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.

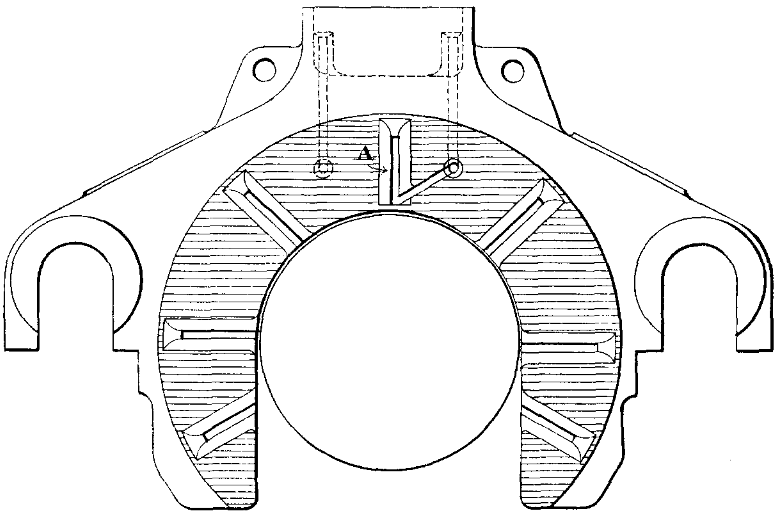
Mr. Gibson.

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.

Mr. Gibson. turbines for ship-propulsion. The Birkenhead Engineering and Shipbuilding Works, with which he was associated, had completed, early in 1913, a geared turbine steamer of 8,000 shaft HP. for cross-channel service, and had fitted thrust-blocks of the usual horseshoe type, with ample surface for reciprocating engines of equivalent power. The rings were water-cooled, and were lubricated by means of oil-boxes cast on top of the rings. Sinuous oil-grooves were cut in the white-metal faces, and the surfaces were fitted with the usual care bestowed on this type of thrust-block. Trouble was experienced owing to the undue heating of the lubricant and the rapid wear of the white-metal, which dragged over and practically obliterated the oil-grooves. The grooves were increased in number and were deepened, and overhead oil-boxes were fitted on the bulkhead, giving a head of about 14 feet. The oil was, however, "pumped" up through the pipes and displaced the worsted siphons. Wood plugs were fitted into the siphon tubes; but even these were driven out by the back-pressure of the oil. The vessel was only kept in service by constant attention to the thrusts, and it was evident that an entirely different system of lubrication was required to meet the case. Some natural law was being opposed, and it was useless to attempt coercion. It occurred to him that if only the oil could be introduced to and insinuated between the surfaces, as in the case of a main-shaft bearing with oil-wells at the sides, all might be well. Accordingly one set of rings was drawn and wide oil-wells were cut radially across the white-metal. These oil-wells were about $1\frac{1}{2}$ inch wide, and their sides were gently chamfered away to join the bearing surface (*Fig. 15*). It would be seen that the thrust surface was considerably reduced; but the result was perfectly satisfactory. The bearing picked up its lubricant automatically from the oil-bath, and ran without any appreciable rise of temperature. The other thrust was altered in a similar manner as soon as convenient, and the overhead oil-boxes were removed. What had been effected would be readily appreciated. An elementary form of Michell thrust bearing had been made, although he was sure the Author would exonerate him from any charge of attempting to infringe a patent. As a matter of fact, he was so convinced of the superiority of the Michell thrust bearing that he was at present making four thrust-blocks of that type for two twin-screw geared-turbine steamers, each of 5,000 shaft HP., which would be running their trials in the Mersey shortly. The necessary abandonment of the old marine thrust-block for geared turbines appeared to be due to the constant steady pressure between the thrust-surfaces caused by the uniform torque of turbine-driven shafting. In reciprocating machinery the thrust varied or pulsated with the impulses produced by the irregular turning-moment of the engines, and consequently the lubricant had some chance of getting in between the thrust-surfaces. But with geared turbine machinery there was no relief once the surfaces came into contact, and the surfaces could not be separated except by some system of forced lubrication where the oil must be introduced at a greater pressure than the unit pressure produced by the thrust. The Michell thrust bearing, however, Mr. Gibson.

Fig. 15.



SKETCH SHOWING OIL GROOVES IN THRUST SHOES

SECTION THROUGH OIL-GROOVE A

overcame the difficulty in an extremely ingenious manner, not by coercion, but by persuasion, and Mr. Gibson heartily thanked the Author for his clear statement of the laws governing this most interesting engineering problem.

