

THE BROWN BOVERI REVIEW



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THE BROWN BOVERI REVIEW

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BEHAVIOUR OF DOVETAILED POLE JOINT UNDER CONDITIONS OF EXCESS STRESS.

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The over-speed test on the large magnet-wheel of a slow-speed alternator for drive by a Kaplan turbine recently gave an opportunity of investigating the stresses in the spider rim and in particular in the dovetailed pole joints. A number of observations made on this occasion are recorded hereafter.

THE purpose of testing machines under excess speed conditions is on the one hand to "run in" the whole assembly of spider and wound poles¹ and on the other to prove the strength, faultlessness, and perfect fit of all parts, so as to ensure 100% reliability at the lower service speed. In contradistinction to the magnet wheels of alternators driven by steam or combustion turbines, Diesel engines, or water turbines of the impulse type, where overspeed tests of no more than 25—40% above the service speed are specified, this figure is pushed up to 280% for the magnet wheels of alternators for drive by Kaplan turbines. This relatively high overspeed is based on the fact that the revolutions of a Kaplan turbine may attain such a value should the entire load suddenly be thrown off with the maximum head of water through the turbine and should the turbine governor simultaneously fail in the most unfavourable blade position, thus resulting in racing of the turbine. Moreover, due to the uncertainty of the calculated speed resulting from these complex conditions a big safety margin is added. Since this case only arises where all of the possible unfavourable

factors occur at one and the same time the probability of its ever coming to pass is extremely slight. In point of fact, to the author's knowledge there has been no single case in recent years where a slow speed turbo-set attained such a high speed in service. This notwithstanding, the magnet wheel has to be designed to withstand the enormous centrifugal forces occurring solely during the over-speed test. The exceptionally large over-speed/service-speed ratio involves very high stresses during the test run, the mechanical stresses being about eight times greater than under normal operating conditions. What is more, these stresses are not evenly distributed throughout the cross-section, as a glance at the design of the magnet wheel in Fig. 1 will show. During rotation the maximum centrifugal stressing of the wheel occurs in the spider rim, being composed of the radial tension in the individual poles and the hoop tension in the rim. The most critical spots lie in the neck of the dovetailed groove in the spider rim, which give a pronounced notched effect, so that stress concentrations are to be anticipated at these points. Whereas the stresses in the rim cross-section are relatively low and can be computed by conventional methods, the peak stresses cannot be accurately calculated and evaluated beforehand. They may, however, be so great that the yield point of the material is attained or exceeded locally. If these stress concentrations were to be specified not to exceed a

¹ See Brown Boveri Rev., 1944, p. 371.

certain percentage of the yield point of the material, the dimensions involved would be far greater than could be economically justified, and the construction of such machines would even be impossible.

The question now is whether and to what extent such peak stresses may exceed the normal admissible stresses in this case and how the material behaves at the endangered points. Since the centrifugal forces are fundamentally of a static nature — the alternating magnetic forces acting on the pole only represent 0.1—0.3% of the centrifugal forces — the corresponding conditions can be arrived at by means of

A. PHOTO-ELASTIC INVESTIGATIONS.

These investigations were carried out on a specially made celluloid model representing a sector of a twentieth part of the circumference of the magnet wheel, i. e. comprising just one pole and the associated part of the spider rim. The linear reduction scale was 2.5 : 1 and the axial thickness of the plane model in the region of the dovetail 7.5 mm. This model was fitted in a steel frame which allowed the uniformly distributed tensile forces to be arranged in the plane of observation. A view of this model is shown in Fig. 2, and it will be noticed that particular attention was paid to uniform

