VOL. XIII FEBRUARY, 1926 No. 2

THE BROWN BOVERI REVIEW

EDITED BY BROWN, BOVERI & COMPANY, LIMITED, BADEN (SWITZERLAND)

CENTOVALLI RAILWAY WHICH CONNECTS LOCARNO AND DOMODOSSOLA (SWITZERLAND-ITALY).
A train at Intragna.

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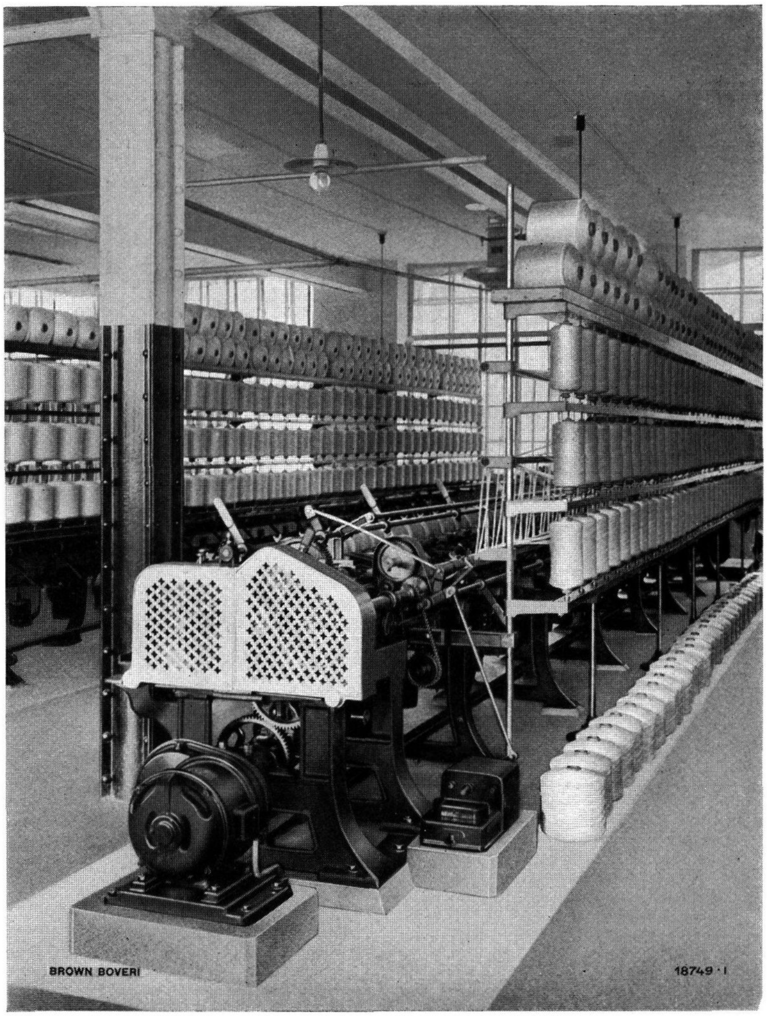
FOR INDUSTRIAL INSTALLATIONS



FILATURES RÉUNIES DE LAINE PEIGNÉE DE SCHAFFHOUSE ET DE DERENDINGEN.
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SPECIAL DRIVES FOR SPINNING AND DOUBLING MACHINES
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THE BROWN BOVERI REVIEW

THE HOUSE JOURNAL OF BROWN, BOVERI & COMPANY, LIMITED, BADEN (SWITZERLAND)

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BROWN BOVERI GEARS.

Decimal index 621. 833.

I. GENERAL.

WITHIN the last few decades, the general tendency of machine design has been to increase the speed of the component units more and more so as to be able to obtain a higher individual output of the machines, together with a saving in weight, cost, and space occupied. The introduction and unprecedented development of the steam turbine rendered increasing the speeds a problem of paramount importance.

De Laval first employed gearing with turbines, which, on account of their speeds ranging from 20,000 to 30,000 r. p. m., could not be directly coupled to other machines. About the same time, Parsons also made a practical turbine with which he used high-speed gearing experimentally. The development of the reaction turbine, in connection with which Brown, Boveri & Co. have always taken a prominent part, subsequently took place in the direction of reducing the speed, and thus dispensing with intermediate gear.

It was only the difficulties presented by the general application of turbines to the propulsion of ships, other than those for high speeds, which emphasised the lack of suitable means of reducing the speed. The conditions arising when direct-current generators are driven by steam turbines are similar, even though the difference between the most favourable speed of each machine is smaller. High-speed compressors, turbo-blowers etc., driven by electric motors also require a speed-changing device in order that each machine may run at its most favourable speed. The development of compressors and blowers direct coupled with steam turbines, led this tendency for increasing speed. As all types of machines do not adapt themselves equally to the augmentation of the speed, means must be sought to utilise the obtainable advantages as far as possible by allowing at least a part of the set, which consists of driving and driven machines, to run at a high speed. Some intermediate means must be used to transmit the power from the prime mover to its drive and at the same time to introduce the necessary difference in speeds. The principal means used are:— electrical, hydraulic, and tooth gearing, because other devices such as belt, rope or chain transmission are either inadequate or introduce serious disadvantages. Toothed gearing has the great advantages of simplicity and higher efficiency than the electrical or hydraulical methods of transmitting power. The introduction of gearing procures many advantages in that it enables standard machines to be employed for the prime mover and driven machine, both of which can consequently be constructed for the speeds to which they are best suited.

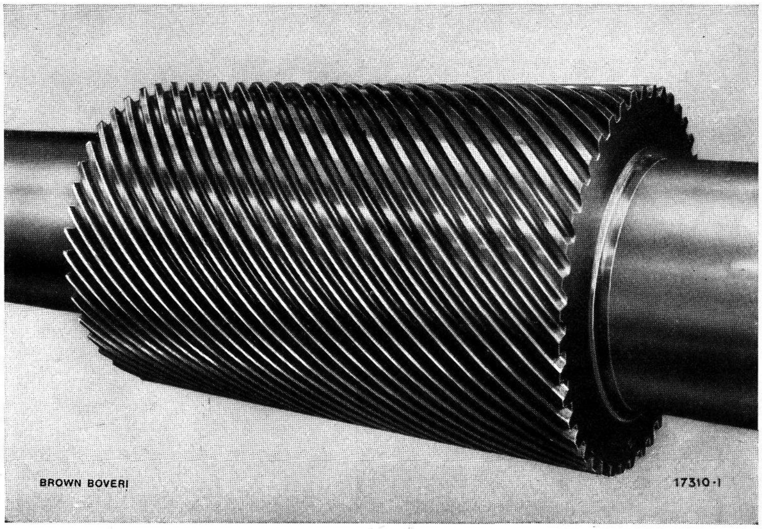


Fig. 1. — Pinion of a Brown Boveri gear.

Helical gears (Fig. 1), the teeth of which have been cut by a generating process on special machines, together with the improvements in the design and methods of manufacture, have gradually made it possible to use toothed gearing for larger powers and higher speeds than formerly.

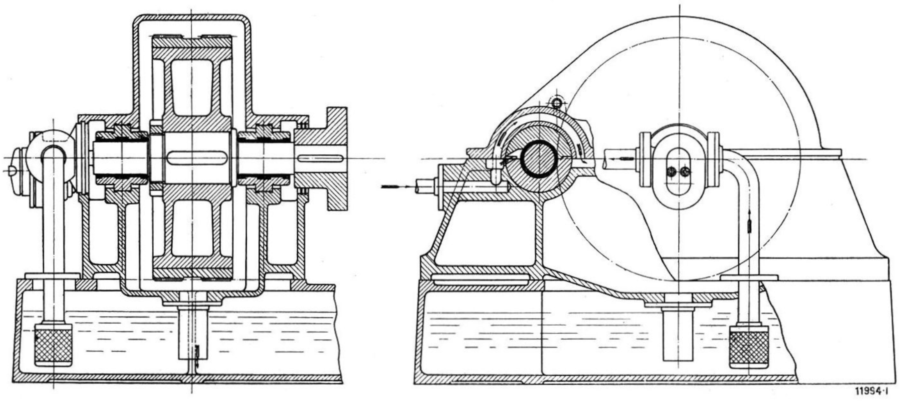


Fig. 2. — Brown Boveri double-helical gear for medium and small outputs.
The base-plate serves as an oil reservoir.

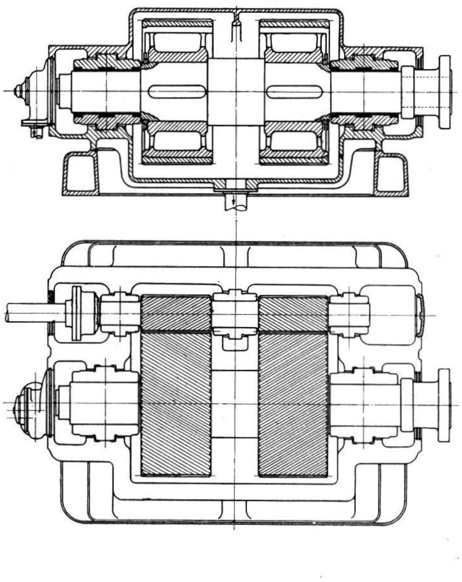
It may be mentioned that Brown, Boveri & Co., due to their activity in the construction of high-speed, rotary machines — e. g., steam turbines, electric generators and motors, turbo-compressors, and blowers etc. — have been continually associated with the development of gears for transmitting large outputs at high speeds, and were among the first firms to manufacture such gears in their own workshops.

II. DESIGN OF BROWN BOVERI GEARS.

The gears constructed by Brown, Boveri & Co. are either single helical or double helical. Both these types have the advantage over spur gears that, on account of their spiral form, teeth are continually in engagement. Furthermore, the duration of the contact between two teeth is longer and the forces occurring only increase gradually from zero to their maximum value and fall back gradually to zero again, whereas with spur gears, the load reaches its maximum value as soon as two teeth are engaged, and then remains almost constant the whole time the contact lasts, until it is suddenly removed when the teeth leave each other.

