The Author. pounds and feet (or inches), whereas chemists and physicists preferred dynes and centimetres. The conversion from one system to the other was a simple matter.[1] If the number expressing the viscosity was only required for comparative purposes the c.g.s. units were preferable, but when the viscosity was required for the purpose of calculating the frictional resistance of a bearing, it was far more convenient to have it expressed in pounds and inches, since these were the units invariably used in this country for engineering purposes. Even those who used the c.g.s. units never had their testing-machines and spring-balances graduated in dynes, but always (in Britain) in pounds or tons. In view of Mr. Evans's kindness in determining the viscosity of the oil used in the tests now under consideration, it might seem ungracious to question the accuracy of his determinations; but everyone familiar with the ordinary type of viscometer knew that the viscosity depended upon the number of seconds taken by a given quantity of oil to flow through the instrument. In reducing the results from seconds to viscosity, one of several reduction formulas was used, but since they did not all agree and no one seemed to know which was right and which wrong, there did appear to be room for doubt. The Michell viscometer, which had been tested at the National Physical Laboratory and stated to give very reliable results, gave in the present instance values differing slightly from those obtained by Mr. Evans. The Author therefore considered that it was desirable to take a mean between the two. Even when that was done it was possible that there was an error of, say, 5 per cent.; but that was a small matter compared with the discrepancy between the theoretical and experimental results.

## Correspondence.

Mr. Boswall. Mr. R. O. Boswall observed that during the past 7 years he had carried out various experiments at the College of Technology, Manchester, with bearings operating under film-lubrication conditions and had also gone fairly thoroughly into the more

[1] Since 1 pound per square foot = 479 dynes per square centimetre, an oil which has a viscosity of 1 lb. per square foot has a viscosity of 479 dynes per square centimetre, and an oil having a viscosity of 1 dyne per square centimetre has a viscosity of $1/479 = 0 \cdot 00209$ lb. per square foot. Also 1 lb. per square inch = 68,970 dynes per square centimetre: hence an oil having a viscosity of 1 dyne per square centimetre has a viscosity of $1/68970 = 0 \cdot 0000145$ lb. per square inch.—J.G.

theoretical aspect of the subject. As a result he had concluded that it was almost impossible to obtain satisfactory film-lubrication conditions when the brass subtended an arc of contact as large as that employed by the Author, namely, 180 degrees. That conclusion appeared to receive some confirmation from the results of the tests described in the Paper. It was naturally difficult to avoid the discrepancies between theory and practice, which might arise not only from the small irregularities in the shape of the bearing-surface caused by defects in mechanical finish or workmanship but also from the distortion produced by temperature-changes and pressure-effects. Apart, however, from the fact that the abrasive action of metallic particles or dirt in the lubricant might possibly cause a certain amount of wear and tear of the bearing-surfaces, a perfectly lubricated surface should not experience the excessive local wear and change in shape that apparently had occurred with the bearing tested by the Author. There appeared to be distinct indications that, although film-lubrication conditions were certainly in existence on the inlet side of the brass, that type of lubrication degenerated into a state of greasy or imperfect lubrication over a region of indeterminate length on the outlet side of the crown of the bearing. The pressure-curves, for instance, Fig. 22, Plate 4, indicated that the film-pressure ceased at a point situated roughly 50 degrees from the outlet edge. Taken by itself that did not necessarily mean that surface contact would occur in the neighbourhood of that point, since the failure of the pressure conditions was probably due to the fact that with a long length of divergent film, such as occurred at the outlet side of the bearing under discussion, a state of negative pressure was almost certain to be set up in the film. The fluid could not, however, withstand pressures which fell below the vapour-pressure condition, and in addition the presence of a negative pressure allowed air to leak into the clearance-space. The combined effect was to break down the continuity of the film. The lubricant, if any, that passed across the last 50 degrees of the brass was therefore of no assistance in carrying loads, and was certainly detrimental from the friction standpoint. The brass, in fact, might as well have been terminated at or near the point where the pressure film ceased to be effective. That would have the effect of reducing the arc of contact to about 130 degrees, of which 90 degrees was on the inlet, and 40 degrees on the outlet side of the crown. The fact that wear took place near the crown seemed to indicate that the surfaces might be virtually in contact or, at any rate, only separated by a very thin layer of lubricant in that region. In that connection Mr. Boswall.

