Correspondence.

Mr. Boswall.

Mr. ROBERT O. BOSWALL considered it to be extremely unlikely that proper film lubrication conditions would be obtained with a bedded brass unless—

(*a*) The brass was loaded eccentrically, that was, the load-line must be arranged so as to pass between the geometrical centre and the outlet edge of the brass.

(*b*) Precautions were taken with regard to the shape of the clearance passage through which the lubricant passed before reaching the actually bedded portion.

Further, the angle over which the brass was bedded should not exceed about 90 degrees.

With regard to Dr. Swift's Paper, the failure of bearings to give satisfactory film lubrication when the arc of embrace for the brass exceeded about 120 degrees had frequently been commented upon, and Dr. Swift's explanation of the sequence of events which preceded the ultimate establishment of the film over the complete angle of the brass was of some interest. It must, however, only be accepted as a rough approximation, owing to the omission of side leakage and temperature effects, which played an important part in the actual operation of any bearing. He was also of opinion that whilst the maximum $\epsilon \cos \phi$ condition assumed by Dr. Swift afforded an apparent solution of the problem, and one which he himself had applied to certain anomalies in connection with the operation of the bedded brass, frictional resistance, shear stress on the lubricant, and heat-distribution throughout the film had also an important influence in determining the actual operating conditions. He suggested that the motion of the journal-centre should have been considered with regard to speed- rather than load-changes, since in practice the load was usually the predetermined factor and what was actually required was some knowledge of the changes that took place whilst the journal was moving from its stationary to its normal running position. Obviously, when the journal was at rest, contact with the brass would occur at the lowest point and the line of centres would be vertical ($\phi = 180$ degrees). For instance, referring to *Fig. 2* (p. 269), the centre of the brass would be at O and the journal centre at C, the distance OC separating them being the radial clearance, and the eccentricity ratio having unit value.

As soon as the journal commenced to rotate, lubricant would be introduced. It was impossible, however, for the film to form completely over the whole arc of embrace until the journal centre had traversed a path which brought it up to the theoretical locus OPQ. The pressure and eccentricity conditions obtained by Dr. Swift for that intermediate phase of incomplete film formation appeared to give a reasonable idea as to what was likely to happen. Mr. Boswall would have preferred, however, to have values of the eccentricity ratio, ϵ, plotted to a base representing $\frac{\lambda U}{P} \cdot \frac{R^2}{r^2}$ instead of the inverse, as shown in *Fig. 7* (p. 301). That would help to indicate more clearly that the eccentricity ratio gradually decreased from initial unit value as the speed, and therefore the criterion $\lambda U/P$, steadily increased from initial zero value. A companion diagram showing changes in the attitude angle (ϕ) would also have been a useful addition. Mr. Boswall.

The effect of side leakage and temperature-changes neglected by limitations of theory would be to shift the curves in *Fig. 7* upwards, the extent of the shift being greater for the large-angle brasses, owing to their greater length/width ratio and therefore greater side leakage. It was probable, for instance, that the curve for eccentricity in the case of an ordinary 90-degree brass should be shifted upwards so as to pass through points giving eccentricity values of 0·9, 0·8, and 0·6 for load criterion values of 6, 3, and 1 respectively.

The pressure conditions shown in *Fig. 6* (p. 300) for the period preceding the attainment by the journal centre of the theoretical locus were also of importance. In the first place it was a noticeable feature that the film restricted its length on the outlet side and thus automatically adjusted itself to a condition of eccentric loading, that was, a condition in which the arc of entry exceeded the arc of departure. It should therefore be obvious that if a bearing operated in practice under conditions such as those indicated by the curves for eccentricities exceeding 0·7, an appreciable extent of the arc of departure was ineffective from a load-carrying point of view and probably added to the frictional resistance. Why not therefore dispense with it altogether? There also seemed to be very little advantage with those high-eccentricity conditions in retaining the inlet 30 degrees of the arc of entry. The data given in the Paper all pointed, in fact, to the advisability of making those alterations and thus reducing the total arc subtended by the brass. That had been confirmed by experiments carried out by Mr. Boswall.

