Discussion.

Sir William Ellis.

Sir WILLIAM ELLIS, Past-President, in congratulating the Authors on their Paper, remarked that he was very anxious that The Institution should more often have mining and colliery matters brought before it. The Paper dealt in a practical manner with an important and somewhat novel subject. Progress in electric winding had to be made very cautiously, because there was probably no branch of engineering where the engineer in charge was more directly responsible for the lives of thousands of men every day. Therefore the question of the introduction of electric winding, in place of the long-established system of winding direct by steam, must demand careful investigation on the part of those who advocated the introduction of that type of machine. The Authors had introduced one or two controversial matters which were sufficiently important to be the subject of separate Papers. He was not prepared to agree or disagree with the explanation which they gave (pp. 109, 110) about a particular type of gearing. He had no reason to think that it was other than excellent, but the subject was very controversial, and he thought it required much fuller investigation. The particular types of winders described in the Paper had been manufactured by a firm which were also eminent as builders of geared turbines for ship-propulsion, and it was evident that their experience of the fine-pitch gearing which had been adopted for turbines had been reflected in their choice of gearing for winders, namely, fine-pitch gearing instead of the larger pitch which had so long been used for machine tools. He would like the Authors to give one or two diagrams which would explain some points in connection with colliery winding which were not perhaps understood by the engineer in general practice. He had touched on one matter in his Address a year ago, namely, the fact that winding in a modern colliery was a case in which the load varied perhaps more widely, in the course of about 45 seconds, than in any other branch of engineering, unless in the case of a locomotive starting a heavy train from rest. It would be of interest to have compared, on a diagram, the varying loads which came on electric and steam winders respectively. With steam winding it was necessary to have such power as would rapidly start a heavy load from rest. The acceleration was very rapid and the deceleration was almost as

Sir William Ellis. rapid. The deceleration or braking in the case of steam winding was done by using the pressure of the steam against the piston. If the power of the winding-engine were plotted against time, as in *Fig.* 2 in his Presidential Address,[1] it would be found that the horse-power developed rose very steeply until the maximum speed was attained; but the period of the whole winding was so short that it was necessary to retard the winder as soon as it had reached the maximum speed, and the consequence was that the power curve fell very rapidly and crossed the zero line, so that the machine had to absorb power from the load in order to stop the cage in time. His object in raising that question was to suggest that the elimination of that braking-loss provided a splendid opportunity of effecting economy in a case where economy was of great importance. It was of course, a difficult question, requiring thought, but the necessarily rapid deceleration provided a source of power which, if it could only be used, would enable something to be put back into the mains. He knew that had not been done to any extent, but he hoped it might be done after more experience had been gained.

Mr. Bailey. Mr. T. H. Bailey observed that he particularly appreciated the Paper because he had been the first colliery-engineer to employ electrical power for underground haulage, in an installation carried out by Messrs. Crompton & Company in 1891. The steam engines had been able to deliver to the pit-bottom about 600 tons a day, and the electrical plant was able with the greatest ease to deliver up the same inclination and the same roads 1,000 tons in the same time. That installation had been at work for many years, and the sole difficulty experienced had been slight initial trouble with the controlling switches, which was a common complaint in those early days. Since then electric underground haulage had made vast strides. The collieries in Staffordshire with which he was now connected had no less than seventy underground electric hauling-engines, and only fifteen ponies were required, instead of the 250 to 300 used in the past. The Paper dealt with one of the most recent applications of electrical power to colliery enterprise, and he was looking forward to the day when the electrical winding-engines would be so perfected that it would be possible to draw from great depths with the utmost ease and without trouble. He would be glad if the Authors would give particulars of the work done by the electrical engines illustrated in the Paper, and of the power which was exerted both at the beginning of and during the wind. One or two difficulties had to be dealt with in winding from great depths.

[1] Minutes of Proceedings Inst. C.E., vol. 221 (1926), p. 10.

The winding-ropes were extremely heavy, the dead weight at the pit-bottom to be constantly lifted, including the weight of the rope, being 30 to 40 tons, and that made it impossible to obtain ropes of light weight which would stand the very high stress put upon them. When at work they acted like springs. That difficulty arose with steam winding, as well as with electric winding, and it had been enhanced by the adoption of conical drums. It was necessary to take special precautions in the winding of men, and all collieries had two sources of electric current in case of an accident happening and one source failing. Another important matter was the character of the engine-men who manipulated the winders. He knew that Sir William Ellis would bear him out when he said that the men who controlled those machines were some of the finest characters in the country. He did not think anybody nowadays would doubt that the electric drive was the best means that had yet been devised for winding from great depths. Lately he had seen a pair of electrical winding-engines erected at Snowdown colliery in Kent, where the depth of the seam was 3,000 feet; and the working of those machines was remarkably sweet and easy. He doubted whether it was advisable to provide no windows for the engine-man to see what was going on at the top of the shaft, and to depend entirely on signals. However, engine-men had told him that they felt no uneasiness in such circumstances. Mr. Bailey.

Mr. H. W. Ravenshaw remarked that he had taken a large number of diagrams, similar to that mentioned by Sir William Ellis, in connection with both very fast steam winding-engines and electric winders, and he agreed that it would be of interest if the Authors gave particulars of acceleration, load, speed, and, particularly, braking-effort. A steam winder with which he was well acquainted performed sixty or seventy winds an hour, and he had noticed that the driver never used the brake, the stop being only about 12 seconds. He did not think that that could be done with electric winding. One of the problems with electric winding concerned the brakes. Were there any special requirements with regard to braking—how the engine was stopped, how long it stood, and whether the brakes had to come on and off every time—in connection with the winders which the Authors described? Could the use of brakes be obviated with the Ward Leonard system? He thought that the energy in that arrangement could be returned, to the advantage of efficiency. He had read a Paper[1] before The Institution nearly 30 years ago on a similar subject, namely, electric Mr. Ravenshaw.