It follows that helical gears are appreciably quieter and smoother running than spur gears on account of the absence of back-lash, and that their life is therefore substantially longer. For the same reason, they can be allowed to run at much higher peripheral speeds and to transmit much heavier loads. These properties are still further enhanced by employing a comparatively small pitch which ensures that a considerable number of teeth are always meshing.

The axial thrust due to the inclination of the teeth can be equalised by providing a double set of helical teeth having opposite inclinations (Figs. 2 and 3). The longitudinal position of the gear wheel and pinion meshing together is exactly determined. Small variations in the teeth, which cannot be avoided even with the most accurate methods of cutting, render it necessary that part of the gear, preferably the lighter pinion, shall, within limits, be free to move in an axial direction. The coupling between the pinion shaft and the shaft connected



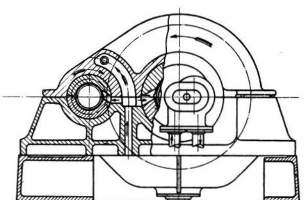


Fig. 3.
Brown Boveri double-helical gear for large outputs.

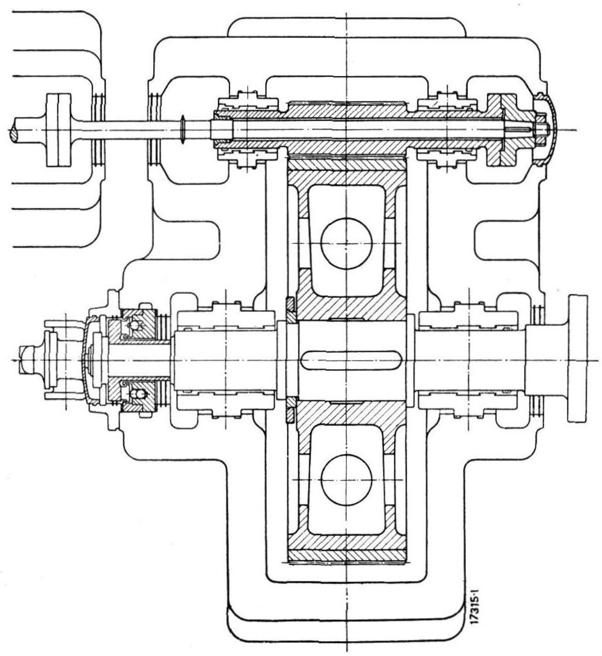


Fig. 4. — Brown Boveri single-helical gear,
fitted with rigid coupling for pinion shaft, with torsion shaft and thrust collar for the gear-wheel shaft.

to it must therefore allow slight longitudinal movement if double-helical gears are employed.

The mutual effect of small inaccuracies in machining, which occasionally come together or are of the same magnitude in the two halves of a double-helical gear, give rise to hunting of the pinion, which, especially for high peripheral speeds, is not without serious consequences. For these reasons, Brown, Boveri & Co. have manufactured single-helical gears almost exclusively for some years past, as with this type the preceding disadvantages are obviated (Figs. 4 and 5). The small unavoidable errors in pitch are corrected by a torsion shaft, generally incorporated with the pinion shaft, which allows a circumferential adjustment. These tangential forces (causing an acceleration or retardation) are resolved into and compensated by an axial displacement with double-helical gears. These forces are entirely missing with single-helical gears, hence such gears run more quietly than those of the herring bone type.

Other advantages in favour of single-helical gears are that they can be machined and fitted more accurately since only one set of teeth is required, and also that the faces of the teeth meshing with one another can adjust themselves more readily to small differences arising either in course of erection or whilst in service.

The axial thrust set up by a single-helical gear may often be used to advantage for balancing that of the machine to which the gear is coupled, — e. g., a steam turbine or compressor, — or the thrust may be compensated by a thrust block or a collar. As the axial thrust is never very large, and modern single-collar thrust bearings are able to bear high specific loads, this problem presents no difficulties. In many cases, and especially when the two machines coupled to the gear must be entirely free from axial thrust, Brown, Boveri & Co. make use of the thrust collar just mentioned. (Figs. 5 and 6.) This collar, which takes the place of an ordinary thrust bearing, is located on the pinion shaft so that its specially-formed thrust face bears against a corresponding face on the rim of the gear wheel, thus equalising the thrust from the teeth directly without transmitting it to one of the two shafts or to the framework of the gearing. Both surfaces which take up the thrust are inclined at a small angle to each other, and each forms a cone, hence a linear contact takes place only in the plane of the two shafts. Within the limits of the overlapping surfaces, a wedge-shaped, pressure zone is formed which converges in the direction of rotation until the line of contact is reached, and then it opens out, as shown in Fig. 6. As generally known, and particularly so for segmental thrust bearings, the oil wedge thus formed between two surfaces inclined to each other is capable of withstanding very high

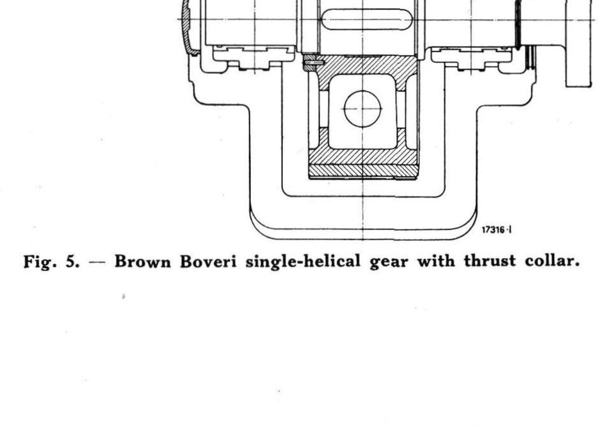


Fig. 5. — Brown Boveri single-helical gear with thrust collar.

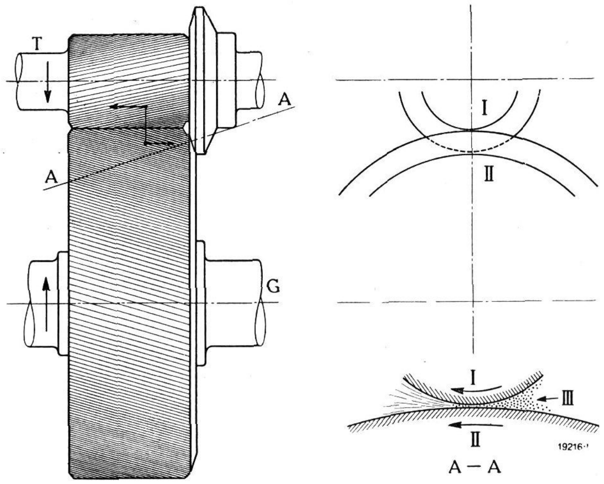


Fig. 6. — Thrust collar for taking the axial thrust of a single-helical gear.

T. Driving shaft. G. Driven shaft.

I. Pinion wheel. II. Gear wheel.

III. Oil.

specific pressures. Thrust collars also have the advantage that the relative speed of the two surfaces is very small, because both surfaces rotate in the same direction. The method of lubricating these surfaces is the most favourable yet conceived, and the design, which has been patented by Brown, Boveri & Co., has been thoroughly proved by special tests and by use on many installations.

Brown Boveri gears are equally well suited either for reducing the speed (as in a geared turbo-generator set), or for increasing the speed (as in a geared motor-compressor set).

Single gears are constructed for transmission ratios up to about 10 to 1 for small and average sizes, and up to about 15 to 1 for large outputs. In exceptional cases, ratios as high as 25 to 1 have been obtained with single gears. Above these values double gears are employed. These can either be formed by two entirely separate gears placed in series, or they can be incorporated in a single casing.

Whenever possible, the gears are arranged to rotate in such a direction that the pressure on the bearings produced by the reaction of the teeth is directed upwards for the pinion shaft and downwards for the wheel shaft. In this way, all liability of unstable conditions being set up at partial loads due to the peripheral forces and the weight of the gear wheel mutually balancing themselves is obviated (Fig.7).

The casing is oil tight and made of cast iron. It is split horizontally at the same level as the centre lines of the shafts. With large gears the bearing shells are constructed separately and are bolted to the upper and lower halves of the casing. The casing is supported on a base-plate or foot plates, the former may be supplied either for the gear alone or for the gear and the machines coupled to it. Cast iron is used for the wheel centres, which in large size wheels are split to avoid setting up strains. The rim and shaft of the gear wheel are made from open-hearth steel. The pinions, except those of large diameters, are forged in one piece with their shafts, special alloy steels being used, as can be seen in the following table.

Special value is laid on the fact that the two parts of a gear meshing together consist of different but well-tried materials.

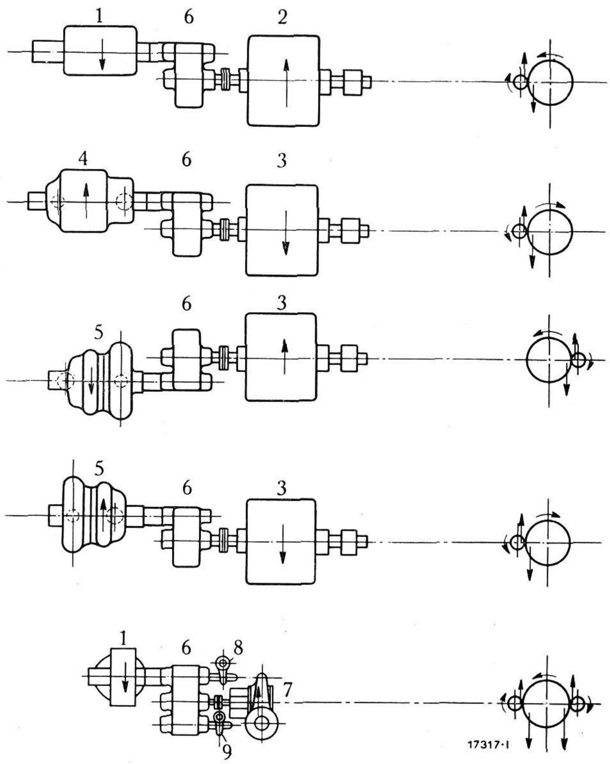


Fig. 7. — Arrangement of a gear with respect to the prime mover and drive.

