The Authors.

automatic take-up gear would have to be provided with friction apparatus instead of ratchets. The latter could give only step-by-step adjustment, and the degree of accuracy depended upon the fineness of pitch of the teeth, or on an arrangement of multiple pawls. The spring-box used in connection with the Whitmore brake had the effect of preventing quick braking from taking place, unless means were provided to render the springs inoperative at that point where quick braking should commence. The floating lever gear on the brake-engine ensured that the engine piston and the brakes took up a position corresponding with the driver's brake-lever; but, unless the brake-gear was maintained in accurate adjustment—not necessarily the case when depending on ratchet teeth—it did not follow that the intensity of braking pressure would correspond with the position of the brakes. On the other hand, the Iversen gear, and the similar devices described in connection with oil brake-systems, ensured that the intensity of the braking pressure corresponded with the hand lever, irrespective of the position of the brake-engine piston. An automatic take-up gear was, however, advisable in connection with these devices, for the reason stated by Mr. Whitmore, namely, to prevent the piston from creeping gradually so that it could reach the bottom of the cylinder before the brakes were fully applied. This was especially apt to occur when wood blocks were used without a lining of friction material, as was frequently the case.

The object of ventilating the brakes, referred to by Mr. Burrows, was mainly to prevent heat from flowing into the drum-sides. It had also been proved that the temperature of the brake-path and lining was reduced by this means, while certain makers of friction material provided ventilation by a corrugated radiator fixed under the lining material, which they claimed ensured a longer life for the lining. It was generally believed that wood brake-blocks were less liable to become charred if the paths were ventilated.

Correspondence.

Mr. Broughton.

Mr. H. H. Broughton observed that few engineering firms had had the opportunity or the courage to build engines of the size and type described in the Paper. The machines illustrated were notable in many ways, but chiefly on account of their great size. Thus the Ward Leonard machine, Figs. 1, Plate 4, was designed to hoist from a vertical depth of 3,000 feet, at a maintained rate of 336 short tons per hour, and the corresponding duty was

Mr. Broughton. 1,008,000 foot-tons per hour. The machine was one of the largest winding-engines in operation, and was worthy of the great workshops in which it had been constructed. The underground machine, Figs. 3, Plate 4, driven by a 2,160-HP. motor, was probably the largest three-phase winder that had been made. It was of little interest mechanically; but for its manipulation a special type of liquid controller had had to be designed, which was capable of dissipating 5,000 HP. continuously. He gave those figures because he had found there were still some engineers who regarded electric winding as an expensive luxury. A more expensive luxury was that of consuming 6 per cent. of the output of coal at the collieries in the uneconomical generation of power; and obsolete steam winders were largely responsible for that state of affairs. At a time when electrical energy was known to be an important factor in industrial progress it was disconcerting to find that no less than 43 per cent. of the collieries in Great Britain could afford, or were compelled, to do without it, and that, with 2,800 mines in operation, only 182 shafts were equipped with electric winding-engines. While a great deal of matter of a controversial nature had rightly been embodied in the Paper, he considered that a number of inaccuracies had crept in, and that certain of the recommendations would have to be modified before they were acceptable to makers and users of winding machinery. In the section on steel-plate drums the Authors referred to the fouling of the ropes by broken rivets, and to "the disintegration of the drum unless rivets are continually replaced." He thought it would enhance the value of the Paper if the Authors would cite examples of disintegrating drums. He knew of only one case where a large number of rivets had had to be replaced; but it was for other reasons than those given. Neither breathing of the drum nor reversal of stress was a sufficient explanation of what the Authors termed shearing of the rivet-heads. No matter how the internal sleeve was connected to the drum-bosses, the sleeve could not be expected to transmit much torque, nor did it do so. Its principal function was to act as a distance-piece, and it was arranged so as to prevent axial movement of the bushes. The latter were stated to be made of gun-metal, whereas, elsewhere in the Paper, it was stated that white metal was superior "without qualification" to all other bearing metals. If that were so, why was the use of inferior metal recommended for the drum-bushes? Apart from what he regarded as faulty design of the expensive "errorless" clutch, he considered that an inspection of Fig. 5, Plate 4, furnished the reason why such clutches had not been used to any extent. He thought that makers, instead of advocating heavy additional expenditure on appliances of doubtful value, should aim at simplifying design and reducing cost of construction. It was unsatisfactory to know that very large orders for winder components had had to be placed on the Continent by British firms in order to reduce the cost of manufacture. Mr. Broughton.