Since the film naturally tended to operate under conditions in

Mr. Boswall. which the arc of entry exceeded the arc of departure, why should not that be arranged for when designing the bearing? That was to say, why not arrange the brass so that the load-line passed through a point lying between the geometrical centre and the outlet

Fig. 10.

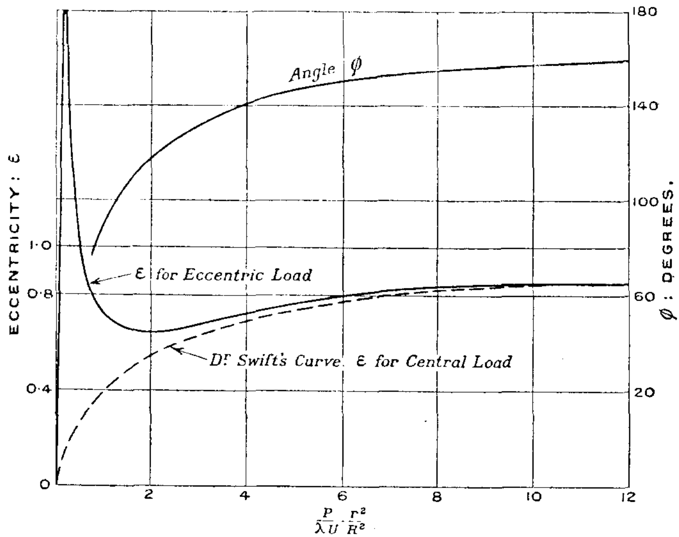
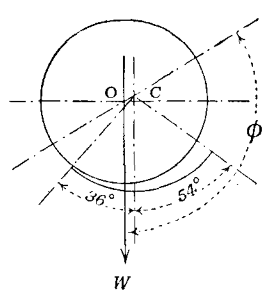


Fig. 11.



edge, instead of through the geometrical centre, as was the usual case. *Fig. 10* showed the relation between eccentricity ratio, ϵ, and load criterion, Δ, for a 90-degree brass loaded in such a way that the load-line passed through a point 54 degrees from the inlet edge (*Fig. 11*). This curve, which neglected side leakage and temperature effects, had been drawn from data given in his book,[1] and, as would be seen, linked up with Dr. Swift's curve for the 90-degree brass in the region of $\Delta = 10$. The attitude curve for angle ϕ showed that with that type of loading the film would become completely convergent over the whole arc of 90 degrees when the load criterion was less than about 5. Mr. Boswall.

That conditions of high local pressure occurring in the early stages of the motion were not detrimental so far as frictional resistance was concerned was clearly indicated by the curves in *Fig. 9* (p. 282). There was, however, a secondary effect which should not pass unnoticed. The small film-thickness which accompanied conditions of high eccentricity would be associated with small quantities of lubricant passing between the bearing surfaces. Temperature-rise of the lubricant might consequently be excessive. That would materially influence the viscosity conditions and increase the discrepancy between theory and practice.

It was not quite clear to him what Dr. Swift implied when he stated that "end leakage . . . provides a valuable means for the disposal of heat generated." The effect of reducing the axial width of a brass was to reduce the film-thickness and quantity of lubricant and to increase the friction coefficient. The rise in temperature of the lubricant would thus increase as the width/length ratio was reduced.