Mr. Boswall. the temperature tests carried out by the Author appeared to have some significance. With perfect film lubrication the frictional work was done upon the lubricant and not upon the surfaces which the lubricant was supposed to separate. The heat was thus generated by the frictional work directly in the lubricant, and any increase that took place in the temperature of the bearing-surfaces and housing must be due either to heat received from the lubricant as it flowed at a relatively high speed through the clearance-space or, alternatively, to heat conducted or radiated from the oil contained in the bath from which the journal picked up its supply. Since it was an essential feature of film lubrication that the flow was viscous, it seemed reasonable to suppose that the greater part of the heat generated would be retained by the lubricant, and it would therefore appear that if the lubrication were perfect the temperature of the oil in the bath would exceed the temperature of the bearing, at any rate in the early stages of rotation or until steady temperature-conditions had been attained by conduction or radiation. The curves given in Fig. 16A, Plate 3, however, indicated a reversal of that condition, the bearing being consistently hotter than the bath. The cooling-device fitted in the oil-bath would, of course, modify the temperature-conditions, and it was not clear from the text whether the oil was being cooled during the period over which the data indicated by the curves in Fig. 16A were being taken. There did, however, seem to be a possibility that the bearing-surfaces were receiving heat in some way other than by conduction from the film, and a natural inference was that heat was being generated over what might be quite a small length of bearing-surface operating under greasy-lubrication conditions. Such heat would naturally be transmitted almost entirely to the bearing-surfaces, and would account for the temperature-conditions recorded. The Author stated that end leakage decreased as the speed increased, and that there was a corresponding increase in the film-thickness. The thickness of the film would certainly increase as the speed increased, but of the total quantity of lubricant drawn into the clearance-space practically the same percentage left in all cases by way of the lateral edges, the small change that occurred in the attitude of the bearing with change of speed having very little effect in that respect. The fact that bearings failed to work satisfactorily under film-lubricated conditions was not entirely due to insufficiency of lubricant, although that was a contributory factor. The actual conditions were somewhat complicated, the trouble being associated also with temperature-effects. As the lubricant passed through the clearance-space its temperature rose continuously, owing to the heat generated by the frictional work, and when the lubricant reached the outlet edges of the surface its temperature might be high and its viscosity correspondingly reduced. If the simple assumption was made that all the heat generated was transmitted to the lubricant, the approximate temperature-rise for any given set of conditions could be calculated, and it could be shown in that way that as the speed increased there was first a decrease and then an increase in the temperature-rise, there being a particular speed for which the temperature-rise was a minimum. For very low and also for very high speeds the temperature-rise might be excessively high and cause a breakdown in the film conditions. With bearings that ran normally at speeds corresponding to reasonable temperature-changes, any excessive temperature conditions would be rapidly passed over during starting, and no serious effects were likely to arise. On the other hand, however, if the normal speed was too low, the film would probably fail to form properly and the lubrication would be of the greasy type, the heat generated being mainly transmitted to the bearing-surfaces and dissipated by radiation or conduction in the usual manner. Although perfect film-lubrication conditions had not been altogether attained in the experiments carried out by the Author, the results obtained were of some considerable value, and from the description given the test-apparatus would appear to be particularly suitable for the work for which it had been designed. He would suggest, however, that subsequent experiments should be carried out with arcs of contact reduced to 120 or even 90 degrees, as with such dimensions it was probable that much more consistent results would be obtainable. Mr. Boswall.

Mr. H. H. Broughton was of the opinion that careful examination of the results of the exhaustive series of tests would be necessary before one could hope to be in a position to discuss the Paper. Inspection of the curves gave an idea of the vast amount of work entailed in the making of the tests. Many months had been spent on other tests of which a résumé was given, but the results were "not regarded as being worth recording." He thought that when engineers had had time to study the Paper properly, it would be found that the research was an important continuation of the Author's earlier work on the subject. Those who had not time to interpret the meaning of the curves would welcome a summary of the results, and he would be glad if the Author would furnish such a summary. Not the least valuable part of the Paper was that which dealt with the thermal characteristics of bearings; and the group of curves, Figs. 16A to 16D, Plate 3, was of special interest to him. The theory of heating and cooling had been investigated by a number of engineers, and had been made intelligible to them by the Mr. Broughton.