1. Turbine.
2. Generator.
3. Motor.
4. Compressor.
5. Blower.
6. Gear.
7. Circulating Pump.
8. Pump for water-jet ejector.
9. Condensate pump.

Strength of materials used for pinions and rims:—

Member	Material	Ultimate strength kg/mm²	Yield point kg/mm²	Percentage elongation on a 10 diam.length	Impact figure metre-kg/cm²
Pinion	Low percentage nickel (or chrome-nickel) steel	65—75	40	18	10
Pinion	Special steel	80—85	45—50	12	6
Rim	Open-hearth steel	52—60	30	18	6
Rim	Special steel	65—75	35—40	14	6

On account of the extreme importance which homogeneity of the material has upon the accuracy of manufacture, pinions and rims are invariably subjected to a thorough examination in the firm's material-testing laboratory.

The gear wheels are made with involute teeth produced by hobbing. This profile is the simplest as the hob has rectilinear flanks, and can consequently be manufactured with the greatest accuracy, and it also has the advantage that wheels with any number of teeth may be produced by the same cutter. This tooth form permits also small variations in the distance between the axes of the wheels and pinion without impairing the meshing of the teeth. The generating process used has the advantage that the cutting of the teeth is a continuous process. The tooth profile is finished in two or three passes, the last cut being of a very fine nature in order to reduce the strains on the machine as much as possible and also to minimise the wear of the tool so as to obtain the greatest possible accurracy of the tooth form. The hobs must be of excellent quality; this can only be assured by using well-tried materials after a thorough examination. The time required for machining the surface of a large gear wheel with broad teeth may be as much as 350 hours, and, during this time, the process should be uninterrupted if possible, or at least the finishing cut should be continuous. For this reason precision of the teeth, of their form as well as of the pitch, is absolutely essential if smooth and noiseless running as well as long life are to be secured. A constant supervision of the process is necessary as well as a minute examination of the finished product. This examination is of the utmost severity, as errors of pitch or of tooth form may be only a hundredth, and in places only a thousandth, of a millimetre. The apparatus necessary for taking measurements is too large a subject to be considered within the scope of the present article.

The module of helical gear wheels is generally very small, usually between 2π and 7π for the normal pitch, the lower value being applicable to small wheels running at high peripheral speeds. For small pinion diameters, the lowest number of teeth per wheel should not be less than 25 in order to avoid undercutting the roots of the teeth. This can also be avoided by moving the pitch circle, thus giving unequal addenda and dedenda to the teeth on the pinion and wheel. Irrespective of this, the addendum is otherwise designed to be approximately 0·8—1·0 times the module, and the root to be 1·0—1·2 times the module, with as great a rounding off at the root as possible.

To obtain the polished tooth flanks, which are produced gradually by service, before leaving the workshops, wheels which will have to mesh can be "run-in", during which process they are run at a small peripheral speed under low load, and lubricated with oil mixed with graphite or some other similar material. Some firms grind the flanks of their spur wheels in order to obtain a fine finish, but this process has not yet been adopted to helical wheels because, with the grinding equipment at present in use, the work would have to be set up on a dividing head, and thus a large part of the advantage of the hobbing method would be lost.

Another occurrence, known as "pitting", often occurs after gears have been put into service. It is usually found near the pitch circle, i.e., at the place where only rolling friction takes place, and consists of small funnel-shaped holes varying in size from that of a needle point to a hole of about a millimetre in diameter, and, in exceptional cases, even larger than this. Divergent opinions are expressed as to the initial cause of the pitting; there is, however, a general tendency to attribute it to recurring local stresses which cause the material to be squeezed out of the flanks of the teeth according to the cones of pressure. As stated, pitting usually forms soon after the gearing has been put into service, but the process normally comes to a standstill after

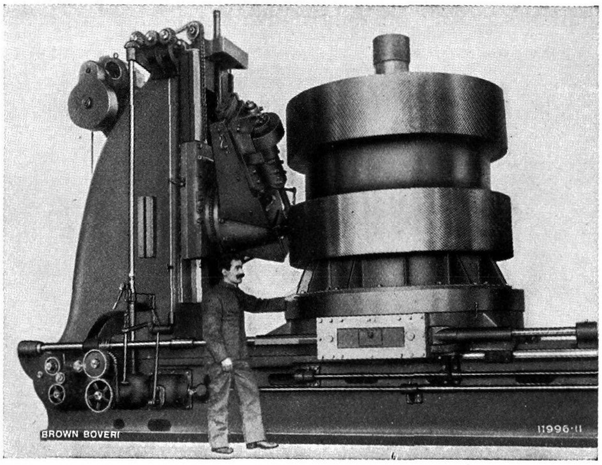


Fig. 8. — Hobbing a gear wheel for marine use.
Output transmitted per pinion 6000 H. P.; per gear, comprising two pinions, 12,000 H. P., speeds 2400/360 r. p. m.

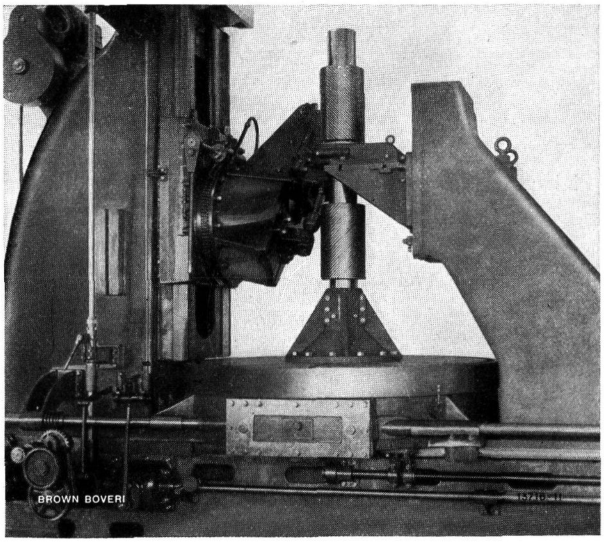


Fig. 9. — Hobbing a pinion for marine use.
Output transmitted per pinion 6000 H. P.; per gear, comprising two pinions, 12,000 H. P., speeds 2400/360 r. p. m.

a short time, and as no disadvantages arise, no importance is placed upon the occurrence.

Figs. 8 to 10 show special hobbing machines on which wheels up to 4·5 metres in diameter and with a breadth of tooth suitable for this dimension, can be cut. Assembled gearing is shown in Figs. 11 and 12. The latest design for the gears is shown in Fig. 12; it will be seen that all the spur wheels have single-helical teeth. As already mentioned, a claw coupling or a pin coupling, permitting longitudinal movement is used to connect the pinion and its shaft with double-helical gears. Single-helical gears have the advantage, especially appreciated for high-speeds, that a rigid coupling may almost invariably be used between the pinion and the shaft.

Forced lubrication is employed for the bearings and teeth. Oil under pressure is normally supplied by an oil pump driven from the gear-wheel shaft. No special oil pump is necessary however, if the gear has to work with machines already possessing an oil pump of sufficient capacity to supply the requisite quantity of oil, or if an independent forced-lubrication system is available. Conversely, the bearings of other machines can be connected to the pressure oil system of the gear to which they are coupled. Such an arrangement offers many advantages with high-speed machines driven through speed-increasing gears. If a separate lubricating pump is provided, oil is drawn from a reservoir, and forced through an oil cooler and, if necessary, through a filter, from which it is distributed to the various points where it is required, and finally returns to the reservoir. With the smallest size of gear, the lower portion of the gear casing, or the base-plate, serves as an oil container. A small auxiliary pump, hand operated, or driven by steam or by a motor, is provided to ensure the lubrication of the bearings

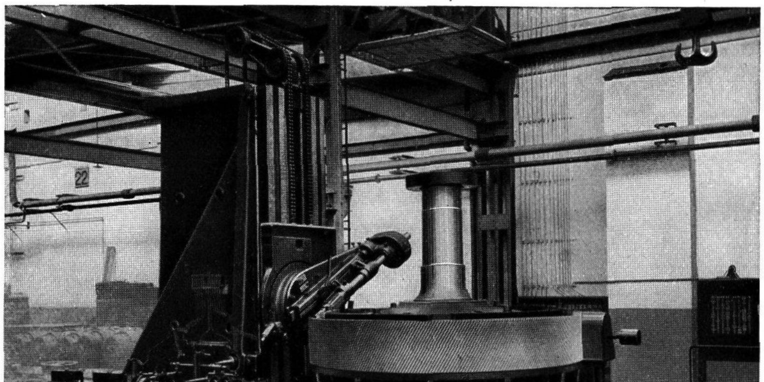


Fig. 10. — Hobbing machine with gear wheel for a merchant vessel.

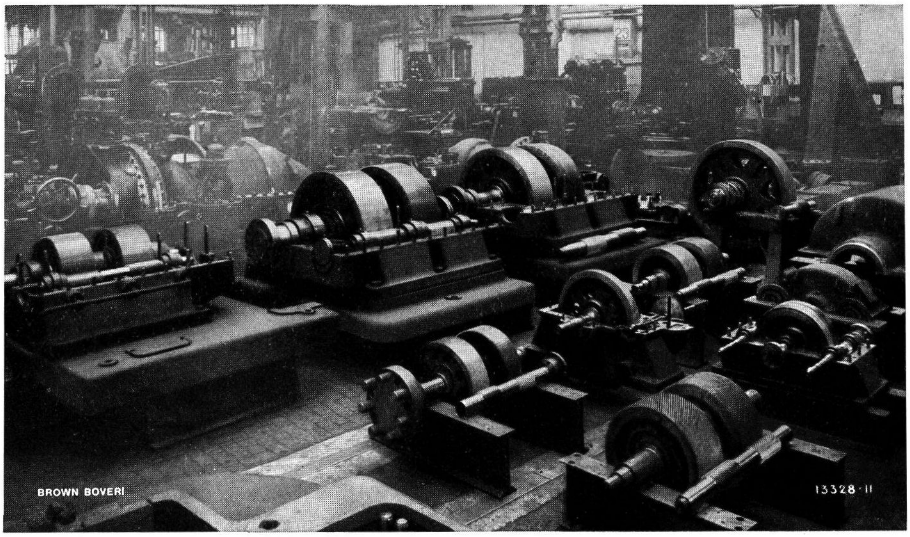


Fig. 11. — View of Brown Boveri gears in the workshops.
Two hobbing machines can be seen in the background.

the casing which serves as an oil reservoir (Fig. 13). The driving motor and gear are either mounted on the same base-plate as shown in many illustrations, or the gearing and motor are provided with independent stools.

