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TECHNICAL MEMORANDUMS

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

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No. 945

EXPERIMENTS ON BALL AND ROLLER BEARINGS UNDER CONDITIONS OF HIGH SPEED AND SMALL OIL SUPPLY

By Günter Getzlaff

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EXPERIMENTS ON BALL AND ROLLER BEARINGS UNDER CONDITIONS OF HIGH SPEED AND SMALL OIL SUPPLY*

By Günter Getzlaff

The author describes a testing machine on which 35-millimeter bearings (bore) can be run at speeds of the order of 21,000 rpm, while the following factors are recorded:

1) Oil circulation through bearing and oil temperature.

2) Maximum temperature of outer bearing ring.

3) Radial and axial load on bearing.

4) Radial, axial, and angular clearance of bearing.

5) Power consumption of bearing.

The experiments show that the lubrication was most reliable and oil consumption lowest when the oil was introduced through a hole in the outer or inner ring of the bearing. In the case of roller bearings the oil circulation could be kept especially low (0.5 liter per hour and less). The temperature of the bearings is chiefly determined by the radial clearance. A satisfactory performance could always be achieved on shafts without overhang, provided the radial clearance was of the order of 30 μ. The axial load also increases the working temperature and it obviously pays to have oil of low viscosity. Increasing the oil flow from 2 to 20 liters per hour, practically doubles the power consumption of the bearing.

I. INTRODUCTION

The general trend in engineering progress is toward smaller dimensions and weights by equal performances, which makes higher rotational speeds obligatory. This need for higher rotational speeds is particularly pressing in the

*"Untersuchungen an Wälzlagern." Jahrbuch 1938 der Deutschen Luftfahrtforschung, pp. II 110-118.

development of aircraft superchargers. Among the engine-design problems, reliable bearings are at present in the foreground of interest. The question of superiority of plain or roller bearing remains an open one. Advance in plain bearing development and lubricant research permit us to hope for further benefits (references 1, 2, 3, and 4). Still the greater part of aircraft superchargers is equipped with roller bearings, which indicates that this type is, for the present, more suitable. Hereby the difficulties of sealing against the passage of greater oil quantities, required by the plain bearing, is a contributing factor. The necessity of keeping the compressed air as free from oil as possible from the supercharger, forms a special requisite for the bearings in airplane superchargers. A small amount of oil entrained as oil mist by the air, as on intercoolers, for instance, may jeopardize the dependability of the whole power plant. For that reason, the realization of the lowest possible oil consumption during the experiments, demanded particular attention. The diameter of the bearings selected for this investigation represent approximately the upper limit of present-day construction, because disturbances are to be expected first on the largest bearings.

II. PRESENT STATE OF RESEARCH AND PROGRESS

The conventional methods of predicting the life and load capacity of roller bearings aim, so far as it is possible to judge on the basis of available literature, toward the determination of the most heavily loaded point by theoretical considerations and, to obtain for this, with the help of the fatigue-resistance curve for the employed material, the mathematical data regarding the number of possible stresses until failure; that is, the fatigue of the material is the starting point. Another empirical possibility for obtaining data on the loadability, are running tests on testing machine, where the bearings rotate under certain controlled load conditions, up to incipient failure and the number of total revolutions is ascertained. The present method of computing the safe load, is based principally upon such practical experience. According to Palmgren (reference 5), the "life" of a bearing is determined at one-fifth of the average life of a test series. Accordingly, the life L is

$$L = \left(\frac{T}{P}\right)^3$$

where

L life in millions of revolutions.

P load in kilograms.

T specific load-carrying capacity = safe radial load for reaching a life of 1,000,000 revolutions.

For single-row, grooved ball bearings, it is

$$T = \frac{4.5\ d_w^2\ z_1^{2/3}\ \cos\alpha}{1 + 0.02\ d_w}\ (\text{kg})$$

where

d_w ball diameter (mm)

z number of balls.

α angle of pressure.

and for roller bearings

$$T = 5\ z_1^{2/3}\ d_w\ l_w\ \cos\alpha\ (\text{kg})$$

with l_w width of roller (mm).

In conformity with the technical state of development, the investigations are, as a rule, extended to include rotational speeds not much over 3,000 rpm; or a coefficient of velocity of 105,000 for a 35-millimeter shaft diameter. Where no direct test data are available, extrapolations are resorted to. As an illustrative example, we show in figure 2, the permissible load plotted against the rpm for a bearing 6207 (35 mm diameter) on the basis of data of two manufacturers. The conversion in these cases was probably carried out along similar lines.

At coefficients of velocity above about 500,000, the operating behavior of roller bearings is no longer exclusively governed by the outside load; in particular, centrifugal force, lubrication, and clearance can decisively influence the operating safety. This effect is not sufficiently considered in the orthodox methods of calculation.

The centrifugal forces of the roller bodies subject the outer bearing surface to additional load besides the axial. For grooved ball bearings and roller bearings, this implies at first no shortening of life, because the inner ring - on account of the less favorable lubricating conditions - is the weakest structural component of this type of bearing.

Figure 3 gives some information on the conditions in commercial roller bearings at high rotational speeds. The centrifugal force of the single balls is plotted for rotations up to 50,000 rpm, on the basis of the theoretical cage revolutions. A comparison between the light and the medium heavy series, discloses more than double the load of the latter. The reason lies in the greater ball diameter and, in addition, in an increased bearing-surface diameter because of the larger inner ring. For comparison, the centrifugal force per ball for 8- and 5-millimeter ball diameters has been included, the inner ring of a bearing 6207 being considered as given. This effect of the ball diameter becomes even more evident in figure 4, since the weight enters into the calculation with the 3d power. As the ball diameter decreases, the relative loadability of the bearing increases, according to Palmgren (fig, 5) (reference 6).

The most common cause of bearing failure, is attributable to insufficient lubrication in the bearings. Experiments made in this direction are described elsewhere in the report. More accurate numerical data on bearing surface, cage stress, and friction are reserved for a subsequent study.

III. TEST SCHEDULE

Owing to the great importance of correct lubricant feed, the development of the bearing testing set-up provided for a multiplicity of oil-feed systems, with a minimum of changes as exemplified in figures 6a to 6e.

a) Oil feed through a hole 1.5 to 2 millimeters in diameter, in the center of the outer ring bearing surface of the roller bearing.

b) Oil feed through inner ring bearing surface of roller bearing.

c) Oil feed through inner and outer ring bearing surface.

d) Oil feed through spray between outer ring and cage by means of nozzles.

e) Oil feed by means of a small paddle wheel.