Mr. Broughton. electrical engineers, who were among the first to recognize its effect on the output of the machinery they designed. In that connection the pioneer work of Mr. E. Oelschläger was of the highest importance and, as far as he knew, it was the first in which the thermal characteristic had been used for predetermining the output of an intermittently-operated machine. Other work, which perhaps appealed more strongly to mechanical engineers, was that of Messrs. C. Bach and E. Roser. There again the thermal characteristic was the foundation of the method, and the method was now in everyday use for determining the loads which worm reducing-gears would transmit. Curves such as those just mentioned enabled engineers to design bearings that would not overheat. The problem was complicated by the large number of variables; but, with a given bearing lubricated with oil of known quality, the load the bearing was capable of carrying without overheating could be ascertained from curves connecting temperature-rise and time. The high speeds at which steam-turbines and electrical machines were run, and the conditions under which the machines were operated, had necessitated a revision of ideas on the subject of high-speed bearings. The very heavy loads imposed on simple low-speed bearings had resulted in designs far removed from the general run of engineering practice, and he thought that the following brief particulars of such a bearing might be of interest. The journal measured 33 inches by 66 inches, which gave a projected area of 2,178 square inches. The load on the bearing was about 250 tons. A speed of 33 revolutions per minute resulted in a journal speed of 285 feet per minute. Depending on the design, the bearing complete with housing would weigh 10 or 15 tons. Between limits, a fair approximation could be made of the heat to be dissipated by the bearing. At the other end of the range could be cited the case of a 57-ton flywheel supported by two 16-inch by 32-inch bearings. The surface speed of the journal was 1,800 feet per minute, and the specific pressure amounted to about 112 lbs. per square inch. Bearing-friction losses of as much as 50 HP. were encountered, and, in view of the relatively small cooling-surface of the bearings, forced lubrication or water cooling, or both, had to be provided to dissipate the heat. He thought it would be of assistance to engineers if the work that had been done on the design of bearings could be collected and collated.

Mr. Hodgkinson. Mr. FRANCIS HODGKINSON observed that the half bearing which was the subject of the experiments had a radial clearance of 0·007 inch per inch of diameter, and with wear or running-in that became 0·006 inch per inch of diameter. The clearance, therefore, was bigger than was general practice. The highest load in most of the tests was 4,000 lbs., corresponding to 222 lbs. per square inch of

Mr. Hodgkinson.

projected area, which was a comparatively high duty for a bearing to be started from rest. He supposed that the bearing had been placed in operation with lighter loads, and that the load had subsequently been increased. The Author properly called attention to the fallacy of certain oft-quoted formulas, and pointed out that the higher the surface speed the heavier the load which might be borne; that, of course, was true, provided means were available to carry away the heat which was the result of the greater rate of shear. The highest speed employed in the experiments corresponded to a rubbing velocity of 11·8 feet per second, which was low in comparison with modern turbine practice of ten times that figure. The Author stated that at very low speeds insufficient oil was drawn into the clearance-space to float the bearing; there was evidence of a failure of the oil-film in the test at 5 revolutions per minute at about 115° F., shown in Fig. 13, Plate 3. Would it not be more proper to state, not that insufficient oil was drawn between the surfaces, but rather that particles between the surfaces, on being squeezed and subjected to centrifugal motion, took a more axial path, so that the film could only be maintained a more and more limited distance in the line of motion of the surfaces as lower speeds were reached? He believed little was known concerning the actual path of the oil-particles with different pressures and speeds, and there was room for useful research on that subject. Fig. 22, Plate 4, showed a nearly constant ratio between the maximum film-pressure and the load per square inch of projected area. The maximum film-pressure ranged from 5·3 to 5·8 times the loading, in pounds per square inch of projected area, for loads between 5,000 and 1,000 lbs., the ratio being 6·5 to 1 for the 500-lb. load. It would seem reasonable to expect that such a constant ratio of pressure would exist for any one given condition of difference in radius. Further experiments to determine this ratio with various differences of radius and viscosities would be of value. A bearing constructed so as to reduce this ratio, rendering the pressure-curves less peaky, would be capable of sustaining heavier loads. He would expect that if an arc of, say, 100 degrees were to be scraped to a radius 0·0007 inch bigger than the journal for this 6-inch journal, more uniform film-pressures would be secured. Film-pressures in Fig. 22 were not given beyond about 130 degrees from the "on" edge, and he imagined it had been assumed that they corresponded to that of the atmosphere. Had the position of the test-bearing been normal, that was, the reverse of the position adopted in the tests, pressures considerably below atmospheric would have been observed. Such might have been observed in the Author's experiments, but perhaps the pressure-gauges did not read sub-atmospheric pressures.