Mr. Albert Kingsbury, of Pittsburg, Pa., confirmed the experience of the Author as to the effectiveness of thrust bearings of the pivoted segmental type. The first bearing built by Mr. Kingsbury on this principle was made in 1898 and was tested at that time under varying loads up to 5,000 lbs. per square inch Mr. Kingsbury.

Mr. Kingsbury. with very satisfactory results. Other bearings were built and tested experimentally later on, the results tending to confirm those of his first experiments. At the present time there were a large number of these in service in the United States. The largest of them were those at the Keokuk plant of the Mississippi River Power Company, the diameter of the bearings being 56 inches, the loads 560,000 lbs. to 575,000 lbs., the normal speed 57·8 revolutions per minute and the maximum speed about 100 revolutions per minute. He had but recently received a copy of Mr. Michell's Paper, referred to on p. 241, and therefore could not speak with confidence with regard to the results of his analysis. It appeared to him, however, that the figures for the relative coefficients of friction for the square block and one of width equal to one-third of the length must be somewhat in error. Bearings constructed in approximately the two given proportions did not behave very differently in service; at least, the ratio of 13 to 1 in the friction did not appear to exist. In most of the bearings designed by Mr. Kingsbury the ratio of length to width of the blocks ranged between 1:1 and 1·5:1. As far as he had been able to obtain reasonably satisfactory data, it did not appear that the friction with these blocks was as much as double that calculated on the assumption of absence of leakage at the sides. There was, however, room for more exact experimental investigation in this respect. A curious result which he had obtained experimentally was that when the blocks were supported at the centre, the load-carrying capacity was fully as great as when they were supported at the theoretically best point, provided that the heat due to friction was dissipated. With the central support, however, the friction was much greater than with the eccentric support. A further conclusion reached by analysis was that the point of support giving minimum friction was not the same as that for the maximum load-carrying capacity, but was somewhat farther away from the centre. The point of support, however, could be moved somewhat from the theoretically best point in either case without greatly affecting the operation.

Mr. Lasche. Mr. O. LASCHE, of Berlin, agreed generally with the theory developed in the Paper, but wished to draw attention to the fact that it was not impossible to design also a solid type of thrust bearing to carry safely specific loads as high as those given on the examples of the Michell type. That would mean, of course, that they must work under similar high efficiency, otherwise they would develop undue heat. For examples he would refer to the work on Turbines for Ships, sect. 115, pp. 217–222, published by

Dr. Bauer and himself.[1] On page 222 of that work data were given for a thrust bearing with the annular type of thrust-block (Fig. 136). The maximum load which one ring could carry continuously at 900 revolutions per minute (42 feet per second circumferential speed) was 15 metric tons, or about 560 lbs. per square inch, with oil at a temperature of 97° C. By testing thrust-collars and blocks of the horseshoe type shown in Fig. 134 (on page 222 an erroneous reference was made to Figs. 136 and 137), it was found that one thrust-block was able to carry at 310 revolutions per minute (16·7 feet per second circumferential speed) more than 30 metric tons, or about 580 lbs. per square inch. While in the first case the load mentioned was the highest limit that could be maintained without exceeding the allowable temperature, in the second case the load was limited by mechanical stresses. At 30 tons, by means of inside water-cooling of the thrust-block and the circulation of 12 litres of oil per minute, the temperature of the oil leaving the thrust-block was only between 40° and 41°, and the sliding faces remained in perfect order. Considering the frictional conditions, there was no doubt that in this case the load could have been forced even higher. With regard to the special means by which it was possible to run the bearings under such high loads, it might be mentioned, as shown in Figs. 134 and 137, that the revolving thrust-collars, and not the thrust-block, were provided with oil-grooves. They were well rounded towards the sliding faces, thus forcing the oil between them. Mr. Lasche.