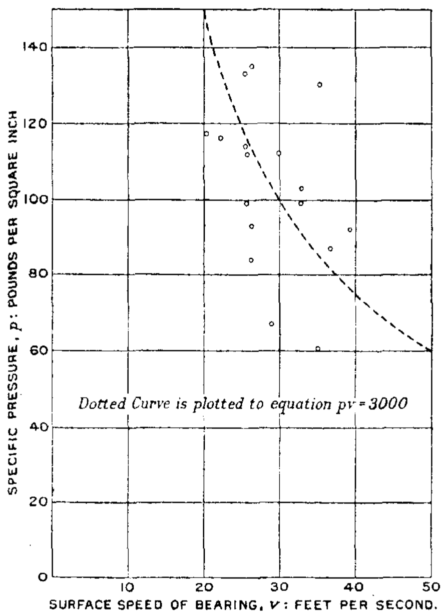
He suggested that the use of the "poise" for unit viscosity was preferable to the foot-pound system. There was undoubtedly a need for standardization of notation and also units of measurement. It was only after some trial calculations that he had discovered that P was expressed in pounds per foot width, instead of pounds per inch width as was usual. It might also be pointed out that the combination of the term r/R with the criterion $P/\lambda U$ and the coefficient of friction was strictly justifiable only when temperature conditions were neglected, as in the case of the data given in the Paper.

Mr. H. H. Broughton was of the opinion that it was difficult, if not impossible, at short notice to discuss either Paper critically, but he believed that the experimental results obtained would be of benefit to future workers in their search for a solution of a highly complicated problem. The amount of work that had been done was impressive, a large number of experiments or observations being needed to make each of the diagrams of results which Professor Goodman had presented. The fact that, notwithstanding the with- Mr. Broughton.

[1] R. O. Boswall, "The Theory of Film Lubrication." London, 1928.

Mr. Broughton. drawal of the Government grant, Professor Goodman had been able to conduct his research single-handed in a garage, was a splendid example not only to his old students, but also to the whole of the engineering profession. Notwithstanding the complexity of the theory and the conflicting conditions, both of which were dealt with in the Paper, plain journal bearings were in use, and they could be relied upon to give satisfactory service under conditions which a few years ago would have been regarded as impossible.

Fig. 12.

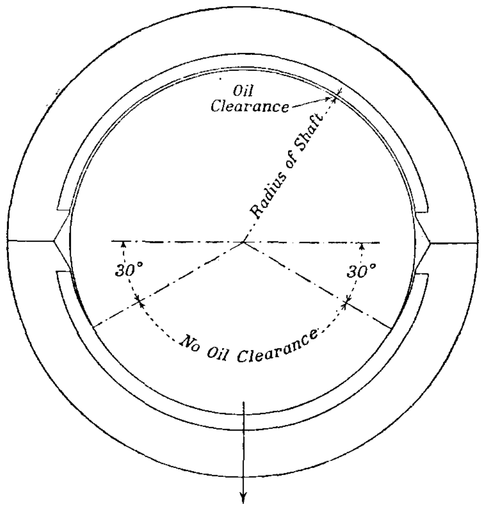


He thought that served to show how designers had by compromise succeeded in combining the desirable elements in a bearing. Like many other time-honoured rules, the *pv* rule was shown to be seriously in error. He had investigated[1] a number of high-speed heavily-loaded bearings made by seven reputable firms. The relation between specific pressure, p, and surface velocity, v, plotted in *Fig. 12*, fully confirmed what Professor Goodman stated; but it should be remembered that designers of high-speed bearings had for many years used a value of specific pressure which was a function of the nth root of the speed, where n itself ranged from 2 to 5,

[1] H. H. Broughton, "Electric Winders." London, 1927.

depending on the speed. In the Paper, if he interpreted it correctly, further research was foreshadowed, and to the items enumerated on p. 265 he would like to see added the type of bearing depicted in *Fig. 13,* with oil clearances as shown in *Fig. 14.* Probably the oil clearance would be increased by 0·005 inch for surface speeds of 40 feet per second, and by 0·01 inch for speeds of 50 feet per second. Had Professor Goodman considered the feasibility of expressing the results of the elaborate series of Mr. Broughton.

Fig. 13.