The design of a brake was figured on emergency conditions, and the severity of the latter, manifestly, depended on the winding-conditions as well as on the position of the cage or cages. Irrespective of the degree of the emergency, however, there was a braking limit which ought not to be exceeded. He thought that limit was reached with a retardation of about 12 feet per second per second, and if the brake-drums were proportioned to resist the high stresses induced by emergency braking there was nò fear of disintegration. The so-called ventilated brake-path described in the Paper was ill adapted for an emergency stopping brake. It had been used by a large number of makers, but chiefly on small machines. The small mass of metal in the path made overheating inevitable, and the effect of the ventilating-ducts was to insulate the path from a large volume of metal which, from its configuration, was well suited for the storage and dissipation of heat. Actually the ducts were still-air pockets which retarded heat-dissipation. In comparison with the parallel-motion brake fitted with trussed shoes, the brake arrangements depicted in the Paper were crude and not sufficiently sensitive for large winding-engines. Users were probably responsible for many of the arrangements which resulted in fierce braking and dangerous oscillation of the braking effort under emergency conditions.

The number of winding-engines fitted with fine-pitch gears was so small, and the application was so recent, that there was no experience to give guidance as to either their suitability or durability. He did not think it had been established that such gearing applied to a winding-engine was more durable and more reliable than accurately-cut gearing of normal pitch. It was, moreover, necessarily more expensive, and the utmost accuracy was requisite in erection. If the results obtained in other applications could be taken as a guide, he thought that fine-pitch gearing would not be generally adopted. Of the three winders illustrated in Plate 4, only one had fine-pitch gearing, and the rating was very conservative for nickel-steel pinions. The teeth were of the ordinary involute form. Irrespective of tooth-form, however, fine-pitch gears were used from necessity, not from choice, for high pitch-line speeds and large ratios. In a winding-engine the simple rule to follow was to avoid gearing, if possible, and to adopt involute teeth of coarse or medium pitch when gearing was imperative. Only in exceptional cases was gearing requisite for Ward Leonard winders; but it was

Mr. Broughton. indispensable for three-phase equipments. In deciding on the type of gearing to use, it was well to remember that winder gears were of low ratio, and that the pitch-line speed was relatively small. He had had occasion to examine a very large number of winding-engines and in several hundred examples he had found that more than 90 per cent. had a gear-ratio of less than 7 to 1, and in no case did the pitch-line speed exceed 3,250 feet per minute. In other words the application was an ideal one for normal-pitch gearing. That that was generally recognized was supported by winding-engines aggregating nearly 500,000 HP. to which such gearing was applied, and some of those gears had been in operation for close on 20 years. The great overall width of fine-pitch gears made it difficult to obtain and maintain correct meshing of the teeth, and the trouble was aggravated by the relatively great distance between the bearings.