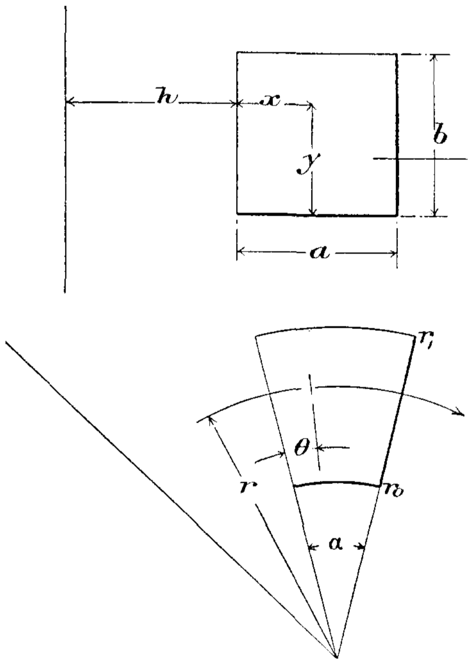
Mr. Robert Matthews observed, with reference to *Figs. 12*, which showed a spherical annular ring E, bearing on the spherical portion of the cross piece F, to compensate for any inaccuracy or want of alignment, that if this were desirable, then in his opinion for a correct bearing the adjusting washer next to the nuts should also have spherical contact with F, and the brass bushes or steps ought to have a spherical centre-piece, to give freedom of movement, in which case it would be necessary to make the cross piece in two parts. It appeared to him that the two tie-bolts attaching F to the turbine would not be equally loaded and would tend to bend the spindle unless the bearing was made as suggested, or unless everything was always absolutely true. There seemed to be no good reason for limiting the rubbing-blocks B to three in number, especially when there was ample room for six of them, which would adjust themselves automatically into three pairs and thus reduce the pressure on each. It would be interesting to know if the designer had tried the rubbing-pieces B as plain disks Mr. Matthews.

[1] Bauer and Lasche, *Schiffsturbinen*, 2nd ed.

Mr. Matthews. held in place by a hollow round pin or axle through the centre through which the oil was forced, and if so, with what results. That seemed to indicate a cheaper mode of production, and owing to the varying speed of rotation of the collar the six small disks would tend to wear themselves true, being free to turn on their axles.

Mr. Michell. Mr. A. G. M. Michell pointed out that the results derived from the theoretical treatment of rectangular bearing-blocks (such as the formulas on page 241 and *Fig. 11*) might be made available as approximations in the more useful case of segmental thrust-blocks, by a suitable transformation. In effect, the segmental block might be regarded as a map-projection of the rectangular block, according to the following laws:—

Figs. 16.



1. To the point (*Figs. 16*), having the rectangular co-ordinates x, y in the rectangular block whose sides were a and b (x and a being measured in the direction of motion), corresponded a point in the segmental block having the polar co-ordinates

$$r = r_0 e^{\frac{\alpha y}{a}},$$

and $$\theta = \frac{\alpha x}{a},$$

α being the angle between the front and rear edges of the block.

2. The thickness of the oil-film at the point x, y of the rectangular block being $c(h + x)$, that of the film at the corresponding point, r, θ, of the segmental block was $c\, r \sin\left(\theta + \frac{\alpha h}{a}\right)$, the face of the block intersecting the plane face of the other bearing surface on the radial line $\theta = -\frac{\alpha h}{a}$.

These conditions being observed, and the angles α and $\frac{\alpha h}{a}$ being not too large, the oil-pressure at each point of the segmental block rotating with angular velocity w, would be, to a first approximation, the same as at the corresponding point in the rectangular block, the velocity of the latter being taken as $\frac{w\,a}{a}$. It resulted that, contrary to what might appear at first sight, the inner and outer portions of the segmental block were equally effective in generating oil-pressure, and both carried their full shares of the load. The approximate distribution of pressure in the segmental block being determined as above, it was easy to find the centre of pressure and to assign the theoretically correct position of the supporting point, and to deduce other properties for the purposes of design. Tests recently made on segmental blocks designed by the method just outlined, had shown a satisfactory correspondence with theoretical results. The leading dimensions of a bearing used in these tests were:— Mr. Michell.

Diameter of thrust-collar	6 inches
Number of pivoted blocks	2
Area of bearing-surface of each block	4·05 square inches
Approximate distance of the centre of the bearing-surface from the axis of rotation (varied, however, slightly in different tests)	2·16 inches
External radiating surface of the bearing (air-cooled in all cases), about	2 square feet.