Mr. Martin. Mr. H. M. MARTIN regretted the publication of the Paper because a journal bearing was singularly ill-suited for the making of quantitative comparisons between theory and observation. In the case of experiments on a Michell block everything was calculable, and, as Dr. von Freudenreich had shown, it was even possible to take account of the effects of the rise of temperature of the oil as it passed under the pad. In the case of a journal bearing it was hardly too much to say that nothing was calculable, because the shape of the oil-film altered with every change in the conditions, partly because the shape of the brass altered with every change in the load, and partly because both the least thickness of the oil-film and the eccentricity of the journal in the brass changed with changing conditions. It was very difficult in practice to eliminate the former source of error, and the elementary theory followed by the Author was inadequate to deal with the latter. As stated in the Paper, extremely small differences in fit made a great difference in the results. As a matter of fact, in the experiments on journal friction carried out at the National Physical Laboratory, it had been found impossible to get reasonable agreement between calculation and observation until the attempt to reproduce practical conditions was abandoned. The diameter of the journal had to be made much less than that of the bush, so that the effective bearing became a very narrow strip. The inherent disabilities of the journal bearing for a comparison of that kind had been aggravated in the Author's case by the use of a brass subtending an angle of no less than 180 degrees. No engineer would willingly use such a brass in practice, and it was also objectionable from the theoretical standpoint. As Mr. Martin pointed out 13 years ago, a negative pressure tended to be established near the trailing ends of large-angle brasses. This tendency increased as the load rose, and something of that sort was probably responsible for the discrepancies referred to on p. 253. Moreover, the wider the angle subtended, the greater was the probability that the brass would deform when the load came on. Another objectionable feature of the Author's apparatus was the narrowness of the brass as measured axially. It was but 3 inches long, or, say, about one-third of the distance between the leading and trailing ends of the brass. Now in the case of a square Michell pad, calculation showed that one-third of the oil escaped at the sides of the block, whilst two-thirds passed out at the trailing edge. With the narrow brass used by the Author, it seemed probable that more oil would escape at the sides than at the trailing edge of the brass. Side leakage was extremely important, but it was totally ignored in the elementary theory adopted by the Author as his basis for comparison.

As matters stood no one seemed yet to have calculated the side leakage from a journal bearing. The differential equations were so involved that probably only an arithmetical solution would be possible; it would take weeks to make, and even then it would only apply to the particular case considered and would have to be repeated for every variation in the conditions. This again showed how ill-adapted a journal bearing was for quantitative comparisons between calculation and observation. Again, the fact that wear occurred suggested that the lubrication was inadequate. Steam-turbine bearings ran for years with no visible wear. A common practice was to bore the bush 0·004 inch per inch larger than the journal diameter. The journal was not bedded in any way, yet perfect running was obtained, although to-day the length of the journal was often not much greater than the journal-diameter. Very ample lubrication was, however, provided for, partly in order to keep down the temperature. Dr. Lasche had proved experimentally that with small clearances and high surface speeds pressure was developed between the top brass and the journal, thus increasing the frictional losses. With large clearances, on the other hand, there might be vibration in starting.[1] His attention had first been called to this by Mr. Eric Brown. It would be interesting to know whether the Author had checked by calculation his hypothesis that the journal embedded itself in the brass. *A priori* it seemed much more probable that the apparent embedding was really due to a bending of the brass. The Author gave 0·0001 inch as the minimum thickness of oil-film consistent with safety. That, however, would probably depend on conditions. Mr. Martin had learnt from Mr. Eric Brown that in some Michell bearings the least thickness of the oil-film was less than 0·004 millimetre, or say 0·00015 inch. The Author had certainly shown great care, patience, and ingenuity in his attempt to overcome the fundamental defects of his apparatus, and this Mr. Martin was glad to recognize, whilst very greatly regretting the ultimate but inevitable failure to attain any useful or informative results. It was an ungracious task to criticize adversely work on which so much care and thought had been expended; but his criticism was reasoned, and when new work was brought forward it should be subjected to the reasoned criticisms of those who had made a special study of the subject. Mr. Martin.