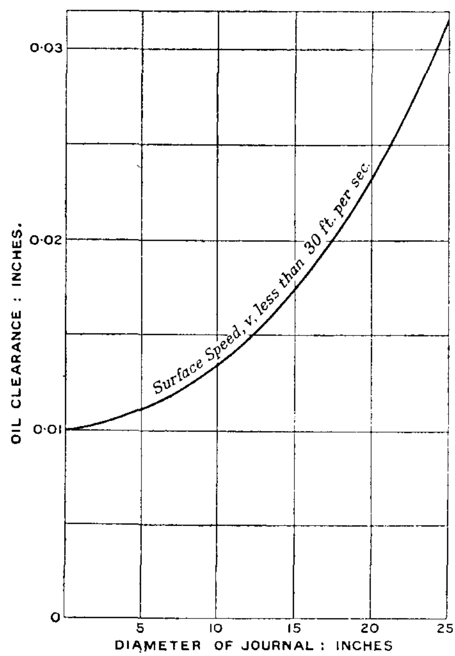
experiments by means of three-dimensional or solid models? If the method could be adopted he thought it would be beneficial.

Mr. Louis Illmer, of Cortland, N.Y., congratulated Professor Goodman upon having made another outstanding contribution in a difficult experimental field in which he had laboured zealously for many years. He believed these newer test findings to be of decided importance in advancing the art of lubrication, particularly so in that they were concerned with two extreme aspects which had not hitherto been adequately investigated, namely, the behaviour of loosely-fitted journals (Part I), and the behaviour of snugly-fitted journals, to which Part II was primarily devoted. Having endeavoured to correlate empirically and make a tabulated survey of friction coefficients obtained by many different research workers,[1] Mr. Illmer.

[1] Journal of the Society of Automotive Engineers, vol. 26 (1930), pp. 67 and 785.

Mr. Illmer. he had found Professor Goodman's series of authoritative tests of decided significance in supplying indispensable data. A brief analysis of Professor Goodman's results on the basis of the survey in question might be of interest. The mode of treatment resorted to in Mr. Ilmer's Papers fixed the friction coefficient for various degrees of journal fit in terms of viscosity and rubbing velocity, the product of which was divided by the projected unit pressure, each

Fig. 14.



such factor being given an individual exponent in accordance with actual test results. As applied to journals that were floatingly upheld by a substantially unbroken oil film the foregoing was expressed by

$$\mu_o = \frac{Cn^3 \lambda^{0\cdot5} U^{0\cdot33}}{7\cdot5 (P/S)^n} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where C denoted a circumstance constant equal to unity for flooded and otherwise most favourable conditions; n was a pressure exponent; P/S denoted the projected pressure, in pounds per square inch, divided by the effective contact factor S, that was, the fraction of journal actually utilized; λ was the absolute viscosity of the lubricant in poises or C.G.S. units; and U was the rubbing-velocity in feet per minute. The contact factor varied with the journal-clearance, being relatively low for a loosely-fitted journal. Applying the given deduction to Professor Goodman's Part I friction determinations, the results were:— Mr. Illmer.

"Castrol A" (heavy) oil $\mu = \frac{\lambda^{0\cdot5}\, U^{0\cdot5}}{260\, P^{0\cdot5}}$

"Castrol F" (medium) oil $\mu = \frac{\lambda^{0\cdot5}\, U^{0\cdot6}}{195\, P^{0\cdot6}}$

Spindle (light) oil $\mu = \frac{\lambda^{0\cdot44}\, U^{0\cdot56}}{230\, P^{0\cdot6}}$

Composite $$\mu = \frac{1}{300}\,\frac{\lambda^{0\cdot5}\, U^{0\cdot56}}{P^{0\cdot56}} = \text{abt.}\,\frac{1}{80}\,\frac{\lambda^{0\cdot5}\, U^{0\cdot33}}{P^{0\cdot56}} \quad . \quad . \qquad (2)$$

Those relations had been found from test data which Professor Goodman had kindly furnished, and the tests had been conducted in the order in which they were listed. The contact factor S for equation (2) was estimated at about 7/16 with a corresponding value of $r = 0{\cdot}013$, instead of the original $S = 3/8$ at $r = 0{\cdot}0172$, thus indicating progressive wear during the test period. As a basis for comparison, the Tower test might be evaluated by giving unity value to each of the C, n and S factors in equation (1). Film-borne-journal friction was characterized by a positive value of the exponent n which decreased in a definite relation to the factor S.