Note: (a to c) - The rings are finished and heat-treated, the holes drilled and then assembled. Because of the mass balance, the inner ring has two holes facing each other.

d) - Because of the oil viscosity when using nozzles, the volume is downwardly restricted. At the usual lubricant pressures, the nozzle cannot arbitrarily be made smaller while on the other hand, with a larger nozzle diameter, the spray in small quantities, of the order of 6 to 7 liters per hour, had little directional force and was subject to the deflecting effects of the entrained air eddies.

e) - Experience with ball bearings in high-speed gears, justify the assumption that a dependable lubrication with minimum oil volume is possible. Probably the oil mist formed by the meshing of the gears at high speeds insures an ample and effective lubrication of the nearby roller bearings. The oiling system developed along these lines is essentially a splash ring designed on the order of centrifugal-pump rotors. It is designed to feed the oil, evenly distributed, in the bearing. The oil leaves on the opposite side of the bearing. If a low pressure in the return line is produced, the then-ensuing transverse flow through the bearing should insure, aside from the cooling effect, complete oil removal. The freedom from oil thus gained on the opposite side, forms an operational necessity for many phases of application as, for instance, for cold-storage plants in the food industry, as well as superchargers.

Bearing clearance and fit required particular attention. Before installation, the roller bearings were tested for radial clearance, the deep-groove ball bearings for radial, axial, and angular clearance (figs. 7a to 7c). The test procedure was as follows: After careful cleaning and lubricating with thin oil, the radial clearance was recorded by minimeter as the radial displacement of the two bearing rings lying in one plane (fig. 7a). The axial clearance comprised the total axial displacement of the two bearing rings toward each other by concentric position of the bearing surfaces, conformable to 7b. The angular displacement is the total deflection of a point on the outer ring when, with inner ring restrained, it swings about the median axis from one end setting to the other (fig. 7c). The employed instruments (minimeter, Orthotest, and dial gage) guaranteed exact measurements four times on the circumference. The test pressure in all measurements was around 5 kilograms.

IV. DESCRIPTION OF BEARING TEST STAND (fig. 8)

The shaft a is carried on the clutch side in roller bearing b; on the other side, in test bearing c, and radially loaded by the balanced disk d. The axial load is introduced in the test bearing c by the calibrated spring e across the axially stressed deep-groove roller bearing f. The radial load can be varied by supplementary weights g. In this case the load is also initiated through bearing f. For the roller-bearing tests, roller bearing b is replaced by a deep-groove ball bearing of the same diameter. The axial spring e and the related bearing f, are removed for these tests. Bearings b and f have smaller diameters than the bearing to be tested. With it grows the probability that disturbances occur first on the test bearing because its coefficients of velocity are highest. The assembled rotor with a roller bearing as test specimen is shown in figure 9.

The oil circulation of the test bearing is completely separated from the remaining oil circulation by the splash rings h, and the tip seals i, so that the amount of oil passing through the bearing can be accurately determined. The tip seals permit, through supply of compressed or suction air, the creation of a positive flow through the bearing. The temperature of the test bearing is recorded with thermocouples at four points of the circumference. In the oil-feed line to the test bearing, a flow volume meter (fig. 10) is mounted, through which the oil volume is checked at the exit with a graduated measuring glass and stop watch. A heater mounted behind the flowmeter, affords controlled oil heat up to about 100° C.

The oil for the test bearing c passes through pipe k. After passing through the bearing, it can pass out at l and m, whereby line l may maintain low pressure for special purposes; for this test, exit m is closed. Fitting n then serves for recording the low pressure with a U-tube (fig. 10).

The drive of the test bearing shaft is by three-phase motor (fig. 11), whose rpm can be regulated to within narrow limits by a series resistance. For great rpm changes, the gear-transmission ratio is modified. By this means, the range from 16,000 to 45,000 rpm can be covered within practically one stage.

Gears and bearing b are oiled automatically by a separate pump circulation. Oil pressure and temperature were kept constant during the test runs, so that the performances could be compared.

The total power absorption (gears and test-bearing shaft) was recorded with an astatic wattmeter. The no-load power absorption ascertained by several gear calibrations, gave as difference the power absorption of the bearings b + c and b + c + f, respectively. The measurement of the bearing temperatures collectively was affected thermoelectrically on the outer ring over a two-way switch and two millivoltmeters.

The oil circulation of the test bearing called for some special arrangements, by means of which the oil volume could be controlled and uninterrupted feed insured (figs. 10 and 11). The oil passes from a tank a across flowmeter b and heater c through pipe k into the bearing. The returned oil is intercepted by funnel d and measured with a graduated measuring glass and stop watch. Then the oil passes through a cooler e to the intermediate tank f, where a pump with strainer g feeds it in any desired amount to the tank a. This tank can be put under compressed air h, thus making it possible to change oil volume and pressure within wide limits, independent of pipe-line resistance and gradient. For oil volumes below 1 liter per hour, a Bosch oiler is used because of its more accurate adjusting power.

V. SCOPE OF TESTS

The scope included:

Arrangement - figure 6a for deep-groove ball bearing (6207).
figure 6b for deep-groove ball bearing (6207) and roller bearing NL 35e.

Revolutions - 16 to 21,000 rpm in four stages.

Radial load steady around 2.4 kilograms.

Axial load 0 to 120 kilograms in seven stages.

Oil volumes 0.05 to 50 liters per hour.

Three grades of oil with 7° E, 2.8° E, and 1.7° E viscosities at 50° C.

Oil inlet temperature 20° to 80° C, viscosity curve according to figure 13.

Radial clearance of new bearings, 6μ to 50 μ.

VI. RESULTS OF TESTS

These findings should not be construed as being all-inclusive, nor as precluding future modifications as additional test data are obtained.

Characteristic of the stress condition of the bearing was the maximum bearing temperature recorded at the outer bearing ring. The steady or final temperatures in the tests were achieved at constant rotational speed, oil volume, and temperature. Several measurements at 10-minute intervals without change were stipulated before proceeding to a new state.

The test values* are illustrated in figures 14 to 24. One intrinsic result is the temperature curve in relation to the hourly oil flow for different rpm, in figure 14. The curves are similar and approach the abscissa asymptotically by decreasing oil volume. Even so, the tests on roller bearings still indicate practically steady temperature values for minimum volumes. The heat in the roller bearings was, on the whole, less than in the deep-groove ball bearings - attributable, no doubt, to the simpler

*Some more recent results will be found in W. von der Nüll's article entitled: "Supercharging Devices for High-Speed Internal-Combustion Engines, particularly Aircraft Engines, published in ATZ, No. 11, 1938.

conditions of movement. However, owing to the effects of clearance and brand of oil, the values of the two types of bearings are not directly comparable.

