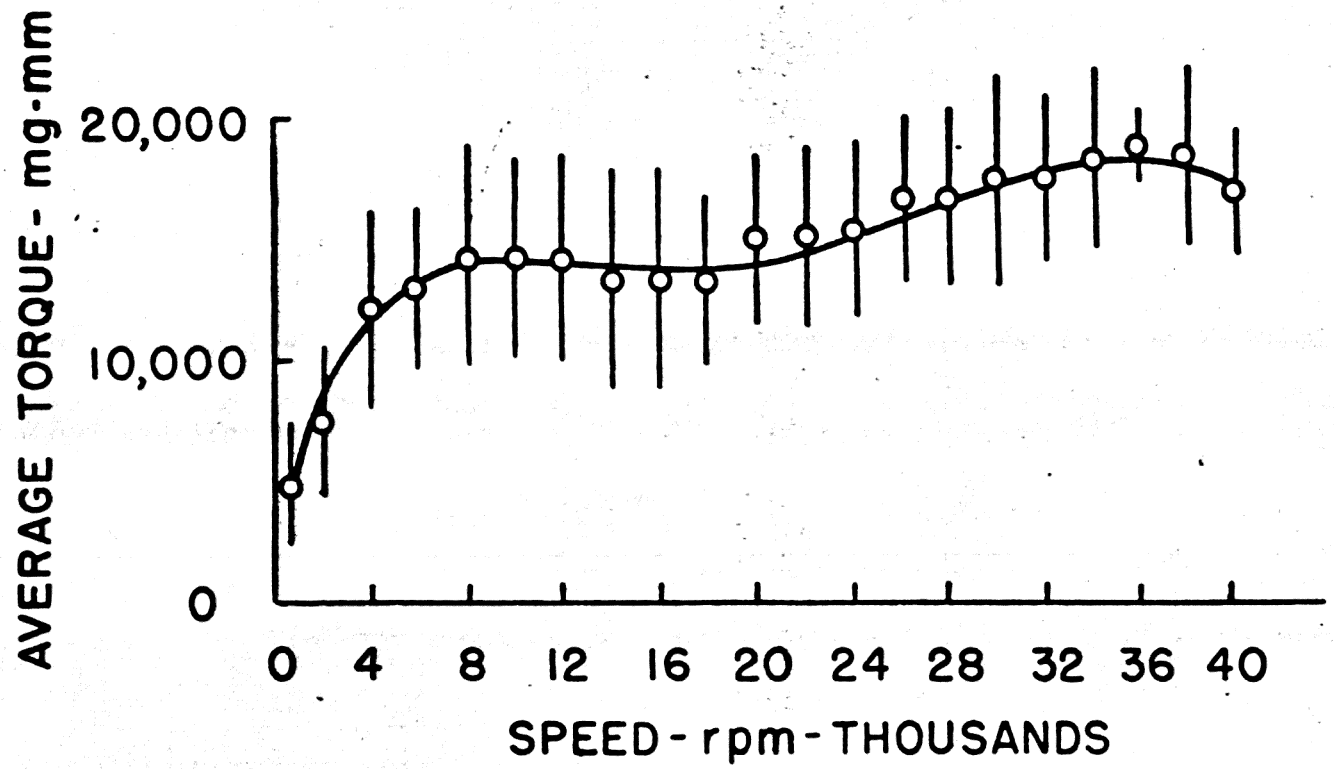

Fig. 17 Factory Oil, Original Run, 2 Axial Load

torque by about 50%,and the 2 axial loading approximately doubled the zero axial loading average torque.

The re-run factory oiled bearings may be compared using Figs. 18 through 20. The results are similar to the original run bearings for axial loadings of 1/4 or less. The bearings were not re-run for higher axial loads.

Torque plots for re-oiled and re-run bearings are shown in Figs. 21 through 23. Only the 1/4 axial loading showed any torque increase, and this amounted to about 20%. It was not known if the increase was caused by contamination during re-oiling or by the axial load alone, or some combination of the two. Re-oiled bearings were not tested at axial loads above 1/4.

From a review of the data shown in Figs. 12 through 23,from another perspective, the average running torques for re-run bearings is seen to be lower than those of the original run bearings for comparable axial loads. Also, re-oiling caused the running torques to be slightly higher than torques for either the original runs or the re-runs with similar axial loading. Because of this consistant torque increase, there is a good possibility of some degree of contamination occuring during re-oiling.

The 1/16 grease pack tests were conducted similarly to the oil tests in that the entire axial loading range was used only with original runs while re-runs were