An unsaponifiable mineral oil which is free from either acid or asphalt is used as lubricant. The most suitable oil for lubricating the teeth of the gear is one with a high viscosity,

and teeth before starting up or when the set is being stopped. The teeth of small sets, transmitting less than 500 H. P., and having peripherial speeds up to about 12 m/sec, can be lubricated directly by immersing the spur wheel in an oil bath. In such instances, ring-lubricated bearings or ball bearings are used, and the oil pump, as well as the forced lubrication, are omitted.

If radiation through the upper part of the casing is not sufficient to disperse the heat generated by friction, a cooling coil is fitted to the lower half of

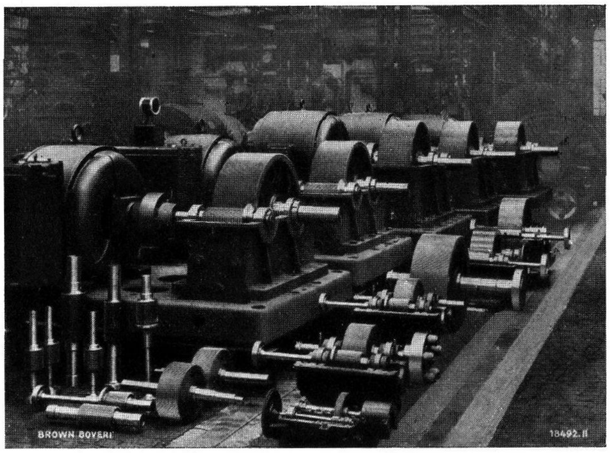


Fig. 12. — Single-helical gear wheels and pinions assembled.

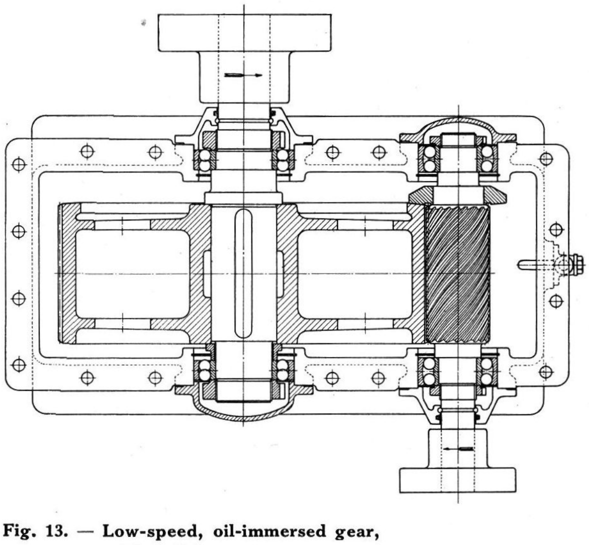
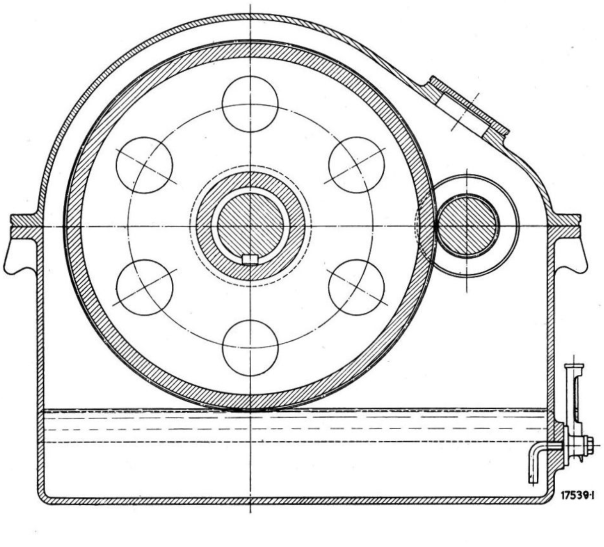


Fig. 13. — Low-speed, oil-immersed gear, with ball bearings.



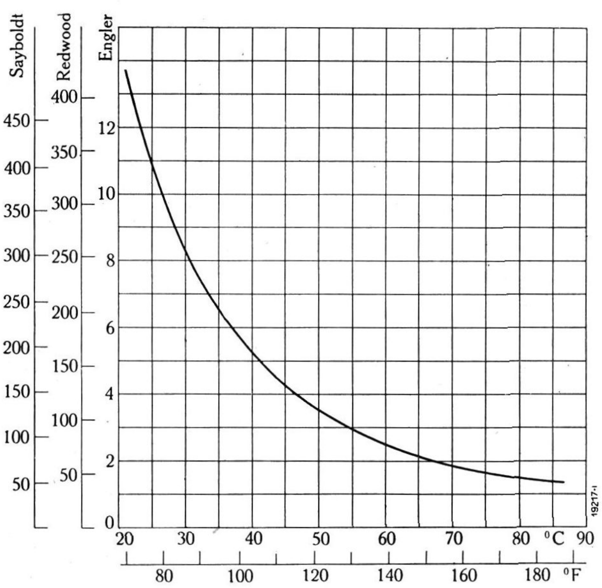


Fig. 14. — **Curve showing the dependence of the viscosity of a lubricating oil on the temperature.**

but the bearings, especially those of high-speed gears, require an oil with a lower viscosity on account of the heat generated by friction losses. It is necessary to find a middle course between the alternatives, as, in practice, it is hardly possible to separate the two lubricating systems. It is general to use an oil having a viscosity of 4—7 degrees on the Engler scale at 50^0 C.

A single lubricating system can be used to satisfy all requirements, if the oil is cooled to different extents for the teeth and the bearings, because the oil is less viscous as the temperature is increased (Fig. 14). If, for example, the oil for the teeth is cooled to 35^0 C, the oil will have a viscosity of $6{\cdot}5^0$ E, as shown by the curve in Fig. 14, while the same oil, used for the bearings, would only be cooled to 50^0 C, the viscosity then being $3{\cdot}5^0$ E. In certain cases, it is sufficient to use the oil for the bearings at the temperature of the mixture.

Whenever intermediate speed-changing devices are employed, considerable importance is usually attached to their efficiency. Compared with all known appliances for this purpose, toothed gearing is the most efficient, figures as high as 98—99 % being attained for large outputs on full load. The efficiency of a set of gearing may be determined as shown in Fig. 15, where two sets are coupled between a motor and a generator, the individual efficiencies of the two machines being exactly known; hence the electrical power lost is equal to the sum of the losses in the drive. This method is extremely accurate, as only the losses are measured, and these are only a small percentage of the total power transmitted.

As the greater part of the losses, especially at high speeds, is due to bearing friction, and these losses do not vary appreciably at constant speed, the efficiency falls off somewhat at fractional loads, for example at half load it amounts to 97—97·5 %. The efficiency is practically the same for a large range of transmission ratios, provided that the lubrication is properly carried out and that the set is suitable for the purpose.

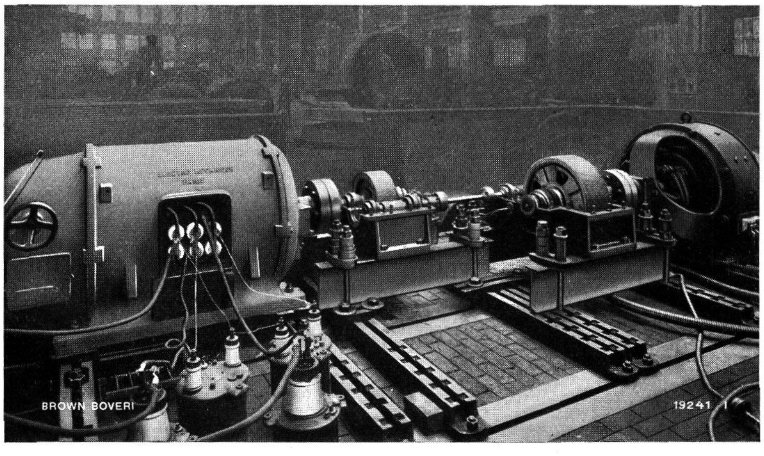


Fig. 15. — **Two gears connected in opposition for determining their efficiency.**

The very small loss in the gearing, amounting to about 1—2 % of the output transmitted, is usually negligible if compared with the advantages obtained. The fact that both of the machines coupled to the gear can be designed to run at the most suitable speed, not only compensates this loss, but renders possible an improvement in the overall efficiency of the set.

(MS 361) *J. Baasch. (J. R. L.)*

(To be continued.)

THE BROWN BOVERI FREE-FALL SAFETY BRAKE FOR WINDING ENGINES.

Decimal index 621. 826 : 622. 6.

ONE of the most important components of a winding machine is its safety brake, the function of which is to bring the machine to rest automatically in the shortest time when any disturbance in the normal working arises, e. g., overwinding of the cage, excessive winding speed, overloading, etc. This it must be able to do without unduly straining either the machine or the rope, without giving rise to rope slip in the case of Koepe pulleys, and without causing any injuries to men who may be travelling in the cage.

As shown in section I, the usual drop-weight brakes only partially fulfil these requirements, with the result that certain safety brakes, instead of protecting the winding service from danger, are actually themselves a source of danger to the plant.