The test data for large volumes are in contradiction with the general opinion that large oil volumes, on account of the expected great friction, should result in high bearing temperatures. If the oil has an opportunity to flow off laterally, it is pushed aside by the circulating system and serves to carry off bearing heat. Rising temperatures were not observed evn by still greater volumes.

Concerning the radial heat gradient in the bearing, no results can as yet be given. No appreciably higher temperatures on the inner bearing ring are likely, for the following reasons:

a) The inner ring is completely unloaded over the greater half of the circumference;

b) The centrifugal force of the roller bodies constantly produces additional strain energy on the whole circumference of the outer ring;

c) Owing to the effect of the centrifugal force on the lubricant, the principal part of the friction is produced on the outer ring.

From the point of view of lubricating technique, there is no noticeable superiority between oiling systems, figure 6a, with hole in outer ring, and figure 6b, with hole in inner ring. The feed through the inner ring is very beneficial because the oil positively touches every wearing spot; operationally, the drawback is that the oil is whirled in the distributor ring groove and specifically heavy parts are deposited as sludge and may obstruct the groove. With this method, the possibility of supervision and cleaning is necessary. The interruption in the bearing surface by the bore has, so far, shown no interference. The experiences regarding it extend over a number of roller and deep-groove ball bearings with individual running periods, up to 250 hours. If sufficiently high axial loads are applied, it need not lead to running in the groove bottom, because the balls are pressed laterally against the bearing surface.

In all measurements, a linear rise in temperature with increasing rpm, conformable to figure 15, was observed.

The curves, extended as far as the ordinate, almost yielded an intersection point which, at zero rpm, has the room temperature as ordinate. By further rpm increase, the approach to limiting values will probably be evidenced by a departure from the linear curve.

The effect of additional axial loading can be seen from figures 16 and 17. The temperature curve plotted against the oil volume, corresponds to figure 14. There is a shift into the region of higher temperatures. The temperature rise at small axial loads is greater (fig. 17). The curves soon tend toward an asymptotic course. This form of curve is in agreement with the behavior of the ball and track deformations. There seems to be a certain relation between radial clearance and relative curve rise at low loading. The greater the radial clearance, the more flat the rise.

The effect of bearing clearance was studied exhaustively. The amount of radial clearance governed the height of the bearing temperature. Greater clearances resulted in lower temperatures, as exemplified in figure 18, where the temperature is shown plotted against the volume for roller bearings with different radial clearance under otherwise identical conditions. A partial explanation is found in the greater sensitivity of the bearings against minor pinching and deformation due to installation or heat effect, which naturally is more effective by little clearance. A second result of small clearances is the greater directional energy needed by the cage. The rollers, in particular, have a tendency to oblique setting by uneven reduction of mobility, and the ensuing additional accelerations increase the production of heat. Because of the squared effect of the speed, the effect increases with increasing rpm.

All cited clearances were measured on the dismantled bearing. Even on a slightly oversized shaft, the widened inner ring can induce measurable clearance reduction. The outer ring is seated with running fit in the housing, thus precluding any further clearance reduction through flattening.

With reference to the seat to be employed on the shaft, a departure from normal appeared appropriate. In accord with present-day practice, a press or driving fit on the shaft is provided for the inner ring to prevent the ring from traveling and damaging the shaft. Decisive for this hard seat is the cold-working of the ring at high outside loads and low rpm. At high rpm, the outside loads are ordinarily lower. The pre-tension effected by the seat on the shaft need only be sufficient to prevent distortion of the inner ring on the shaft at the highest accelerations and decelerations. Since these forces, referred to the load at low rpm, are substantially lower, the required springing of the inner ring - that is, the oversize of the shaft can be less with respect to the diameter of the bore. However, neither the clearance reduction through the seat nor the enlargement of the inner ring through the centrifugal force should be overlooked. Past experiences on supercharger shafts with very jerky operation, have shown that for shaft diameter of 35 millimeters, an oversize of 4μ is ample for velocity coefficients up to 735,000. In some bearings of 20-millimeter diameter, even a sliding fit proved acceptable. The friction at the side faces of the bearing was obviously sufficient.

On the basis of measurements with overhung shafts for deep-groove ball and roller bearings (35 mm diameter), the radial clearance should preferably be no less than 30μ which, compared with the usual values of 4μ to 18μ (reference 4), means an increase of more than twice as much. The operation disclosed no appreciable clearance increase through wear; hence no running in is to be expected on bearings mounted with too little clearance. Table I gives the related test values of the three clearances of several bearings; as these are mutually dependent upon the design of the bearing surfaces, however, the data represent only approximate values.

TABLE I. Related Test Values of Clearances of Several Deep-Groove Bearings 6207 in μ

Radial clearance sp_{rad}	Axial clearance sp_{ax}	Angular clearance sp_{wi} test diameter, 66 mm
8	120	270
15	120	210
25	100	300
30	160	260
40	120	260
50	150	270

Attention should be given to the high absolute values of the axial clearance, since they may induce design requirements for bearing installation: for instance, with regard to the disposition of tip sealing and clearance as well as considerations involving potential impact stresses for pressure and bearing changes. The amount of angular clearance in connection with the axial clearance shows, moreover, that a direct susceptibility of the deep-groove ball bearing to alinement errors need not occur - that is, a harsh running bearing at mounting, is at once a sign of a great error. Minor inaccuracies which the bearing can absorb, will be evidenced by a rise in temperature.

The lubricant employed has a profound effect on the operating conditions of roller as well as of deep-groove ball bearings. Higher oil viscosity effects a higher proportion of the fluid friction as supplemental producer of heat. That this increase is considerable can be seen on the roller bearing in figure 19, where the temperature difference, solely through a change of oil is, in the extreme case, 23° C. The definite thermic superiority of oil of minimum viscosity extends even to the regions of smallest quantities. As the oil temperature rises, the viscosity differences become considerably less because of the relation between temperature and viscosity, according to figure 13. However, the more favorable action of the oil with flattest viscosity curve persists even at high oil inlet temperatures, as indicated in figure 20 for an 80° C inlet temperature. These effects are greater for bearings with small radial clearance.

As the oil inlet temperature rises, the bearing temperature rises also (fig. 21). With decreasing volume and for amounts less than around 2 liters per hour, the differences are no longer measurable. This means that with oil volumes below this limit (intersection of curve), practically the entire heat created by the bearing is carried off through the housing and by radiation. With larger volumes, the oil still participates in the heat removal and can dissipate less on account of the decreased temperature difference through the higher inlet temperature. This makes a rise in temperature level obligatory until equilibrium condition has been re-established.