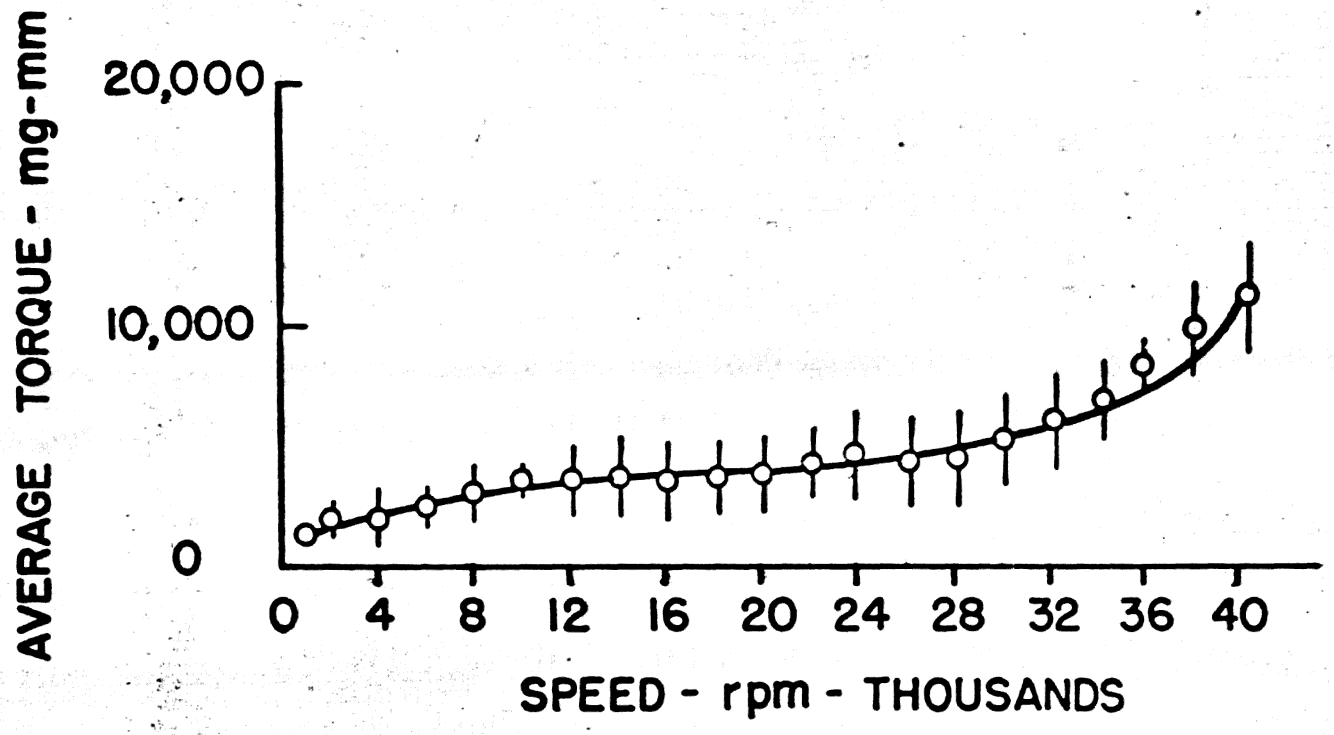

Fig. 18 Factory Oil, Re-run, 0 Axial Load

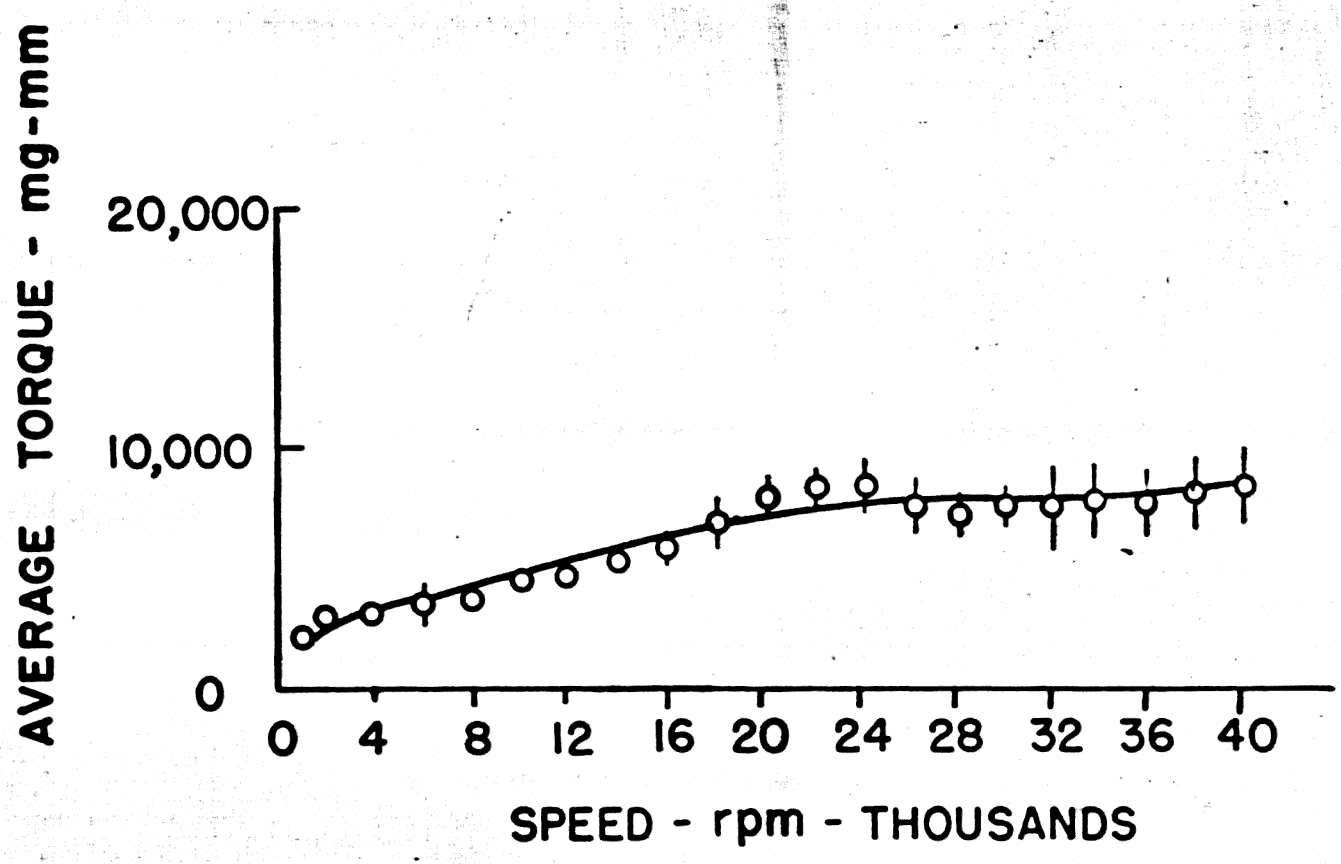

Fig. 19 Factory Oil, Re-run, 1/8 Axial Load

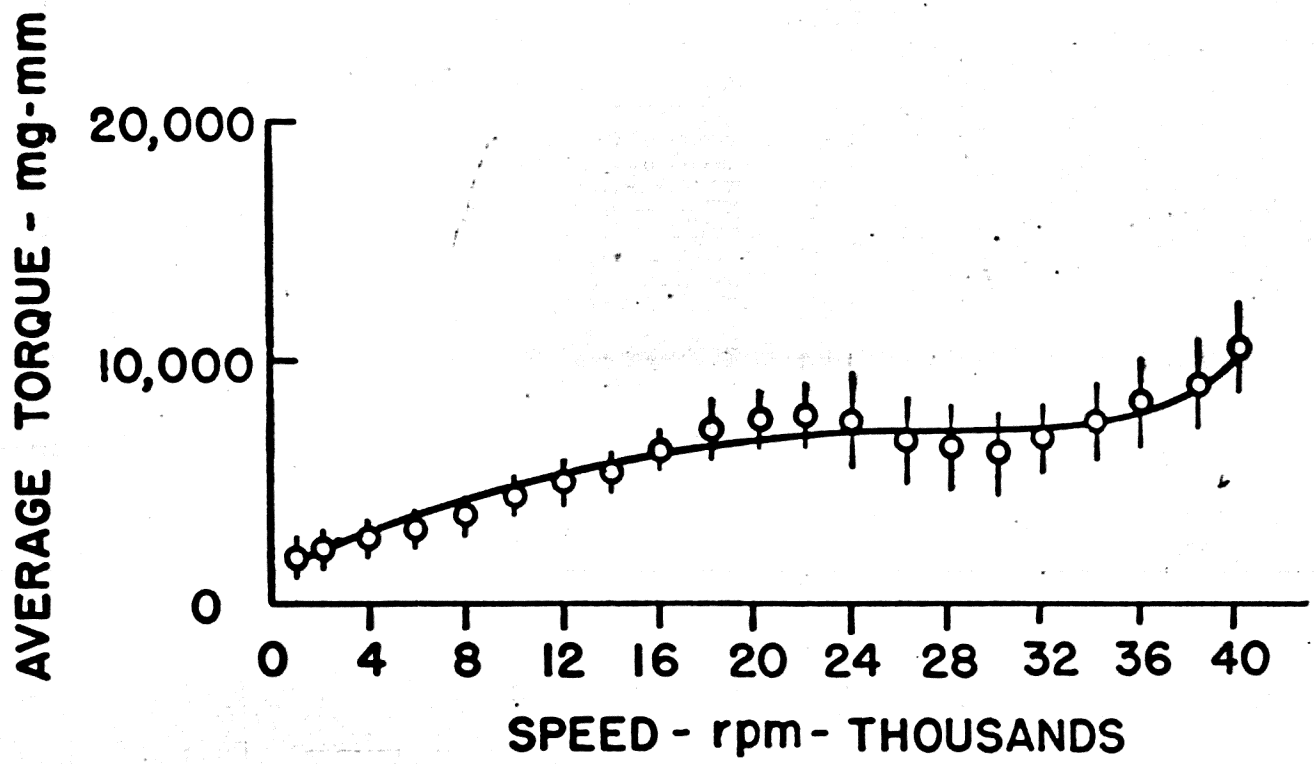

Fig. 20 Factory Oil, Re-run, 1/4 Axial Load

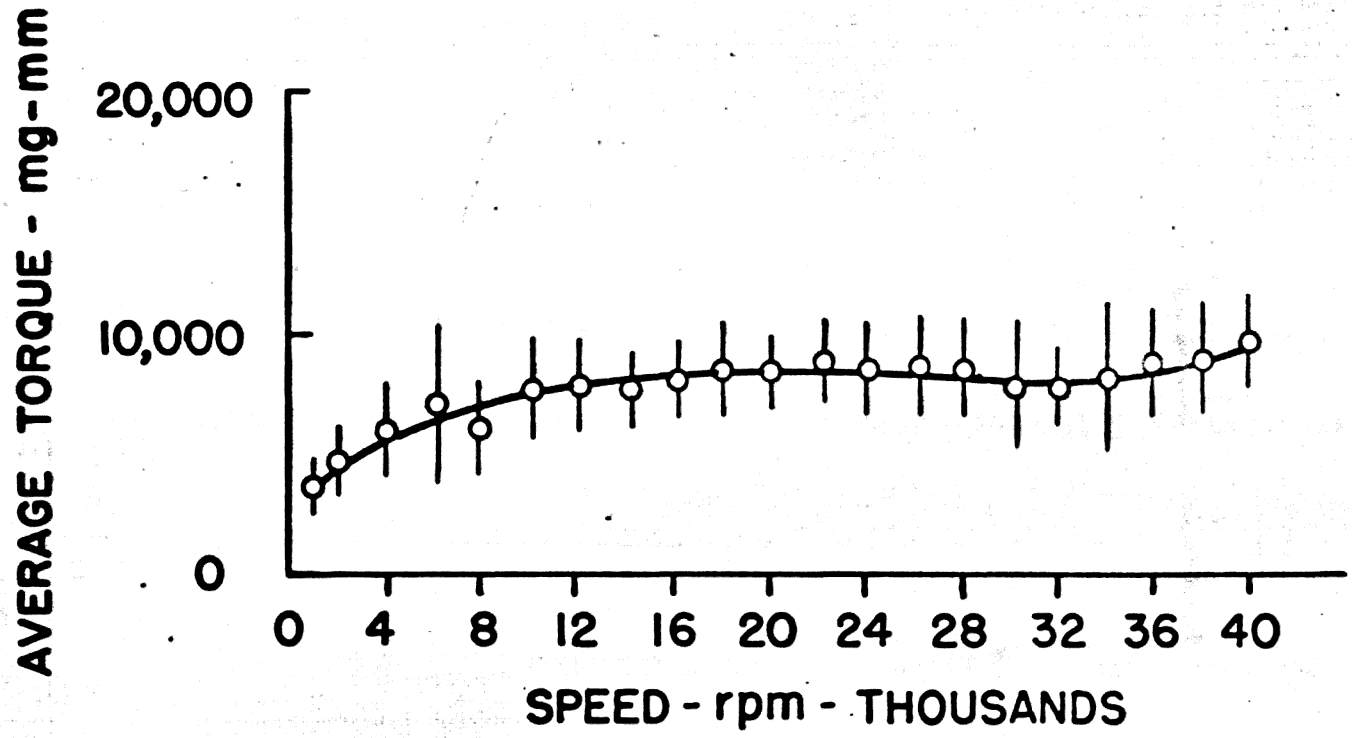

Fig. 21 Re-oil, Re-run, 0 Axial Load

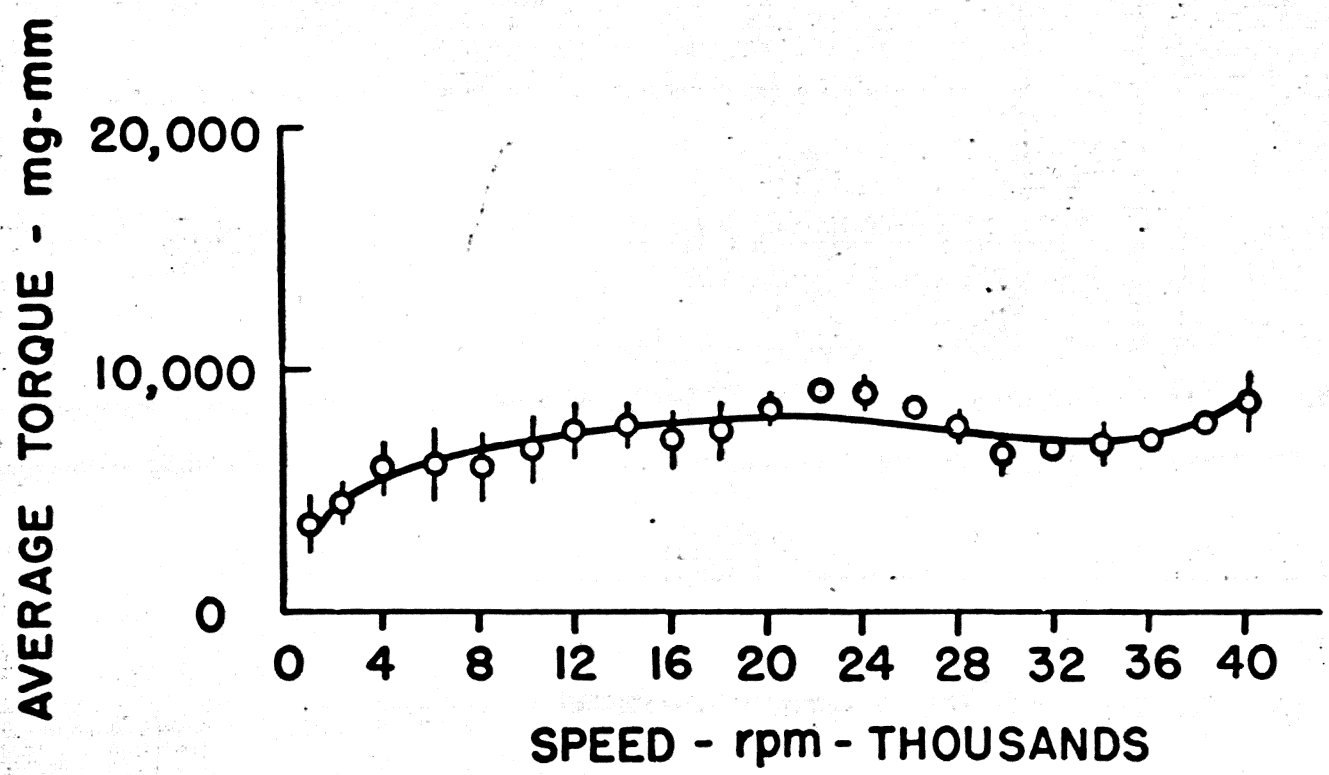

Fig. 22 Re-oil, Re-run, 1/8 Axial Load

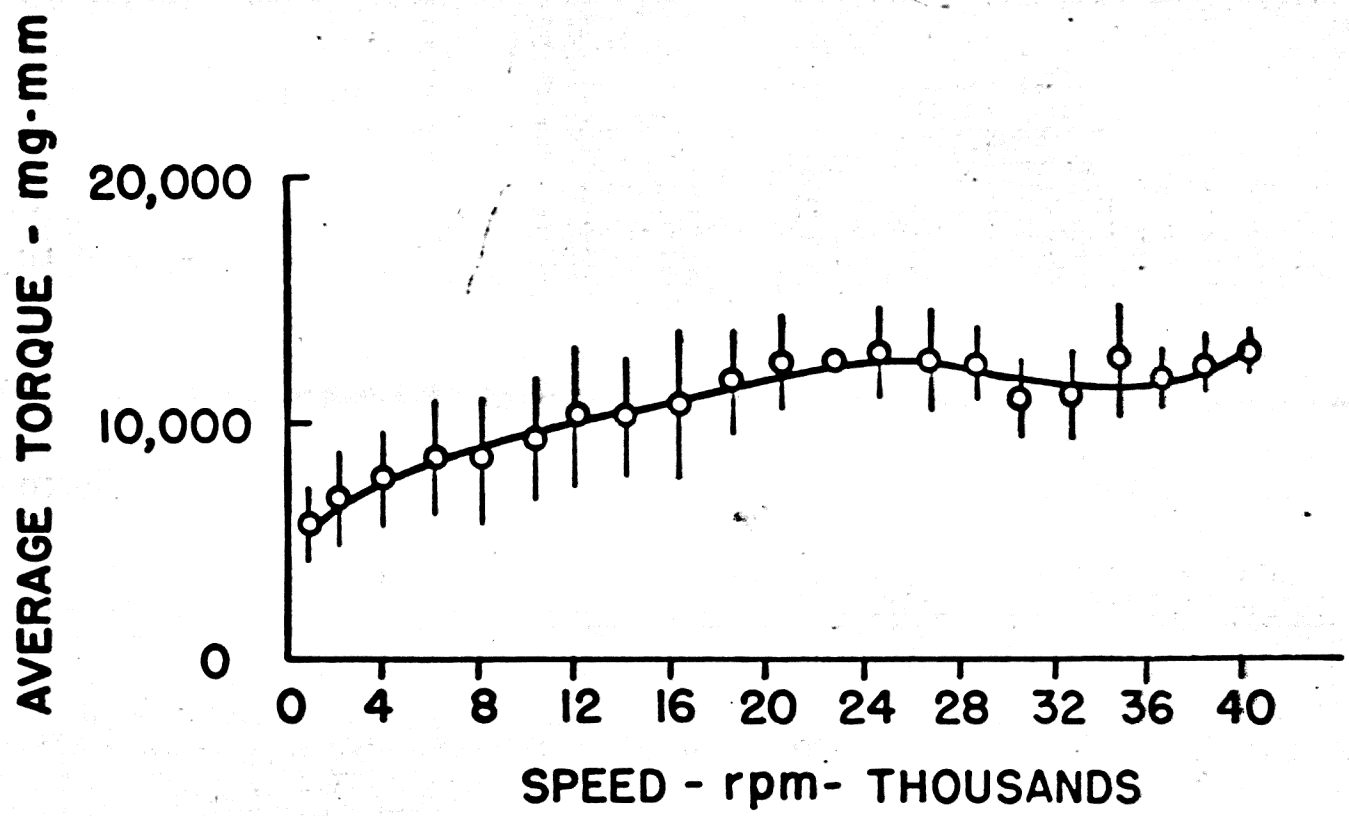

Fig. 23 Re-oil, Re-run, 1/4 Axial Load

conducted with axial loadings up to 1/4.

The 1/16 grease pack results are shown in Figs. 24 through 29 for the original runs. Bearings with light axial loads showed running torques very close to each other. However, an upward shift of the average torque was noticeable when the axial load reached 1. Here, the torque curve exceeded the zero axial load curve by approximately 20 %. For an axial load of 2, the average torque was about 40% higher than the zero axial load torque curve.

The curves for the 1/16 grease pack re-run bearings are shown in Figs. 30 through 32. The curves were very close over the entire speed range for the axial loads tested. The only exception was that the zero axial load curve was slightly higher in the mid-speed range than curves for the 1/8 and 1/4 axial loadings. Considerable overlapping of the sample standard deviations gives some indication as to the agreement among the curves.

In comparison of the original runs against the re-runs, the re-runs exhibit higher running torques in the lower speed ranges seemingly regardless of axial loading. This is not quite what would be expected, especially with grease lubricated bearings. It would seem that the torques would be slightly higher when first run while the lubricant was being displaced from the raceways. Lower torques should result in subsequent runs.