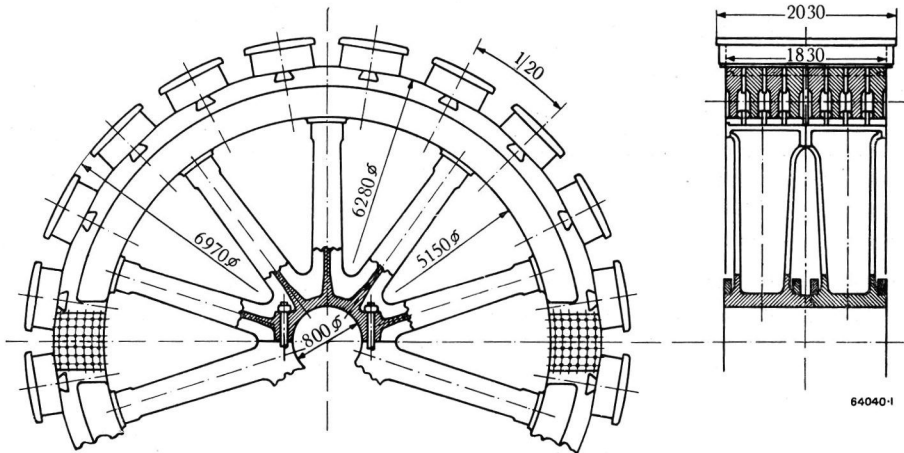


Fig. 1. — Dimensions and construction of magnet wheel investigated.

simple static load tests and the resultant stresses referred to the yield point of the material. It might further be pointed out that the conditions are entirely different when the ratio of the alternating stresses to the total stress is higher, since the notches, i. e. the concentrations of stress, then influence the admissible stress in a fundamentally different manner and to a much greater extent.

The more important dimensions of the magnet wheel investigated are shown in Fig. 1. The service speed is 100 r.p.m., the overspeed test revolutions 280 r.p.m., the weight of a complete pole 3500 kg, the tensile force on the pole at the overspeed test revolutions 1,065,000 kg.

The investigations carried out can be split up under the following three headings:—

- A. Photo-elastic investigation of dovetailed joint.
- B. Mechanical investigation of dovetailed joint on steel model.
- C. Measurements made during overspeed test on magnet wheel.

distribution of the forces. For the sake of simplicity the radial forces acting on the inside of the wheel section were substituted for the massed centrifugal forces on the sector of the wheel. The photo-elastic tests were entrusted to the Swiss Federal Materials Testing Institute at Zurich and the following results are taken from the report of this authority.

The principle and technique of photo-elastic tests will not be gone into here. It might, however, be recalled that changes in colour or intensity of the interference bands visible in polarized light in a mechanically stressed celluloid plate are a direct indication of the direction of the principal stresses and the difference between the principal stresses at every point, as well as of the contour stresses on edges which are subjected to no forces.

The form of the pole investigated is shown in Fig. 3, in which the dimensions of the actual magnet wheel are inserted. As already stated the celluloid model was 0.4 times smaller. This pole was subjected to various stresses, which are shown diagrammatically in Fig. 4. The same load was employed for all of the

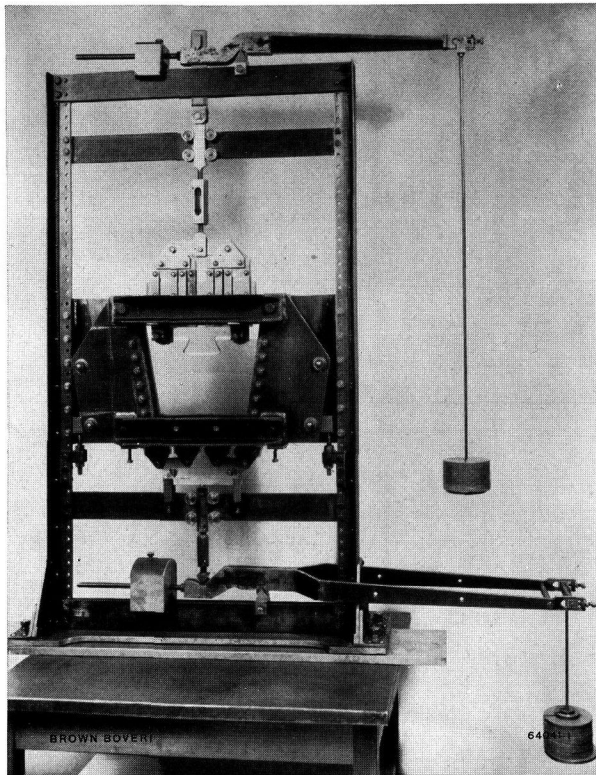


Fig. 2. — View of celluloid model fitted in the loading device for the photo-elastic test.