[1] Minutes of Proceedings Inst. C.E., vol. cxxx (1897), p. 11.

Mr. Raven-shaw. lifts and cranes, and at that time engineers asked for particulars of the actual work done: he thought similar figures would be of advantage in connection with the present Paper. Some of the winding-engines he had tested had an acceleration of 4 feet per second per second, and he knew of retardations as high as 8 feet per second per second. Such rates put a very high stress on the the mechanism, and it was generally arranged that the acceleration should be lower with an electric winder; but the retardation might be equally high, because, in order to complete a wind in a reasonable period, it was necessary to pull up very sharply indeed. The positions of the control-levers in the old steam-engines were arranged to suit the maker's convenience. He knew of one old steam-engine which had to be driven by a separate gang because some of the levers worked in the opposite direction to those of the other engines; and it was not considered safe for the same men to work both types of engine. He would suggest that the British Engineering Standards Association might take up the question of the positions of control-levers, so that any driver could be familiar enough with any engine to remove all fear of his doing the wrong thing.

Mr. Whitmore. Mr. F. L. WHITMORE observed that the Authors referred to spiral grooving (p. 98), but no mention was made of parallel grooving. Where two or more layers of rope were coiled on the drum, recent practice introduced concentric or parallel grooving, rather than spiral, because while spiral grooving made a smooth lead of the rope across the drum while running in one direction, it made a bad lead when running backwards over the first layer. The second layer acted as a exaggerated "drunken" thread: the rope receded and advanced spasmodically all the way across the drum, because during part of the revolution the spiral of the first layer was going backwards; and for the rest of the revolution the rope was pushed forward by the preceding coil over the first layer. That action agitated the rope and tended to cause it to jump a coil, which was detrimental in winding. He would like to know the opinion of other makers regarding that matter, because it was a moot question. The Authors explained (p. 101) that a distance-piece was put between the drum-cheeks to help to take the torsion off the drum-shell, but he questioned whether very much relief could be given by a sleeve so near the shaft, and he imagined that its principal function was to keep the drum-sides at a certain distance apart, and to retain the grease in the drum-bushes. With regard to clutches, he suggested that the Paterson clutch was a means which had been introduced, he would not say unnecessarily, but perhaps before it was wanted. The ordinary toothed clutch, so far as he was aware, was quite capable of making all the adjustment to the rope that was necessary. It was mentioned by the Authors that in a 30-foot drum the rope error might be as much as 24 inches. There was no reason why the teeth on the clutch should not be of a considerably finer pitch than they were to-day. He knew of instances where the pitch of teeth had been reduced to about 2 inches, and if such a pitch were used on a 10-foot clutch, the error would not exceed about 7 inches, which was near enough for practically all ropes on such large drums. The simple clutch avoided the complication that must necessarily follow with the Paterson type. The gearing inside the clutch-boss was detrimental: it was very highly stressed because it was so near the shaft, and it also required very accurate machining. The only object attained was the possibility of shifting the clutch to such a point that it would enter the portion on the drum. Was that clutch operated from the driver's platform? His experience was that brakes of the anchored-post type gave the best all-round results. They were cheaper than the centrally-suspended brake, and the effect on the brake-blocks was not very different. He thought it would be found that the action on the brake-block at the bottom of the shoe was not very different from that at the top. The anchorage was placed well under the brake-drum, and therefore the movement of the brake-block at the lower end of the shoe was directly radial, whereas at the top of the shoe it was partly tangential, and the resultant wear through the shoe was practically uniform. It seemed that since the shoe was supported at the bottom, and not between links which were not rigid in themselves, it was the best form of brake. The Iversen gear was intended to overcome the same trouble that the Whitmore brake was designed to avoid. It gave an elastic brake in which the braking effort depended solely upon the distance that the driver threw over the control-lever. The Whitmore brake had a spring in a large spring-box which had to be compressed in order to put the load on the brakes. The Iversen gear compressed a spring by emission of air and arrived at practically the same result, but the position of the piston did not correspond to the load that was on the brakes. That was not important, except that there might be times when the piston might be gradually creeping down and getting near the end of the range of movement and, unknown to the driver, the piston might be on the bottom of the cylinder so that the full force of the brake was not applied. That would also happen with the Whitmore brake-engine, but for an automatic take-up gear which was provided, so that, as the brake wore, the wear was automatically taken up, and Mr. Whitmore.