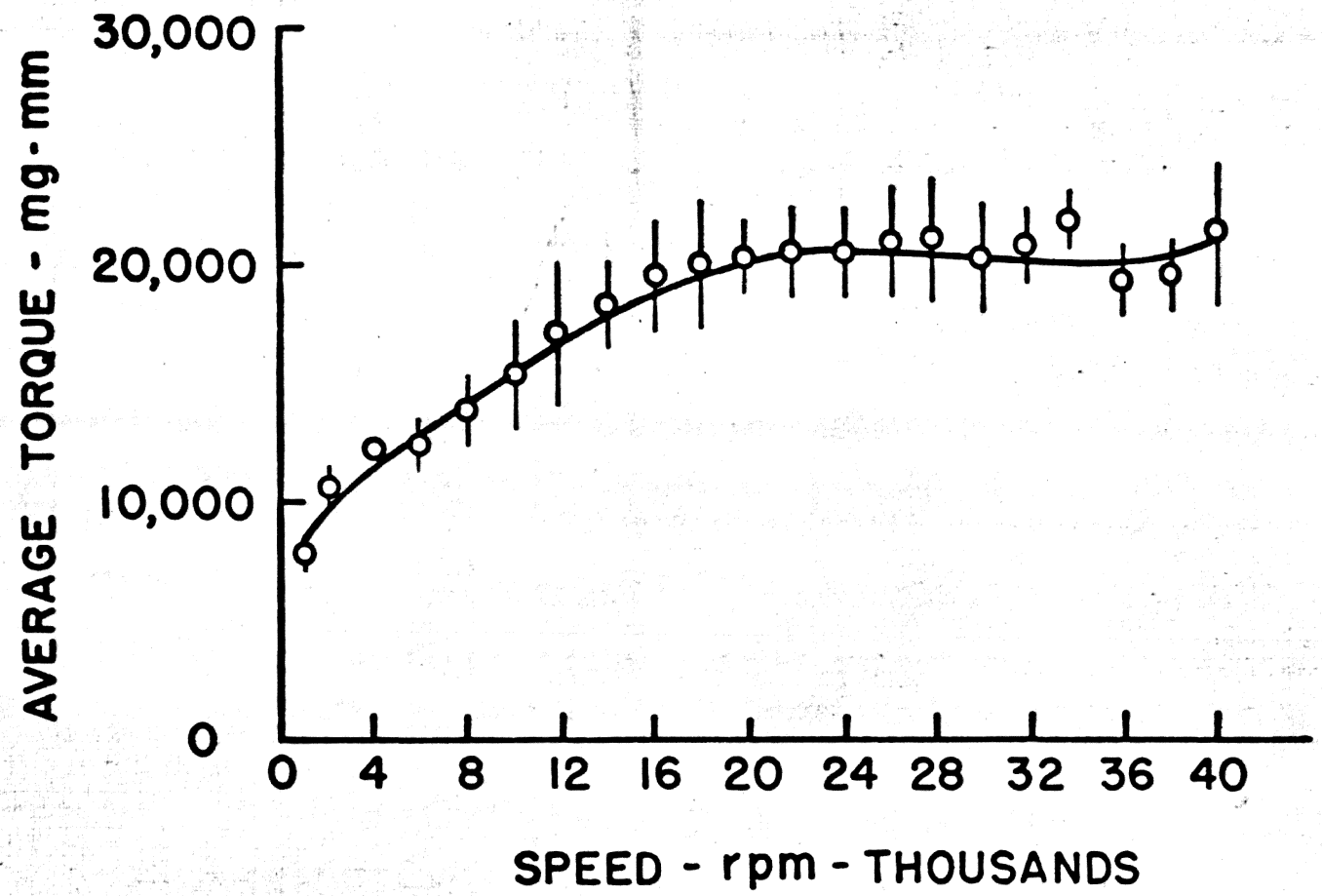

Fig. 24 1/16 Grease Pack, Original Run, 0 Axial Load

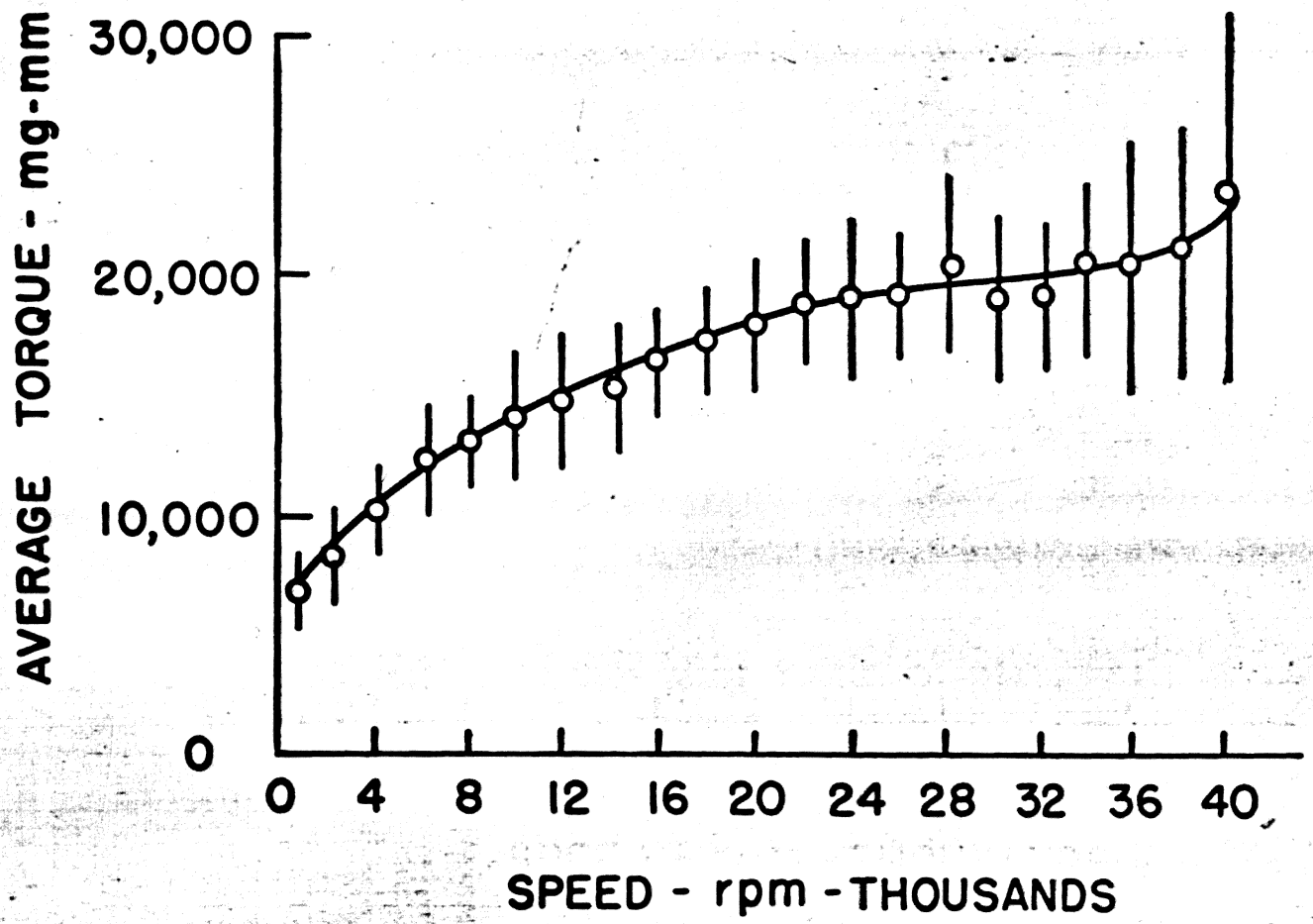

Fig. 25 1/16 Grease Pack, Original Run, 1/8 Axial Load

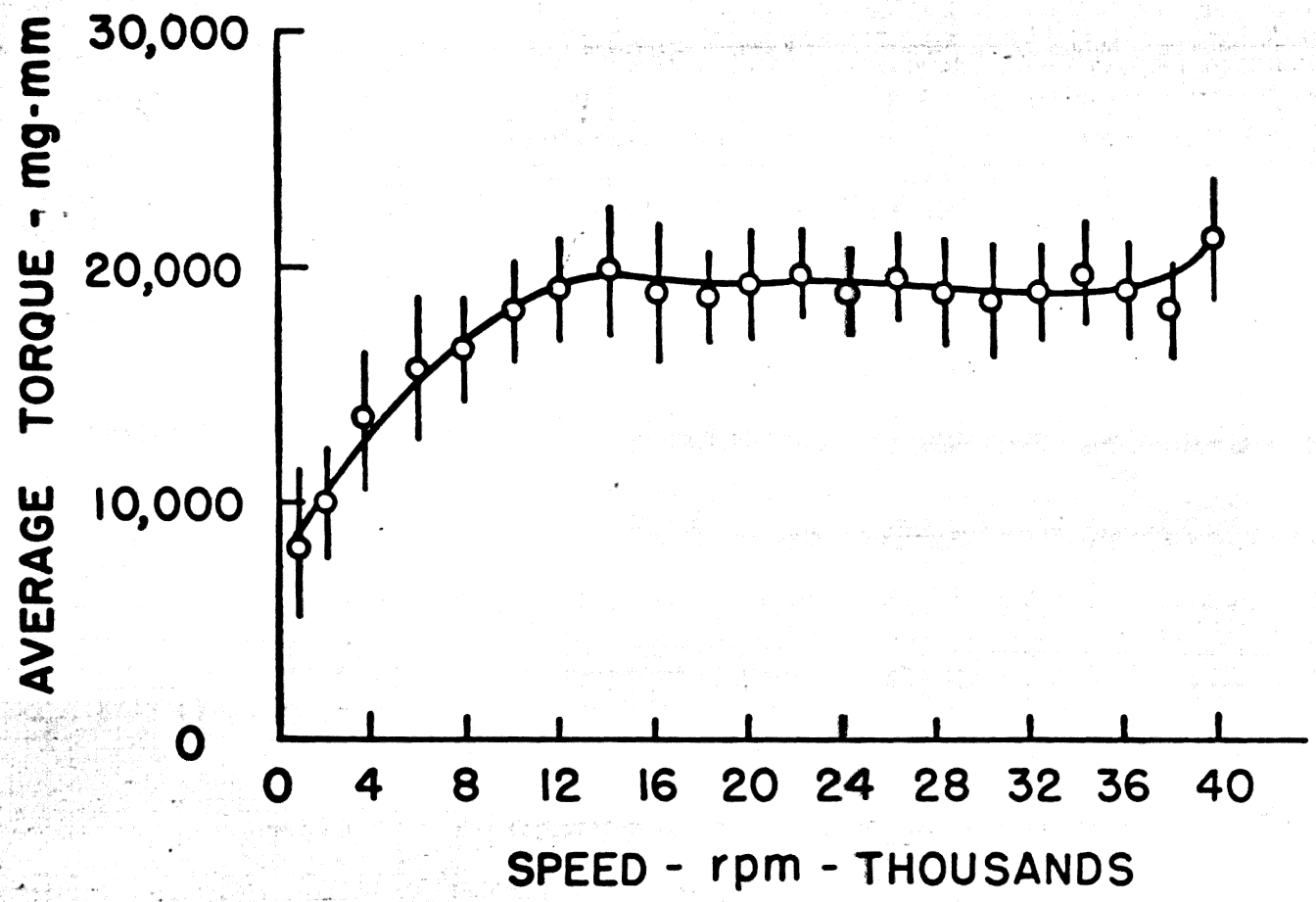

Fig. 26 1/16 Grease Pack, Original Run, 1/4 Axial Load

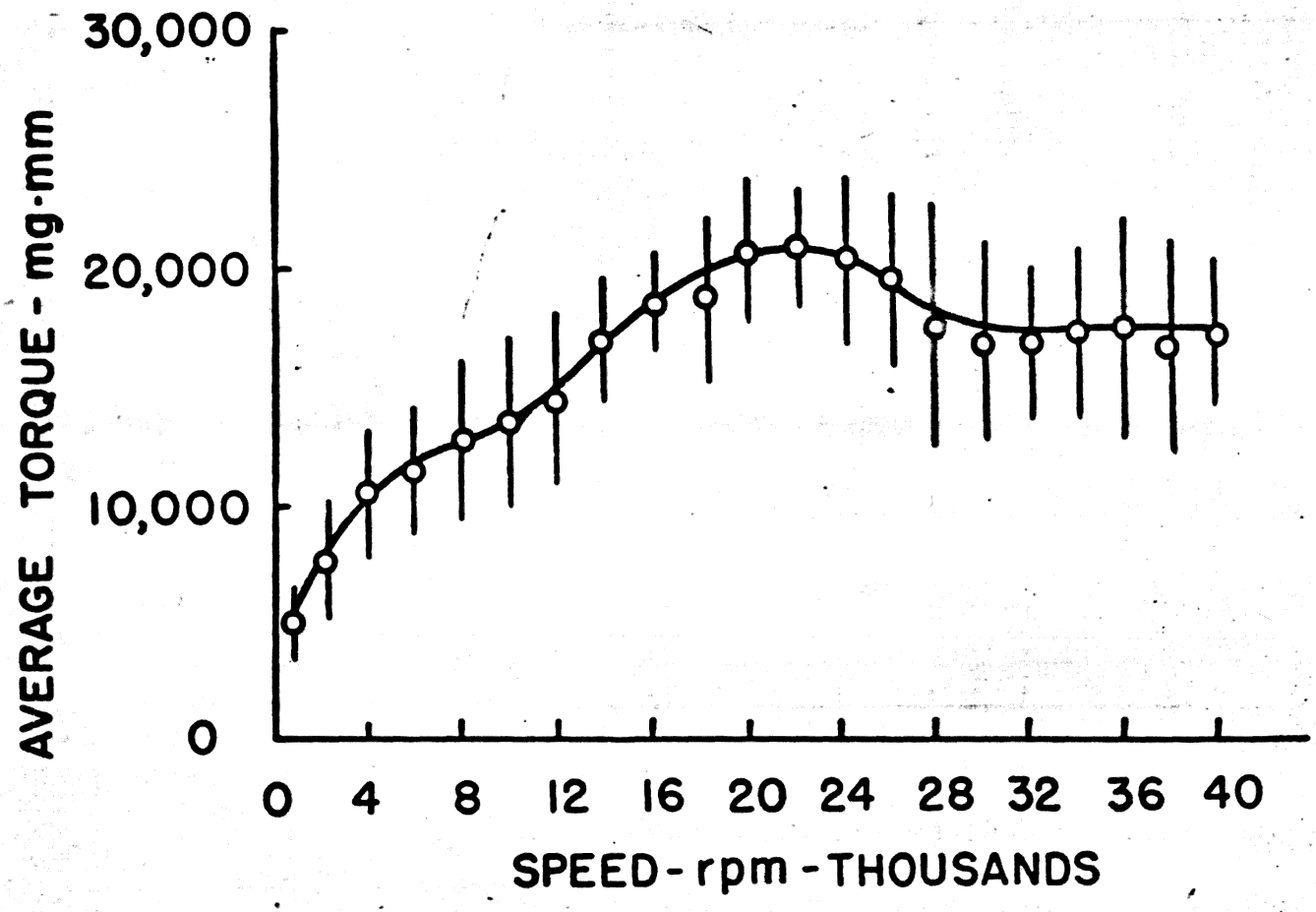

Fig. 27 1/16 Grease Pack, Original Run, 1/2 Axial Load

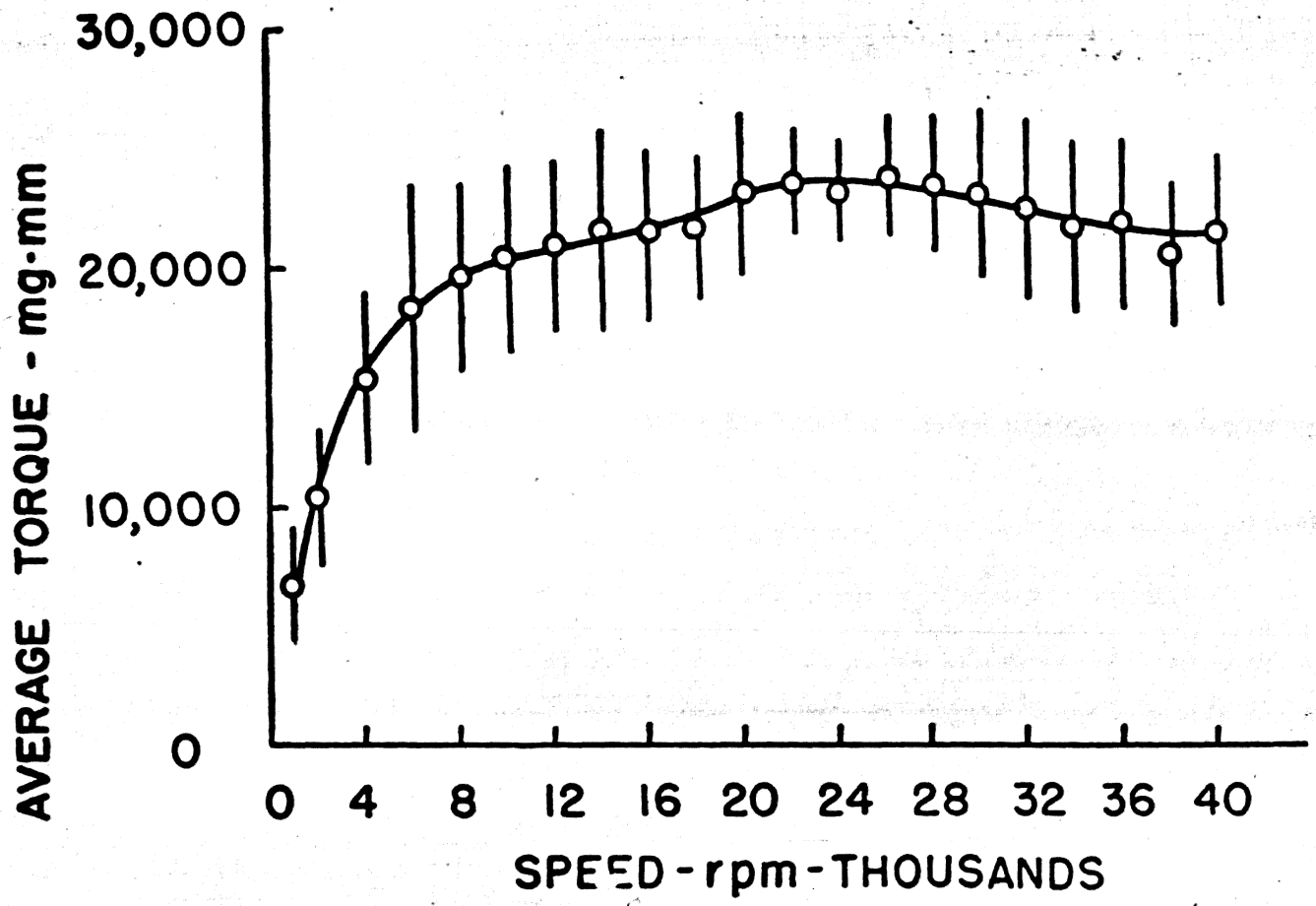

Fig. 28 1/16 Grease Pack, Original Run, 1 Axial Load

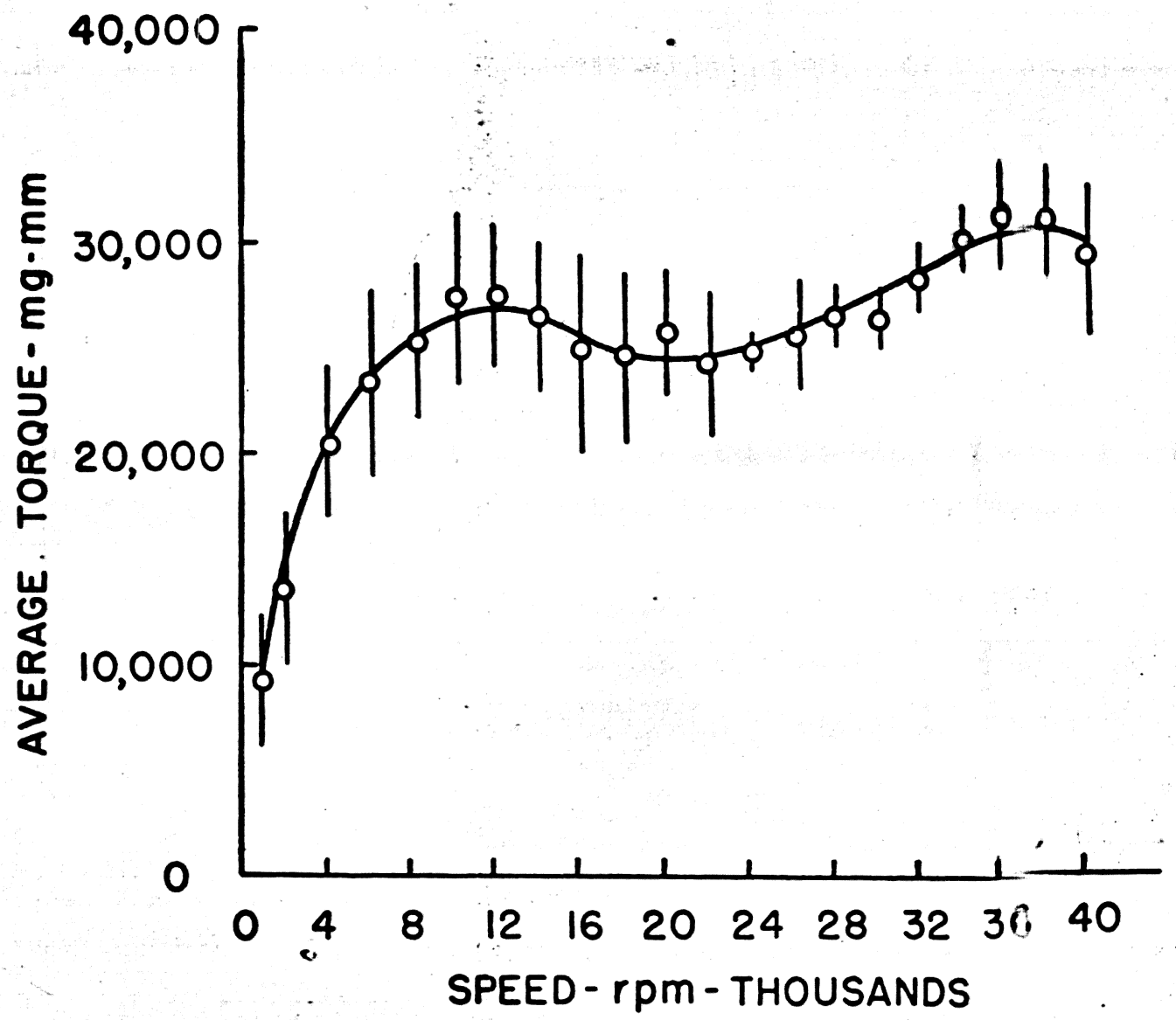

Fig. 29 1/16 Grease Pack, Original Run, 2 Axial Load

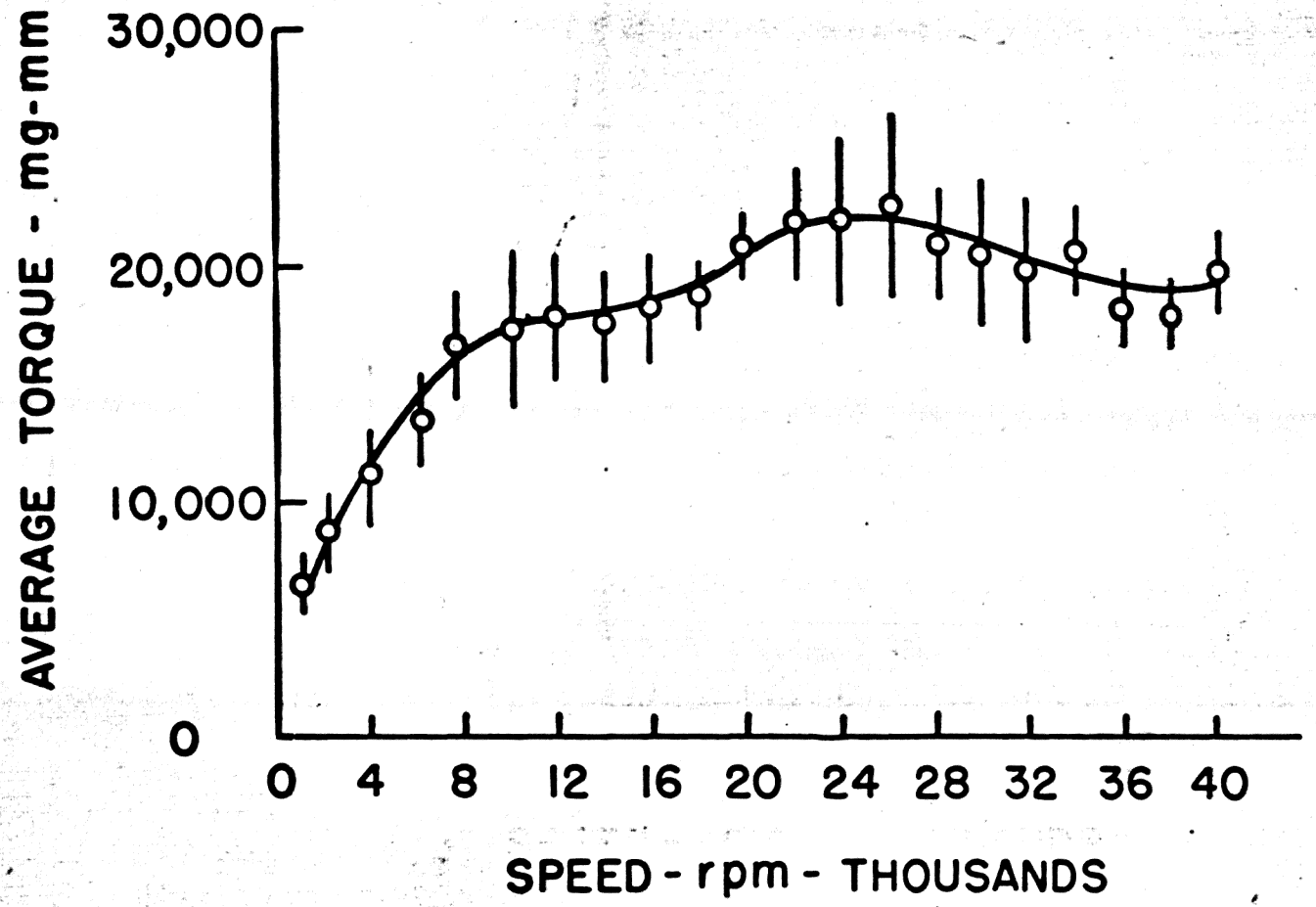

Fig. 30 1/16 Grease Pack, Re-run, 0 Axial Load

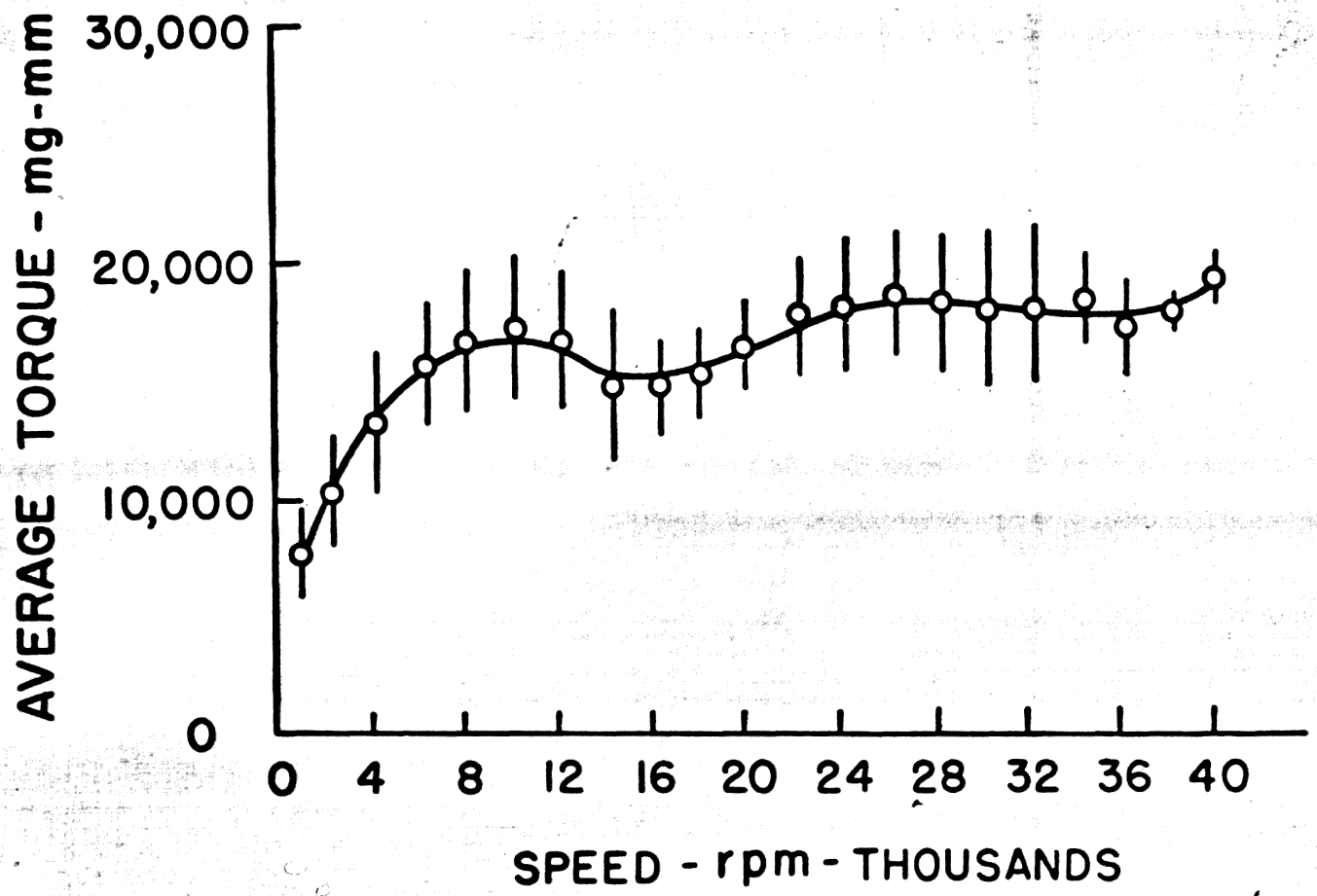

Fig. 31 1/16 Grease Pack, Re-run, 1/3 Axial Load

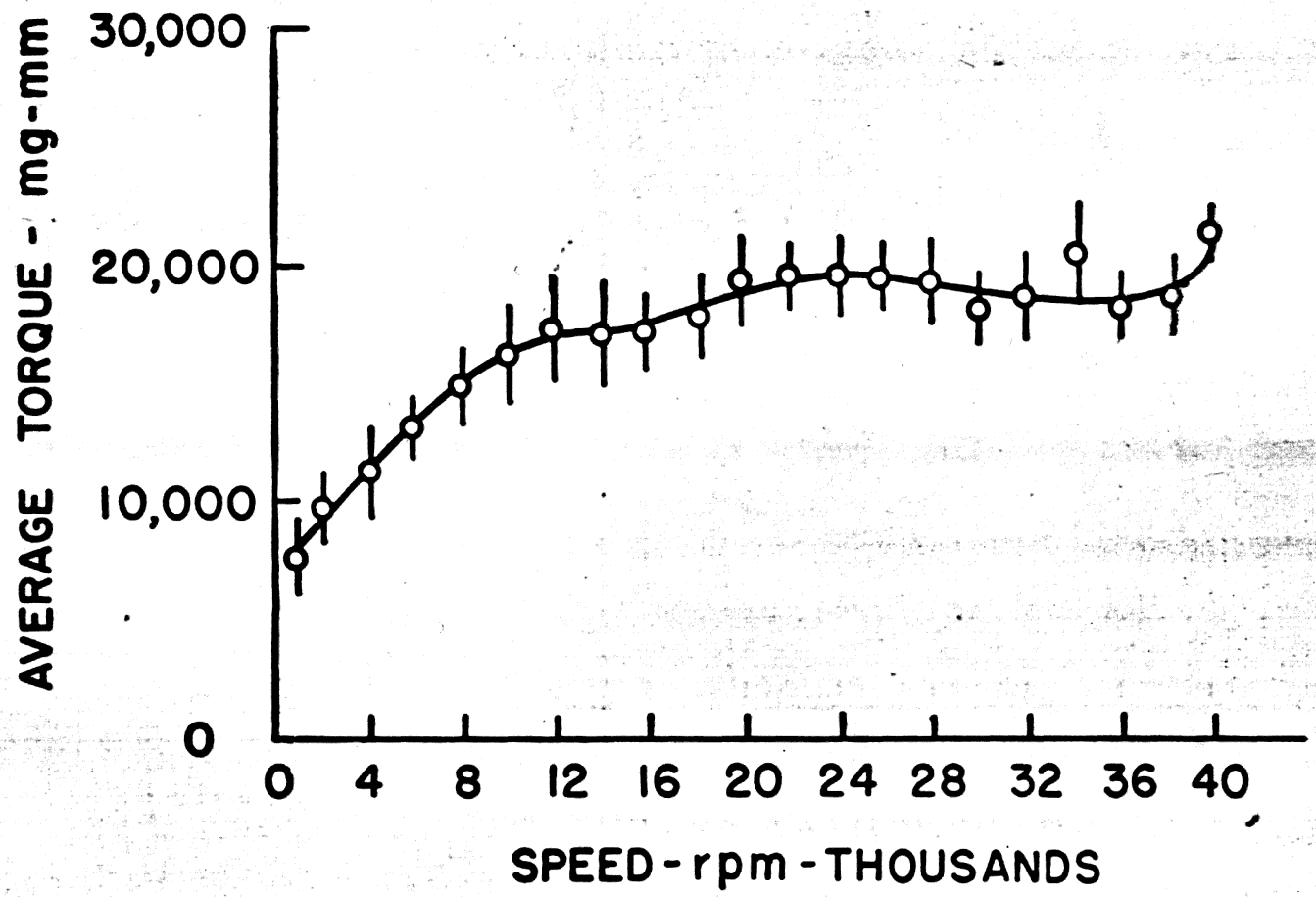

Fig. 32 1/16 Grease Pack, Re-run, 1/4 Axial Load

Actually, the opposite sequence was observed. There is some question about a chemical change occurring in the grease either due to atmospheric exposure, "working" from the action of the balls, or some combination of both. However, a definite answer was not found to this unusual torque sequence.

Figs. 33 through 38 represent the data for 1/8 grease pack bearings during original runs. Axial loadings of 1/8, 1/4, and 1/2 produced running torques only slightly greater than the running torque for zero axial load. Any differences were very slight as is indicated to some extent by the overlapping sample standard deviations at every speed point. An axial load of 1, however, increased the average torque approximately 30% above the zero axial load values. Further, an axial load of 2 increased the torque some 70% above the zero readings.

The re-run data is shown in Figs. 39 through 41 for 1/8 grease pack. As with the 1/16 pack bearings, these also showed a definite tendency for the average torques of the re-run bearings to exceed the average torques for the original run bearings. This tendency seemed to more pronounced in the mid-speed range than at the upper or lower speeds. Nevertheless, the original runs and re-runs were in reasonable agreement with each other.