VII. POWER ABSORPTION

Only approximate values can be given for the power absorption. The method applied, yielded considerable scattering for such small powers, especially in the operating power which could not be sufficiently accurately defined. In the no-load calibrations, a specified internal heat condition in the gears could not be maintained constantly. However, the investigations are supplemented by measurements with the friction balance. The power increases about linearly with the rpm (fig. 22), which corresponds with the curve of the temperature (fig. 15). The relation of power to axial load also is similar to the corresponding temperature curve (fig. 17) as exemplified in figure 23. The power increase with increasing oil volume (fig. 24) is accounted for by the growing share of fluid friction. In connection with the action of the temperature in relation to volume (figs. 14 and 16), it is seen that the greater oil volume is of little practical use for the bearing temperature. To illustrate (fig. 24): from the curve $n = 26{,}000$; $P_{ax} = 60$ kilograms for a volume $Q = 20$ liters per hour, 1 horsepower is required. At an oil inlet temperature of $t_e = 20^\circ$ C, and an outlet temperature of $t_a = 85^\circ$ C, the oil is able to carry off the following heat:

$$G_s\ c_p(t_a-t_e)\ \frac{1}{A\ 75} = \frac{20}{3600} \times 0.40(85-20) \times \frac{427}{75} = 0.82 \text{ hp}$$

The other 0.18 horsepower is removed by radiation and through the housing. On the other hand, this last proportion can, for the concerned temperature gradient, amount to 0.36 horsepower on the basis of measurements. The share for the pure rolling friction amounts to about 0.5 horsepower by extrapolation of the curve $(Q = 0)$, hence leaving for the lubrication a dissipating portion of $0.5 - 0.36 = 0.14$ horsepower. But $0.82 - 0.50 = 0.32$ horsepower is removed; i.e., the lubricating efficiency is $\frac{0.14}{0.32} \times 100 =$ about 44 percent. In other words, the bearing - lubricated at the rate of 20 liters per hour - would be substantially cooler if the oil could be removed from the bearing with less internal friction. Here the cage design might account for a considerable share.

In high-speed roller bearings, the conventional cage shape forms the first cause of breakdown. With the usual system of oiling from the outside through the gap between inner race, outer race, and cage, insufficient oil reaches the actual wear spots in the bearings at high speeds. Particularly troublesome are the eddy rings caused by the rotating ball-cage system at both sides and which, as observation teaches, are in the position of largely deflecting the oil spray directed toward the gap. Externally smooth cages, and the half-round rivet heads replaced by flush rivets, might insure better results. In some designs the interspace between inner race, outer race, and cage can be enlarged for easier oil induction by reducing the cage diameter.. The use of light alloy, provided no other mechanical difficulties occur, may conceivably bring about better heat conditions because of smaller accelerative forces. The extent to which roller-body guides in the common cage shape are at all practical for further speed increases, is being studied. Guidages developed on the basis of the special requirements for high speed, are probably more appropriate.

Conversion of the measurements is briefly referred to. The absolute values are different for each installation because heat conduction and radiation are affected by the design and material of housing and shaft. If the bearing is situated within the zone of flowing mediums, the bearing heat can be considerably less on account of the greater heat removal. The explanation of this problem is being undertaken.

VIII. CONCLUSIONS

The effect of bearing installation, oiling system, oil feed and oil volume, also bearing fit and clearance on the running condition of bearings are described on the basis of test runs with single-row, deep-groove, and roller bearings with 35-millimeter bearings (6207, NL, 35e) at speeds between 16 and 21,000 rpm, corresponding to velocity coefficients of from 560,000 to 735,000. With consideration of the requirement for minimum oil volume so essential in airplane superchargers, it was found that dependable operation could be insured with oil volumes of from 5 to 0.5 liters per hour (and less in many cases) when the oil was fed through a hole in the inner or outer race. The lowest figures are for roller bearings. The radial clearance is decisive for the operating temperature. Small clearances give higher temperatures. With radial clearances over 30μ the operation was not always satisfactory on shafts without overhang. The reduction in clearance through the seat of the inner race must also be considered, even when the shaft is little oversize. Additional axial loads up to 120 kilograms are accompanied by temperature rise. Oil viscosity has an appreciable effect on the bearing temperature. The lowest viscosity gave the least heat, even by rising oil-inlet temperatures. The power required was approximately defined and amounted, for example, to about 1 horsepower at 21,000 rpm and 20 liters-per-hour oil volume. Inspection of the suitability of commercially listed bearings resulted in the preference for the light series, whereby the rising internal load on large roller body diameters is pointed out. A change in the present cage shapes so as to facilitate the entry of the oil into the bearing might make it possible to simplify the oiling system.

Translation by J. Vanier,
National Advisory Committee
for Aeronautics.

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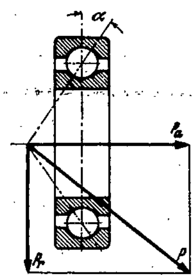


Figure 1.- Pressure angle α of a ball bearing.

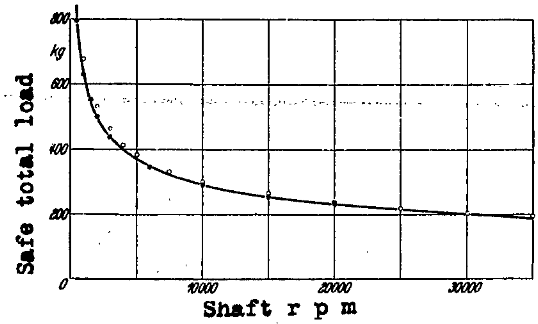


Figure 2.- Total load of bearing 6207 (35 mm diameter) plotted against rotational speed (manufacturer's list).

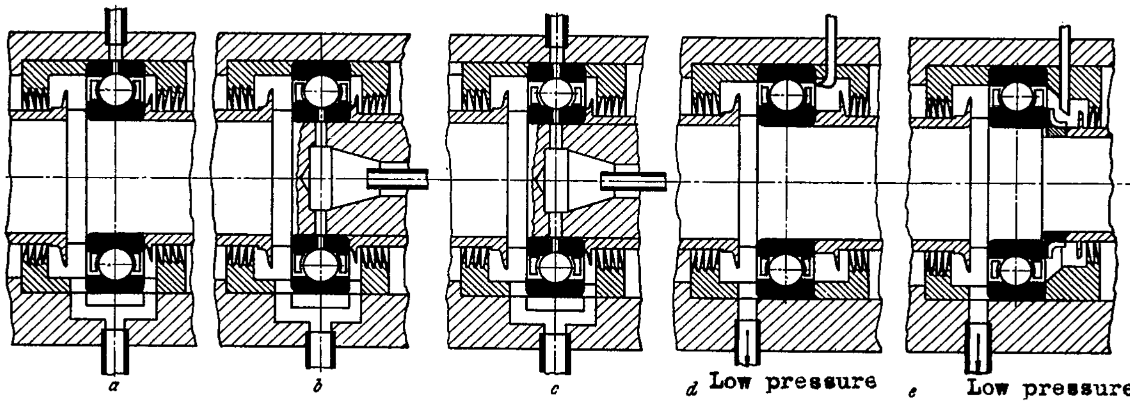


Figure 6.- Different arrangements of oil feed.