Mr. Whitmore. it was impossible for the brake-engine piston to touch the bottom of the cylinder. He was naturally in favour of gears of coarse pitch and narrow face, because he had had experience in that line. The fine-pitch gears advocated by the Authors appeared to be not entirely suited to winding: firstly the clearance was very fine; and, secondly, the width of the face suggested the possibility of deflection in the pinion. Winding-engines did not receive the attention that was given to many other classes of machines. Bearings might run very sweetly on a winding-engine, and as long as that was so, it was quite possible there would not be any trouble with fine-pitch gears. But bearings did wear, and the wear was not taken up as soon as it should be in many collieries where there was little time to attend to them. The result was that a little neglect, which of course was harmful to coarse-pitch gearing, was very detrimental to gears with fine clearances and fine-pitch teeth. The firm he was associated with had for many years used swivel bearings for both the pinion- and the main- or drum-shaft of electric winding-engines, and had found no occasion to put in any other type of bearing. It was advisable, in his opinion, to bolt the pedestals independently on bed-plates and not to incorporate all the bearings in one large casting. It was suggested in the Paper that, as no adjustment was required, the latter was the better method. That type of bearing required very accurate machining and erection. Adjustment by keys overcame those requirements and was not, in his opinion, detrimental. One of the large mining-companies in South Africa had specified that the drum-shaft and pinion-shaft pedestals should be parts of the same casting in the case of winders for several mines in different places. It had been feared that, if adjustment were provided, the various engineers might be liable to make an adjustment for some purpose without recognizing that it was doing something else, and would probably do more harm than good to the engines. That apparently had not been borne out recently, because that company now called for the pedestals to be separate. He had found that ring lubrication was satisfactory in all bearings in electric winders, and no other method had been necessary. In South Wales there was a cylindro-conical drum, 14 to 21 feet in diameter, weighing 65 tons, driven by a direct-coupled 1,330-HP. motor on each end of the drum-shaft. In watching the drum at work he had noticed there was a side movement of the drum-shaft of about $\frac{1}{8}$ inch during the wind. In that instance the driver did not touch the brakes, and when the cage came to rest at the bank the drum and shaft swayed from side to side for several seconds after the winding was stopped, due to the angular action of the rope.

He thought that was an interesting indication of the frictionless condition of the bearings. Mr. Whitmore.

Mr. G. P. W. SIMS remarked that the Paper was particularly valuable at the present juncture when the economical and efficient equipment of collieries was so much under discussion. He would draw attention to the somewhat arbitrary comparison between what the Authors termed coarse—which he preferred to call normal—pitches and fine pitches for gearing. That comparison was valid for straight-tooth gearing, but did not necessarily hold for the double helical type. The angle at which the teeth were cut had also to be taken into consideration. That point was not dealt with in comparing the two types of gearing mentioned in the Paper. The number of teeth in mesh was also stated, somewhat arbitrarily, as being only one. In the very large number of winding gears of normal pitch with which he had had to deal, the number of teeth in contact was usually three for such ratios as were ordinarily met with in big winding practice, namely 6 or 8 to 1. The influence of the number of teeth in mesh upon the starting conditions was also dealt with in the Paper, where it was stated that coarse-tooth gearing was at a disadvantage as compared with gears having fine pitch. In starting an electric winder the conditions of stress were static rather than dynamic. The speed was virtually nil, and strength rather than wear had to be considered. In the normal-pitch type of gear illustrated in Figs. 3, Plate 4, the reinforcement of the teeth required to meet the severe starting-conditions was obtained by shrouding. Could that be done with the type of gearing having enveloping teeth? Coarse pitches might apparently involve a certain disadvantage in gears of larger diameter, but that point had been dealt with at length by Mr. Whitmore. Gears having a fine pitch must of necessity have a wide face in order to get the necessary strength, and such wide faces might involve widely-spaced pinion-shaft bearings, and consequently there was the danger of deflection. *Figs. 11* (p. 111) purported to show the superiority of the fine-pitch gears in relation to their increased rolling contact as compared with sliding contact. That, of course, was a text-book maxim and held good, in his experience, for dry surfaces; for lubricated surfaces the conditions were somewhat different. Two teeth coming into mesh with a film of lubricant between approximated to the conditions found in the Michell bearing, which enabled the oil-film to be drawn in only so long as there was relative sliding motion between the two surfaces. It had been suggested by certain eminent authorities on the subject that to obtain the best results in connection with lubrication of Mr. Sims.

Mr. Sims. gears sliding action was virtually essential. He had had experience of a very large set of gears in connection with a mine in South Wales, where, owing to the pinion-shaft being bent, it could be assumed that at no moment was there true rolling contact, even at the pitch circle. He had examined those gears at various times and he had never seen a set in which the condition of the teeth was better after 4 years' use. It was insisted upon in the Paper that the fine-pitch gear necessitated very careful adjustment and maintenance ; that was obvious, and it seemed to him that, under the exigencies of colliery practice, it was somewhat of a disadvantage. Colliery plant had to run for protracted periods, and adjustments had to be left for the first possible opportunity. The satisfactory results obtained with the enveloping-tooth gear were apparently the results of extensive shop tests, and also presumably of lengthy experience, and it would be interesting to have figures regarding the Authors' experience obtained with that class of gear. The following figures related to horse-power transmitted on electric-winder installations by the coarse-tooth type of gear made by one particular firm. In 1919, 49,000 HP. was being transmitted through that type of gearing ; in 1920, 68,000 HP. ; in 1921, 76,000 HP. ; in 1922, 80,000 HP. ; in 1923, 103,000 HP., and in 1924, 112,000 HP.