Mr. H. T. NEWBIGIN asked why the load on the bearing was limited to 5,000 lbs., which was only 280 lbs. per square inch of projected area; if a true pressure oil-film had been formed, there should have been no difficulty in loading it up to the capacity Mr. Newbigin.

[1] *Engineering*, vol. 109 (1920), p. 600.

Mr. Newbigin. of the testing-machine. It appeared that the method of generating pressure in the oil-film by a radius-difference between the shaft and the brass had the objection that in any given bearing the higher the load the smaller the arc over which useful pressure was generated. For example, referring to *Figs. 9*, the point of nearest approach, A, depended upon the radius-difference and the load, coming nearer to the vertical load line as either or both of these were increased. No effective oil-pressure was generated beyond A, and the angle of oil-pressure might be taken as being the angle B beyond the load line plus twice that angle in front of it. If the sum of those angles exceeded 60 degrees there was a risk of failure at a much lower load than could be applied when the oil-wedge was not limited by the radius-difference required for its formation. In the case of railway-axles the brass was made (or wore to) the same radius as the shaft, and the oil-wedge was formed by a slight movement in the position of the centre of the axle, due not to a radius-difference but to the fact that the brass subtended an angle of about 60 degrees only. Where a wider angle of support was desired the arc might be increased to 120 degrees by fitting two brasses or pads of 60 degrees capable of lifting at their leading edges under the wedging action of the oil by reason of their being pivoted in the housing. He had occasion a few years ago to make a comparative test between two types of bearings, each embracing the shaft over 120 degrees. Those bearings were fitted to a cold-rolling mill, and there were four bearings, each 10 inches in diameter, in the mill; the first type had solid brasses, while the second had pairs of pivoted pads, each pad having an arc of 60 degrees. All the bearings had been carefully hand-scraped to ground roll-necks. The speed was 30 revolutions per minute, and the pressure per square inch of projected area was 725 lbs. in the first type, and 1,100 lbs. in the second. The power absorbed in bearing friction (as arrived at from the electric power absorbed in driving the mill after deducting the work done by the mill itself in rolling the material) was in the first case 45 HP., against only 2 HP. in the second case. The reason for the great difference was of course that in the first case the resistance followed the laws of greasy friction because of the absence of clearance, while in the second there was true pressure oil-film lubrication, because the oil-film was able to form the correct wedge by the lifting of the leading edges of the pads. While the method of obtaining film lubrication in journal bearings by giving a clearance between the shaft and the brass met most requirements, he did not think the conditions for formation of the film were nearly as perfect as was commonly supposed.

