Discussion.

The CHAIRMAN, in moving a vote of thanks to the Author, remarked that The Institution was much indebted to him for having brought before it a difficult and complex problem, and for having shown to what extent he and others had solved it by their researches and experiments. He hoped that those who had carried out similar experiments would give the members the benefit of their experience. The Chairman.

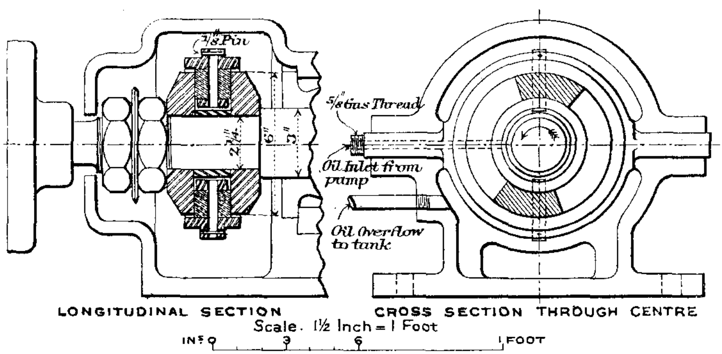
Mr. W. H. ELLIS shared the Chairman's view that The Institution had received a contribution of considerable value, which had involved much study and research. It was naturally a difficult subject to discuss in detail, especially for one whose experience of the problem had been rather in the manufacture of large thrust-shafts for marine work than with the question of friction involved in their subsequent use. The Author's investigation would, however, concern the manufacturer, in so far as it tended to do away with the very uneconomical method which involved the removal, in some cases, of 6 to 8 tons from a single shaft in order to make room for the spaces between the collars, the surfaces of which carried the thrust.[1] No doubt the students of the turbine system of propulsion were conscious that in abandoning the reciprocating engine they were at the same time introducing a reciprocal action as between the thrust inwards of the propeller and the blading of the turbine, being of the opposite hand to that of the propeller, thereby balancing to a large extent the thrust of the propeller, and thus greatly simplifying the thrust problem. What would have happened by now with the large horse-powers—as much as 70,000 and 80,000 HP.—in Atlantic liners and warships it was difficult to say, because the problem of the manufacture of large crank-shafts, in itself sufficiently difficult, would often have been much complicated if it had been necessary also to provide for the absorption of the enormous thrust which would have been involved by such horse-powers. That had to some extent been reduced by the reciprocal action of turbines to which he had referred. The Author showed in *Fig. 12* a thrust-bearing he had designed on that principle, and it would be useful if an illustration could be given of the same design applied where the horse-power was so great as to necessitate more than one ring. He hoped the Author would also indicate the dividing line between multiplication of rings and increase in diameter. It appeared to him that on the blocks Mr. Ellis.

[1] The problem would have become much more acute if turbine propelling-machinery had not been introduced.—W. H. E.

Mr. Ellis. shown there must be a point at which the Author would hesitate to increase the diameter, and would recommend the multiplication of rings, or possibly two or three rings. It would be desirable if the Author could add a few words in further explanation of a point referred to at the top of p. 233, namely, the calculations necessary to determine what proportion would give the balance between the in-going and the out-going oil. He desired to add, in reference to the balancing action of turbine propelling-machinery, that the problem of the large thrust bearing was to some extent revived by the introduction of the geared turbine. It appeared to him, therefore, that the subject must be one of special interest to marine engineers, if they again had to face the absorption of thrust entirely by thrust bearings, without help from the blading of the turbines.