Mr. Burrows. Mr. W. Burrows observed that there was a minor point in connection with brakes to which he would call attention. The Authors stated that in order to dissipate the heat generated by frequent application of the brakes it was advisable to ventilate the treads. That was a general practice, but there was difference of opinion as to what object was gained by the ventilation. Some makers of brake material said that the object of rapid radiation would be defeated by the recesses and holes, and that if a brake-path were made in continuous contact with the drum-side, the heat would be radiated far more quickly, because the heat-transference through cast iron was much more rapid than it was through air. Apparently the system of ducts described by the Authors was provided in order to dissipate the heat safely, because when the heat was allowed to flow quickly into cast-iron drum-sides there was a risk of cracking. That appeared to be the real object, and not to get the heat away in the quickest manner. The enveloping tooth gears described appeared to depend on sound principles. The surface in contact would be very greatly increased in that form of gearing, and the confinement of the contact to the recess angle meant that there was no reversal of sliding stress such as occurred in the ordinary involute

Mr. Burrows.

double helical gears when crossing the pitch-line, and sometimes caused pitting. At the same time, the fine-pitch gears were certainly disadvantageous for heavy work. The Authors specified that the bearings must be specially lubricated to eliminate wear, and he agreed that, if wear were eliminated, a principal cause of shafts getting out of alignment would be eliminated also. He did not think that the Authors had gone far enough to produce a bearing free from wear. They did not state the oil-pressure used, but from the description and illustration it looked as if it might be a nominal pressure of, perhaps, 15 or 20 lbs. per square inch, which was generally called pressure lubrication; but, in his experience, that pressure should be considerably increased. Unless the pressure of the oil exceeded the bearing pressure, the shaft would not be separated from the bearing at the start: it would need rotation of the shaft at some considerable speed before the oil-film was properly formed, and during that time wear was taking place. He had carried out much research work in the application of pressure lubrication to heavy-duty bearings up to 26 inches in diameter, with loads up to 80 tons per bearing, and could confirm its great advantages, especially when an oil-pressure in excess of the bearing pressure per square inch of projected area was adopted. Under such conditions and with specially grooved and fitted bearings, the oil-film was maintained even when the shaft was stationary, and therefore the friction at starting was even less than when rotating. Thus in this respect the same characteristics existed as in a ball bearing, and wear of the bearing and shaft was eliminated, as borne out in extensive practice. A bearing "flood" lubricated, or with forced oil-feed under a comparatively low pressure, would, no doubt, maintain a satisfactory film of oil between the surfaces while the shaft was rotating above a certain minimum speed, but at the lower speeds the film broke down, and thus in the case of shafts running intermittently, as in winders, the opposing metallic surfaces would be in contact a few seconds after the rotating parts came to rest; accordingly, the coefficient of friction under conditions of lubrication with only moderate oil-pressure, or with ring lubrication, might be as high as 0·15 at starting. With high-pressure lubrication, under an oil-pressure about 20 per cent. in excess of the bearing pressure per square inch of projected area, tests carried out on a 12-inch diameter bearing, with a loading of 300 lbs. per square inch, at 1,000 revolutions per minute showed a coefficient of friction of 0·0037, which fell to 0·0022, with 150 lbs. per square inch at 300 revolutions per minute. Starting from

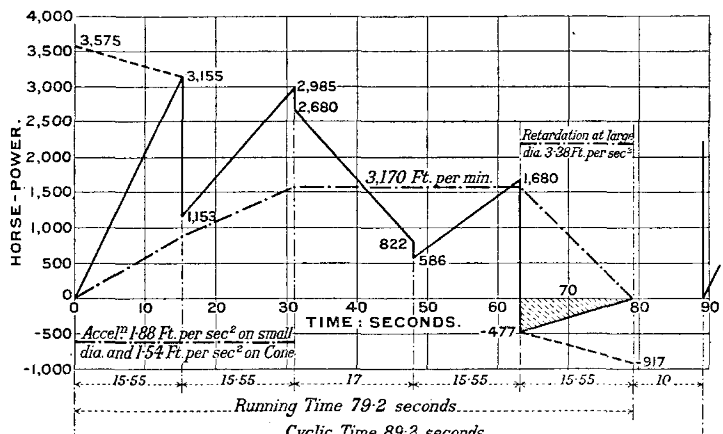
Mr. Burrows. rest with the latter bearing pressure the coefficient was 0·000011. These figures for running conditions were, he believed, lower than were obtainable under the alternative lubricating systems mentioned, while the figure for the friction at starting was much less than that of a ball or roller bearing. That had been confirmed repeatedly by many tests on other sizes of bearings, and incidentally enabled the static balancing of high-speed steel flywheels, used in connection with motor-generator equalizer sets for large Ward Leonard winders, to be carried out with the wheel and shaft mounted in its own bearings, for an amount out of balance of only 1 ounce at 6 feet radius would cause rotation of a 30-ton wheel carried in 12-inch diameter bearings under high-pressure lubrication conditions, and was thus a far more sensitive method than that of balancing on "parallels" which was often used. What oil-pressure had the Authors adopted? The thickness of oil-film mentioned in the Paper—1/3000 to 1/1000 inch—seemed to be very small, and could be obtained by a bearing lubricated by rings alone. With high-pressure lubrication a film not less than 0·005 inch thick was obtained, and it was usually in the region of 0·01 inch. The Authors specified a maximum rope-angle of 1 in 40 to ensure close coiling on the drum. His experience was that, even with the angle reduced to 1 in 70, open coils resulted at the commencement of the wind, and theoretically, for close coiling the angle should not exceed the helical angle of a closely-coiled rope, which ranged from 1 in 250 to 1 in 300 for the usual drum-to-rope ratios. With a drum 12 feet in diameter, a distance to head-sheaves of, say, 100 feet, and $1\frac{1}{2}$-inch rope, a lead angle of 1 in 40 would produce open coils for five revolutions of the drum with a travel of 26 inches across the drum during this period, and the first coil would be of about $9\frac{1}{2}$ inches pitch. Such open coiling was not of much consequence with a single layer, provided the final lead angle (at the end of the wind) was not thereby rendered excessive. With drums carrying a second layer, however, any open coiling of the first layers would be detrimental to smooth coiling of the second, and it was therefore preferable to have a grooved drum when required to carry more than one layer. He was surprised that the Authors referred only to grooving of a drum spirally, as some of the latest large winders, having two or more rope layers, had discontinuous grooves turned on the barrel, which method was claimed to give better coiling of the second lap than did spiral grooving. Could the Authors give any information on this practice?