Mr. Southcombe. Mr. J. E. Southcombe remarked that the Author had conducted a difficult and valuable research very thoroughly. Direct measurements of the thickness of the oil-film were of the utmost importance in the study of lubrication, and the Author's work in that direction would rank as a classic. The thickness of the film at low speeds was of particular interest, and Figs. 13 and 15, Plate 3, the curves in which showed a sharp rise in the friction at about 5 revolutions per minute (approximately 8 feet per minute), afforded confirmation of previous experiments and indicated that what had been called the "boundary" effect became manifest at that speed. It was rather surprising, however, that the film at that speed was of considerable thickness according to the Author's figures (Fig. 7, Plate 3). Was that due in some way to the "indentation" of the bearing? The film at the point of closest approach was very much thinner in Sir Thomas Stanton's experiments. The use of oils of different chemical composition with the Author's apparatus at low speeds would no doubt give information of great importance in "boundary" lubrication. The Author referred to measurements made at low values of $K\lambda U/P$, and it would be interesting to know whether he found any signs of a sharp rise in the value of $\mu$ when $K\lambda U/P$ was very small—in confirmation of the curves obtained by Messrs. R. E. Wilson and D. P. Barnard from Professor Stribeck's figures? Some further information on the chemical composition and physical constants of the oils employed by the Author would be valuable. Exact knowledge on lubrication could only be developed by correlation of mechanical results with the chemical and physical properties of the oil so far as they could be determined. Viscosity alone was not sufficient. Were they hydrocarbon or compounded oils, and, if the latter, what were the glycerides, fatty acids, etc.? Mr. Southcombe.

Dr. H. W. Swift observed that from time to time during the past 4 years he had been in touch with the work described in the Paper. He had been able to appreciate the circumstantial and intrinsic difficulties of that work, and he felt that they were too modestly stated in the Paper, which gave to the reader only a partial impression of the experimental ingenuity and technique which had made possible the direct measurement of the oil-film between a running journal and its bearing. The results were known to be divergent from the theories of Professor Osborne Reynolds, and it had therefore been necessary to take quite extraordinary precautions to establish their accuracy. The scrupulous care with which observations had been taken, and the patience with which tests had been repeated and results confirmed, could only be known to those who were in touch with the work. There were three main directions in which the work advanced knowledge. It showed beyond doubt that the hydrodynamic theory of the infinitely long bearing could only be applied in a qualitative way to a short bearing running Dr. Swift.

Dr. Swift. under substantially practical conditions; that this qualitative agreement only held over a limited range; and that even over this range the elastic indentation was sufficient to exercise an important influence on the conditions of operation. Incidentally, also, it emphasized the impossibility of avoiding wear in any bearing required to maintain any load during starting or reversal of motion. The importance in experimental work of the elastic—or in some cases plastic—indentation of a bearing appeared to be so considerable that it was remarkable that no earlier worker had paid attention to it. It was to be expected that even when an oil-film existed the indentation might still be considerable under the high pressures which has been observed in some cases. In Sir Thomas Stanton's work local oil-pressures exceeding 3 tons per square inch had been recorded, and the close agreement which he had been able to obtain with theory could only be explained by the large radial clearances (2 per cent. and 6 per cent.) between his journal and bearings. With the smaller clearances and the less rigid bearings more commonly encountered in practice it was to be expected that this effect might easily invalidate any simple application of the hydrodynamic theory. Another interesting speculation to which the Paper gave rise was the effect of end leakage. The theory, of course, neglected any leakage of oil in an axial direction, and was therefore only applicable to a bearing of "infinite" length. It was only natural that this theory should require correction for bearings of normal length, and particularly for the comparatively short bearings used by the Author. Leakage would tend to reduce the thickness of the film in the region of high pressure, and would also tend to increase, for a given load, the intensity of pressure at the centre of the bearing. It would therefore be likely to increase the bearing friction. The results shown in Fig. 21, Plate 4, supported this view as regarded both film-thickness and friction. In further confirmation, it was notable that the discrepancy from theory, as shown in Figs. 17A and 17B, Plate 4, was greatest at high pressures and tended to become less at higher speeds. The theory of a flat inclined slide-block, due to Mr. Michell, showed that the friction of a square block was twice that of an infinitely wide block, and that of a block whose width was only half its length was nearly thirty times as great. The effect of end leakage in this case was therefore all important, both to the friction and to the pressure-distribution. A further point to which attention seemed to be invited by the Paper was the difficulty of applying a correct value to the viscosity of the lubricant under the conditions of operation. The effect upon the viscosity of the oil of the high pressures which existed locally in an oil-film was known to be considerable, but it appeared to have been neglected by all workers, even by Sir Thomas Stanton, to whom knowledge of the effect was due. The difficulty of evaluating the viscosity under high pressures and the added difficulty of estimating its effective mean value were admittedly great, but they must be taken into account before any close quantitative agreement with theory could be expected, unless the fortunate dispensation of Nature which made the friction in some cases insensitive to changes in viscosity was of general application. Dr. Swift.