Capt. Sankey. Captain H. RIALL SANKEY thought that the Author's reason for writing the Paper, stated in the first paragraph, might have been a little extended. He had shown by *Figs. 8*, *9* and *10* that, in order to obtain a film of oil between two surfaces which were being pressed together, they must be inclined to each other, and further, that in the case of ordinary journals the inclination was obtained automatically by an almost fortuitous arrangement of a slight shifting of the shaft as explained by the Author. A formula was given on p. 241 which related the angle of inclination of the line $d\,g$ in *Fig. 10* to the line $f\,j$, and there were one or two points in connection with that formula to which he desired to call attention. First, although it might be somewhat hypercritical, the Author stated that c denoted the angle of inclination, but it actually denoted the tangent of the angle of inclination, for on p. 242 the Author gave that angle in some cases as being 1/3,000. Bearing in mind that, when the line $d\,g$ was 1 inch, as in the first example given of a bearing made for Messrs. Belliss and Morcom, the difference between the height at one end and at the other was only one-third of the thickness of a piece of cigarette-paper, it was quite obvious it would be impossible to adjust the inclination of the two planes to a sufficient degree of accuracy, and, in fact, this was stated to be the *raison d'être* of the Michell bearing. He did not think, however, the Author had made his case as strong as he might, because that angle depended, as would be seen by the formula on p. 241, on λ, the coefficient of viscosity. Viscosity was there expressed in absolute units; he presumed the Author meant at a temperature of 20° C., or at a temperature of 60° F., according to the units used. As was well known, the coefficient of viscosity varied greatly with the temperature. In the case of the Belliss–Morcom bearing the temperature was stated to be 126° F. He presumed that the temperature of the oil would be slightly more than that, certainly not less. Capt. Sankey. If 4·8, the figure given on p. 245 for the absolute viscosity of the oil, was at 60° F., then at 126° F. it would not be much more than 0·4. Inserting the figure 4·8 in the formula under discussion, c was found to be $\frac{1}{1700}$, but when the viscosity was reduced to 0·4 it became $\frac{1}{6000}$. The point he wished to make was that not only must the inclination between the two surfaces be extraordinarily small, and therefore require extraordinary accuracy in manufacture, but it had to vary from the time the bearing started running, and the slightest alteration would prevent the bearing from running properly: the film of oil would not be obtained which was necessary for "perfect film lubrication." It was obvious there must be some automatic means of adjusting the inclination between those surfaces,

Figs. 14.



and that was precisely what Mr. Michell had provided by using movable disks or plates. He was glad to say he had had an opportunity of using some of those bearings in connection with Marconi discharge disks. These were steel disks a little over 4 feet in diameter, rotating at about 2,600 revolutions per minute, and carrying copper studs; and it was necessary to have a thrust bearing to balance the thrust of the driving machinery. A bearing proposed by the Author was 7 inches over all, having two friction disks placed at opposite diameters. They were each about 2½ square inches in area, and the thrust provided for was 2,000 lbs., or 400 lbs. per square inch. The bearing was slightly different from those described in the Paper, inasmuch as it took the thrust in both directions. As a result, the disk, instead of being pivoted on the back, was pivoted through the centre by means of a bolt, as would be seen from *Figs. 14*, and could therefore adjust itself according to the direction

Capt. Sankey. from which the thrust came. It was stated on p. 230 that the saving in the use of roller and ball bearings was in their low static friction rather than in their low running friction. He had had an opportunity of proving that statement with the Marconi disks to which he had referred. Originally, ball bearings were used for the shaft-bearings, but at the present time plain phosphor-bronze bearings under forced lubrication were running quite satisfactorily, and, as far as he could see, there was no difference in the amount of friction.