A third method of comparing the preceding data, from Fig. 12 through Fig.41, might be on the basis of

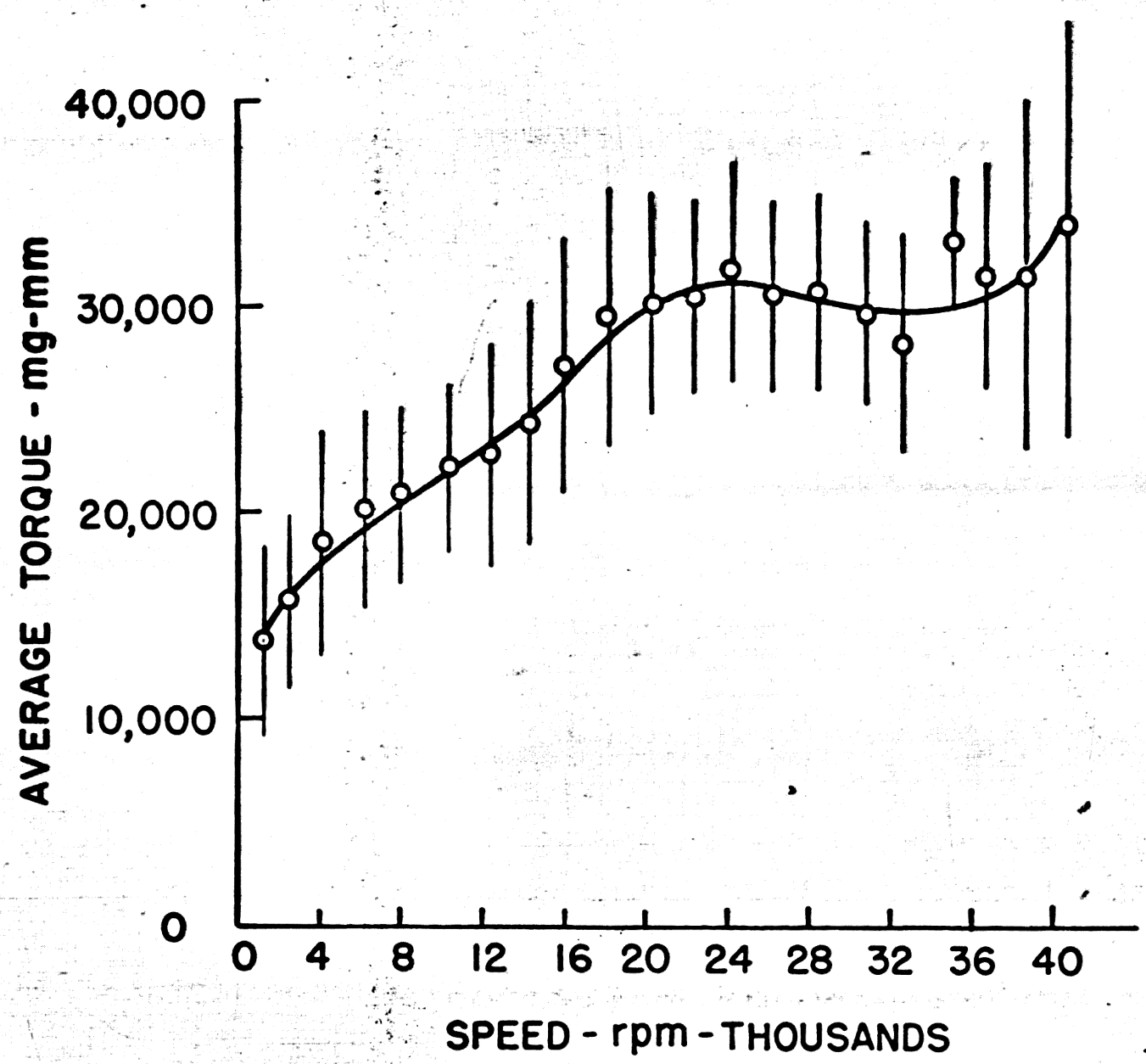

Fig. 33 1/8 Grease Pack, Original Run, 0 Axial Load

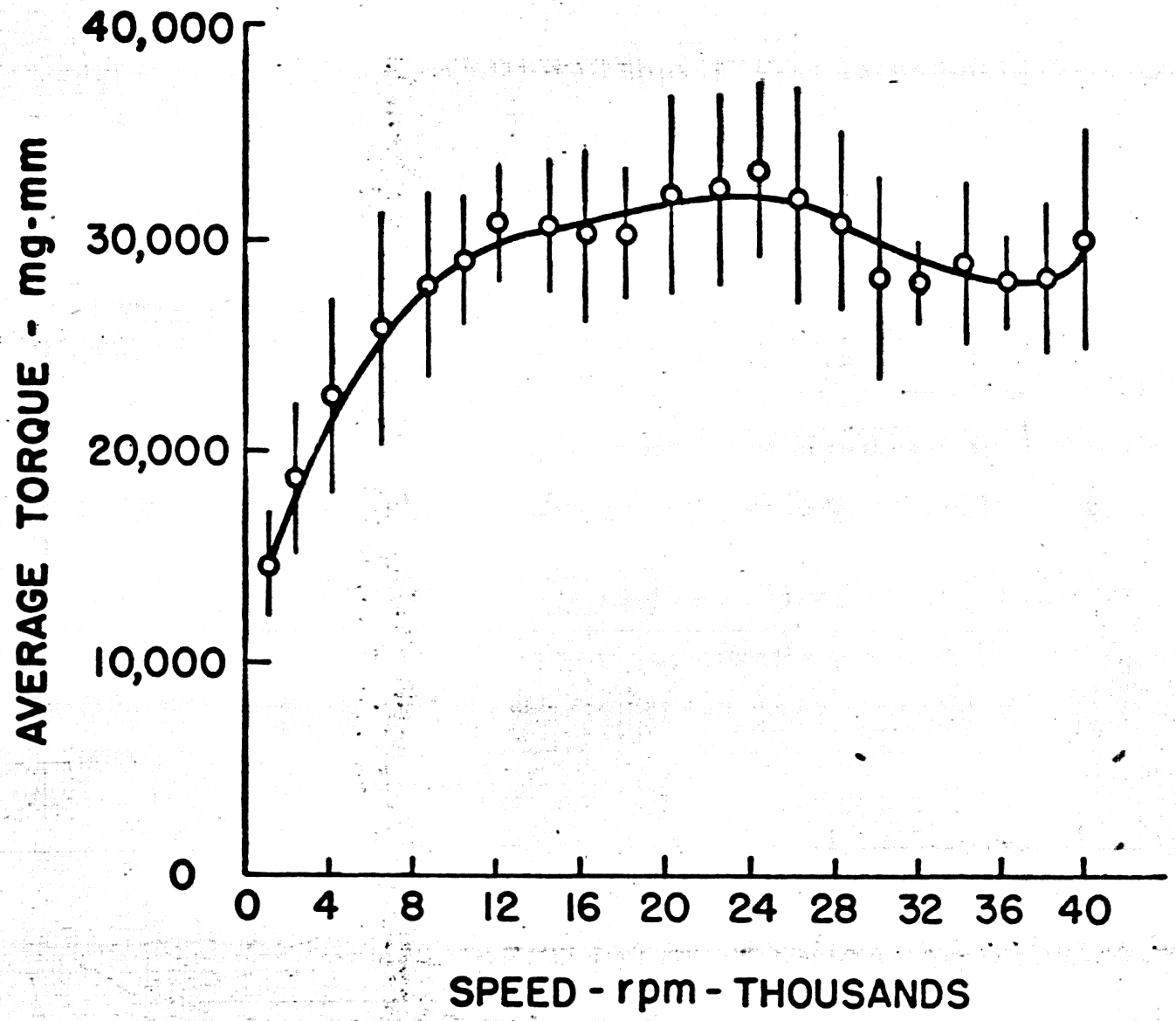

Fig. 34 1/8 Grease Pack, Original Run, 1/8 Axial Load

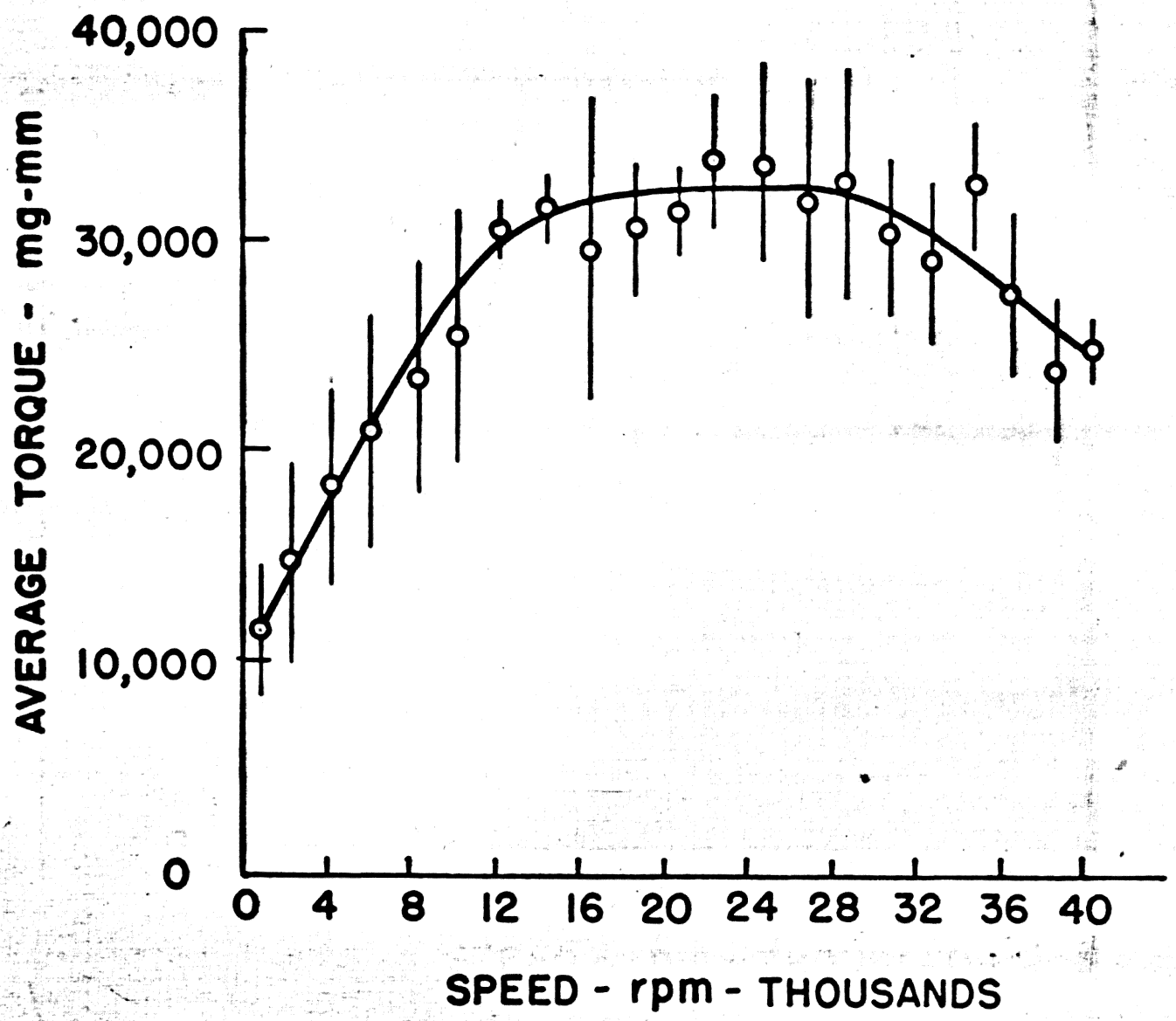

Fig. 35 1/8 Grease Pack, Original Run, 1/4 Axial Load

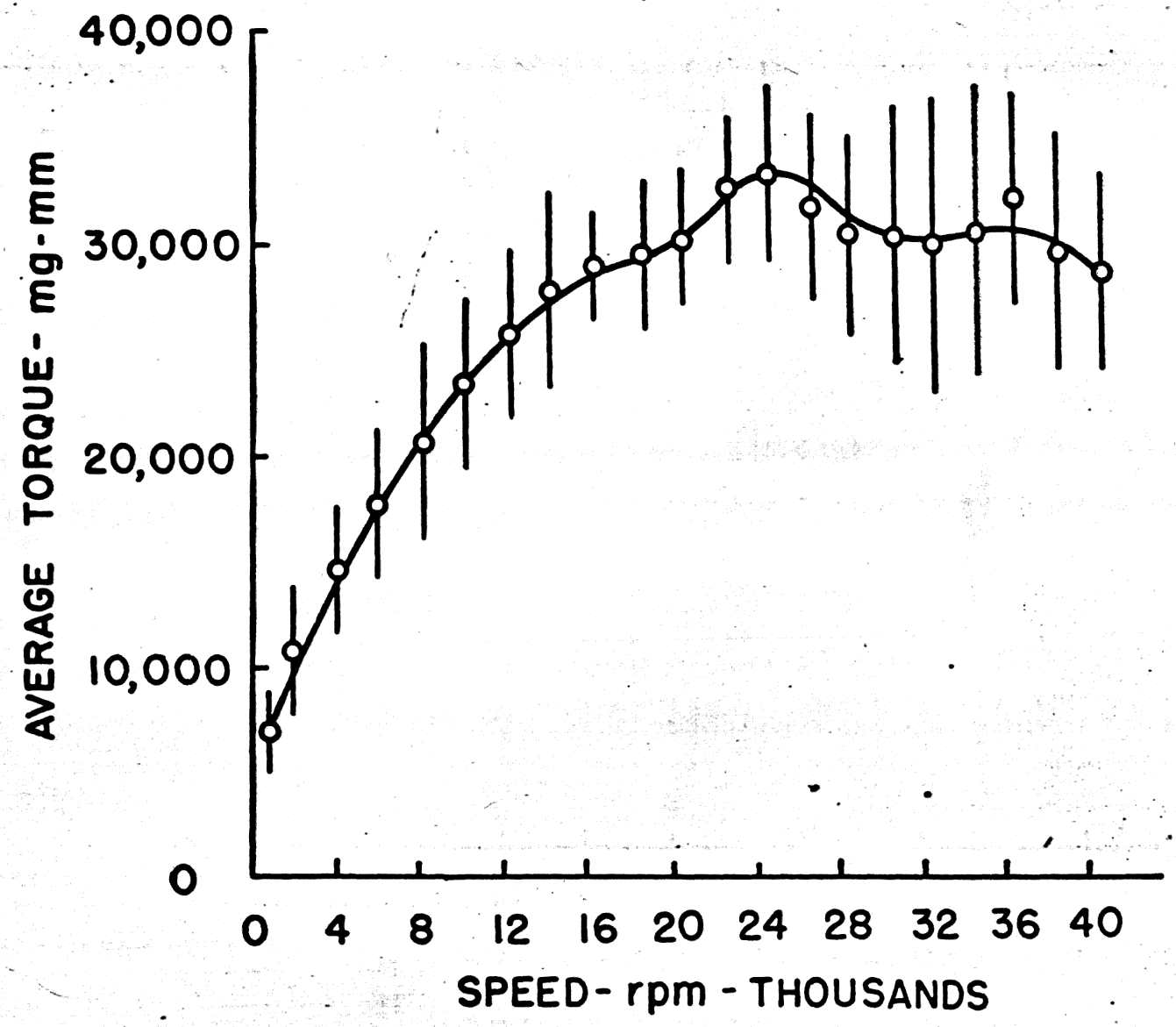

Fig. 36 1/8 Grease Pack, Original Run, 1/2 Axial Load

93

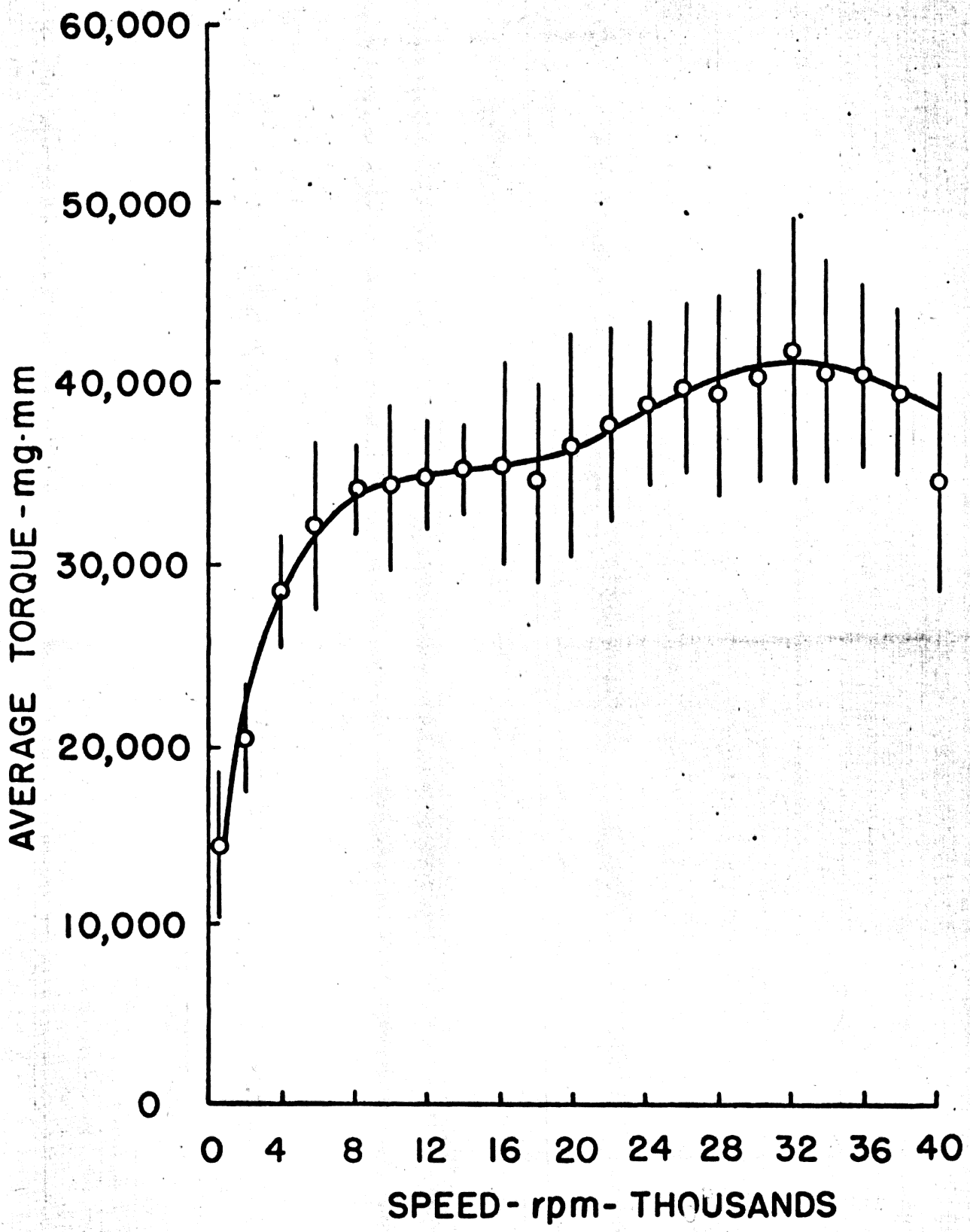

Fig. 37 1/8 Grease Pack, Original Run, 1 Axial Load

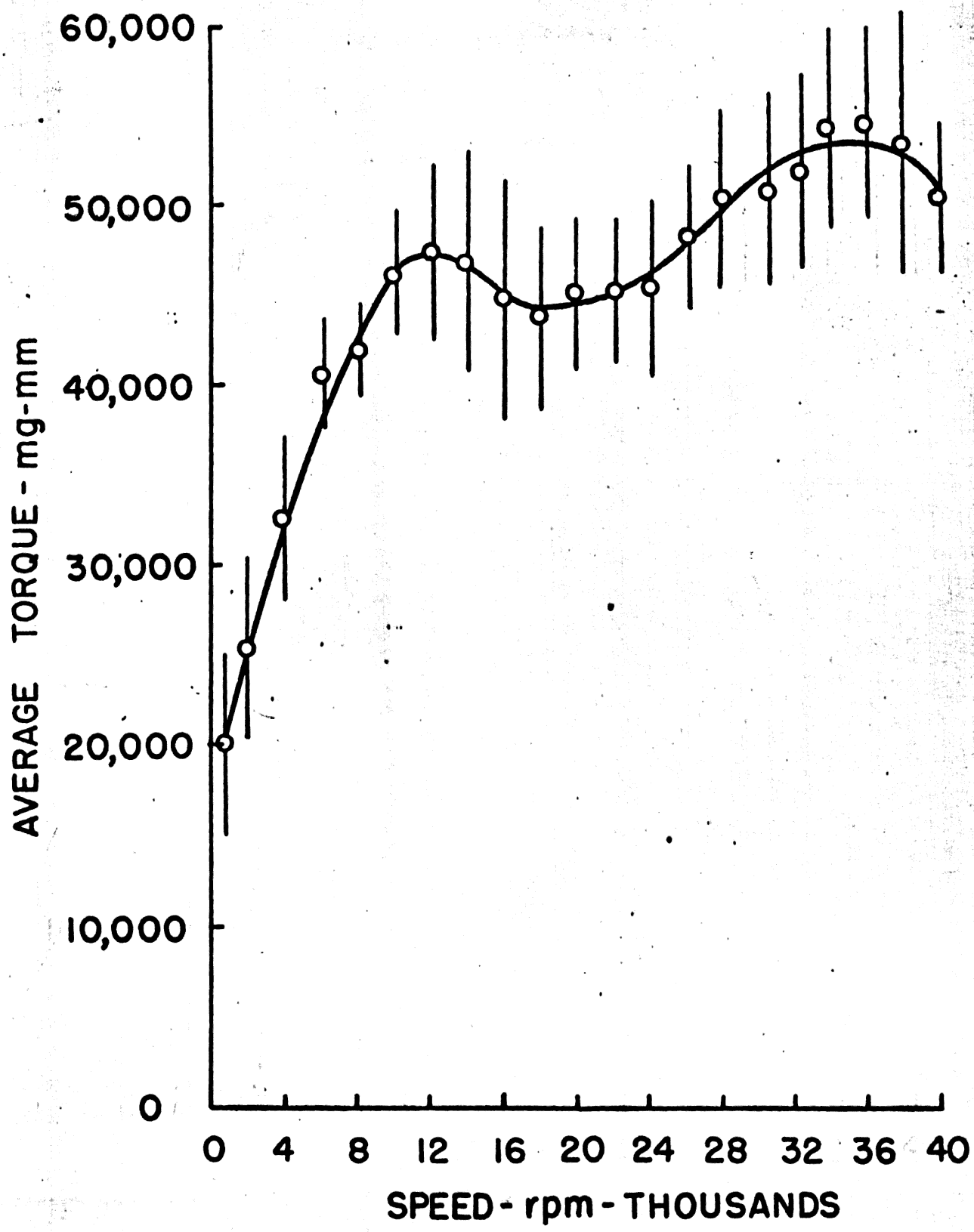

Fig. 38 1/8 Grease Pack, Original Run, 2 Axial Load

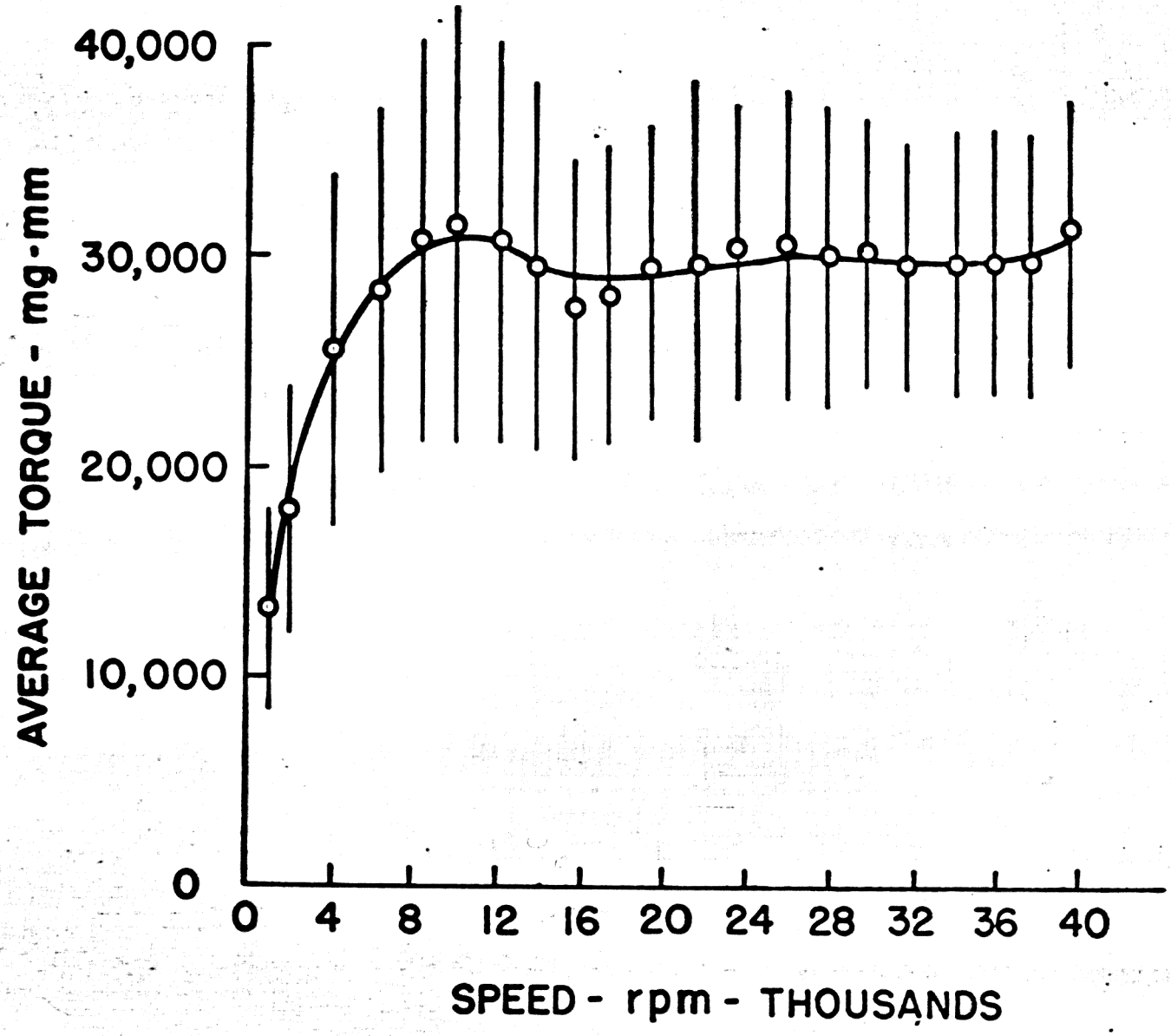

Fig. 39 1/8 Grease Pack, Re-run, 0 Axial Load

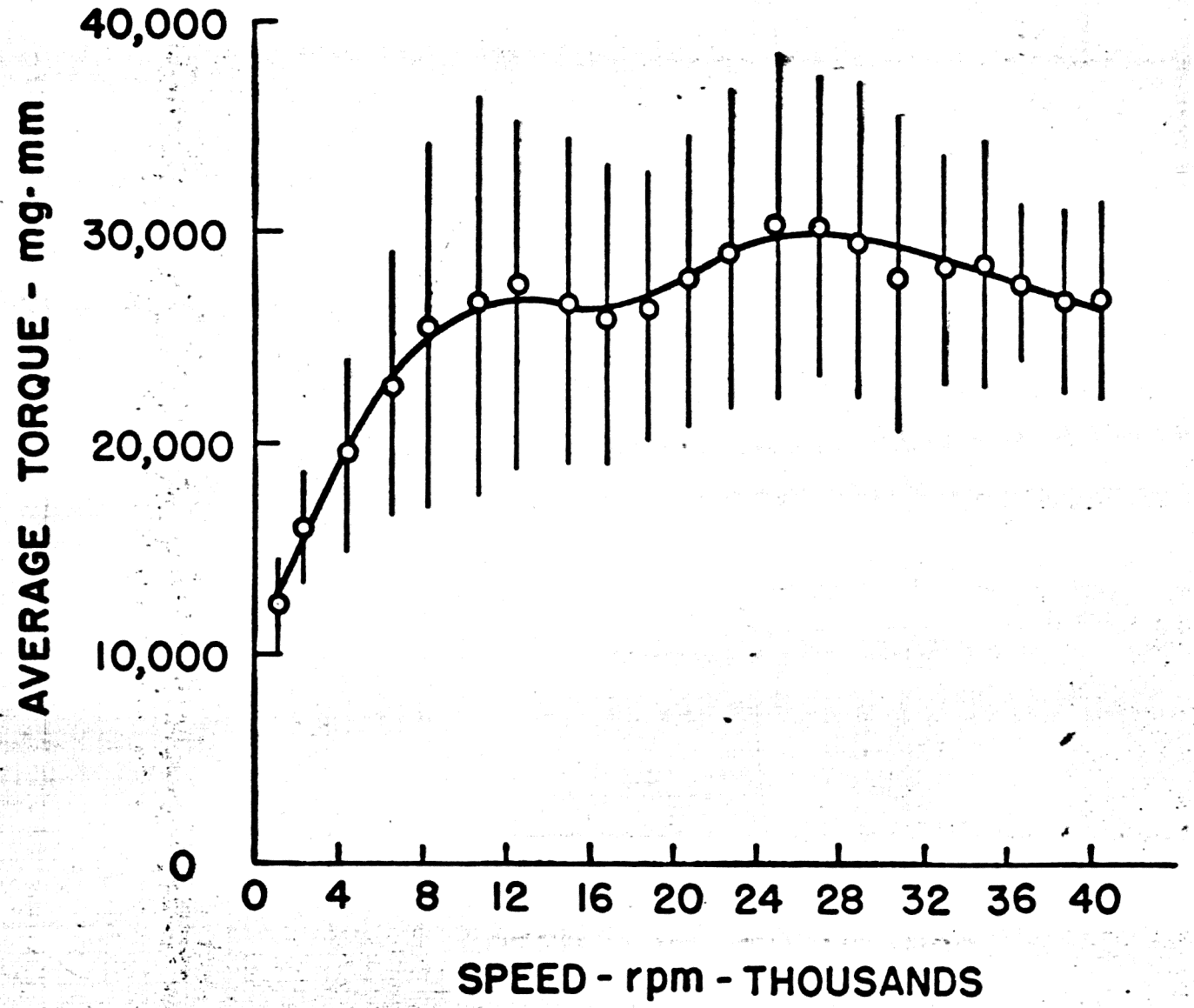