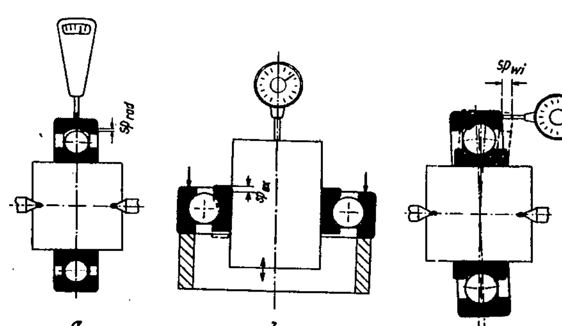


Figure 7.- Clearances of a roller bearing
(a) Radial clearance
(b) Axial clearance
(c) Angular clearance

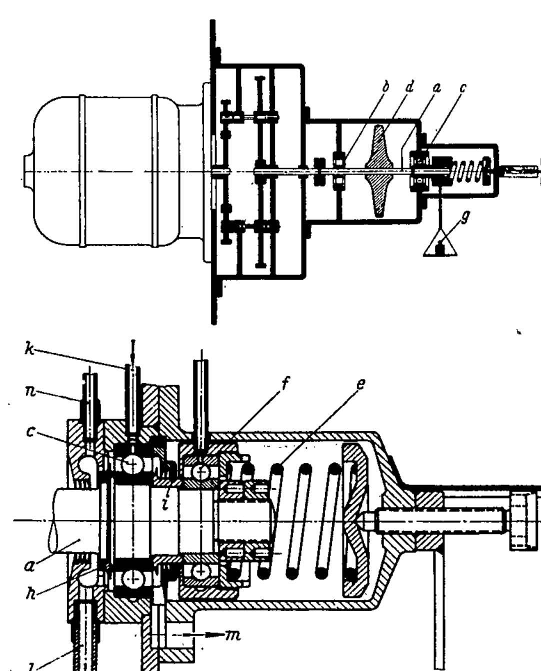


Figure 8.- Bearing testing machine.

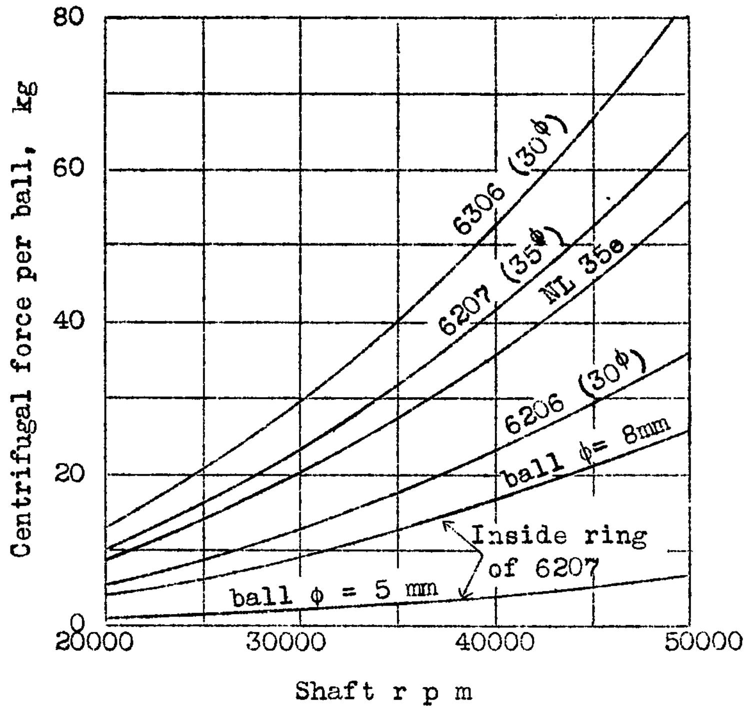


Figure 3.- Centrifugal force per ball plotted against shaft r p m.

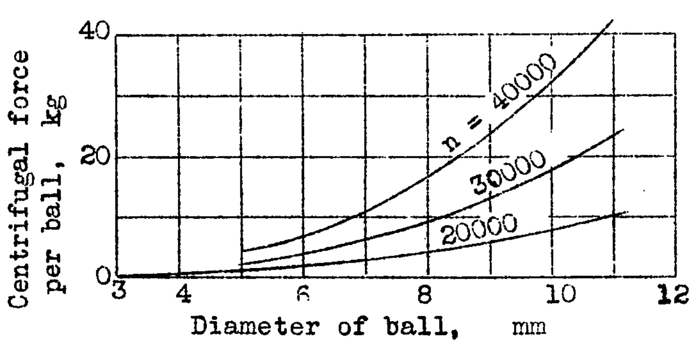


Figure 4.- Centrifugal force plotted against diameter of ball by equal inner race (6207).

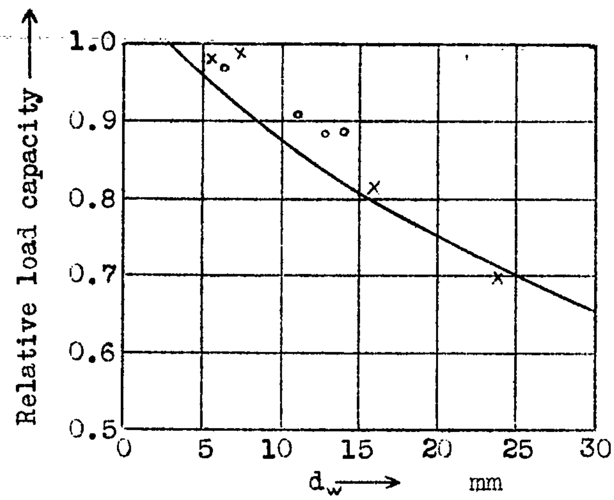
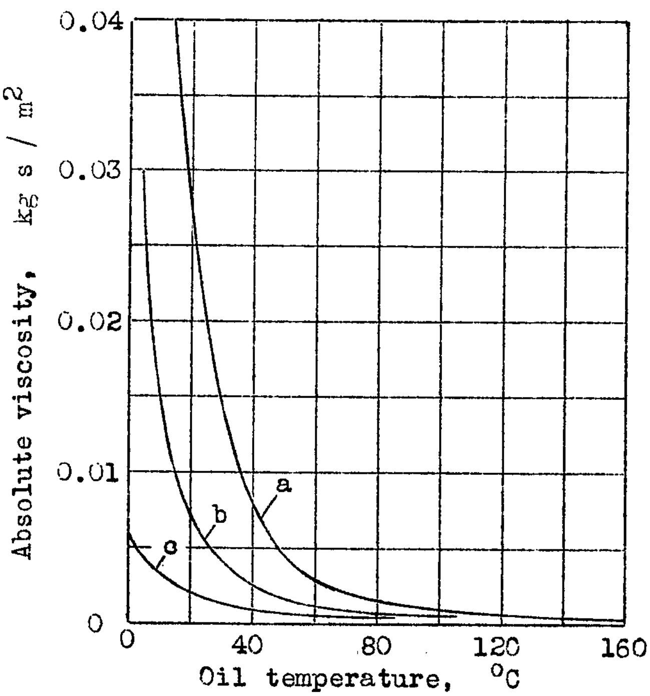