Mr. Stoney. Mr. G. GERALD STONEY remarked that thrust bearings such as those referred to in the Paper would make possible a geared turbine ship with each shaft passing perhaps 40,000 HP. The term "coefficient of friction" was frequently used in the Paper. He suggested that "resistance per square inch of projected bearing surface" be substituted. That expression was used in Tables I and III, and if worked out it would be found that in Table II the resistance per square inch, which was the product of the coefficient of friction by the pressure, was practically constant and equal to 0·36. The resistance per square inch was, in a perfectly lubricated bearing, independent of the area and pressure, and for low speeds such as the Author had dealt with it varied directly as the velocity. As the speed increased, according to Mr. Lasche's experiments, carried out on high-speed bearings, the resistance gradually altered until it varied as the square root of the velocity, and finally at high speeds, about 30 feet per second, it was practically independent of the velocity. That meant that the average thickness of the oil-film must be independent of the pressure per square inch; that it thickened as the speed increased; and that at high speeds the thickness of the oil-film was practically proportional to the surface speed of the bearing. The Author stated on p. 243 that the pivoted thrust bearing was the mechanical equivalent of a journal bearing. In a journal bearing the distribution of pressure was as shown in *Fig. 6*, where the centre of pressure was practically the centre of figure. The coefficient of friction was very small, perhaps 1/1,000, in such a bearing; and in a bearing of 4 inches diameter, such as Mr. Tower experimented with, the distance of that centre of pressure from the centre of figure would only be an angle equal to the coefficient of friction. Thus it would be somewhere about 2/1,000 inch from the centre of figure, or practically an inappreciable amount. When the arc of contact was shortened, as given in Professor Goodman's test in Table II, practically the same form of oil-film was obtained as in a pivoted thrust-block. There a wedge action was obtained similar to that in a pivoted thrust-block, and the centre of figure was to all intents and purposes the centre of pressure—that was to say, instead of the distribution of pressure being as shown in *Fig. 11*, it was as shown in *Fig. 6*. Experiments further showed that it was practically immaterial where the pivot was put in such a thrust-block; it worked equally well whether it was centrally pivoted, as has been proposed by Sir Charles Parsons, or put out of centre as Mr. Michell did. The whole theory was exceedingly difficult, and he did not think it had yet been properly solved by mathematicians. The pressures that could be worked to were very great; personally he had worked up to 4,000 lbs. per square inch at 52 feet per second, and Mr. Kingsbury in America had reached 7,000 per square inch. At both those pressures the bearing failed, owing to the white-metal squeezing out. The strength of the white-metal limited the pressures that could be used, not failure of the oil-film. It was certain that such bearings could be used up to speeds of 50 or 60 feet per second at 500 lbs. per square inch with a considerable margin of safety. A notable example of such a bearing was in the cross-channel boat "Paris," where there were thrust-blocks on each shaft, with centrally pivoted bearings taking 24 tons each, at about 300 revolutions per minute. Mr. Stoney.

Professor Goodman.

Professor JOHN GOODMAN thanked the Author for the very clear way in which he had placed the subject before The Institution. It might be of interest to refer briefly to the history of the subject. Some 30 years ago, when Mr. Tower made his experiments, ultra-practical men asked what was the use of making tests on bearings flooded with oil, because in practice no bearings were treated in that manner. But what had been the result? At the present day almost every high-speed engine in the world was fitted with an oil-bath or its equivalent. Mr. Tower's results also led Professor Osborne Reynolds to go very thoroughly into the theory of the subject, and, as was well known, the results of his investigation were in remarkable agreement with the experiments of the former. The latest outcome of that theory was Mr. Michell's work on the thrust bearing. He thought the illustrations given in the Paper were hardly fair to the old thrust bearing. For example, in the bearing illustrated in *Fig. 3* the oil-holes were so arranged that the oil fell on the outside of the collars, whereas it was well known that the oil should be applied to the bottom of the collars. Then by centrifugal force the oil was carried outwards and distributed over the collars themselves, which made a much more satisfactory bearing. Again, the ordinary pivot or thrust bearing for high speeds was always fitted with loose collars at the bottom, and the oil was supplied through the middle of the collars. He had himself made bearings of that type and tested them up to 340 lbs. per