tests and was selected to give pure elastic stressing of the celluloid model. The loads on the magnet wheel at the overspeed revolutions (given in brackets) are obtained by multiplication by the similarity or model factor. In the case of the stress measurements the values measured on the model were likewise converted to the stresses on the wheel at the overspeed revolutions by means of the model conversion factor, and inserted

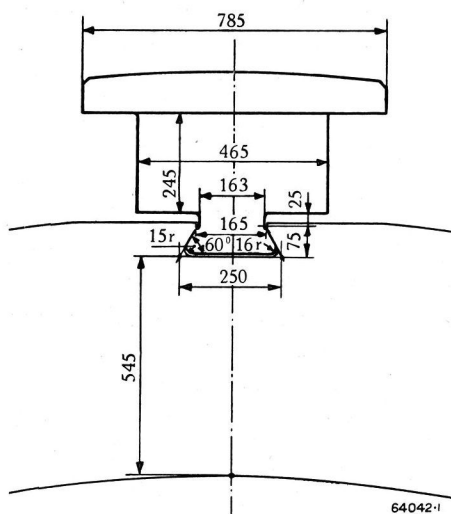


Fig. 3. — Type of pole joint investigated (the dimensions given must be multiplied by 0.4 for the photo-elastic model).

in the diagrams. The directions and differences of the stresses were determined only on the spider rim, since due to the hoop tension this is much more highly stressed than the pole.

Figs. 5, 6, and 7 give the lines of the principal stresses for the three loading cases shown in Fig. 4. At every point they give the direction of the mutually perpendicular principal stresses σ_1 and σ_2 . The magnitude of the stresses at any given point cannot be determined from the lines, while their density is chosen absolutely arbitrarily and is no indication of the local stressing. Nevertheless, the three figures prove that

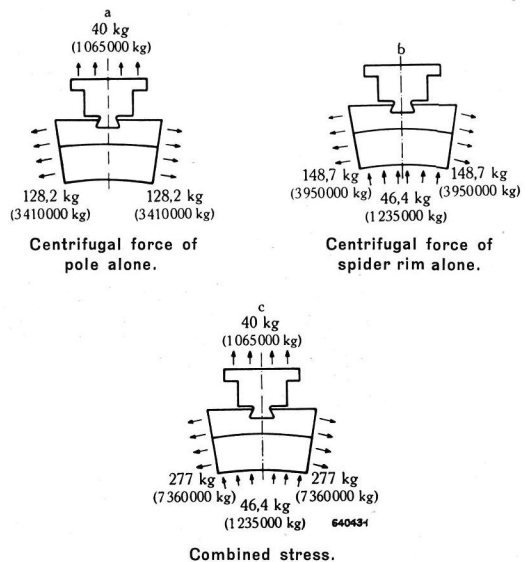


Fig. 4. — Load diagram for photo-elastic measurements.

(The figures in brackets refer to the loads on the full-size magnet wheel.)

the directions of stress are greatly deformed by the dovetailed grooves. When the spider rim is subjected exclusively to the tangential hoop tension, i. e. without pole tension (Fig. 6), the neck of the dovetailed groove is virtually unstressed, and the notch effect and in consequence the stress gradient are thus much less pronounced than for the case of pole tension in Fig. 5. As a result, it is found that the stress characteristic for combined loads in Fig. 7 differs qualitatively very little from that for the case of pure tensile load in Fig. 5, i. e. both diagrams are virtually the same.

The differences between the principal stresses were only determined for the most unfavourable case of combined stress as shown in Fig. 4c. This difference between the principal stresses gives the maximum shear stress $\tau_{\max} = \frac{\sigma_1 - \sigma_2}{2}$ kg/mm². This stress was determined point by point in a right-angle line system

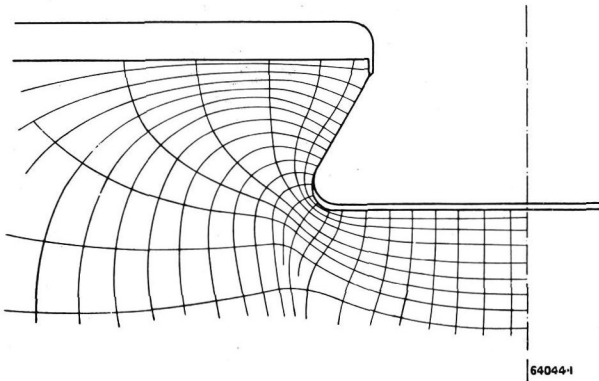


Fig. 5. — Principal stress lines for the load resulting from the centrifugal force of the pole corresponding to Fig. 4 a. (Determined from photo-elastic tests.)