As a more important result, the stabilized friction coefficients reported in Part II and pertaining to the embedded type of journal had been adequately investigated for the first time. The following relation co-ordinated Professor Goodman's findings:—

$$\left.\begin{matrix}\text{Bedded journal}\\ @\ r = 0 - \text{bath}\end{matrix}\right\}\ \mu = \frac{1}{7\cdot45}\,\frac{\lambda^{0\cdot33}\, U^{0\cdot25}}{P^{0\cdot75}} = \text{abt.}\,\frac{1}{13\cdot3}\,\frac{\lambda^{0\cdot33}\, U^{0\cdot33}}{P^{0\cdot75}} \quad . \quad . \qquad (3)$$

As compared with a flooded journal operating with a sufficiently large radial clearance to provide for a reduced pressure-exponent of $n = 3/4$, the bedded journal showed a 4/3 higher friction-coefficient. On the basis of a unity n factor as in the Tower tests having an estimated radial clearance of about 0·002 inch, the bedded journal revealed a coefficient about twice as large. On the other hand, the friction for a more loosely-fitted journal, as derived from

Mr. Illmer. equation (2), was materially in excess of that of the embedded journal. No consistent equation could be derived for the embedded-pad lubrication tests.

Referring to Fig. 86, Plate 1, in which the bedded bearing friction was plotted against $\lambda U/P^2$, that was hardly justified, notwithstanding that a reasonably smooth curve was obtained. Equation (3) made it evident that the friction coefficient actually fell at a much lower rate with pressure-increase. Attention might also be directed to Fig. 85, Plate 7, in which the bedded bearing friction was plotted upon the conventional $\lambda U/P$ base, but in which no rational co-ordination was obtained. The repeated plotting of test results on such a grouped basis was open to objection. In the publication already referred to, issue had been taken with this prevalent method, and Mr. Illmer seriously questioned whether the generally accepted $\lambda U/P$ relation constituted a correct measure for friction coefficients, since they were almost invariably found to vary approximately as the square root of oil-viscosity and some root or logarithmic function of the journal rubbing movement. Equations (1) to (3), which were based upon Professor Goodman's own test results, likewise went to prove that contention.

Mr. McKee. Mr. S. A. McKee, of Washington, D.C., observed that Professor Goodman had presented valuable data which added confirmation of the fundamental ideas upon which the hydrodynamic theory of journal bearings was based. If practicable, it would be of interest to compare the results of his tests with some of the mathematical treatments that took account of side leakage. The oil-pressure distribution curves (Figs. 50–60, Plates 5 and 6) were of particular interest to Mr. McKee, being of the same general shape as those obtained [1] at the Bureau of Standards, using a 360-degree bearing widely different in design. The most marked difference between the two sets of data was that the 360-degree bearing had a region of limited extent where the pressure was below atmospheric. With reference to Professor Goodman's discussion of the general lubrication equation, Mr. McKee cited some experiments [2] conducted at the Bureau of Standards on the frictional characteristics of 360-degree journal bearings having the same diameter but various clearances and lengths. The results when plotted in the form of $(f - ZN/P)$ curves (where f denoted the coefficient of friction,

[1] S. A. McKee and T. R. McKee, "Pressure Distribution in Oil Films of Journal Bearings." Am. Soc. Mech. Eng., December, 1931.

[2] S. A. McKee and T. R. McKee, "Friction of Journal Bearings as Influenced by Clearance and Length." Trans. Am. Soc. Mech. Eng., A.P.M., p. 161; Mech. Eng., vol. 51, p. 593.

Z the absolute viscosity, N the speed of the journal, and P the pressure on the projected area) indicated not only that f was a function of ZN/P for a given bearing, but also that the effects of changes in clearance upon the $(f - ZN/P)$ curve were in general of the same order of magnitude and in the same direction as had been predicted by the hydrodynamic equations developed by Sommerfeld or Harrison for 360-degree bearings. Mr. McKee.