Professor Goodman. square inch, and had not experienced the slightest difficulty in keeping them cool. He did not know exactly what the coefficient of friction was, because it was a rather difficult matter to determine it in a bearing of that type. Coulomb's laws of friction were known to be very inaccurate, largely due to the fact that his experiments were made on a toy scale; and when similar tests were carried out on a larger scale, totally different results were obtained. The friction was not independent of the speed; with dry surfaces it fell off rapidly with the speed. It was not independent of the area; it fell off somewhat rapidly as the area was decreased. It was not proportional to the force pressing the two surfaces together; it fell off somewhat rapidly. It was, however, tolerably true that it was dependent only on the nature of the surfaces. He thought it was important to call attention to that point, because many blunders had arisen from the assumption that the Coulomb laws were correct. The Author referred on p. 226 to the coefficient of friction between smooth dry surfaces of anti-friction white-metals, but surely a bearing was never intended to run dry. The coefficients obtained, even with a very slightly greasy shaft, were much lower—somewhere in the neighbourhood of one-tenth of the values given by the Author. Professor Goodman was still of opinion that the ball bearing could compete successfully with a bearing of the type described in the Paper. He had seen many failures of ball bearings, but he believed their failure in every instance had been due to bad design. He further believed that the coefficient of friction with a well-designed ball bearing would be lower than with a bearing of the Michell type, although he was perfectly willing to admit that in some cases a bearing of the latter type would be better suited to the work than a bearing of the former type. Captain Sankey had referred to a ball bearing which had been replaced by a phosphor-bronze bearing, and it would be interesting to have particulars of the ball bearing which failed. About 13 years ago some very careful experiments were made to investigate the manner in which a bearing wore. It was found that when the brass subtended about 180° the wear was always on the "off" side, exactly as would be expected from Mr. Tower's experiments. The angle embraced was then reduced step by step, and it was still found that the wear occurred on the "off" side of the brass. When the brass subtended about 90°, however, the wear was equal on both sides. With less than about 90° the wear took place on the "on" side of the brass and not on the "off" side. A narrow brass such as that was very similar to the block used in the Michell bearing. He was not prepared to say that the Michell theory was wrong; but it was curious that in the case of a

Professor Goodman.

brass subtending only a very small arc the wear should be on the "on" side, and not on the "off" side, as would be expected from that theory. It was an interesting point that might well be gone into somewhat more fully. He had observed it many times since in other experiments, and was therefore quite sure it was the case. It would be a great convenience if the Author in his reply would give the viscosity in English units, as it was confusing to have the C.G.S. system on one line and the British system on the next. No one admired Mr. Tower's work more than he did, but it was to be regretted that Mr. Tower had no artificial means of keeping his bearing at a constant temperature; hence all his results obtained were more or less defective, because the temperature of the bearing and of the oil varied from time to time. It was given as 90°, or 100°, or 120°, but as a matter of fact it varied considerably during the tests. He did not make that statement without having definite proof; moreover, there was internal evidence that the temperature of the film did vary, and that had been pointed out by Professor Osborne Reynolds. He believed that Mr. Lasche's experiments, in which it appeared that the friction remained constant at very high speeds as the speed increased, were to a certain extent open to the same objection as Mr. Tower's. He was well aware that Mr. Lasche used some means of keeping his bearing at constant temperature, but he could not believe that the temperature was actually kept constant. Very shortly after Mr. Lasche's experiments were published, Professor Goodman to a large extent repeated them, and, although he could not get up to the same speeds, he found very definitely, as the theory would indicate, that the friction increased directly as the speed, even up to the highest speed to which he could attain—of course not including the irregularities which always occurred at low speeds.

Capt. Sankey.

Captain H. RIALL SANKEY desired to say, in answer to Professor Goodman's question, that the ball bearing to which he had referred did not fail in the ordinary sense, but owing to high-frequency currents a certain amount of electrolytic action took place which pitted the balls. The failure, therefore, had nothing to do with the ball bearings as such.

Mr. Ferranti.

Mr. S. Z. DE FERRANTI had been interested for some years in trying to obtain an end thrust-bearing which would carry very heavy loads efficiently at high speeds, namely, the speeds of reaction turbines. The loads which he wished it to carry were very considerable, and he made a number of experiments with a view to taking those loads by different kinds of thrust bearings instead of by a dummy piston. In one case, 80 tons had to be carried con-