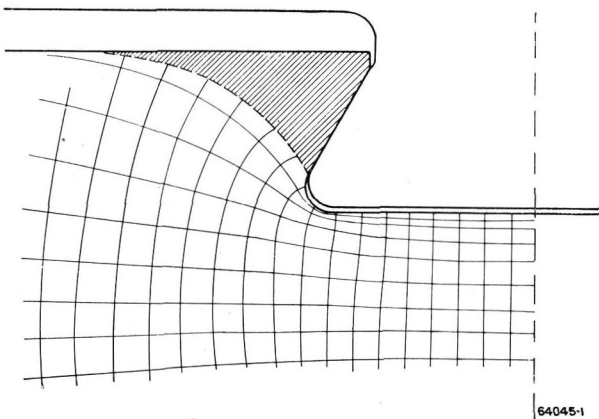


Fig. 6. — Principal stress lines for the load resulting from the centrifugal force of the spider rim corresponding to Fig. 4 b. (Determined from photo-elastic tests.)

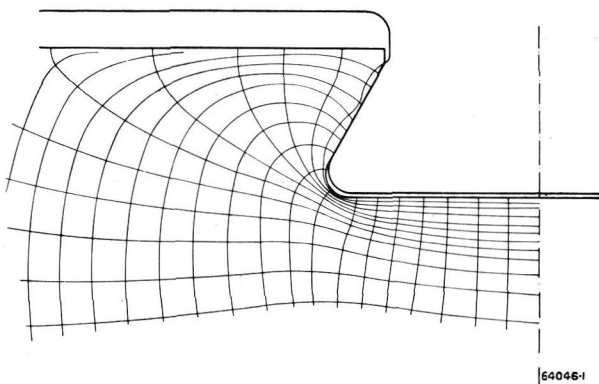


Fig. 7. — Principal stress lines for the combined load corresponding to Fig. 4 c. (Determined from photo-elastic tests.)

for one half of the groove in the spider rim, the results being given in Fig. 8 in the form of lines of equal shear stress. This illustration shows that the highest stresses occur in the rounded off part of the corner

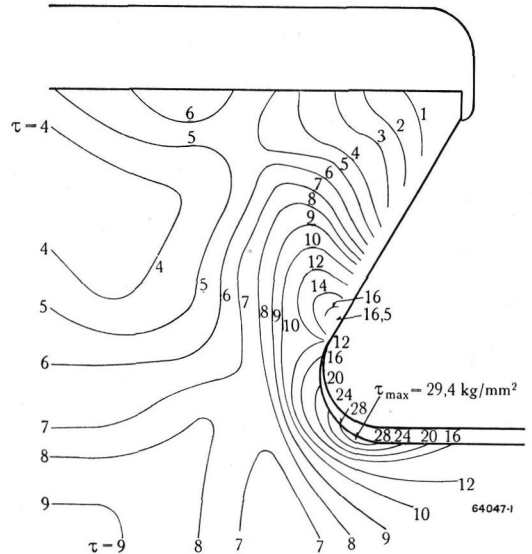


Fig. 8. — Lines of constant maximum shear stress ($\tau_{max} = \frac{\sigma_1 - \sigma_2}{2}$) around the dovetail groove of the wheel for the combined stress at the over-speed revolutions. (Determined from photo-elastic tests.)

of the groove and then after a drop to a minimum again attain a maximum at the top at about 60—64 degrees to the horizontal. A point worthy of note is the rapidity of the drop from the highest peak on the rounded off corner towards the interior; only 6 mm from the edge the shear stress concentration has dropped from 29.4 to 16 kg/mm², i. e. to about half. In contradistinction to the curves of the principal stresses these lines only give an idea of the magnitude, but not of the direction of the stresses.