The Authors.

The Authors, in reply, observed that Sir William Ellis's allusion to the necessity for care in winding, in view of the responsibility for men's lives involved, was fully realized by manufacturers, as witnessed by the continual progress in safety provision referred to in the Paper. Mr. Bailey's remarks regarding the character of winder-drivers generally, would be endorsed by everyone who had come in contact with those men. With regard to the opinion expressed that drivers should be able to see the pit-top from the platform, the tendency was now opposed to this, with the present perfection of safety arrangements, and it was considered both unnecessary and undesirable to attract the driver's attention to anything outside the winder-house.

Figs. 24, 25, and *26* gave horse-power—time diagrams of the duties of the three winders illustrated in Figs. 1, 2, and 3, Plate 4.

Fig. 24.

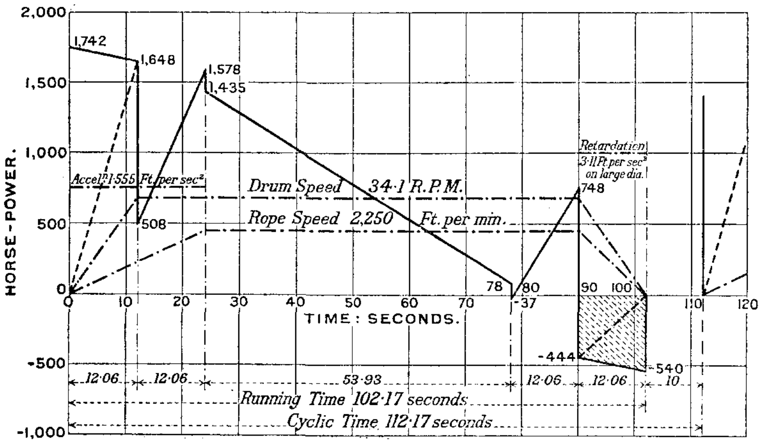


Bi-cylindro-conical Drum 14 feet and 26 feet in Diameter. (Figs. 1, Plate 4.)

Net load hoisted	tons	$7\frac{1}{2}$
Weight of each cage and chains	,,	10
Weight of tubs per cage	,,	$3\frac{3}{8}$
Depth of shaft (vertical)	feet	3,000
Maximum hoisting speed	feet per minute	3,170
Speed of drum	r.p.m.	38·6
Revolutions of drum per wind		40·93
Inertia of drum and gears (WR^2/g)	pounds-feet² (approx.)	550,000
R.M.S. power of each of two motors	horse-power	1,200
Peak ,, ,, ,,	,,	3,000
Speed of motors	r.p.m.	350
Weight of $2\frac{1}{8}$-inch diameter locked-coil rope	pounds per foot	10·8

The Authors.

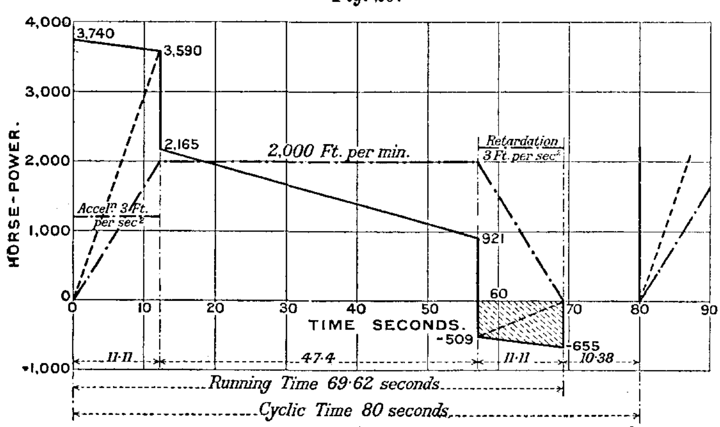
Fig. 25.



BI-CYLINDRO-CONICAL DRUM 10 FEET 6 INCHES AND 21 FEET IN DIAMETER. (Figs. 2, Plate 4.)

Net load hoisted	pounds 8,000
Weight of trucks	„ 3,200
Weight of cage	„ 11,000
Depth of shaft (vertical)	feet 3,152
Maximum hoisting speed	feet per minute 2,250
Speed of drum	r.p.m. 34·1
Revolutions of drum per wind	51·23
Inertia of drum and gears (WR^2/g)	pounds-feet² (approx.) 340,500
R.M.S. power of motor	horse-power 1,107
Peak „ „	„ 1,742
Speed of motor	r.p.m. {245, 250}
Weight of $1\frac{5}{8}$-inch diameter flattened strand rope	pounds per foot 4·9166

Fig. 26.



TWO PARALLEL DRUMS 16 FEET IN DIAMETER. (Figs. 3, Plate 4.)

The Authors.