Mr. Ferranti. tinuously at 1,500 revolutions. His experiments had led him to conclude that, with the means available, he could not carry that load on a single collar, and that it was necessary to have a multiple collar somewhat of the nature of the old marine thrust bearing. As a matter of fact, he used in that case ten rings, and worked up to a pressure of about 500 lbs. per square inch. The lubrication was arranged in the ordinary way, just as for the ordinary bearings of a turbine—that was, the oil was cooled artificially, and was supplied at a moderate pressure by a pump. It appeared to him that the vital point with such high pressures was to make sure, where the collars were multiple, that an approximately even load was obtained on each of the collars, or, rather, that no collar had more than its proper load on its surface. He had tried to devise all sorts of lever arrangements for equalizing the load between the collars, and eventually he had arrived at a simple spring support so arranged that, if more than the correct load came on any one collar, it collapsed a few thousandths of an inch and equalized the loads on the others. The result was that, notwithstanding the stretch of the shaft holding the collars (for they could only be of a moderate diameter), the collapse of the box that held the standing collars, and inaccuracies of workmanship and fitting, there was still an even distribution of the load between the various surfaces, and the thrust worked well in consequence. The Author had put before the members the scientific side of the working of thrust bearings. Unfortunately, more invention was often necessary for the production of those scientific explanations than was required in the development and building up of the original invention. He did not wish to go quite as far as that in the present case; but some of the opinions that had been expressed did not fit in very well. In dealing with the delicate question of getting the right amount of angling on the two surfaces on which it was necessary to take in the oil, he took a ring surface just like an ordinary marine thrust ring, but continuous all round instead of being half a circle, and grooved it deeply with grooves about $\frac{1}{4}$ inch wide. That made a number of segmental surfaces on which the load was carried. The edges of the surfaces were rounded where the oil had to enter, so that it would not be scraped off, and the surfaces were bevelled off to the extent of about the thickness of a piece of cigarette-paper. He made very careful experiments because he had two thrust collars being pressed up against a running collar, and that formed a very good brake. He was thus able to weigh very accurately the friction resulting from any particular load, and to observe the quality and temperature of the oil, and other conditions. In the result he found he could readily vary the angle which the Author had calculated theoretically. The essential thing was to round the edges, and it was necessary to shorten the length of the surface in the direction of motion—that was, round the circle in that case, and to make the length of the individual surface small. It did not matter exactly what it was, but it had to be small to ensure good results. Under those conditions satisfactory results in working were secured. He obtained coefficients of friction as low as the best that had been referred to that evening; in fact, a non-pivoted, or plane surface cut up by oil-ways worked perfectly well, provided pains were not taken to rub the oil off the surface. In engineering work one of the most difficult problems he had had to face was to prevent oil from getting where it was not wanted. When the way in which the surfaces obtained oil by passing the grooves at very short intervals was borne in mind, it was not at all surprising the method worked perfectly well. He worked up to pressures of only 1,000 lbs. per square inch because he was afraid of his white-metal surfaces. The white-metal surfaces did not actually fail, but he was expecting them to do so. The oil, as Mr. Stoney had said, never failed. The problem was one of getting a non-seizing metal surface opposing a steel surface which would not collapse or crumble away. Then, if the pressures were distributed between the various collars, so that an excessive pressure was not obtained, and the heat generated was taken off, a thrust could be produced which would carry any amount of load. Mr. Ferranti.

Mr. H. S. Broom was of opinion that the Author had underestimated the difficulty of securing perfect lubrication with a journal bearing. This could only be obtained with bath or forced lubrication, and in a bearing subject to reciprocating motion the $\frac{3}{1000}$ to $\frac{5}{1000}$ inch clearance required for perfect lubrication would cause a knock which, when started, would prevent an oil-film being obtained between the two surfaces. Mr. Michell had lately applied the tipping-block principle to journal bearings for turbines, reducing the width of the bearing by two-thirds, and as the friction absorbed by the bearing was proportionate to its surface, a great saving in friction in the overall length of the machines was effected and the oil-temperature was reduced. The length of bearings fitted to turbines, which was one source of difficulty in obtaining perfect lubrication, might be reduced by this method. Michell thrust bearings had been tested to a pressure of 12,000 lbs. per square inch without failure of the oil-film. He believed the reason for the lower coefficient of friction in a Michell bearing was that it could be bedded up to surface-plate with a much closer approach to a perfectly flat Mr. Broom.