It must further be stated that the internal stresses usually inherent in photo-elastic models were not eliminated here. These occur chiefly on the edge through the deformation of the surface and edges during machining, as well as due to ageing or subsequent hardening processes in the synthetic material employed. They can attain quite considerable values, but were not determined here.

Around the contour of the model where no transmission of forces perpendicular to it is possible, one principal stress runs parallel to the contour, while the second principal stress vanishes; from the difference between the principal stresses, therefore, the principal tensile stresses in the corner can be determined. These contour stresses are plotted along the limiting line of the rim groove in Fig. 9. The peak stress of 58.8 kg/mm² occurs approximately on the bisection of the angle. Since heavy forces are transmitted by the straight flanks of the groove against which the pole bears, the contour

stresses for this part of the groove outline cannot be determined by this method; the same applies to Fig. 8.

The stresses on the magnet wheel derived from the photo-elastic measurements will only be correct when the wheel material behaves just as elastically as the celluloid during the tests on the model up to the highest stress concentration, i. e. when the strain is always proportional to the stress. If, however, at certain points the elastic range, i. e. the elastic limit of the steel, is attained or exceeded, this assumption no longer applies. The question then is whether greater strain may not occur which under certain conditions might lead to permanent deformation or even fissures in the material. We shall see from the following section what actually happens in practice.

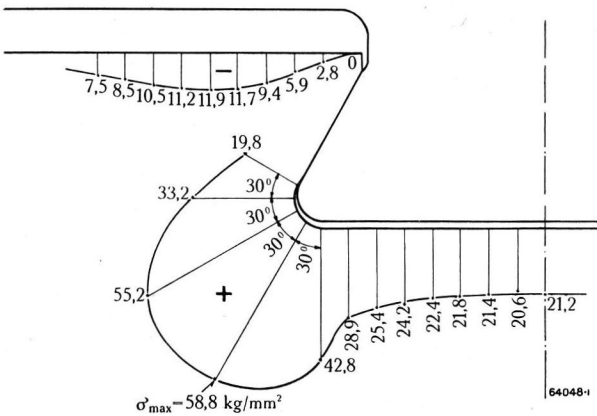


Fig. 9. — Contour stresses around dovetailed groove of the wheel for the combined stress at the overspeed revolutions.

(Determined from photo-elastic tests.)

B. MECHANICAL INVESTIGATION ON CAST STEEL MODEL.

With a view to elucidating these conditions on the tough and relatively ductile working steel, the above-described photo-elastic tests of the Swiss Federal Materials Testing Institute were supplemented by the following investigations in the Company's own materials testing laboratory. Plane models of the pole and grooved section of the spider rim were made from cast steel of the same quality as used for the magnet wheel (i. e. with a tensile strength of 50—60 kg/mm²) to a scale of 1:2. The cast steel plates employed were 20 mm thick. Fig. 10 shows the cast steel model which was fitted in the tensile testing machine by means of the threaded journals top and bottom, like a giant tensile test specimen. Altogether six such cast steel models were tested as follows:—

Tensile and impact tests on the material.
Load and pulling-out tests on the pole joint.
Observation of resulting distortion.
Strain and stress measurements.

Since in this case the material has the same modulus of elasticity and yield point as the magnet wheel itself the strains and stresses on the latter calculated from the model do actually represent the behaviour of the pole and spider rim both as a whole and at individual points. It must nevertheless be remembered that the stressing under these test conditions does not conform to the actual distribution of the forces. The forces here only act in the radial direction, whereas under service conditions the pole tension and the centrifugal

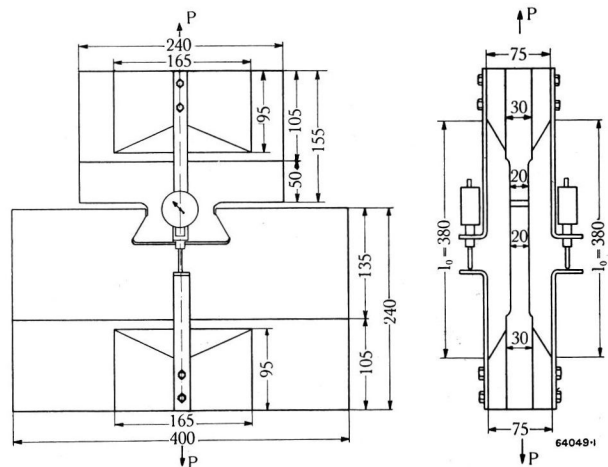


Fig. 10. — Cast steel model to scale of 1:2.