Net load hoisted	tons	8
Weight of skip	,,	4·5
Depth of shaft (vertical)	feet	1,948
Maximum hoisting speed	feet per minute	2,000
Speed of drum	r.p.m.	39·29
Revolutions of drum per wind		38·5
Inertia of drum and gears (WR^2/g)	pounds-feet^2 (approx.)	275,000
R.M.S. power of motor	horse-power	2,160
Peak ,, ,,	,,	3,740
Speed of motor	r.p.m.	246
Weight of 2-inch diameter rope	pounds per foot	6·5

These diagrams indicated, below the zero line, the power returned to the winder by reason of the descending load overbalancing the load ascending. The diagrams would be similar for steam and electric winders, but in the case of the former, the negative power during retardation would be obtained by actually admitting steam against the engine. With electric winders on the Ward Leonard system of control regenerative braking might be used when negative peaks existed in the winding-diagram; the direct-current winder-motor would then be driven as a generator by the descending load and thus supply energy to the direct-current generator (running for the time as a motor) of the motor-generator set, either speeding up the fly-wheel or returning energy to the line, dependent upon the quantity of energy in the negative peak and the speed of the fly-wheel set at the instant of that peak's occurrence. The shaded portions of the diagrams showed the negative powers referred to. With three-phase winders reverse-current braking might be used, when the energy was dissipated in the winder-motor's liquid controller, with the disadvantage that the rotor voltage was increased to twice the stand-still value. It was also possible with alternating-current winders to use the regenerative method, where, by allowing the motor to attain super-synchronous speed, the latter operated as an induction generator and returned power to the line; this method, however, could not be used to bring the cage to rest and was therefore unusual, because it necessitated very careful operation by the winder-driver. The only really practical conditions under which energy was returned to the line occurred in the case of a straightforward Ward Leonard equipment, where any tendency of the induction motor to run over its synchronous speed resulted in its operating as an induction generator, returning energy to the system.

The comparative merits of steam and electric winding varied in practically every case with the different conditions prevailing in

The Authors. individual mines, but there was no doubt that, in view of modern developments, electric winding was the most advantageous from all points of view in the majority of cases. The best illustration, naturally, was afforded by considering the following actual examples.

The first example was an English colliery using comparatively small electric winders.

	No. 1 Shaft.	No. 2 Shaft.	No. 3 Shaft.
Depth of shaft . . . . feet	217	480	900
Weight of cage and chains . . . . . . tons	1½	2½	1½
Weight of tubs . . . . cwt.	12	24	36
Load of coal raised . . . . cwt.	20	40	60
Approximate weight of rope . . . lbs. per foot	3½	3½	3½
R.M.S. horse-power of motor.	86	150	300
Gear-ratio	11·8/1	11·2/1	10·4/1
Drum diameter . . . . feet	9·5	9·5	10·5
Peak horse-power of motor	130	235	530
Maximum working torque . . . . lbs.-feet	2,080	3,600	6,400
Width of drum . . . . feet	4	4	8
Circumference of rope . . . . inches	4½	4½	5
Speed of motor . . . . r.p.m.	284	284	328

Comparison of Systems.

		Approx. Total Capital Expenditure.
(1)	Three electric winders and equipment. Power supplied by Power Company.	£11,000
(2)	Three electric winders and equipment. Power supplied by central generating station installed at colliery (sufficient boiler plant being already available).	£22,000
(3)	Three steam-driven winders	£10,500

The report showed that the power cost for lifting 1 ton of coal, including interest and depreciation charges, but excluding all colliery royalties, rentals, labour, overhead and general charges, etc., worked out at :—

(1) 1.8*d.*
(2) 0.89*d.*
(3) 1.6*d.*

and consequently, though the initial capital cost for scheme (2) was high, it was much more economical when taken over a period of years than either of the other two schemes.

A second example was provided by an English colliery using large winders. The following figures were reproduced by the courtesy of the Metropolitan-Vickers Electrical Company, Ltd., for whom the Authors' firm built the winders referred to:— The Authors.

Output per annum	tons 894,000
Net load per wind	tons 7·5
Depth of shaft	yards 1,000
Auxiliary load, no coal winding taking place	kilowatts 750
Auxiliary load during coal winding period	kilowatts 1,300
Standing charges, repairs, maintenance, stores, rent, rates, and taxes	per cent. of total 18
Necessary buildings, depreciation and maintenance	per cent. of total 11
Water charges, hauling, etc., treatment	pence for thousand gallons 6
Coal, including price for handling	per ton 7*s*. 6*d*.
Labour charges as per current rates.	

Comparisons:—Tables I and II, drawn up on the basis of the conditions and programme actually existing in the colliery, compared the following cases: (1) Stubbs-Perry system with direct-current motors, and turbo-flywheel set. (2) Exhaust steam-electric, with steam-engines and mixed-pressure turbo-alternator. (3) Entire power from public supply. (4) Partial public supply with winder driven by turbo-flywheel set.

TABLE I.

Case.	Coal Burnt.		Cost Per Ton Wound.					
	Winder.	Aux. Load.	Winder.	Aux. Load.	Annual Standing Charges.	Water Charges.	Labour Charges	Total.
	Per Cent. Per Ton Wound.	Per Cent. Per Ton Wound.	Pence.	Pence.	Pence.	Pence.	Pence.	Pence.
1	0·475	1·05	0·426	0·946	5·94	0·125	0·767	8·204
2	1·350	0·50	1·215	0·449	5·90	0·163	0·809	8·536

In Case 1 the quantity of coal burnt per kilowatt-hour was 2·314 lbs. In Cases 3 and 4—entire and partial public supply—all conditions were as for Table I, the tariffs being 7*s*. per month per kilovolt-ampere maximum demand, and 0·375*d*. per unit consumed.

TABLE II.