Mr. Broom. surface than with a journal bearing bedded to the more or less irregular surface of the shaft. With both high speeds and heavy loads, the Michell bearing possessed great advantages over the ball bearing, as at high speeds the centrifugal action threw the balls on the outer side of the race, and with heavy loads and low speeds the period of compression of the ball bearing was long enough to destroy the molecular construction of the material. After constructing numerous bearings, he had found that the most satisfactory manner of carrying these was with the two blocks carried in a half gimbal, ensuring both blocks taking equal care of the load. Where the surfaces between the rubbing parts were well finished and the materials suitable to the loads, the bearings never failed. The most suitable materials for high speeds were case-hardened steel and a special close-grained mixture of cast iron, not white-metal as suggested. Cast iron apparently held a film of oil better than any other material when standing under load, and no wear occurred before a speed of 10 feet per minute was attained—the starting point of perfect lubrication. The most satisfactory arrangement for lubrication at high speeds was to core the housing of the bearing so as to allow the thrust-block to act as a propeller pumping the oil round to the centre of the bearing.

The Author. The AUTHOR, in reply, said it was quite practicable to design a multiple-collar thrust bearing with pivoted segmental blocks provided with means for distributing the load equally among them, but there had been no difficulty in dealing with any load carried so far on a single collar. Bearings of this type had been made in America over 4 feet in diameter carrying 240 tons. As to the point of pivoting, in a journal bearing, owing to the surfaces being curved and to there being a slight difference between the radius of the shaft and that of the brass, the thickness of the oil-pressure film increased somewhat on leaving the point of nearest approach to the shaft; therefore the oil-pressure figure was almost symmetrical and the centre of resultant pressure was practically in the centre-line of the bearing, as pointed out by Mr. Stoney; but in the case of plane surfaces the greatest pressure was much nearer the "off" side, as was also the point of resultant pressure at which it should be pivoted. He had made some tests as to the effect of varying the pivoting point, in conjunction with Messrs. Belliss and Morcom, in June, 1910. With the block pivoted $\frac{1}{16}$ inch in front of the centre, the temperature was 135° F.; pivoted at the centre, 132° F.; pivoted $\frac{1}{16}$ inch behind the centre, 126° F.; and pivoted $\frac{1}{8}$ inch behind the centre, which was about the theoretical point, 124° F. There was a difference in temperature, and consequently

The Author.

a corresponding fall in friction, amounting to 11°, due to the correct pivoting of the block. The blocks were tested by means of a thermometer in the blocks themselves—that was, a hole was drilled in the block so that the actual temperature was obtained as near the face as possible. The bearing was running at 1,750 revolutions per minute, and the load on the blocks was 500 lbs. per square inch. The explanation of a block pivoted in front of the theoretical point working at all was probably that the rise in temperature in the oil-film passing under the block reduced its viscosity and thereby its ability to support the trailing edge. In the large bearings made in America the blocks were approximately square and pivoted behind the centre, so that practical experience there appeared to confirm the correctness of Mr. Michell's theory. These bearings had been tested by the Westinghouse Company to 10,000 lbs. per square inch, at which pressure they ran cool but failed by the white-metal flowing. In reply to Captain Sankey, the viscosity of the oil must be taken at the working temperature. In the case of the Belliss–Morcom tests the viscosity given was at the working temperature and was taken from the absolute-viscosity curve of the oil. With reference to the question of dividing a plane rubbing-surface into a number of fixed instead of pivoted planes. It was quite possible to get an oil-pressure film between the surfaces in this way; there was, however, the objection that the small amount of inclination necessary was apt to wear off. Coulomb's laws had been referred to. The figures given in the Paper had been obtained from Messrs. Archbutt and Deeley's work on Lubrication, and he was not responsible for them. For the mathematics of the flow of viscous substances, alluded to by Mr. Ellis and others, he would refer to Professor Osborne Reynolds's Paper,[1] which went into the subject very fully. The inclination of the blocks was given by him as 2·2 : 1, at the "on" side and "off" side respectively, to get the best results. [The Author here exhibited a bearing which had been running at 1,500 revolutions per minute carrying a load of 1,200 lbs.]

Correspondence.

Mr. Gibson.

Mr. J. Hamilton Gibson observed that the inefficiency of the ordinary marine type of thrust-block had been forced upon the notice of the engineering world by the introduction of geared

[1] Phil. Trans. R. Soc., vol. 177, p. 157.