I. THE SAFETY BRAKE AS GENERALLY EMPLOYED.

With these brakes the drop-weight is rigidly fixed to the brake rod. If the brake weight were allowed to fall absolutely freely it is true that the cages would be stopped within a very short distance, but the braking shocks ensuing would give rise to stresses of inadmissible magnitude. As soon as the brake blocks touch the rim, the velocity of the weight is reduced almost instantaneously to zero, and the kinetic energy, which has been accumulated during the unresisted falling of the brake weight, produces, in the brake rod, a tensile force which may rise to a very high value owing to the low extensibility of the rod (and therefore on account of the very small distance in which the kinetic energy must be given up). In the following instant, the contraction of the rod causes the brake weight to be suddenly jerked back. This cycle repeats itself until the kinetic energy of the weight has been completely absorbed. To obtain a true picture of exactly what occurs in practice, exhaustive braking tests were made on a Koepe-pulley winding machine, designed for winding a net load of 4000 kg from a depth of 560 m at a speed of about 9 m/sec. Pressure curves were traced by the aid of a pressure indicator. Figs. 1—4 show the brake-pressure diagrams obtained when the safety brake was released with the machine running at full speed. In diagrams 1—3 the usual type of drop-weight brake was employed, whereas Fig. 4 was obtained from a Brown Boveri free-fall safety brake.

The pressure diagram, Fig. 1, shows clearly the heavy shock produced when the brake blocks meet the brake rim, and the subsequent oscillations which occur in the braking pressure before the steady final value is reached. The magnitude of the oscillations in the braking pressure will be still greater in practice, when, owing to wear, the blocks have a longer travel than 3 mm as used in the experiments. In this case the distance through which the weight drops is correspondingly greater, and hence, also the amount of kinetic energy accumulated during its fall.

Braking, accompanied by shocks and oscillations of this character, is particularly dangerous, as there is the possibility of overstraining the machine and winding rope; the liability of injury to men travelling in the cage; and also, with Koepe-pulley machines, rope slip may be set up owing to the sudden rise in the braking pressure exceeding the friction of the rope.

With this type of safety brake, therefore, it becomes necessary to provide a damped falling of the brake weight, which reduces its speed at the moment when the brake blocks make contact and thereby diminishes the braking shock. Figs. 2 and 3 show the braking-pressure diagrams for various degrees of damping. It is true that by adopting this expedient it is possible to reduce the excess stresses to within permissible limits, but it brings in its train the serious

Figs. 1—3. Braking-pressure diagrams for a safety brake with rigidly-suspended drop-weight.

Braking-pressure diagram with air release fully opened.
Duration of braking 7·3 sec.
Braking distance 32·8 m.

Braking-pressure diagram with air release only 1/5th. open.
Duration of braking 9·0 sec.
Braking distance 40·5 m.
Normal final pressure attained in about 3 sec.

Braking-pressure diagram with air release only 1/20th. open.
Duration of braking 16·8 sec.
Braking distance 75·5 m.
Normal final pressure attained in about 14 sec.

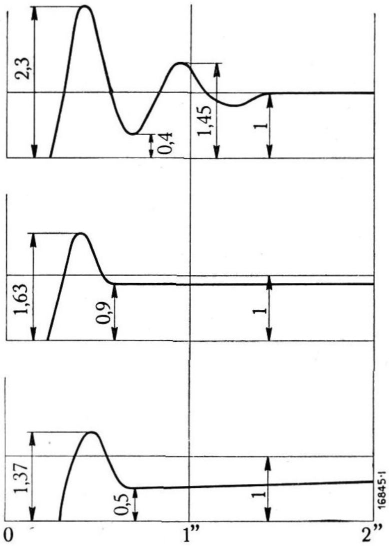
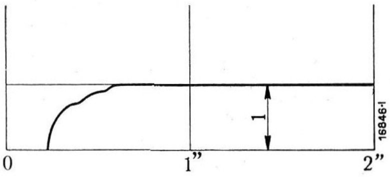


Fig. 4. — Braking-pressure diagram for the Brown Boveri free-fall safety brake.
Duration of braking 7·5 sec.
Braking distance 33·8 m.
1. Normal final pressure.



disadvantage that, on account of the damping, the distance through which the cages continue to travel subsequent to the releasing of the brake weight is considerably prolonged. This increased travel may easily be the cause of an accident which might have been avoided had the machine been brought more rapidly to rest.

II. THE BROWN BOVERI FREE-FALL SAFETY BRAKE.

The principal feature of this brake lies in the fact that the brake weight falls absolutely unresisted up to the moment that the brake blocks meet the brake rim. It is not until then that the motion of the weight is retarded by means of a special damping device in such a way that the force exerted by the weight is transmitted without shock to the brake blocks.

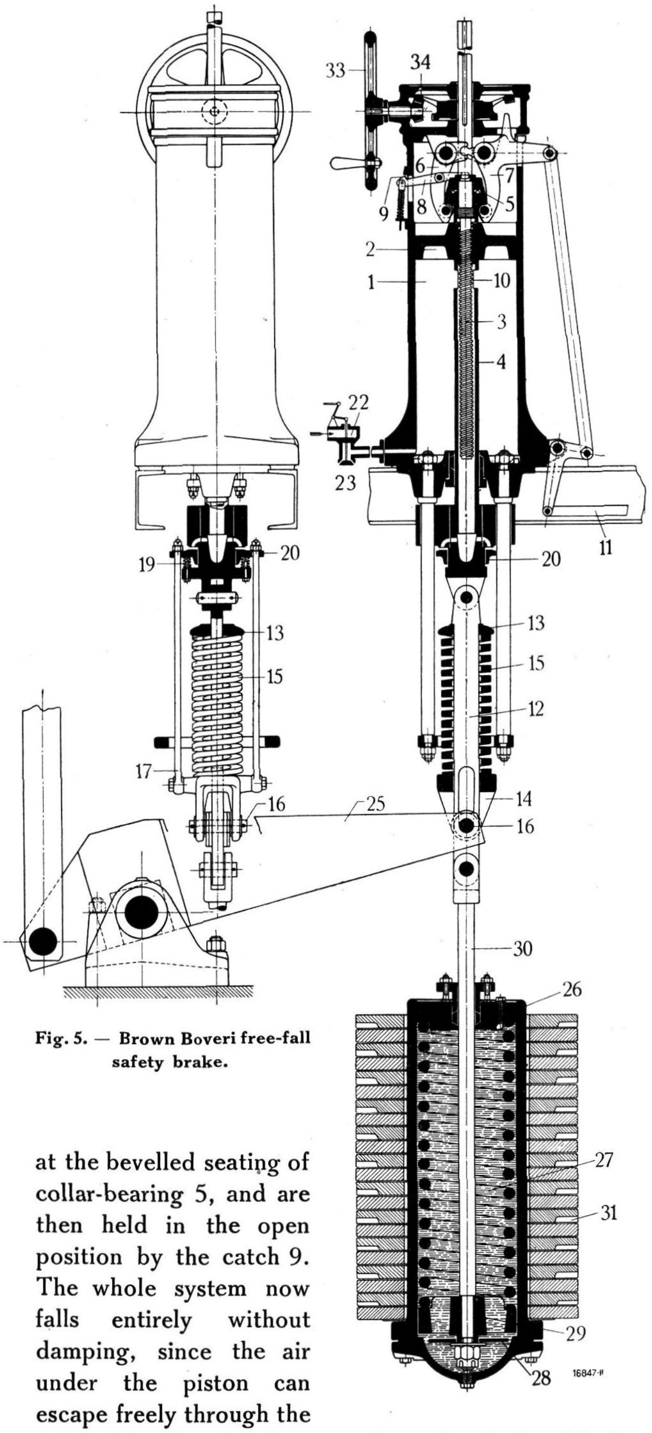


Fig. 5. — Brown Boveri free-fall safety brake.

Construction. The Brown Boveri free-fall safety brake is shown in Fig. 5. The brake-weight supporting rod 12 is rigidly connected to the hollow piston rod 4, and this again with the piston 2. The latter is made in the form of a nut in which the lifting-gear spindle 3 can turn. The spindle is supported by the collar bearing 5, bevelled on its under surface, which rests upon two roller-tipped pawls 6 and 7. When the brake is lifted, these are held in position by the electro-magnet mounted on the operating stand. The brake weight 31 is supported upon the weight buffer 26—30 which is attached to the brake-weight supporting rod 12. The connection of the latter to the brake lever is effected through the compression spring 15, the upper end of which presses against the brake-weight supporting rod, whilst its lower end rests on the brake lever. The interior of the cylinder is in communication with the atmosphere through the holes 10 in the hollow piston rod, through the valve 20, and also by means of the throttled opening of valve 23. In the upper portion of the safety brake, a hand-operated lifting device for the drop-weight is also mounted; this is described later.

The brake-weight buffer consists of the closed cylinder 26, the piston rod 30, the piston 29 provided with valve 28, and a spring which is interposed between the piston and the upper portion of the cylinder. The cylinder is filled with oil.

Operation. The brake weight is released if, for example, the current circuit of the electro-magnet is interrupted by the functioning of one of the safety devices (overwind switch, centrifugal switch, etc.). As soon as the magnet ceases to exercise its force on the rod 11, the pawls 6 and 7 are separated by the pressure due to the drop weight acting at the bevelled seating of collar-bearing 5, and are then held in the open position by the catch 9. The whole system now falls entirely without damping, since the air under the piston can escape freely through the valve 20. At the instant when the brake blocks meet the brake drum, the brake lever 25 is brought to rest, but the brake weight and the supporting rod continue to move still further, compressing the spring 15. As a result of this the rods 17, which are connected to the brake lever, are brought to rest in space and thereby release the springs 19, causing valve 20 to close. As soon as this has occurred, a very small further motion of the piston gives rise to a rapidly increasing air pressure on its underside, which checks the falling motion and takes up both the static force due to drop weight and the kinetic force resulting from its deceleration. While the compressed air is escaping through the throttled valve 23, the brake-weight pressure, now partially carried upon the air cushion, is transferred without shock, through the spring 15, to the brake lever. Not until the air pressure under the piston has again become atmospheric does the pressure in the spring correspond to the full force of the brake weight, and only then is the full braking pressure finally attained

The gradual bringing of the brake weight to rest, after the brake blocks have come up against the brake rim, is effected by means of the weight buffer which absorbs the kinetic energy of the brake weight in compressing the spring 27. The re-extension of this spring is opposed by the action of a liquid cataract in such a way that it can take place without reacting on the braking pressure. Thus no vibratory shocks can be imposed upon the brake lever. The work stored up in the spring 27 is dissipated in overcoming the resistance of eddy currents set up in the oil in its passage through the very-nearly-closed valve 28 as the cylinder 26 moves upwards.