As long as certain elementary principles were observed in the design, the type of bearing was not of much consequence, though the less complicated the type the better. The four-part bearing was the least suitable, and was entirely out of place in an electric winding-engine. Parallelism of the two shafts, which was indispensable for the proper meshing of gears, could only be effected by angular movement of the pedestals. Leakage of the oil at the joints in important parts of the bearing was another disadvantage. If they disregarded fine-pitch gearing, which they were entitled to do, since it was only used to a very limited extent, the largest users of electric winding-engines preferred two-part self-aligning bearings to the rigid type. Ring lubrication was desirable for both the pinion- and drum-shaft bearings. The Authors rightly laid stress on the importance of ensuring the creation and maintenance of an oil-film in a bearing. The running-clearance, which was one of the vital factors, was a function of the shaft-diameter, and depended on, among other things, the surface speed of the journal, as well as on the nature and temperature of the lubricant. For a journal-speed of less than 25 feet per second, a running-clearance of about 1/40 inch was requisite for a 20-inch shaft. When the load acted vertically downwards the clearance was provided in the upper half of the bearing, and about two-thirds of the lower half was bored to the net diameter of the shaft. Although it was stated in the Paper that the bearings described ensured an oil-film on principles " which enable the friction to be reduced to one-twentieth of the figure previously obtained," that could not be taken to mean that the bearings described were more efficient than other well-designed bearings. That was not the case, because all bearings that were run under similar conditions and in which the oil-film was maintained were known to be equally efficient. It was a little difficult to follow the Authors when in one place they stated that the specific load on the bearing did not exceed 200 lbs. per square inch, and in another that the product of specific load and surface speed could safely be made 600,000, at which figure there was a factor of safety of about 10. A drum-shaft of which he had knowledge was supported by a 24-inch by 48-inch bearing, and the surface speed of the journal was 300 feet per minute. The bearing, which was pressure lubricated and correctly designed, carried a total load of 242,000 lbs., or 210 lbs. per square inch, whereas, according to the Authors' formula, it could have been loaded to 600,000/300 = 2,000 lbs. per square inch, and even at the latter load would have a factor of safety of 10. He was not questioning the accuracy of the research that had been made, but he doubted the validity of applying it to cases lying outside the range of the experiments. If the Authors had faith in their formula, it would be of interest to know why the formula had not been used in designing any of the bearings illustrated in the Paper, the more so since some of those bearings weighed several tons each. None of the bearing arrangements described prevented deflection of the shafts, although they were said to do so. The word "rigid" applied to a bearing was a misnomer, because no bearing was sufficiently rigid to prevent deflection. Even if it were, the fact that the journal was a loose fit in the bearing (and it had to be for the oil-film to be established) made the shaft nearly as flexible with one type of bearing as with another. The usually accepted reason for fitting adjustable folding wedges under the centre bearing differed from that given in the Paper. It was mentioned that the load on that bearing was liable to be excessive; and the object of the wedges was not to counteract wear, but to enable the bearing to be set off-centre so as to adjust the load on the bearing to an amount which, in practice, it was capable of carrying without overheating. If the wedge adjustment was used for compensating for wear, why was not similar provision made for the other bearings? Mr. Broughton

The primary objects of a flexible coupling between motor and pinion were not stated in the Paper. Briefly, such a coupling was necessary in order to absorb the mechanical shock that occured on starting the motor, and to enable the rotor to pull itself into the magnetic centre independently of the pinion.

While he did not agree with many of the conclusions on the subject of control-gear, he would not go further into detail. There was, nevertheless, a matter to which reference ought to be made. Figs. 15, 18, and 19, Plate 6, had been prepared under his instructions in 1922 with the object of illustrating in as simple a manner as possible what should be avoided.

Mr. Burns. Mr. SYDNEY BURNS congratulated the Authors in that they had dealt with the subject of drums and incidental apparatus along interesting as well as instructive lines. He was not competent to discuss much that was given in the Paper in regard to constructional details of design, but he did know that the Authors were mistaken in their estimate that correct rope-lapping might be relied upon—whether the drum were plain steel barrelled or wood lagged —with an " outside " angle corresponding to 1 in 40. An unpleasant " rope-lapping " experience had led him to investigate this subject carefully. He had collected data for every electric winder in the Northumberland and Durham coal-fields. It would be out of place here to review his conclusions[1] at length, but he was confident that with plain cylindrical drums and under winding-conditions such as those encountered in the North of England, the " outside " angle through which the lead from the headgear pulley to the first live turn was deflected from the straight lead should not be greater than that corresponding to 1 in 75 if correct lapping was to be ensured. Similarly, if undue wear between adjacent coils of the rope towards the end of the wind was to be avoided, the "inside" angle should not exceed that corresponding to 1 in 50. If the drum was of the bi-cylindro-conical type the width at its larger diameter should not be less than the cage-centres dimension. It was frequently argued that it was beneficial to set the headgear pulleys in planes which were not at right angles to the axis of the drum-shaft, but the horizontally projected dimension of the arc of contact of the winding-rope at the pulley was so small in comparison with the rope-plane dimension that, unless he was mistaken, the inclining of pulleys in that way had negligible effect in so far as the correction of imperfect lapping was concerned. Ventilated brake-treads were fashionable, but he wondered whether the designer's natural striving for effective heat-dissipation had sometimes led to a less robust construction than was desirable. There was a tendency to reduce the thickness of metal unduly, and he had in mind more than one instance where the treads had been brightened (by rubbing contact with the brake-blocks) only at those points which were immediately above the supporting ribs on the brake-rim. He recalled one case where this defect had led to serious inconvenience. It should also be borne in mind that the brake mechanism of a bi-cylindro-conical drum winder should be so designed that a fully-loaded cage might be held in suspension above banking-out level, that was, with the taut rope at the larger diameter of the drum. Under those