Figure 5.- Relative load capacity against ball diameter d_w.



(a) Oil No. 1 ~ 7° E at 50° C
(b) " " 2 2.8° E " "
(c) " " 3 ~ 1.7° E " "

Figure 13.- Viscosity curve of employed oils.

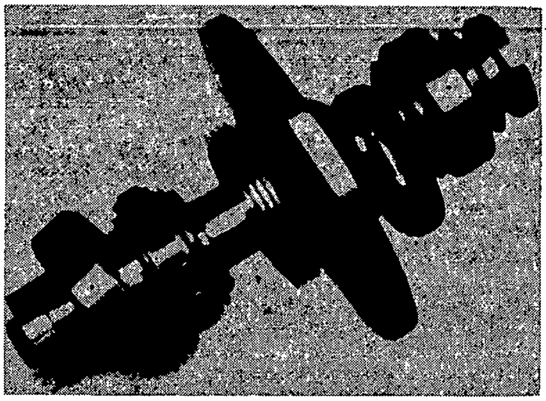


Figure 9.- Rotor assembly.

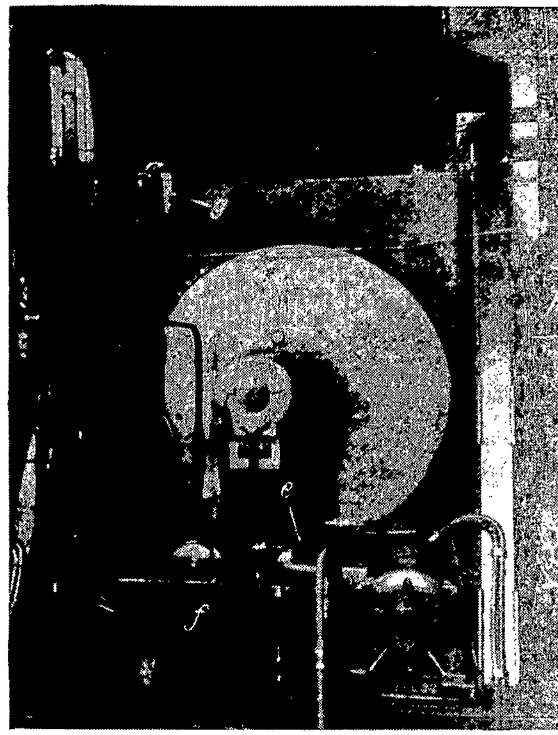


Figure 11.- Rear view of testing machine.

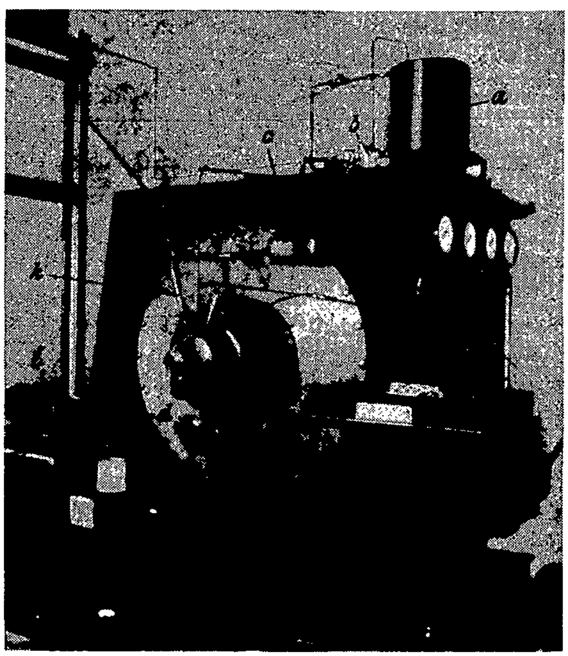


Figure 10.- View of testing machine.
(a) Oil tank
(b) Flow volume meter
(c) Heater
(d) Disc
(e) Oil cooler
(f) Intermediate tank
(g) Oil strainer
(h) Compressed air tank
(i) Bosch oiler with motor
(k) Lead to test bearing

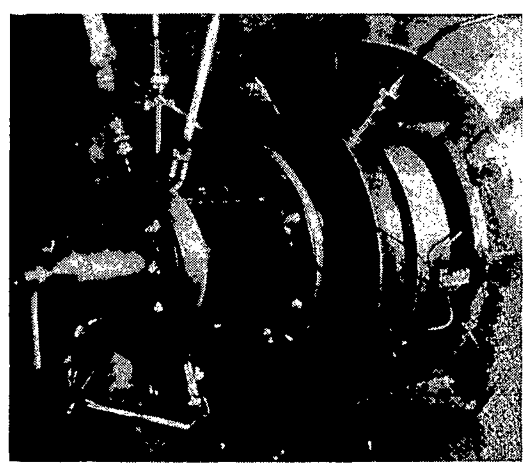


Figure 12.- Rotor and gear case.