Mr. Henry Shore, of New York, remarked that Dr. Swift pointed out the difficulties entailed in a complete mathematical analysis of journal lubrication. But a start in the problem could be made by introducing some simplifying assumption. Fortunately, the problem need not be so highly idealized as to offer little help in actual design work, for science now had the tool and means to attack complex differential equations which formerly were incapable of solution or else entailed excessive computation. He referred to the differential analyser.[1] With that machine, solutions in the form of plotted curves were given for ordinary differential equations for specified boundary conditions. To set up the machine for a given problem required but a few hours, and when once that had been done, only 10 minutes was required for each solution corresponding to a given set of boundary conditions. Thus means were available for obtaining complete curves of bearing performance with a minimum of effort, which would be of incalculable value to the designer of bearings. He understood from Dr. V. Bush, of the Massachusetts Institute of Technology, where the analyser had been developed and built, that the machine was available in the Department of Electrical Engineering for any industries desiring to make use of it. The use of the differential analyser would permit of the variation of viscosity with pressure being introduced into the solution: with certain lubricants that variation might be as high as 100 per cent. for the pressures encountered in some bearings.[2] It would also permit of taking into account the effect of end leakage, thus placing more complete design data in the hands of the engineer. Mr. Shore.

He would like to call attention to Papers[3] by Dr. M. D. Hersey

[1] V. Bush, "The Differential Analyser: a new Machine for solving Differential Equations." Journal of the Franklin Institute, vol. 212, p. 447. ["Engineering Abstracts," No. 51 (April, 1932), p. 15.]

[2] M. D. Hersey and H. Shore, "Viscosity of Lubricants under Pressure." Mechanical Engineering, Vol. 50 (1928), p. 221.

[3] "The Laws of Lubrication of Horizontal Journal Bearings." Journal of Washington Academy of Science, vol. 4, No. 19, November 19, 1914.

"On the Laws of Lubrication of Journal Bearings." Trans. Am. Soc. Mech. Eng., vol. 37 (1915), p. 167.

"Problems of Lubrication Research." Journal Am. Soc. Naval Eng., vol. 35 (1923), p. 648.

Mr. Shore

on the laws of lubrication, which should provide a powerful tool to extend the excellent experimental data obtained by Professor Goodman, who recognized that an extension of the theory contained in his Paper was necessary to embrace the field thoroughly. The Papers referred to made use of dimensional reasoning, which, when combined with experimental results, afforded an elegant and potent means of attack on the general problem of journal-bearing design and so supplied the extension for which Professor Goodman asked.

Professor Goodman.

Professor GOODMAN, in reply, thought Mr. Boswall had overlooked the fact that " proper film lubrication conditions " had actually been obtained with a bedded bearing centrally loaded and embracing an arc much greater than 90 degrees, namely, 146 degrees (p. 257). It had been suggested that the chamfered approach and recess channels enabled film conditions to occur which would have been impossible if the bedded surface had terminated abruptly. Since the reading of the Paper the chamfered channels had been cut away, leaving square abrupt entrance and exit edges to the bedded surface. Tests under those conditions had given to all intents and purposes the same results as before, thus showing that the chamfered oil channels had no appreciable effect as regarded securing true film lubrication. A sharp-edged scraper had then been fitted to the inlet edge of the bearing and kept in contact with the shaft by means of a light spring. It had been realized that such a procedure was very drastic and that a great risk would be run of damaging the shaft and bearing; but as it was the extreme condition that might occur with a bedded bearing it had been deemed worth running the risk. Seizing occurred and both shaft and bearing were damaged.