[1] " Winding Costs: A Plea to the Purchaser of Electric Winding Plant for the Withdrawal of Stipulations which are not conducive to Economical Working." Trans. Inst. Mining Eng., vol. 71 (1926), p. 141.

conditions the unloaded cage would be resting upon the balks, and the rope associated with that cage would be slack. There was a minor brake-gear problem about which it would be interesting to hear the Authors' opinion. It was conceivable that a men-raising or men-lowering trip might be interrupted, owing to failure of supply and the consequent automatic application of the air- or oil-operated brakes. In such an event it might be felt to be desirable either to lower the cage to the flat sheets or to raise it to bank (as the static conditions might permit) by careful hand-manœuvring of the brakes. Had a reliable appliance been developed whereby to effect this operation? He did not think there was mention in the Paper of the statutory regulation with respect to the boring of drum-shafts where the diameter was 10 inches or more. Did that regulation apply abroad? Mr. Burns.

Mr. J. HAMILTON GIBSON observed that the Authors gave the maximum allowable pressure on the projected area of the journals as 200 lbs. per square inch; and presumably that was the pressure adopted in the "improved" or special design of bearing (Figs. 12, Plate 5), which was stated to embody special features to ensure the maintenance of the necessary oil-film on principles similar to those employed in the Michell bearing. There was no indication of this in Figs. 12, unless it was assumed that the brasses were bored out slightly larger than the shaft, with a thin liner at the joint, which was removed before assembly. That would leave the brasses slightly open at the sides where the oil entered, and would produce a quasi-Michell tapered pressure film at the top and bottom centres only; thus following the practice established in naval engineering for main bearings a quarter of a century ago. If that was so, there was nothing startlingly novel in this "special design," although it might be an advance on previous winding-engine practice. The Michell system was now sufficiently well known to require no detailed description, and its principles had already been the subject of a Paper[1] read before The Institution. Referring again to Figs. 12, each top and bottom brass would be replaced by at least three pivoted segments or pads, and the lubricating oil—which might be either "circulated" from an outside system or drawn in automatically by light collars clipped to the shaft, from a reservoir in a self-contained pedestal—formed an unbreakable film under every pad. Loads up to 600 lbs. per square inch of projected area had been carried, and the axial length of the bearing could be reduced in proportion. A big marine turbine installation carrying this load showed no signs of wear or metallic contact after 8 years service. A bearing of that type, with oil-films Mr. Gibson.

[1] Minutes of Proceedings Inst. C.E., vol. cxcvi (1914), p. 223.

Mr. Gibson. all round, like an elastic garter, instead of at one or two points only, was ideal for small-toothed gearing such as was described in the Paper, and was independent of the direction of the load. Many such bearings were fitted in marine turbine gears, for both single and double reduction, the advantage being that the wheel and pinion centres were rigidly maintained. The rocking pivots had an axial length of only about one-half of the length of the pad, which, in conjunction with the shortness of the bearing, provided for the sag of a heavily loaded shaft. Spherical self-aligning bearings, such as those shown in Figs. 14, Plate 5, were thus not only unnecessary, but obsolete in the best engineering practice. The extended ends shown could be dispensed with, and there was no need for the archaic ring lubrication or for oil-grooves cut in the brasses.

Mr. Williams. Mr. IVOR WILLIAMS remarked that there were a few points on which he would like to have the Authors' comments. One concerned the advisability of putting in a centre spider on drums, which, of necessity, transferred the deflection of the shaft to the drum-shell and caused "breathing" of that portion of the drum. He had experienced that disadvantage on a large winding-drum, and it would be interesting to have comments on experience elsewhere in that respect. With reference to the grooving of the drum-shell, was it preferable to have the grooves parallel, rather than helical, when there was more than one layer of rope? He understood that parallel grooving tended to reduce lateral vibration when the second coil was being wound. In the case of a cylindro-conical drum, what did the Authors consider to be the minimum width of the parallel portion on the large diameter of the drum, relative to the centres of the pit-head sheaves, so as to ensure a lead from the conical to the parallel portion of the drum? With reference to the Authors' experience as to the maximum size of moulded cone-plates, namely, 16 feet, his firm were using excellent drums made by various manufacturers in this country with diameters up to 18 feet 6 inches, and all joints had proved to be perfect. He was interested in the remarks regarding the advisability of using small-pitch gearing. His own experience, however, tended to be in favour of the coarse pitch and narrow faces of the triple helical design. In the case of some of the largest alternating-current geared winding-engines in the country, small-pitch gears with comparatively broad faces were all showing signs of pitting, whereas in the case of the narrow wheel with coarser pitch, the teeth were in excellent condition. He wondered whether that was confirmed by the experience of other engineers. He agreed that forced lubrication was necessary with gearing, but he suggested that, in order to ensure that the gears were properly lubricated at the beginning of the wind, when the load was heaviest, the pump drive should be independent of the winder. Mr. Williams.