The AUTHOR, in reply, remarked that he agreed with Mr. Boswall in thinking that it was almost impossible to get satisfactory results with a bearing embracing 180 degrees, but something had been accomplished if that point had been established by these experiments. He believed that the excessive local wear of the bearing used in the tests now under review was largely, if not wholly, due to the constant stopping, starting, and reversing of the machine; but it appeared to be impossible to avoid that when making measurements of the film-thickness. The practice now adopted of relieving the bearing of load before starting had reduced this evil. In some of the Author's early tests the temperature of the bath, as predicted by Mr. Boswall, was higher than that of the bearing; but that was attributed to a probable error in the thermo-junction galvanometer, since the reverse occurred when a more reliable instrument was installed. Thus it appeared possible that the first results were correct, and that the condition of the bearing had altered when the later tests were made. No cooling of the oil by artificial means was adopted while the temperature tests were being made. A summary of the results obtained had been asked for by Mr. Broughton; the subject was developing so rapidly that it would be premature to attempt to summarize at this stage. The Author was fully alive to the desirability of making such a summary when he wrote the Paper. Such tests as those suggested by Mr. Hodgkinson were now in progress, and he hoped to report upon them at no distant date. The criticisms of Mr. Martin were somewhat severe, but they were highly valued and appreciated because they were very suggestive and helpful. He admitted that a rigid analytical treatment of the subject appeared to be almost impossible at the present time; but that was no reason why experiments should not throw light on it. A close analogy was found in the treatment of the flow of water over a notch in a thin plate: mathematicians admitted that they were quite unable to get a rigid solution of that problem; but, as every engineer knew, the commonly used approximate solution, when corrected by an experimental coefficient, gave very reliable results. It was the experimental determination of a corresponding coefficient for bearings that the Author was aiming The Author.

The Author. at getting by his researches. The determination of such a coefficient at present appeared to be shrouded in difficulty; but he hoped to accomplish something of value in the future. The defects of the measuring-apparatus and the method of procedure were freely admitted. The criticisms received would be helpful in avoiding them in future. The end-leakage determinations mentioned in the reply to Mr. Rowell's contribution to the oral discussion showed that about three-fourths of the oil escaped by end leakage, but Mr. Martin was mistaken in thinking that end leakage had been "totally ignored" by the Author, since the vertical and horizontal shifts of the bearing as measured were profoundly affected by the end leakage. Had such shifts been calculated without regard to the end leakage, his strictures would have been entirely justified. The lubrication could hardly be described as "inadequate," since the shaft for nearly one half of its diameter dipped into oil, and shields had to be used to prevent it from flying in all directions. No attempt had been made to check the bearing-indentation by calculation, since it was an exceedingly difficult mathematical problem; the indentation had been measured by the same Geneva gauges as were used for getting the thickness of the oil-film; the bending of the brass, if any, did not affect the measurements. Apparently Mr. Brown's measurement of the least thickness of the film agreed well with that found by the Author, namely 0·00015 inch and 0·00010 inch respectively. In the opinion of others interested in the subject, the Author's work, though far from being perfect, had already yielded "useful and informative" results. In some of his early experiments in this series a bearing was used which subtended only 60 degrees; but it was so unstable laterally that no film-thickness results could be obtained. The making of tests at very high speeds and loads resolved itself very largely into a question of expense in running. Unfortunately he had not the unlimited funds of a public institution to draw upon. He was unable to supply information about the chemical composition and physical constants of the oils used; but since the oils were of well known standard brands, such information could doubtless be obtained by any chemist conversant with oil technology. He wished to place on record his appreciation of the help he had received from Dr. Swift, who had very kindly undertaken the preparation of the bearings in his laboratory, which required a higher degree of accuracy in workmanship than was available in most engineering workshops. A pupil of Dr. Swift's, Mr. Ralph Townson, had also been of great assistance in taking observations and otherwise helping the Author; his work had been characterized by extreme care and precision.