In some early tests made by him a white-metal bearing having a normal amount of clearance was tested with bath lubrication. Under light loads up to about 300 lbs. per square inch the oil-pressure diagrams were quite normal, such as those given in Fig. 55, Plate 6, the maximum pressure increasing proportionally to the load on the bearing. At higher loads the white metal began to flow and the maximum pressure fell below that corresponding to 300 lbs. per square inch. The diagrams bulged out sideways. When the load on the bearing amounted to about 700 lbs. per square inch the metal had flowed to such an extent that the bearing was a perfect fit—or was perfectly bedded on the shaft. The oil-pressure diagram then resembled those shown in Fig. 80, Plate 7. The pressure at the inlet and outlet edges was very high and was approximately atmospheric at the crown. The actual diagrams were reproduced in the Proceedings of The Institution of Mechanical Engineers.[1]

[1] Vol. 122 (1932), p. 549.

Mr. Broughton's suggestion that bearings of the type shown in *Fig. 13* (p. 315) should be submitted to tests was well worth the attention of some other research worker. His own programme for future work on half bearings would take many years to complete, hence it was improbable that he would be able to undertake tests on full bearings. It was hoped that three-dimensional models might be constructed, when the available experimental data were more complete, but at present they were insufficient for the purpose. Professor Goodman.

The remarks by the American contributors were very welcome. Apparently much more interest was taken in experimental work by scientific men in the United States than by European workers, who seemed to regard theories as of more importance than facts. Mr. Illmer has done very valuable work in reducing the experimental results of many workers; it was the first real step towards putting modern research results into a form in which they could be utilized by practical men and designers.

Dr. SWIFT appreciated the thoughtful comments of Mr. Boswall, who, however, was mistaken in supposing that the stability theory referred only to the preliminary stages of the development of an oil film. The theory was founded on the basic mechanical principles of stable equilibrium, and was confirmed descriptively by experimental results obtained under steady running-conditions, which showed that at higher eccentricities, as Mr. Boswall himself admitted, a continuous film never became established over the complete angle of the brass. Mr. Boswall's preference for speed rather than load as the predominant factor in the criterion $\frac{P}{\lambda U} \frac{r^2}{R^2}$ could be understood so long as he confined his interest to the transient periods of starting and stopping; but the designer was more concerned to specify the correct loading or clearance for a bearing to run at a predetermined speed, and in that case the examples on pp. 284–286 suggested that the load criterion was convenient as it stood. The polar diagrams showed clearly the relationship between the attitude angle ϕ and the eccentricity, and would give Mr. Boswall all the information necessary to plot ϕ against any other relevant quantity. Dr. Swift.

He would agree with the suggestion, arising from the stability theory, that bearings should be loaded forward of the centre if they were required to run in one direction only and were compelled to run at an eccentricity exceeding 0·6 or thereabouts. But the most efficient conditions should be obtained with an eccentricity in the neighbourhood of 0·4, when the whole bearing arc would be effective, and the case for non-central loading had in those circumstances to be based on other premises. *Fig. 10* showed that the advantage of reducing the bearing angle could not, as Mr. Boswall

Dr. Swift. suggested, be held as proved by the data in the Paper. He maintained his view that side leakage provided a valuable means for the disposal of heat generated. It was not found in practice that long bearings gave lower coefficients of friction than short ones; modern practice tended to adopt short bearings for the contrary reason; and the shorter the bearing the greater would be the proportional end leakage of oil and therefore of heat. There was unquestionably a need for the standardization of notation and units. When dimensionless criteria were employed, as they were with great advantage in problems of film lubrication, it was essential, if full advantage was to be taken of the generalizations and simplifications which they offered, to express all relevant quantities in a single consistent system of units. The foot-pound-second system was accepted by British engineers and had therefore been adopted in the present treatment. The only practicable alternatives for the British engineer in lubrication calculations were either to convert viscosity once for all to British units by a single factor and work entirely in British engineering units, or to work in C.G.S. units and subsequently convert every result from that system. The former alternative was obviously preferable.

He was indebted to Mr. Henry Shore for his reference to the differential analyser. He only hoped that such a machine might be available for the patient enthusiast who should ultimately undertake the theoretical study of the short journal bearing.