The pivots of the blocks were approximately $\frac{1}{8}$ inch behind the centres of the bearing-surfaces, and nearly in the position corresponding with *Fig. 11* (p. 242). The bearing was driven by a small electric motor, of which the output was determined under the various conditions of the experiments. Some results of the experiments were given in the following Table:—

Revolutions per Minute.	Total Load.	Intensity of Load.	Kind of Oil.	Viscosity of Oil C.G.S. at 60° F.	Temperature of Oil in Bearing.		Power Absorbed.	Coefficient of Friction.
					°F.	°C.		
	Lbs.	Lbs. per Sq. In.					HP.	
508	3,000	370	Petroleum	0·017	72	22·2	0·049	0·00096
418	10,000	1,235	Mineral colza	0·08	71	21·7	0·137	0·00095
525	5,000	617	Mineral colza	0·08	74	23·3	0·075	0·00084
1,350	5,000	617	Mineral colza	0·08	115	46·1	0·372	0·00161
744	5,000	617	Red mineral	1·08	114	45·6	0·164	0·0014
744	10,000	1,235	Red mineral	1·08	116	46·7	0·228	0·00096

Mr. Michell. In the foregoing results the diminution of the coefficient of friction with increase of load, decrease of speed and decrease of

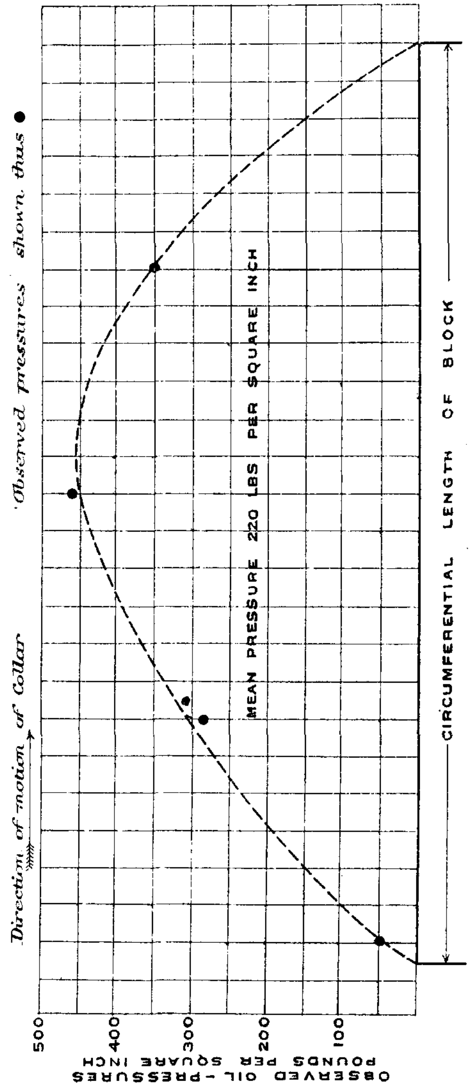


Fig. 17.

viscosity were clearly apparent, and the experimental values of the coefficient approached fairly closely to the theoretical values calcu-

Mr. Michell.

lated by the process described. In other experiments on thrust-blocks of the same size pressure-tubes were used, on the lines of Mr. Beauchamp Tower's experiments. *Fig. 17* showed one of the curves of pressure obtained, the pivot of the block being situated at 0·42 of the length from the rear end, as indicated by theory. The effect on the curves of pressure due to varying the position of the pivot was also examined, and it was found that a definite distribution of pressure only occurred when the pivot was behind the centre of the block, as indicated by the theory. In some cases, however, lower and irregular pressure-readings were obtained with the pivot at, or in front of, the centre of the block when the block was not rigid or its face not truly plane. It was evident that in such cases pressures might be generated under portions of the blocks which happened to form suitably inclined surfaces, and it was doubtless the undesigned formation of such surfaces on portions of the working-faces which enabled ordinary non-pivoted bearings to operate at all.

The Author.

The AUTHOR, in reply to Mr. Kingsbury, said Mr. Michell's statement, that the coefficient of friction in a block three times its breadth long was thirteen times that in a square block, was based on theory only, and was made on the assumption that the angle between the surfaces remained constant. If the angle were increased the pressure must be reduced, and it was more convenient to assume a constant angle than a constant pressure. The actual friction in the longer block might not be so much higher, but the oil-film would carry less pressure owing to the leakage at its sides. In reply to Mr. Matthews, the cross piece F in the bearing shown by *Figs. 12* was set to suit its journal bearing. The case D was adjusted by the ring E moving eccentrically to the shaft S. The object of limiting the number of blocks in this case had been to reduce the friction, which was proportional to the area of the rubbing surfaces. It would not be advisable to drill holes in the centre of the blocks in the manner suggested, as the oil-film was at its highest pressure there and the hole would allow it to escape. The bearing generated and maintained its own oil-pressure: no forced lubrication was required.