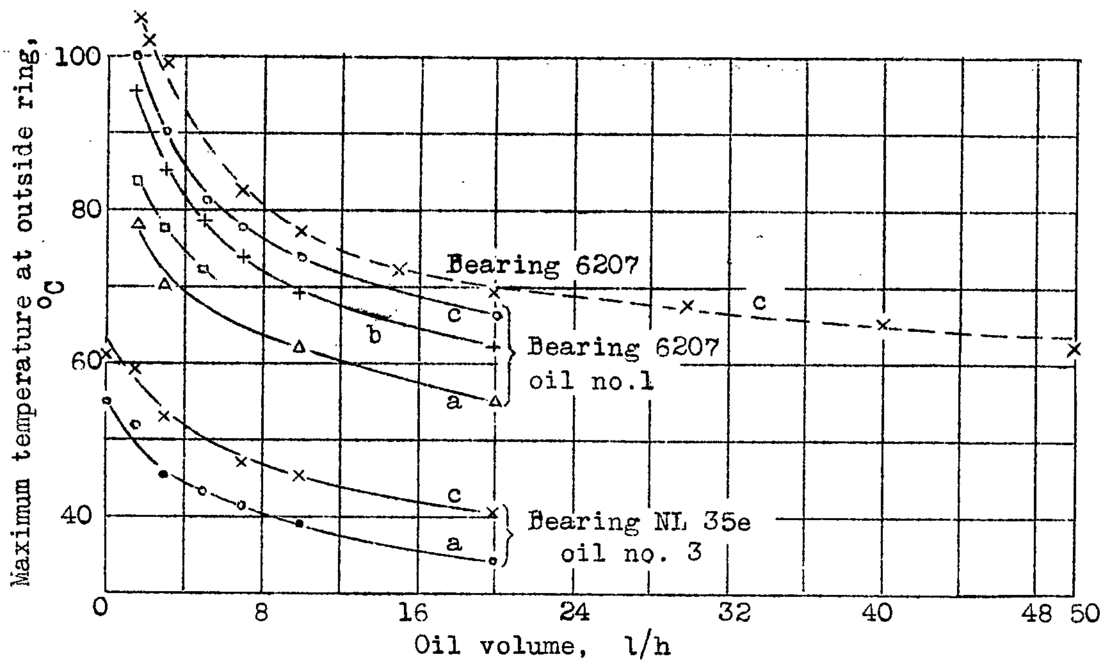


Figure 14.- Oil volume against bearing temperature.
Radial load $P_{rad} \sim 2.4$ kg, axial load $P_{ax} = 0$, oil inlet temperature $t_e = 20^\circ$ C. an = 16000 r p m , bn = 19250 r p m , cn = 21000 r p m.

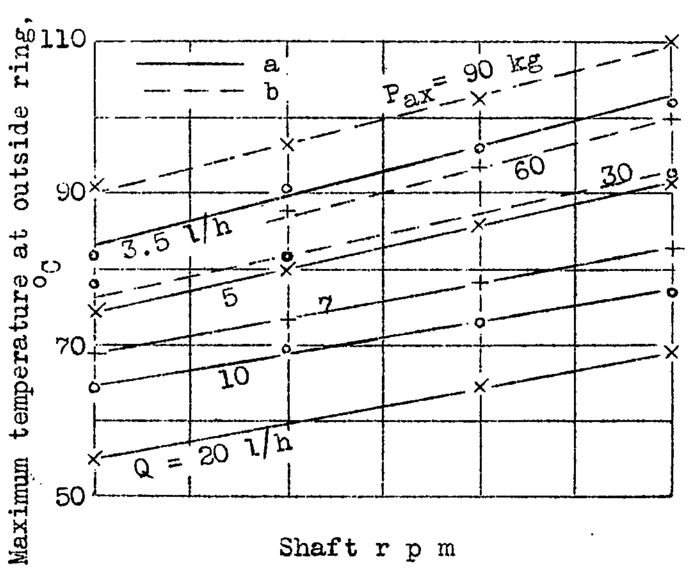


Figure 15.- Bearing temperature against r p m. Bearing 6207
$P_{rad} \sim 2.4$ kg, a $P_{ax} = 0$ kg, b oil volume Q = 10 l/h.
From left to right the ordinates with test points corresponding to n = 16000 17650. 19250 . 21000 r p m plotted on abscissa.

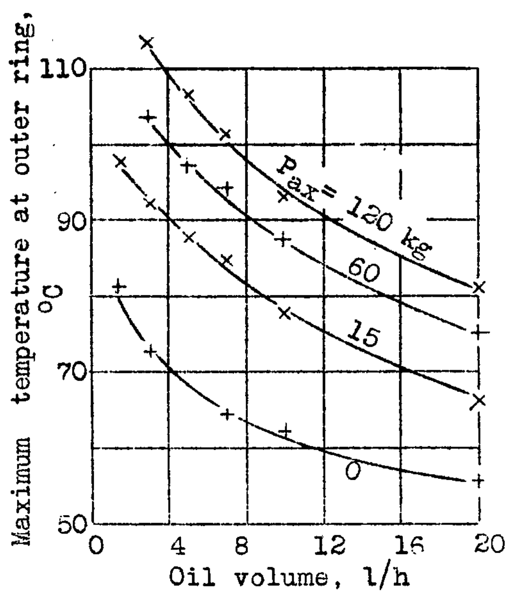


Figure 16.- Effect of additional axial load against oil volume. Bearing 6207, $P_{rad} \sim 2.4$ kg, n = 16000.

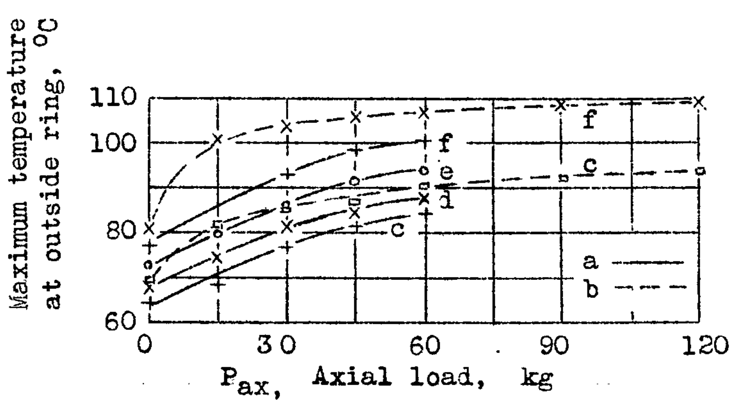


Figure 17.-Axial load against temperature.
$P_{rad} \sim 2.4$ kg, Q = 10 l/h, a radial clearance $sp_{rad} = 50\ \mu$
b " " $sp_{rad} \sim 8\ \mu$
c n = 16000 r p m e n = 19250 r p m
d n = 17650 r p m f n = 21000 r p m

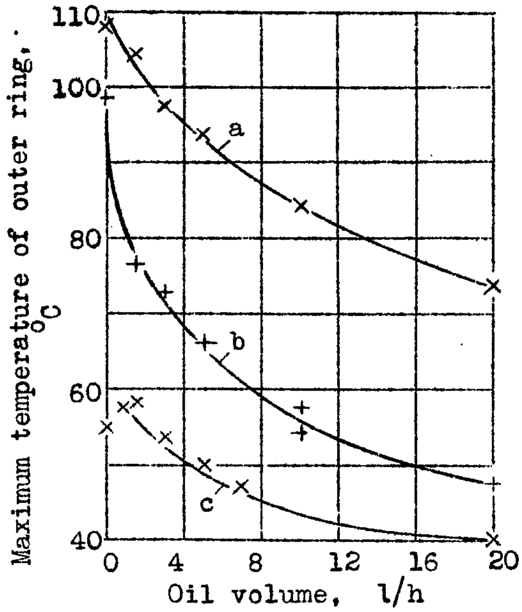


Figure 18.- Effect of radial clearance.
Bearing NL 35e, n = 21000 r p m, $P_{rad} \sim 2.4$ kg, oil no. 3.
a $sp_{rad} \sim 8$ μ, b $sp_{rad} = 30$ μ, c $sp_{rad} = 40$ μ.