Fig. 40 1/8 Grease Pack, Re-run, 1/8 Axial Load

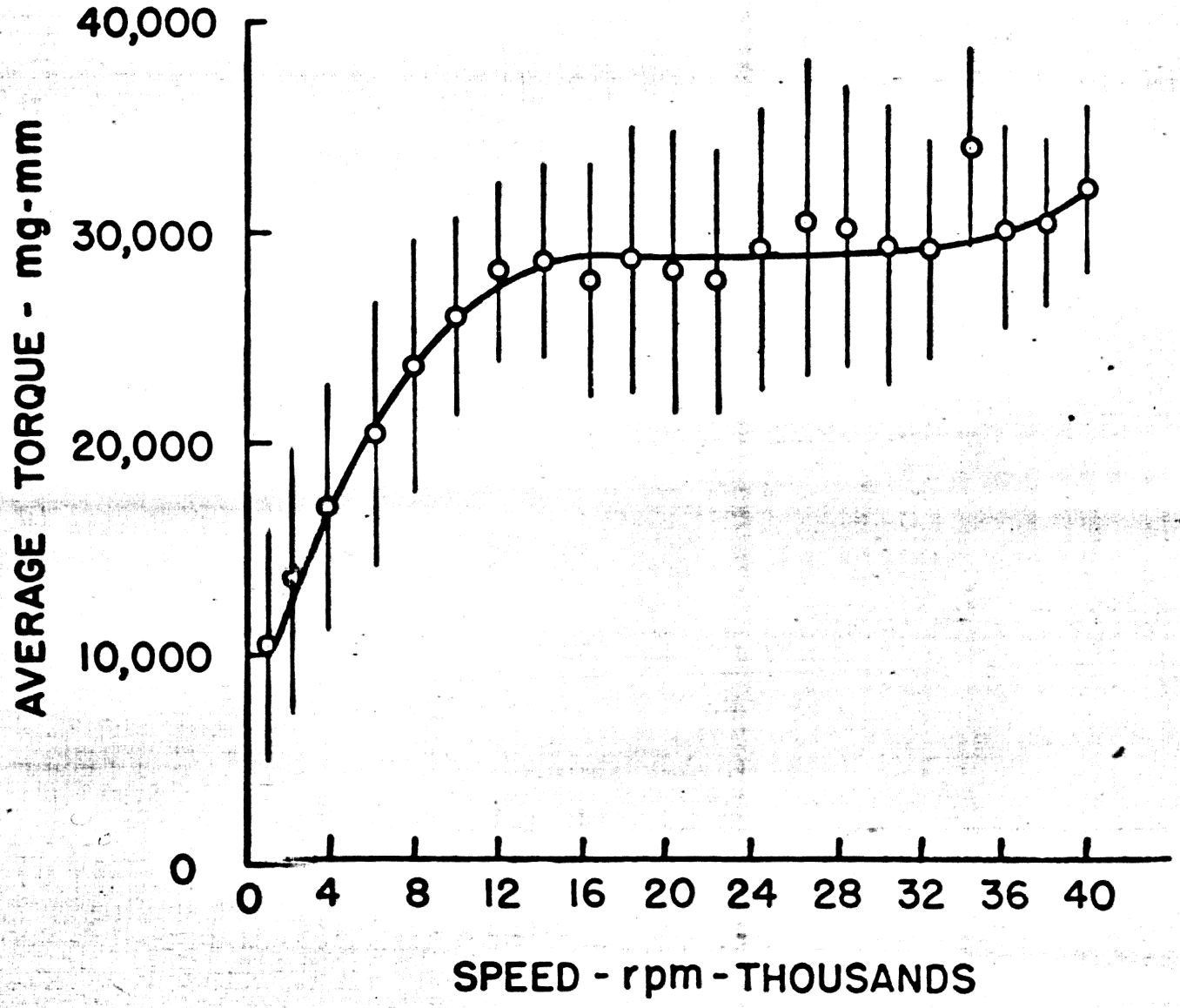

Fig. 41 1/8 Grease Pack, Re-run, 1/4 Axial Load

type and quantity of lubricant used. From this point of view, the oiled bearings very definitely exhibited the lowest running torques over the entire speed range. They had maximum values of running torque in the region of 10,000 mg-mm. When lubricated with a similar weight of grease, 0.0087 grams of grease versus 0.0076 grams of oil, the bearings exhibited running torques approximately twice as great as the oiled bearing torques. When the weight of grease was doubled, as in going from 1/16 pack to 1/8 pack, the running torques rose from approximately 20,000 mg-mm to over 30,000 mg-mm. This represented an increase of more than 50%. All of these comparisons were made for bearings with an axial load of 1/4 or less.