Case.	Cost Per Ton Wound.				
	Winder and Aux. Load.	Annual Standing Charges.	Water Charges.	Labour Charges.	Total.
	Pence.	Pence.	Pence.	Pence.	Pence.
3	8·438	2·67	—	0·441	11·549
4	4·876	4·32	0·040	0·562	9·798

The Authors. The capital costs shown in Table I were approximately equal, as reflected by the standing charges, while Case 1 showed an economy of 0·332*d*. per ton hoisted. The steam-engines in Case 2 were less efficient than the turbines in Case 1. That was to be expected, since the engines were non-condensing and operated on a spasmodic programme due to the winding cycle, whereas the turbine was condensing, and its steam-demand was equalized by means of a fly-wheel. These figures were influenced by the question of load-factor, and the case given in the Tables was particularly favourable to the steam winder, because, owing to the full-load conditions on the winder, a correspondingly large quantity of exhaust steam was available.

The figures given in Table II, with the tariffs quoted, were very uneconomical in comparison with the two cases given in Table I. In Case 4, where the winding was on the Stubbs-Perry system, a very considerable economy in working-costs had been effected. From the foregoing it was obvious that public supply companies must quote very low tariffs in order to become attractive to collieries. Naturally there were numerous pits close to large power supplies where it was economical to purchase power. The elimination of boiler-plant, with its coal-, ash-, and water-handling, repairs and labour, was desirable under certain conditions, and, although the cost of power might be higher than that at which the colliery could generate, that was outbalanced by other considerations. With suitably arranged tariffs, the most economical scheme for a large number of collieries would be to use the supply for their auxiliary load—pumps, fans, haulages, etc.—and to install the winders on the Stubbs-Perry system.

The positions of the control-levers on the platform and their direction of movement for the various operations had received attention from both makers and users of electric winders; standardization in these respects, although unofficial, already practically existed.

The question raised by Mr. Whitmore of helical versus parallel grooving on the rope-treads of drums had come into prominence during the last two years, and although it was generally admitted that parallel grooving gave a better lead for the second layer of rope, this advantage was gained at the expense of the rope in the first layer on the drum-barrel. With parallel grooves each coil of rope in the first layer had, at one part of the circumference, to cross over the crown into the next groove, and it was drawn tightly on to the crown by the tension on the rope. The second layer of rope, passing over these crossings, doubled the compression of the under

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layer on the crowns. Consequently, the under rope suffered in the first place by having to cross, under tension, the crowns between the grooves, and further, by being compressed at those points.

Referring to Mr. Bailey's suggestion that cylindro-conical drums involved greater stresses in the ropes than did parallel drums, the Authors were of the opinion that such was not the case, provided the small rope treads were of ample diameter to suit the type of rope in use.

The rope-angle of 1 in 40 was stated in the Paper as a maximum and in connection with more than one layer of rope; it was usual with such conditions to groove the drum-barrel. On smooth barrels an angle of 1 in 70 to 75 should be arranged.

The distance-piece between the drum-bosses might, at first sight, appear to be of little assistance in relieving the drum-shell of torsional stresses. There were, nevertheless, cases on record where the bolts which secured the drum-barrel had suffered in the absence of a distance-piece, while the trouble ceased after a distance-sleeve was fitted. If the drum-shaft was made exceptionally large, there was no doubt the distance-piece could be dispensed with, although, as suggested by Mr. Whitmore, one of its functions was to maintain the sides of a loose drum at their correct distance apart.

Referring to the question of clutch adjustment for the purpose of mitigating or eliminating rope error, the suggested reduction of the tooth-pitch to 2 inches, would still enable the rope error to amount to 12 inches in the case of a large drum, which was sufficient to be very objectionable; while the sectional area of each tooth would be insufficient to sustain the total loading on the clutch (a condition frequently specified) unless very highly stressed. The adjusting gear in the Paterson clutch admittedly required very accurate machining, but the mechanism contained no parts which could get out of order, and there was no difficulty in making all the parts of ample strength to withstand any stresses to which they could be subjected; further, all thrusts due to driving were contained within the clutch, and none was transmitted to the shaft or drum. The Paterson clutch could be operated from the platform to the same extent as the ordinary clutch. From the point of view of engagement the Paterson clutch possessed the advantage that the rotation of the drum-shaft required with other types was unnecessary.

The practice of bolting and keying bearings to the bedplate possessed advantages, especially from the manufacturers' point of view, over the method of casting them solid, as referred to on p. 117. The former method required less care and time in machining, while adjustments were possible if and when required.

The Authors. The successful results obtained with fine-pitch gearing in marine turbine work gave confidence in the maintained efficiency of this type of gear for driving winding-engines, particularly because the equivalent tooth-contact was larger than could be obtained with a coarse pitch. It would be appreciated that large tooth-contact was of special importance in winders, where the maximum tooth pressure was exerted every time the machine started in either direction. The question of the superiority of coarse- or fine-pitch gearing was very controversial. The contention that fine clearances were necessary was, of course, correct, and it had been found advisable to provide bearings of special design. Those bearings were very simple, they needed no adjustment when once set accurately, and they were less liable to get out of order and to require constant attention than was any other type of bearing. Gravity lubrication was really essential, and, when it was installed, less attention was required than with ring oiling or grease, while there was less risk of neglect of lubrication. No appreciable wear would take place over very long periods in bearings correctly designed and lubricated. It would be generally agreed that pressure or gravity lubrication was preferable to any other method for the drum-bearings, pinion-bearings, and gearing of winders, and there were no objectionable features in a system working with a low pressure.