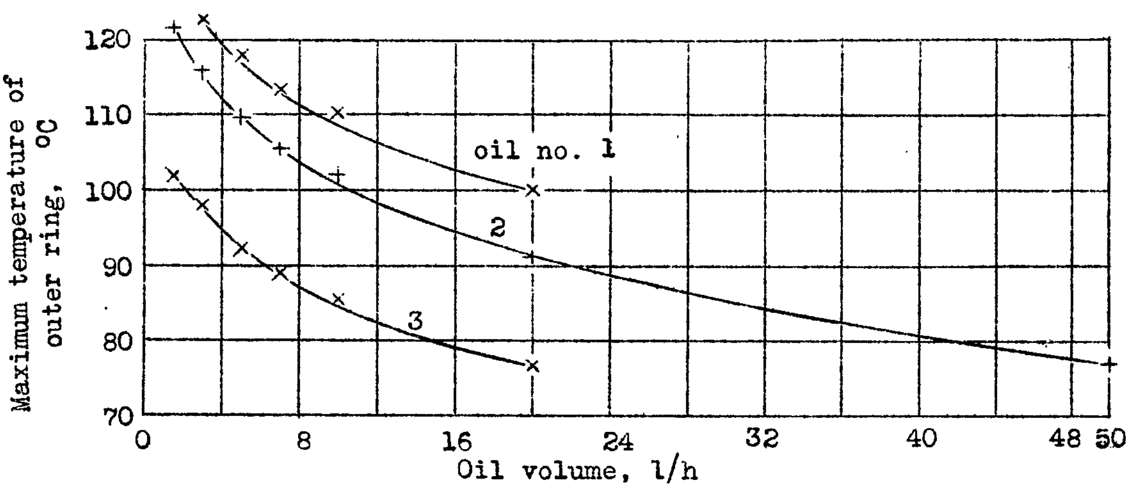


Figure 19.- Effect of viscosity.
Bearing NL 35e, oil inlet temperature $t_e = 20°C$,
n = 21000 r p m, $P_{rad} \sim 2.4$ kg.

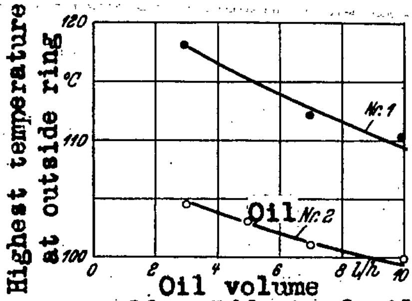


Figure 20.- Effect of oil viscosity at high inlet temperatures. Bearing NL 35e; t_e=80°C; n = 16,000 r p m; P_{rad}~2.4 kg.

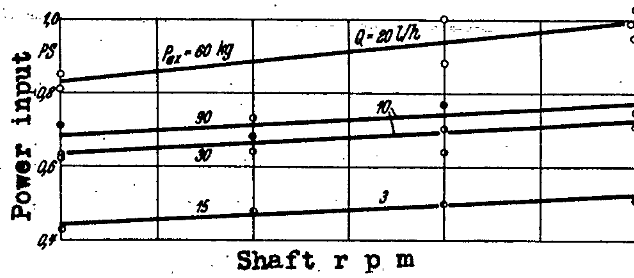


Figure 22.- Power absorbed for bearing 6207 against r p m. P_{rad}~2.4 kg ; n = 16,000; 17,650; 19,250; 21,000 r p m .

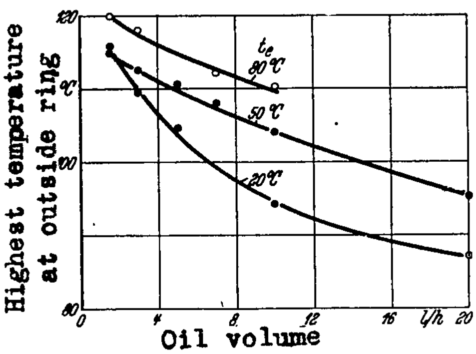


Figure 21.- Effect of oil inlet temperature. Oil number 1; n = 16,000 r p m P_{rad}~2.4 kg.

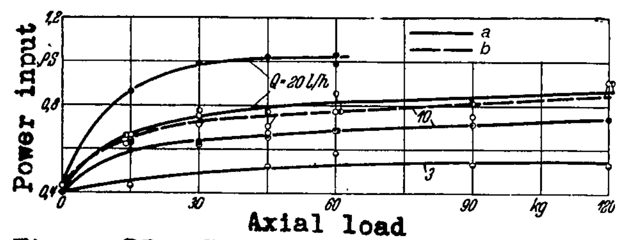


Figure 23.- Power absorbed for bearing 6207 against axial load. P_{rad} 2.4 kg ; a n = 16,000 r p m; b n = 21,000 r p m.

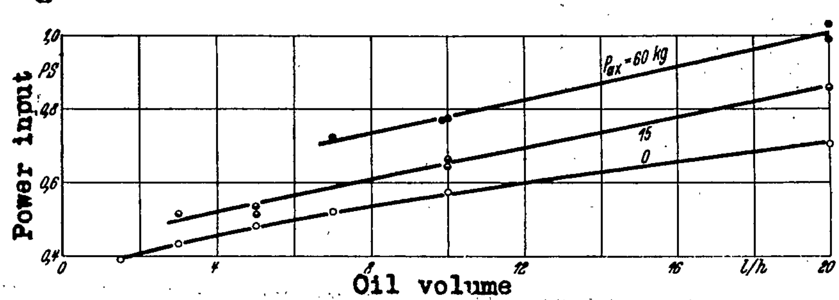


Figure 24.- Power absorbed for bearing 6207 against oil volume. P_{rad}~ 2.4 kg ; n = 21,000 r p m.