It was possible to make direct comparisons with the work of H.H. Mabie on the same bearings under several of the same conditions. Comparative plots of Mabie's data with the author's data is shown in Figs. 42 through 48.

In comparing the two sets of data under conditions of factory oil lubrication shown in Figs. 42 through 44, the plots may be seen to correspond very closely in both shape and magnitude. Although the sample standard deviations for these oil curves are small, they still overlap in nearly all instances. A slight tendency can be seen for the author's data to be slightly greater in magnitude in the mid- to low-speed range and slightly lower in the mid- to upper speed range.

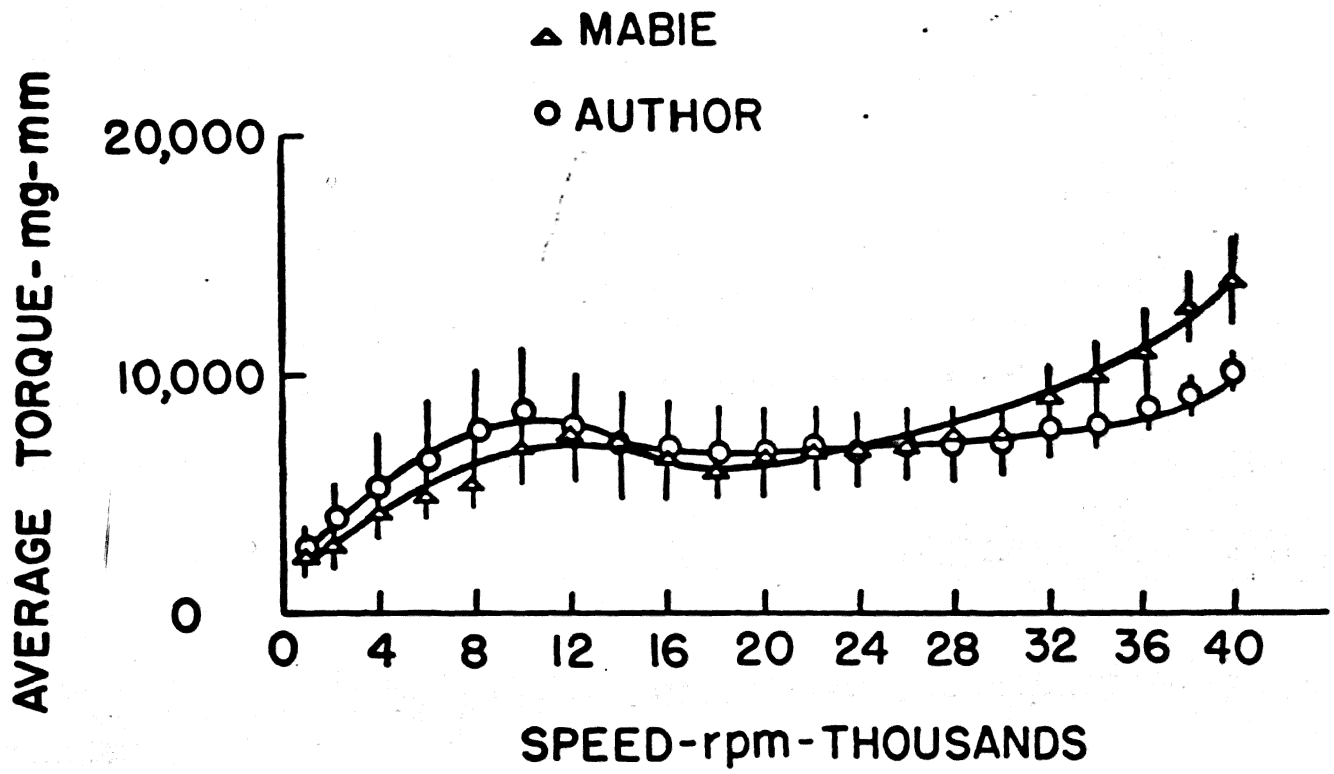

Fig. 42 Comparison of Factory Oil, Original Run, 0 Axial Load

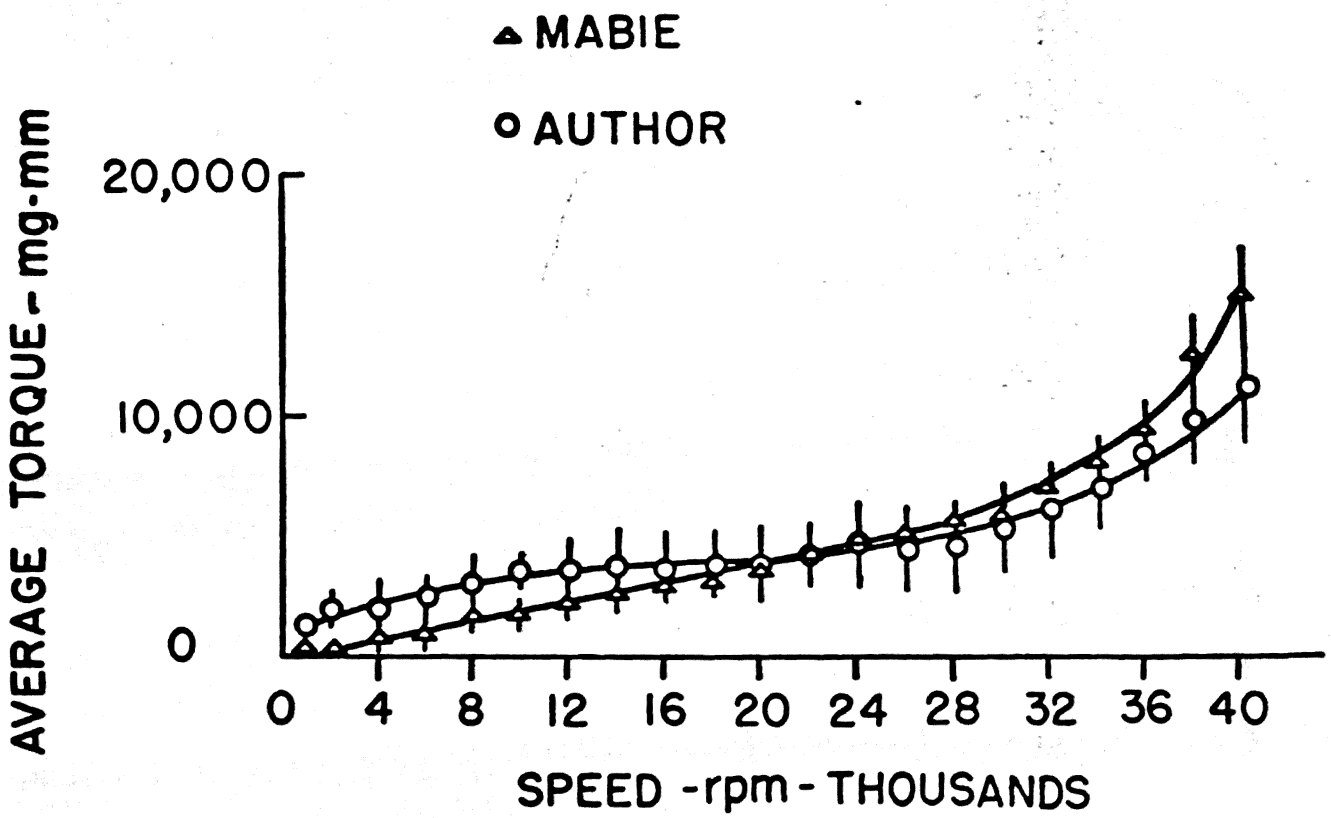

Fig. 43 Comparison of Factory Oil, Re-run, 0 Axial Load

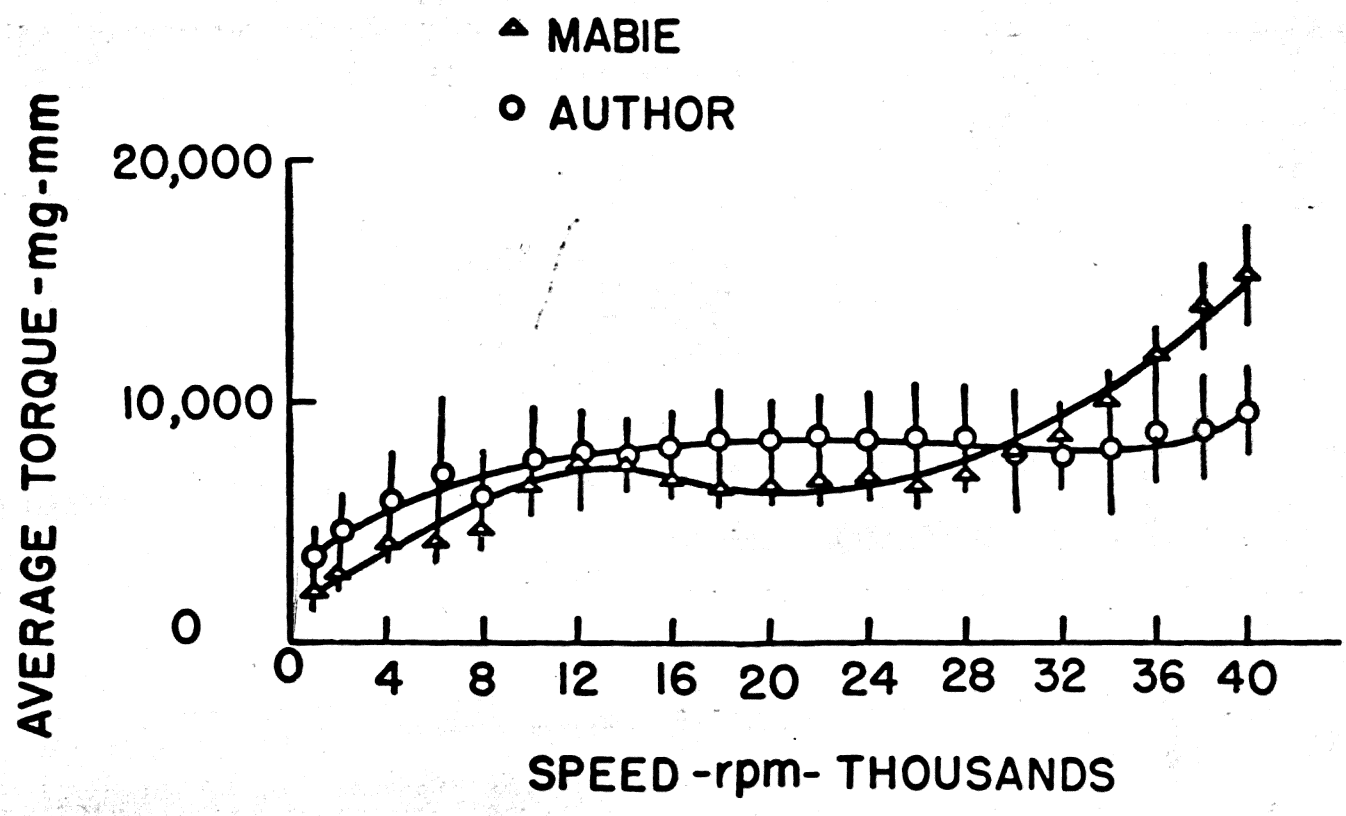

Fig. 44 Comparison of Re-oil, Re-run, 0 Axial Load

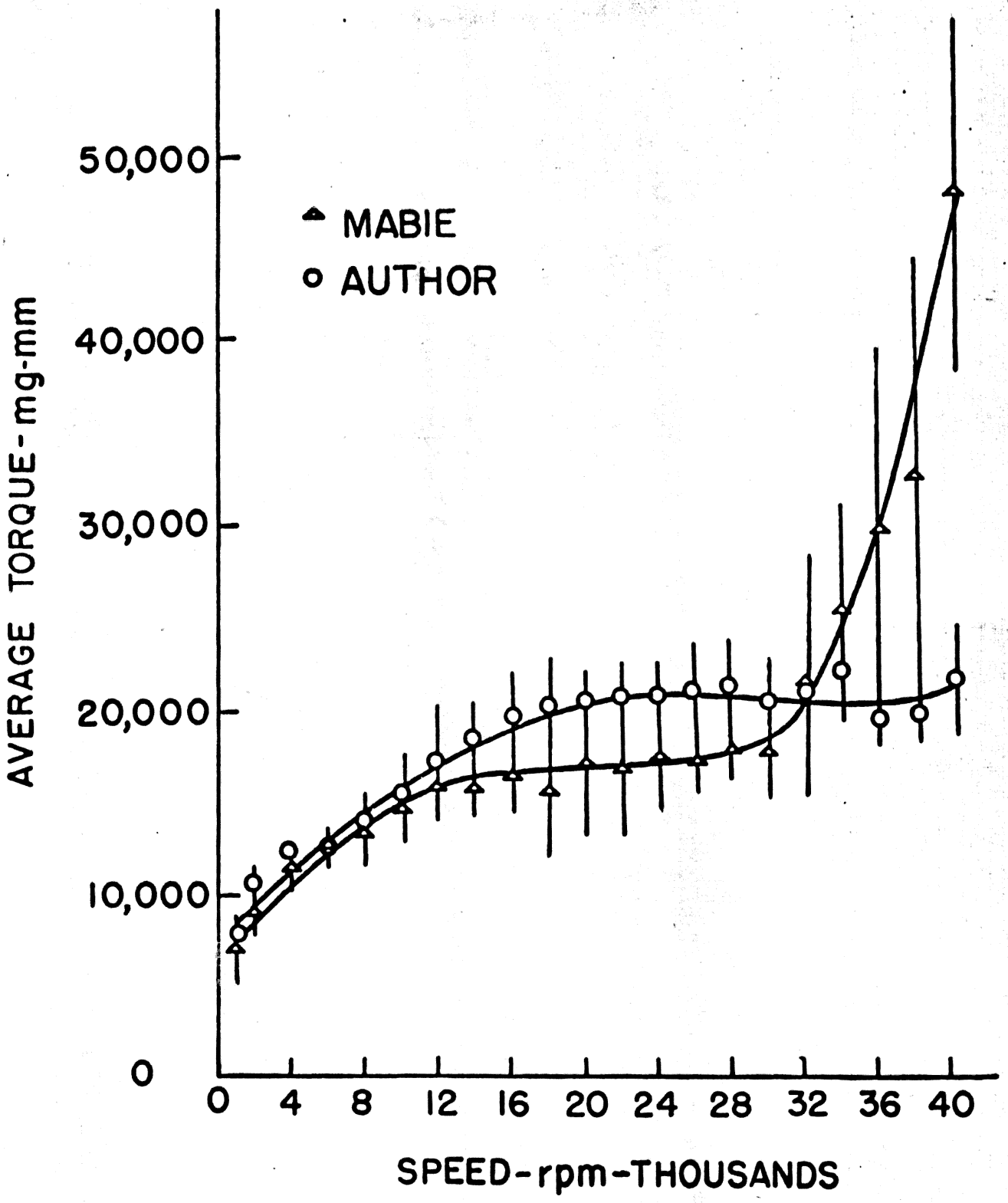

Fig. 45 Comparison of 1/16 Pack, Original Run, 0 Axial Load

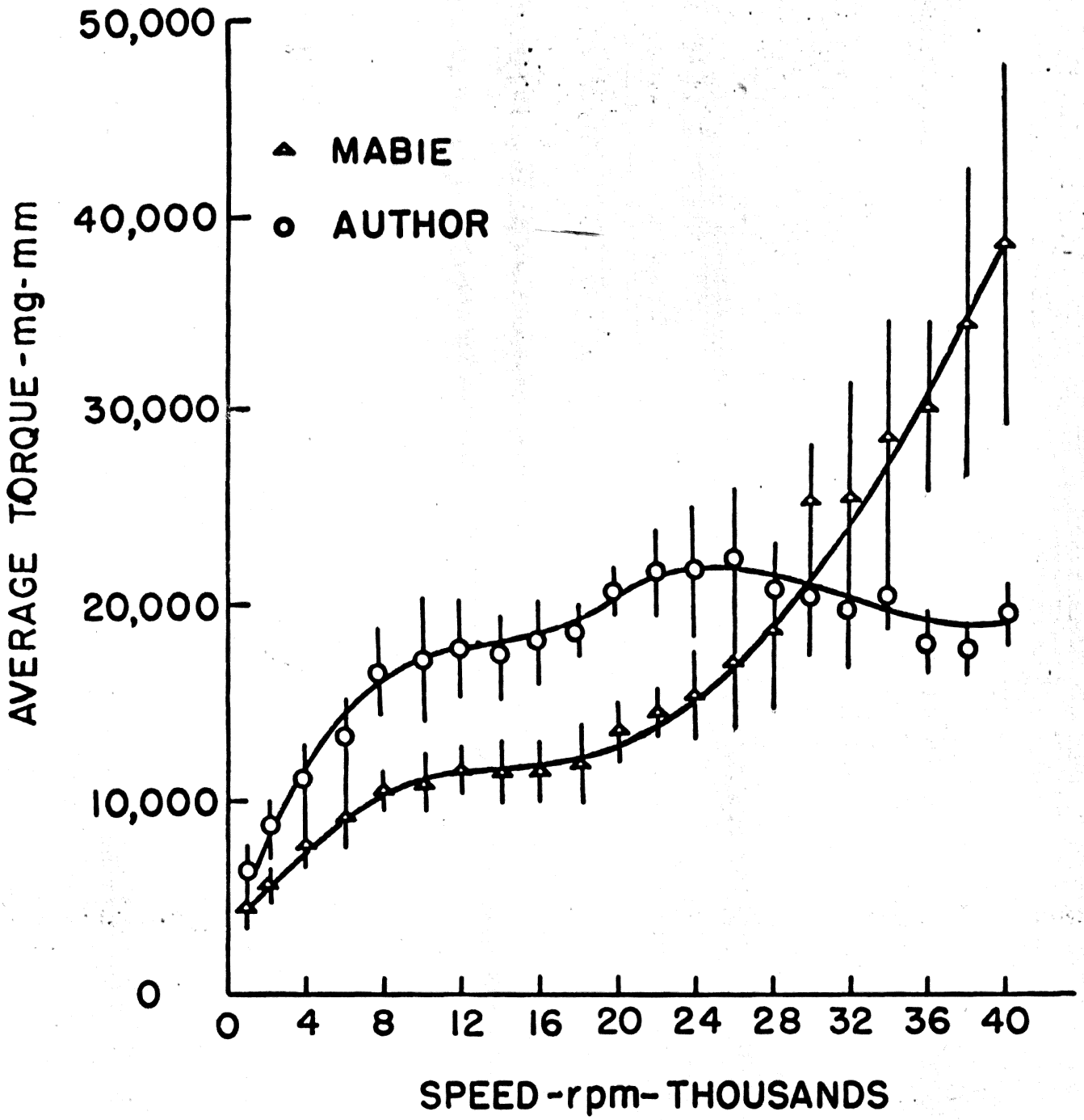

Fig. 46 Comparison of 1/16 Pack, Re-run, 0 Axial Load

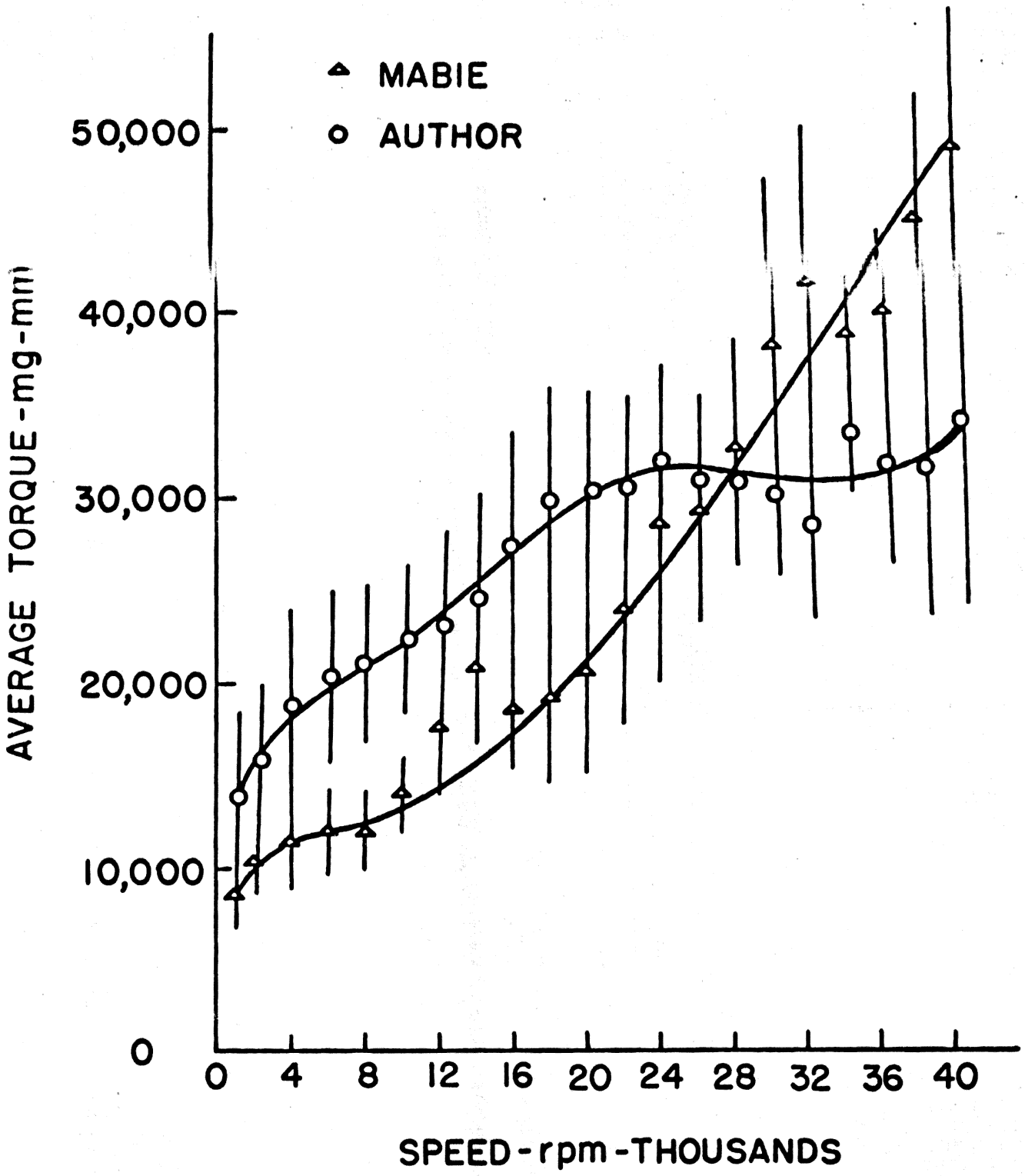

Fig. 47 Comparison of 1/8 Pack, Original Run, 0 Axial Load

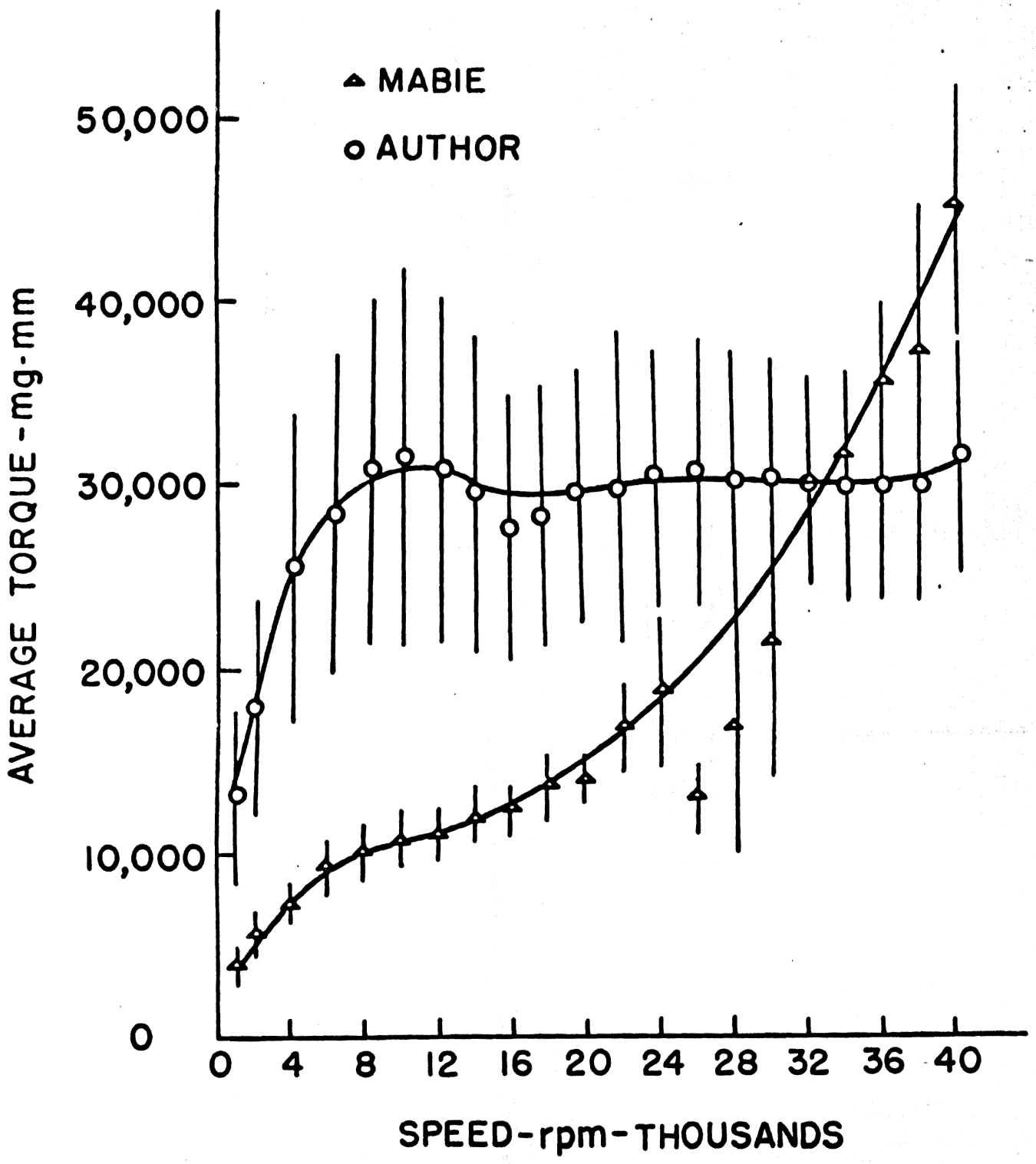

Fig. 48 Comparison of 1/8 Pack, Re-run, 0 Axial Load

In comparing the grease lubrication tests, the same tendency noted above, for the torques to vary slightly as some function of speed, is displayed but in a more pronounced manner. The comparison between Mabie's curves and the author's curves for 1/16 and 1/8 grease pack original runs agree reasonably well. However, the data for re-run grease lubricated bearings as seen in Figs. 46 and 48 contrast appreciably. The author's data is seen to rise rapidly to torques considerably above Mabie's data, but the author's data reaches a plateau and remains essentially constant while Mabie's data continues to rise considerably above the author's values. The differences for re-run greased bearings are seen to be as great as 66% for 1/16 grease pack shown in Fig. 46.

Although there are a few points of difference between Mabie's work and that of the author, the data compared is in reasonable agreement. From this agreement with other experimental data, a reliable basis was established for conclusions concerning the effects of axial loads.

VI. CONCLUSIONS

From the results of the subject investigation, the following conclusions can be drawn:

1. the strain gage method of torque sensing will accurately measure the running torque of R-3 size ball bearings at ambient temperatures.
2. the results presented are in substantial agreement with investigations[5] working in the same speed and temperature range.
3. the effect of axial loading of a ball bearing loaded with 47 grams radially is negligible until the axial load equals or exceeds the radial loading.
4. grease lubricated bearings demonstrate running torques approximately twice as great as bearings lubricated with a similar quantity of oil.

VII. RECOMMENDATIONS

1. It is recommended that larger values of radial load and corresponding axial load ratios be tried since 47 grams is far below the expected capacity of an R-3 bearing.
2. It is recommended that in future investigations re-runs and re-lubricated runs be discontinued because of the consistent similarity to the original runs.

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THE MEASUREMENT OF THE RUNNING TORQUE OF OIL AND GREASE LUBRICATED INSTRUMENT BALL BEARINGS UNDER COMBINED RADIAL AND AXIAL LOADS

by

George Edward Clarke III

ABSTRACT

Although many studies have been made on the operating characteristics of instrument bearings, most were conducted at two rpm or less and with thrust load only. A study by H.H. Mabie tested the running torques of radially loaded bearings from 1,000 to 40,000 rpm. The purpose of this investigation was to study the running torques of R-3 size instrument ball bearings at speeds up to 40,000 rpm while under combined radial and axial load.

Much of this investigation was devoted to the construction of an accurate torque sensing device. The method employed relied on the amplification of strain gage signals by a strain gage indicator and an x-y plotter. The strain gages were used to detect the strain at the base of a small beam that prevented rotation of the outer race of a test bearing while the inner race was driven at test speed by an air turbine. The accumulated data was the result of 30 test series, with each series being

constituted of a test sample of six ball bearings.

From the study, it was consluded that the strain gage method of torque sensing accurately measured the running torque of R-3 size ball bearings at ambient temperatures. It was also concluded that the effect of axial loading of an R-3 ball bearing loaded with 47 grams radially is negligible until the axial load equals or exceeds the radial loading.

By comparing lubricants, it was concluded that grease lubricated ball bearings demonstrate running torques approximately twice as great as bearings lubricated with a similar weight of oil.