Fig. 4 shows the pressure diagram given by the Brown Boveri free-fall safety brake. The machine was brought to rest just as quickly as in the case of Fig. 1. It will be seen, however, that with this brake, shocks in the braking pressure are completely eliminated, the brake-weight pressure being transferred to the brake blocks in such a way that the braking pressure grows quite gradually from zero up to its full normal value.

The conical form of the under surface of the collar bearing 5 has the advantage that the brake weight tends to release itself, so that it falls immediately the electro-magnet ceases to exert its holding force. In this way, extremely rapid and absolutely reliable operation is assured, as, on account of the conical under surface of the collar bearing, the pawls cannot remain jammed. The safety brake can be released mechanically by hand at any time, through simply withdrawing the magnet armature by means of a lever provided for the purpose.

The electro-magnet, which holds the pawls in position when the brake is lifted, may also be mounted directly on the safety brake itself, as shown in Fig. 6.

Lifting the brake weight. The resetting of the brake may be accomplished either by compressed air or by hand.

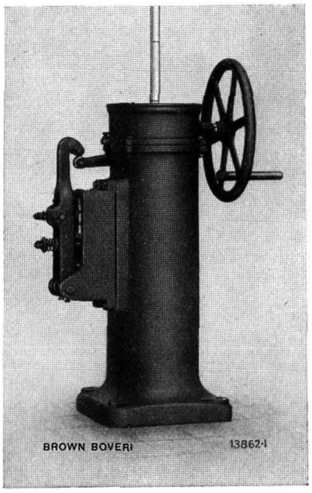


Fig. 6. — Free-fall safety brake with built-on supporting magnet and releasing lever.

Lifting by compressed air is effected by opening the valve 22 and allowing compressed air to operate on the piston. The whole system is lifted until the supporting collar strikes the lever 8, whereby the catch 9 is released, allowing the pawls to grip again under the collar bearing. On closing the valve 22, the compressed air in the cylinder escapes through the valves 20 and 23. If, therefore, the brake be again released, the weight can once more fall quite freely, i.e., without being damped.

In raising the brake weight by hand, the spindle 3 is first screwed up by means of the hand wheel 33 and the pair of bevel-wheels 34, the piston, piston rod and brake weight still remaining in the lower position. Assuming that the cause of release has in the meantime been removed, the magnet will again actuate the pawls 6 and 7, causing the latter to support the spindle in its upper position. Thus, on turning the handwheel in the opposite direction, the piston and brake weight can be screwed up the spindle (now held in the upper position) until the brake is fully lifted.

The raising of the brake weight can only take place if the holding magnet is again excited, i.e., if the cause of release has been removed. Further, automatic release of the brake weight can always take place, even during the relifting of the brake, which represents an important advance on brakes of the usual types.

The device for lifting the brake weight by hand offers the advantage that, by means of the safety brake alone, loads can be lowered at any desired speed. The braking force can be regulated to any desired value on account of the flexible connection between the brake-weight supporting rod and the brake lever, so that it is possible to lower a load gently and with absolute certainty.

(MS 352) *H. Winter. (A. H.)*

Centovalli Railway; train composed of motor coach and two trailers.

FURTHER COMMENTS ON THE GROWTH OF CAST IRON AT HIGH TEMPERATURES.

Decimal Index 620. 112. 1 : 620.17.

THE damage caused by the growth of cast iron with particular reference to steam turbines has been the subject of much recent investigation, particularly in Germany. The growth, liable to occur in grey cast iron may give rise to serious disturbances. The reliability of operation in steam-power stations is of the greatest importance, it is, therefore, essential that the cause of this phenomenon should be ascertained and suitable precautionary measures adopted. An article, published in the Brown Boveri Review 1925, No. 10, discussed this subject and critically reviewed the report of the investigations which had been carried out up to date.

As made clear in the previous article, it is approaching the problem from the wrong side to imagine that specifying the alloy will prevent the growth. The present article takes into consideration the results of the latest chemical researches upon the structure of the different substances. It was formerly assumed that any given substance had well defined properties by which it could be classified, but the latest researches upon colloids have shown that between the forms of a substance previously referred to as allotropic modifications there exist gradual differences which are essentially dependant upon the conditions of formation. Hence, the idea of allotropic modifications can be advantageously replaced by that of phases, and it has been suggested that a reaction product is the result of a certain development which cannot be understood if considered with regard to the chemical or physical structure only. In this connection, Kohlschütter[1] has referred to a sort of physiology of the structure to supplement the morphology described. If the product of any physical or chemical action be considered entirely as a phase then it is essential to take into consideration the circumstances influencing the formation of the substance, in order to determine their importance with regard to the final product. The circumstances prevailing when the substance was produced provoke a large series of new products which take part in further reactions. If this idea is

[1] Die Naturwissenschaften, 1923, No. 11, p. 865.

developed, it is clear that these factors must have an important bearing upon natural phenomena.

Amongst the most important factors in a reaction affecting the allotropism, the place of formation is preponderant and explains a considerable number of the so called topo-chemical reactions.

These brief theoretical remarks will be illustrated by an example of great importance affecting the composition and properties of cast iron, namely graphitic iron. Until recently, carbon was believed to exist in at least two forms other than diamonds, viz., amorphous carbon and graphite. Both of these two classes were again subdivided into different groups. As a criterion, graphite may be characterised by its ability to enter into reactions, and by its behaviour in the presence of certain reagents, with which it can form various oxides and graphitic acid. By these means two subdivisions of graphite were obtained:— graphite proper and graphitite. Detailed research has shown that graphite, as generally defined, is by no means a uniform substance. It has likewise been found that amorphous carbon is able to form different secondary modifications derived from the same initial state, and also that it does not essentially differ from graphite, but only the structure of the carbon changes. The structure in graphite is mostly floccular whereas in amorphous carbon it is tumulary. Graphite is a typical allotropic form of carbon and is almost entirely dependent on the place in which it was produced. It is, all the more like graphite as the surface is smoother, it passes over to amorphous carbon as the surface gets rougher. Cast iron provides an excellent example of the various phases of carbon, namely graphite and cementite either of which can be obtained experimentally according to the conditions prevailing when the cast iron is produced. It is emphasised that the so called graphite in cast iron has no strictly definite structure. The different forms occurring in cast iron have different capacities for entering into reactions; the cementite is very easily oxidised, but the graphite is only affected by powerful oxidising agents. A considerable increase in the strength characterises a tumulary structure of the cast iron; whereas the normal tensile strength of grey cast iron is about 20 kg/mm^2, for instance, that of cementite is about twice this value. One of the problems met with when producing grey cast iron consists in controlling the separation of the carbon so that cementite is formed, without it being necessary to produce the same by subsequent heat treatment. The research of Goerens on the temperatures of melting and solidification of both cementite and graphite eutectoids have shown that this is possible in principle. Consequently, a great diversity in the nature of cast iron can occur, according to the circumstances in which the iron is formed. The conditions become considerably more complicated when it is considered that the carbon in the cast iron, which is described as graphite, can originate in several ways, e. g., by crumbling of the graphite eutectoid or of the carbide, and also through the mutual influence of the different alloyed constituents on the process of disintegration. For example under certain circumstances, silicon may come into consideration as a structure forming factor. The phase of the graphite has a considerable effect upon the other properties of the cast iron. The phase in which the fundamental materials are found is also just as important. The latest researches of Schüz[1] on graphite eutectoids have shown that irons which have a high silicon content can exhibit very good qualities, especially that they can be very stable. Schüz has applied for a patent for the manufacture of cast iron with 3·0—3·5 % silicon, having a tensile strength of about 36 kg/mm^2. The chief advantage of such an iron is the manner in which the structure withstands the effect of heat, and even after having been heated to between 800 and 850° C, the metal remains ductile and its tensile strength is not reduced. This example merely serves to illustrate what different influences certain constituents in an alloy can cause under definite conditions of formation. The electro or pearlitic cast iron, produced in the Brown Boveri Foundry, has very good mechanical properties, and above all, it has a very close structure.[2] In general the laminated structure of the pearlitic cast iron can be discerned when the specimen has been magnified 500 diameters, this fact shows the extremely fine division of the structure. This cast iron, obtained by special manufacturing process, is remarkable for its extraordinary stability.

This short discussion should result in drawing attention to another method of regarding the problem of the growth of cast iron. In particular, it is impossible to arrive at a satisfactory solution by merely specifying the composition of the iron without taking into account the possible action of the constituents as factors influencing the formation of the structure, and with the resultant effects they have on the properties of the final product. These two facts are of importance and are entirely separate from the conditions of casting and their influence on the finished product.

[1] Stahl und Eisen, 1925, No. 45, page 144.

[2] Die Giesserei, 1923, No. 10, page 287.

(MS 352) *Dr. H. Stäger. (J. R. L.)*

NOTES.

Arrangements for removing tar from turbo gas-exhausters.

Decimal index 621. 63.

IT is a well-known fact that Brown Boveri turbo-blowers for gases containing tar, such as are used in coke oven installations are fitted with drain connections at the lowest point of each stage for the purpose of leading off the tar, naphthalene and water deposited in the blower by the centrifugal action. To prevent loss of gas, water seals are provided as shown in Fig. 1.