Mr. Sims had mentioned that the strength of the gear had to be considered when a winder was started. Contact pressure between teeth, however, was a more important consideration, and, if the teeth were so proportioned that the contact pressure was kept within moderate limits, the strength of the teeth would then be such that they would stand without fracture more than forty times the load permitted by considerations of contact pressure. The fine-pitch gears, including the "V.B.B." gears mentioned in the Paper, were cut by the hobbing process to ensure the greatest accuracy, and therefore shrouding could not be done effectively. As explained, however, the tooth strength was so much in excess of the requirements that shrouding was unnecessary.

Deflection in pinion-shafts could be avoided by making the shaft between the teeth and the bearings practically equal in diameter to the pinion at the bottom of the teeth, and by making the bearing span a minimum.

Mr. Sims's reference to the lubricating conditions of rolling and sliding contact was scarcely in accordance with practical experience. It was well known that lubrication was twice as efficient when surfaces rolled over each other as when they slid. The sliding contact of the teeth, instead of drawing in the lubricant as suggested, had the opposite effect of squeezing the oil out; and, in the case of involute teeth with line contact, it caused intense metallic pressure, resulting in pitting, which originated just below the pitch-line, where the whole of the addendum portion of one tooth slid over a small portion of the dedendum of the other. Mr. Burrows mentioned, in connection with "V.B.B." gearing, the limitation of the contact within the arc of recess. This was an important point in favour of this gear; it eliminated the sliding contact in each direction towards the pitch line which occurred with involute teeth, whereby the metal was swept to this line, there to be subjected to intense pressure, which action caused pitting. This trouble, with involute gearing, had been partially overcome, in some cases, by actually reducing the height of the teeth, so as to minimize the sliding action between the gears. The Authors

Regarding the suggestion that the necessity for careful adjustment and maintenance of fine-pitch gearing was a disadvantage in colliery practice, it had already been pointed out that, if such gears were properly equipped and installed, less attention was needed, and the risk of damage resulting from neglect was also less, than with other types of gearing. Up to the present time four winders, totalling 7,000 HP. had been fitted with fine-pitch involute gears. The "V.B.B." gearing (a more recent development) had so far been fitted to two 850-HP. winders only, but the following sets of this gear had been supplied and were working with very satisfactory results.

Ten	sets	120	HP.	each,	driving pumps.
Three	,,	400	,,	,,	,, general machinery.
Five	,,	4000	,,	,,	for ships.

In referring to pressure-lubricated bearings, Mr. Burrows assumed at the outset that the oil-pressure must exceed the bearing pressure in order to obtain an oil-film. It had been found, however, by extensive and thorough research and experiment that, if the bearings were correctly designed with regard to the arrangement and proportions of the oil-inlet ducts and oil clearances to produce the correct wedging action, a film-pressure of 10,000 lbs. per square inch could be obtained with a supply pressure of 5 lbs. per square inch. It was necessary, however, that there should be nothing in the form of oil-grooves, which had the effect of destroying the film. The gravity lubricating-tanks indicated in the Figures were placed 20 feet above the winder-house floor, and that head was ample for the purpose. A serious disadvantage in high-pressure lubricating systems was the difficulty in providing oil-tight packings where the shafts projected through the bearings and gear-case;

The Authors. that trouble was practically absent with a low oil-pressure. If the bearings were fitted with oil-grooves, it was necessary, in order to maintain an oil-film, that the oil-pressure should exceed the bearing pressure. If there were no oil-grooves, the oil-pressure need not exceed 5 lbs. per square inch, provided the bearings were correctly designed. The film-thickness stated in the Paper was correct and had been found to be satisfactory.

The brakes on the winder illustrated in Figs. 1, Plate 4, were of the anchored-post type, applied by a dead weight of 8,000 lbs. They were released and controlled by a compressed-air engine having an air-cylinder 14 inches in diameter and an oil cataract cylinder 8 inches in diameter, both with a stroke of 18 inches. The engine pistons and rods weighed 910 lbs., making the total application weight 8,910 lbs. The leverage between the dead weight and the brake-blocks at the drum-centre was $42\frac{1}{2}$ to 1; therefore, the pressure exerted by each of the four brake-blocks was 94,668 lbs., the total braking pressure being 378,672 lbs. Taking the worst condition, with the full cage unbalanced being lowered slowly and the rope just beginning to wind off the drum-cone, that was, with the cage 870 feet from the pit-bottom, the weight of the cage, tubs, net load, and 2,130 feet of rope was 69,900 lbs. at 13 feet radius. The tangential force required at the 8-foot radius brake-path to hold this weight was therefore 113,587 lbs. Taking the coefficient of friction between blocks and brake-path as 0·3, the radial force required on the brake-path was 378,623 lbs. An air-pressure of 59 lbs. per square inch was required in a 14-inch diameter cylinder to lift a weight of 8,910 lbs. In emergency, pressure was admitted above the engine piston, where the cross-sectional area of the cylinder was reduced to 75·5 square inches, to augment the dead weight. The frequency of operation of the brakes varied with the proficiency of the driver. Some drivers rarely used the brakes in normal working except to stop the cage previous to kepping. The rates of acceleration and retardation mentioned in the discussion were high compared with those given in *Figs. 24* to *26*, which represented standard practice.

In the anchored-post type of brake, owing to the varying pressure along the surface of the blocks, the wear must obviously be less uniform than in the suspended type. Further, if the blocks were not lined with friction material, and were allowed to wear too much, it was necessary to adjust the anchor-brackets in order to prevent the lever positions from being distorted, a duty likely to be neglected owing to inaccessibility. Suspended-post brakes threw off better than the anchored-post type.

In order to give results equal to the Iversen gear, the Whitmore

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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