P = Tensile force.

force in the rim are offset by the tangential forces in the rim in the form of a combined load as shown in Fig. 4c. In the following notes and curves the values for the magnet wheel converted from measurements on the model are again given, the tensile forces being stated in kg per cm pole length subjected to the force, to obtain smaller figures. The total tensile force on the pole always results from multiplication of the pole force per cm by the length of 156 cm.

1. Properties of the Material.

The average properties of the cast steel of the model were as follows:—

Yield point kg/mm ²	Tensile strength kg/mm ²	Elongation 5 d %	Reduction of area %	Impact mkg/cm ²
27.5—28.6	51.3—53.0	21.2—24.6	31—35	6.5—7.8

These values are absolutely the same as found by test at different points on the actual spider rim.

2. Pulling-out Tests.

During these tests the radial elongation or displacement of the pole in the direction of the tensile force relative to the spider rim was measured as a function of the tensile force per cm pole length. The result is shown in Fig. 11. It will be noticed that there is quite a definite "yield point of the pole joint", when the radial displacement between the pole and spider is of the order of 1 mm. The yield point is attained with a tensile force of 11,000 kg/cm. As the test proceeds the load increases still further and the tensile

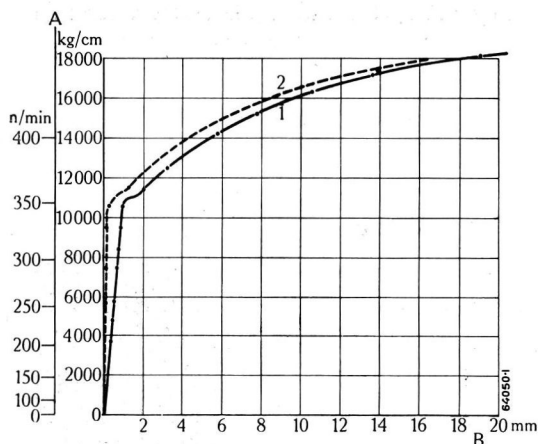


Fig. 11. — Pulling-out tests on cast steel model.

Radial displacement = f (tensile force per cm pole length).

- A. Tensile force on pole in kg/cm and r.p.m.
 B. Radial displacement of pole out of groove. Δl in mm.
 1. Total displacement Δl under load.
 2. Permanent displacement after unloading.

force on the pole reaches a maximum of 19,200 kg per cm pole length. With a simultaneous drop in load a heavy sliding movement begins and the pole slips out of the groove. During an overspeed run of the actual magnet wheel, however, the tensile force would remain constant and the pole fly out of the groove immediately the maximum load is attained, instead of gradually sliding out. Moreover, under overspeed conditions the tensile force is only 6820 kg/cm, i. e. far from attains the above-mentioned values. Without taking the circumferential forces in the spider rim into consideration, therefore, a factor of safety of 1.62 is obtained, referred to the yield point of the pole joint, and 2.8 against the poles flying out.

3. Observation of Distortion.

These pulling-out tests enabled the initial permanent deformation, i. e. the points of maximum local strain, to be observed on the plane model. With the naked

eye the Lüders' lines, i. e. the points of maximum shear stress, can be clearly recognized as dull curved lines on the highly polished surface, but make a very poor photograph. Fig. 12 gives an idea of these strain lines and shows the maximum shear stress directly across the grip tooth at an angle of about 60° , as already seen in Fig. 8. The strain lines in the corner of the groove occurred after those in this shear surface.