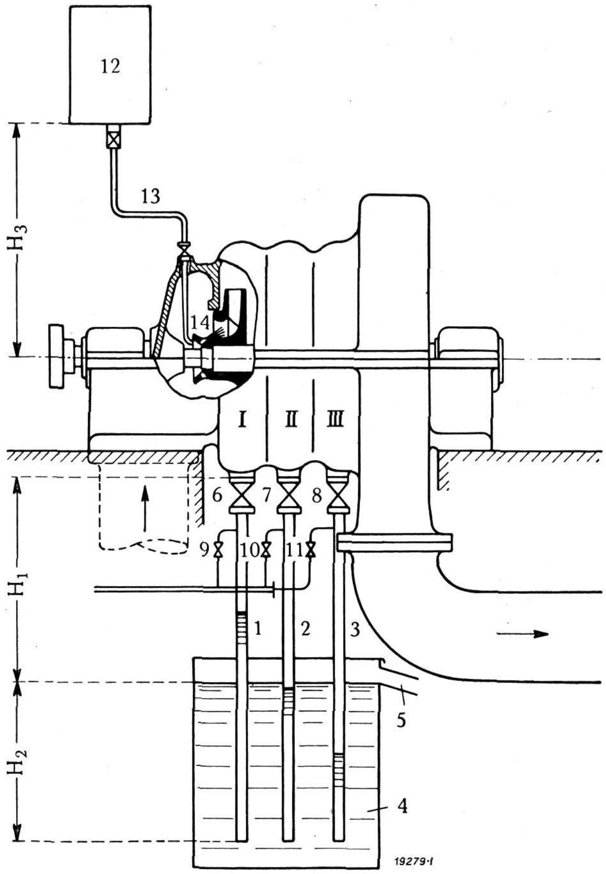


Fig. 1. — Leading off the tar through water seals.

I, II, III. Stages of blower.
1, 2, 3. Tar drains.
4. Water container.
5. Tar drain.
6, 7. 8. Shut-off valves.
9, 10, 11. Steam connections.
12. Elevated tank for anthracene oil.
13. Oil pipe.
14. End-nut for atomising the oil.

The arrangement of the syphon pipes and their accessories, which as a rule are not supplied by Messrs. Brown, Boveri & Co., often leaves something to be desired, and, moreover, the quiet running of the blower is largely dependent upon the arrangement of these pipes. It, therefore, appears advisable to re-state the principles which must be followed when designing these mains and to supplement this information by reference to the latest improvements.

The following points must be observed:—

1. The tar drains 1, 2, 3 (Fig. 1) must be vertical or have an inclination of at least 45°.
2. The height H_1 must not be less than the highest vacuum occurring during service, and the depth of *immersion* H_2 must exceed the maximum gas pressure. When determining H_1 for a given case the possibility of throttling on the suction side must be considered. Generally H_1 may be between two and three times the normal suction head *without throttling*.
3. The tar pipes must be provided with shut-off cocks 6, 7, 8; these should be located as high as possible and are usually open while the blower is working.
4. Steam connections 9, 10, 11, must be arranged directly *after the shut off valves*, so that when the latter are closed the tar drains can be thoroughly cleaned by blowing through with steam.

To maintain the tar in a sufficiently fluid condition recent practice favours the injection of anthracene oil into the suction side of the blower. The anthracene oil may be contained in the elevated tank 12 and led to the blower through the pipe 13. To ensure that the oil is uniformly distributed, it is passed over, 14, the specially formed end-nut of the impeller, where it is converted into a fine spray. The injection of oil should take place at regular intervals during working, and particularly just before shutting down when it should be performed thoroughly.

These measures fulfil their purpose better than, for example, the steam-heated syphon traps shown in Fig. 2. The steam heating causes the lighter oils to distill, and the residue, consisting mainly of pitch, simply solidifies in the pipes. For this reason, steam should only be used in the manner shown by Fig. 1, i.e., simply as a medium for cleaning the pipes by scavenging.

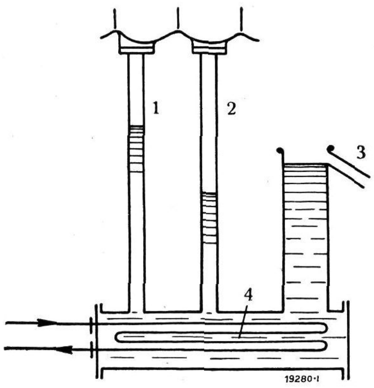


Fig. 2. — Steam-heated syphon trap.

1, 2. Tar drains.
3. Tar overflow.
4. Steam heating.

For the rest, reference should be made to the instructions for erection and operation supplied by Brown, Boveri & Co. with all turbo-exhausters.

(MS 368) *A. Baumann. (J. R. L.)*

The Lima Tramways.

Decimal index 621. 331. 34 (85).

THE very extensive tramway system of the Peruvian capital are under the management of the Lima, Light, Power & Tramway Company, and comprises both urban and suburban lines. The streets in Lima are mostly very

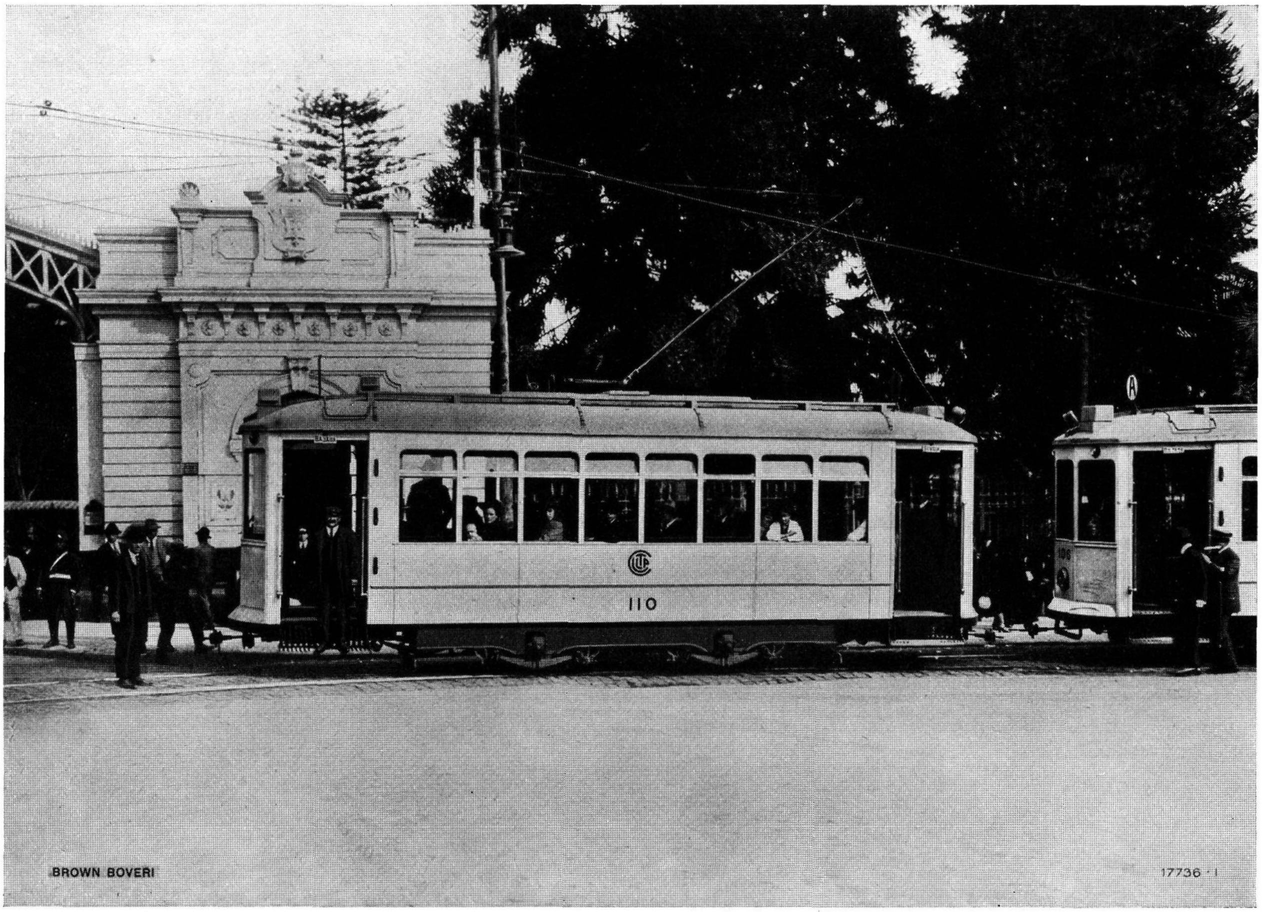


Fig. 1. — Lima Tramways, one of the two-axle motor coaches.

narrow and for this reason only one line can be installed in a street. Trams can, therefore, proceed only in one direction, going one way and returning by another street.

Three suburban lines connect Lima on one side with its port of Callao, 15·3 km distant, and on the other side with the important townships Chorrilos (13·8 km) and Magdalena (7 km). The first two lines consist of double tracks, while the third is only a single track, although a double line is being installed.

All the lines are standard gauge, the distance between the inner edges of the rail flanges being 1435 m. In the town the gradients vary between 1·0% and 1·5%; for 150 metres there is one gradient of 5·0%. On the suburban lines the maximum gradient is 1·8%. The curves are rather sharp, particularly in Lima, where the smallest radius is only 10 metres.

Direct current is used for driving the cars and is supplied by four substations. The station in Lima supplies the town section and the line to Magdalena. Two substations one in La Legna and one in Callao, supply the line to Callao, and the fourth, in Miraflores, supplies the line to Chorrillos. All these substations are supplied with three-phase current at 10,000 V, 60 cycles, and convert it to direct current at 600 V by rotary converter sets.

The traffic on the greater part of the system is very heavy. On the line to Callao, for example, trains consisting of a motor coach and trailer coaches follow each other at intervals of 5 minutes. On level stretches they travel at 55 km/hour. When the tramways passed into the hands of the Lima, Light, Power and Tramway Company, it was found necessary, in view of the great increase of traffic during the last two years, to provide 54 new motor coaches with two axles and 20 with four axles.

The following are the main features of these coaches:—

1. Two-axle motor coaches, Fig. 1.

Length of body (including platforms)	9100 mm
Width of body	2430 mm
Wheel base	2300 mm
Diameter of wheels	850 mm
Seating accommodation	32
Standing accommodation	26
Total accommodation	58
Weight of coach, empty	12·85 tons

Each coach is equipped with two direct-current totally enclosed series motors, Type GTM 11, having nose suspension, and with a one-hour rating of 42 B. H. P. at 560 r. p. m. and 500 V. The reduction-gear ratio is 5·54 : 1. The motors have four poles and the motor and axle bearings are plain bearings with pad lubrication.

The *controllers* are of the usual tramway type. They have five series, four parallel and six braking positions.