The Authors. The AUTHORS, in reply, remarked that, referring to the question of disintegration of built-up drums through rivets failing, they had been informed of instances where rivets had constantly to be replaced in new winders installed in devastated areas in France, and they knew of actual occurrences in this country. It would, however, in their opinion, be unwise to cite examples. The question of the internal sleeve in a drum transmitting torque had been proved in actual practice; one of the Authors knew of an instance where loose drums had been fitted without distance-sleeves and trouble had been experienced with the bolts which secured the barrel-plates to the drum-cheeks. Distance-sleeves were then fitted and the trouble ceased.

The Paper made sufficiently clear the different functions of gun-metal and white metal, while it should be remembered that gun-metal bushes might be white-metal lined.

The errorless clutch described was the only one, so far as the Authors were aware, which was absolutely errorless, and therefore the question of its extra cost depended upon the value set upon this feature. Although accurate machining was absolutely essential, the design was quite simple; it would appear useless to criticize it as of faulty design without specifying wherein the faults lay.

The Authors agreed that it was very unsatisfactory that winder components were ordered from the Continent, especially in view of the ample facilities possessed by firms in this country for undertaking the largest class of work by the most approved methods.

The braking limit duty specified for large winders was usually about 6 feet per second per second, while 10 feet per second per second was stipulated very occasionally. The ventilating-ducts in brakes could be so designed that, contrary to acting as still-air pockets, they would actually create a circulation of air, and under such conditions it was conceivable that ventilation was of benefit. The braking arrangements described in the Paper were in continuous successful operation on a number of the largest winders working. Modern braking-appliances would provide for the contingency mentioned by Mr. Burns.

The Authors had nothing to add to what they had stated in the Paper, and, in reply to the discussion on the vexed question of fine-pitch versus coarse-pitch gears, beyond the fact that progress in the direction of more scientifically designed gearing would inevitably force its way. Small-pitch gears frequently pitted slightly at first when running in, but directly any high spots present were self

The Authors. corrected, the pitting ceased and did not recur. Experience had proved that after some years of work the condition of the tooth-faces was actually improved.

It did not appear to be realized in some quarters that the special bearings described in the Paper bore, in effect, the same relation to ordinary journal bearings that Michell thrust bearings bore to ordinary thrusts. The figure of 200 lbs. per square inch bearing-pressure did not necessarily apply only to the special bearings described.

The control-gears illustrated by Figs. 18 and 19, Plate 6, had been fitted by one of the Authors to winders built in England between 1910 and 1912 for India, and had given satisfaction. That shown in Fig. 18 had been repeated for the same mine. There were, no doubt, many other instances of the use of these control arrangements even before, and certainly after, these dates. The gear shown in Fig. 15 was generally similar to that used in the majority of single-drum winders during the past 16 years.

The Authors agreed that a rope angle corresponding to 1 in 40 was a limit, and one to be avoided if circumstances permitted. The question of parallel versus helical rope-grooves had already been dealt with in reply to the discussion.

The regulation respecting the boring of drum-shafts 10 inches in diameter and more, did not apply generally abroad; it was frequently specified, however, by British engineers there, and reputable manufacturers preferred to do it in any case in order to relieve internal stresses and also to permit of a more thorough inspection of the shaft material.

The drum-shaft should be so proportioned that its deflection under load was so small as to produce no appreciable "breathing" of the drum through the use of a centre spider. The minimum width of the parallel portion of the main barrel of a cylindro-conical drum should be the distance between the cage-centres. Moulded cone-plates could be made exceeding 16 feet in diameter, of segmental construction, in cast iron; the difficulty at the joints mentioned in the Paper referred more particularly to cast steel.

It was the usual practice to drive by an independent motor the pump which returned the oil to the gravity lubrication-tank. This was indicated by Figs. 13, Plate 5. Safety switches were introduced in the system, as described in the Paper, to ensure that oil was fed to the bearings and gears before the winders could be started.