In the last parallel position the field coils are shunted by a resistance which weakens the motor field by 20% and increases the speed of the coach. The resistances are of the grid type and are mounted under the floor. The coaches are provided with compressed-air brakes, pressure being supplied by a motor compressor set for 2·2 kW, 650 V, 1000 r. p. m., delivering 230 litres of air per minute at a pressure of 4·5 to 6 atmospheres. This set is controlled by a pressure switch which closes at 4·5 atmospheres and opens at 6 atmospheres.

2. Four-axle coaches.

Length of body (including platforms)	14600 mm
Width of body	2570 mm
Distance between bogie pivots . .	7900 mm
Wheel base of one bogie	1400 mm
Seating accommodation	56
Standing accommodation	22
Total accommodation	78
Weight of coach empty	25·95 tons

Each coach is equipped with four direct-current self-ventilated series-motors, Type GDTM 28/4·24 having nose suspension, and with a one-hour rating of 53 B. H. P. at 810 r. p. m., and a continuous rating of 37 B. H. P. at 970 r. p. m. and 600 V. The reduction gear ratio of these motors is 4·4:1. As with the two-axle coaches, these motors have four poles and are provided with the same type of bearings and lubrications. Similarly the controllers possess the same features as those of the previous type of coach, the only difference being that it has six series positions, five parallel positions, the last of which serves to shunt the motors, and seven braking positions. The shunt weakens the field 20% as before. The resistances are of the grid type and mounted under the floor of the coach. The compressed air for the brake is supplied on each vehicle by a motor compressor set of 4·5 kW, 750 r. p. m., 600 V giving 590 litres of air per minute at a pressure of 4·5 to 6 atmospheres. The starting and stopping of the compressor is effected in the same way as for the two-axle coaches.

The mechanical part of the new two and four-axle motor coaches was made by the firm of Ernesto Breda in Sesto-San Giovanni (Italy) while the electrical equipment was supplied by Brown, Boveri & Co., Baden, who also assembled the coaches.

Most of the new motor coaches have already been in service since 1924 and very satisfactory results are being obtained.

(MS 374) *M. Hiertzeler. (J. R. L.)*

The Seebach Substation of the Swiss Federal Railways.

Decimal index 621. 312. 63 : 621. 331. 32 (49.4).

THE Seebach Substation, which first came into service on August 1, 1925, when the line between Zurich and Winterthur was electrified, is situated about 1200 m to the north-east of Oerlikon Railway Station. This substation, which will ultimately supply the railway lines between

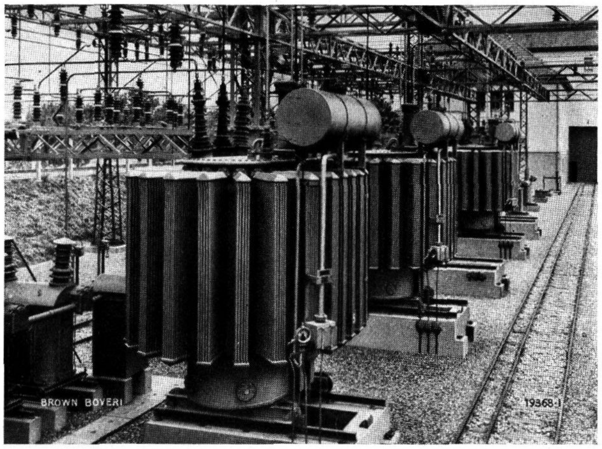
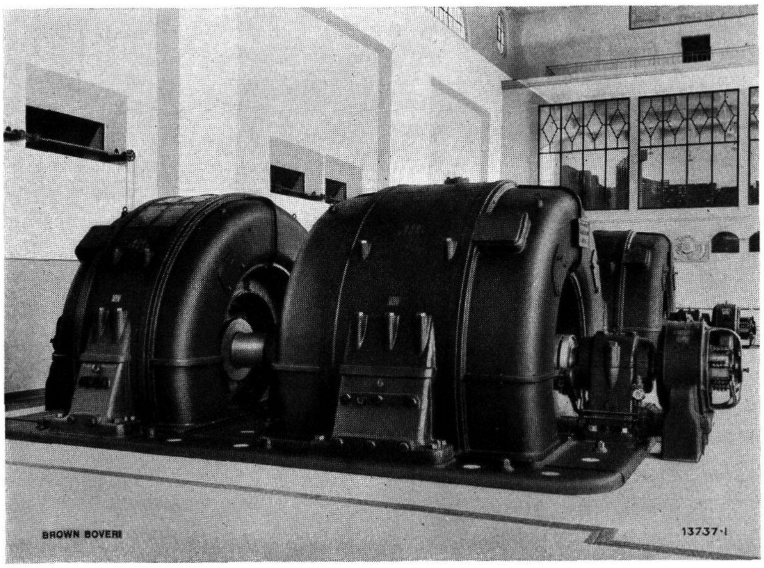


Fig. 1. — The Seebach Substation of the Swiss Federal Railways.
General view of the three single-phase, oil-immersed, outdoor transformers with natural cooling, 3000 kVA, 60/15 kV, 16²/₃ cycles.

Zurich-Winterthur, Bülach, Uster and Wettingen as well as Oerlikon Station, is an outdoor installation and is equipped with 3000-kVA outdoor transformers with a pressure ratio of 60/15 kV at 16²/₃ cycles. It was equipped by Brown, Boveri & Co., Baden, and the transformers resemble those already delivered for the outdoor stations at Puidoux, Emmenbrücke, Brugg and Bussigny. Only three transformers are supplied provisionally, but the station has been designed for six transformers.

(MS 371) *C. Giudici. (J. R. L.)*



Muhleberg Central Station of the Bernese Power Supply Co., Berne.
Two frequency and phase converters, each of 5000 kVA, changing three-phase current at 16,000—17,000 V, and 50 cycles to single-phase current at 15,000—16,000 V, and 16²/₃ cycles; 500 r. p. m., for supplying the contact wires of the Swiss Federal Railways.

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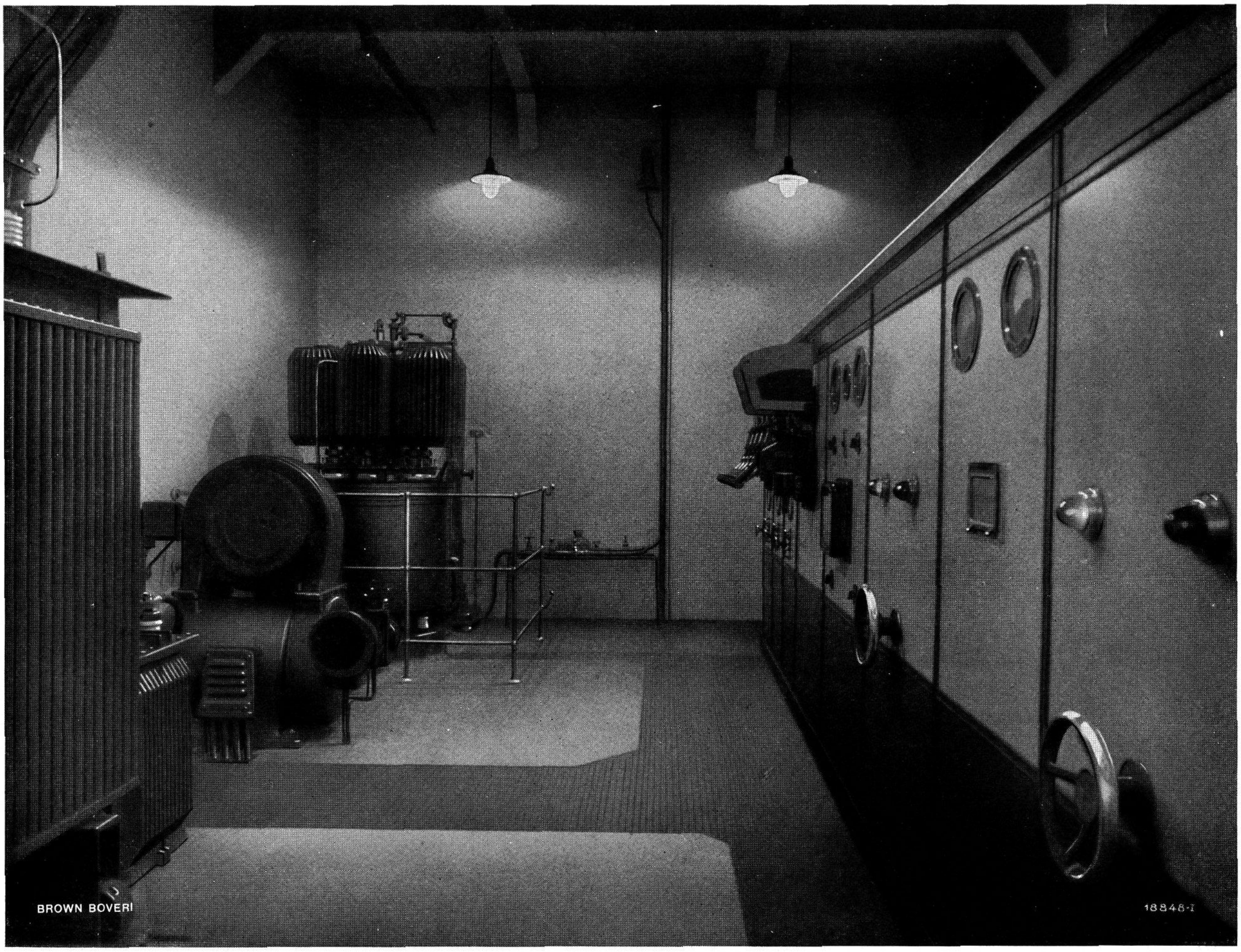
ELECTRICAL EQUIPMENT FOR POWER AND SUBSTATIONS



COMPAGNIE CENTRALE D'ÉNERGIE ÉLECTRIQUE, ROUEN, FRANCE.

Completely underground converter station with Brown Boveri mercury arc rectifier, 300 kW, 270 V (three wire installation 2×135 V). Automatic control of the pressure by induction regulators. Supplied with three-phase current 5000 V, 50 cycles.