An even more demonstrative method, especially where low stresses are concerned, is to cover the surface of the model with a thin coat of an adhesive but brittle varnish. Devoid of practically all properties of plasticity this immediately cracks when the model is subjected to stress and the specific strain reaches a certain value. These readily visible cracks always run perpendicular to the greatest principal tensile

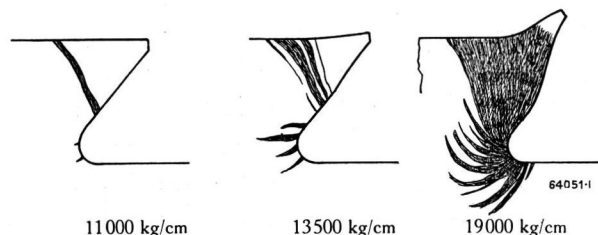


Fig. 12. — Lüders lines around groove at various stages of pulling-out test.

stress. The stress or strain at which the cracks first appear depends on the composition, state of dryness, and thickness of the varnish coating. They are no criterion for the yield point of the material. Fig. 13 shows the initial cracks in the varnish, which again form the lines of the maximum shear stresses occurring here in the grip tooth at an angle of 64° to the horizontal. Each individual crack runs exactly according to theory at an angle of 45° to the direction of the shear stress, i. e. perpendicular to the greatest principal stress. In Figs. 14, 15, 16, and 17 the cracks increase in both size and number with increasing pole tension. The brittle varnish is not able to follow heavy deformations of the steel model and flakes off over whole sections of the surface (Figs. 16 and 17). In Fig. 16 the lines already take the form of real stress curves; if they are compared with the curves in Figs. 5 and 7 determined photo-elastically (the cracks naturally only show one of the two groups of principal stresses) the agreement between the two will be found to be good. Here, too, the shear stress zone in the grip tooth appears earlier than the stresses along the edges of the groove.

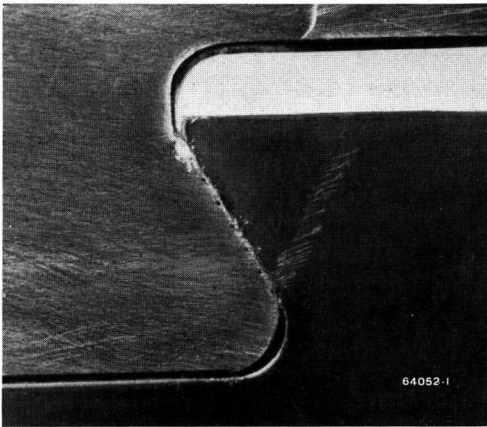


Fig. 13. — Cracks in varnish coating during pulling-out test.
Load on pole = 9000 kg/cm.

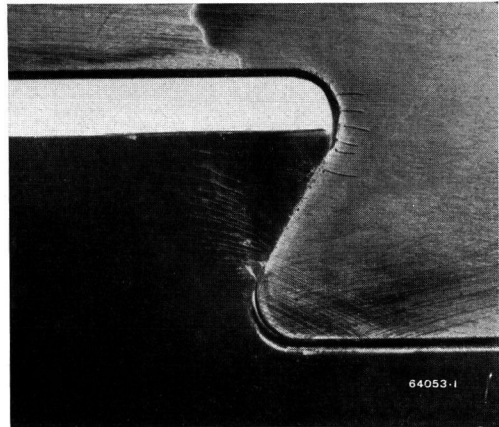


Fig. 14. — Cracks in varnish coating during pulling-out test.
Load on pole = 12,000 kg/cm.

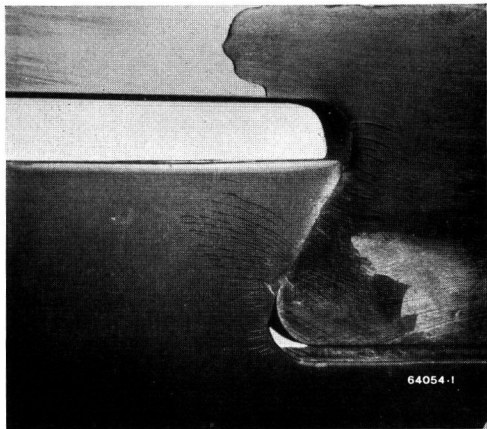


Fig. 15. — Cracks in varnish coating during pulling-out test.
Load on pole = 14,000 kg/cm.

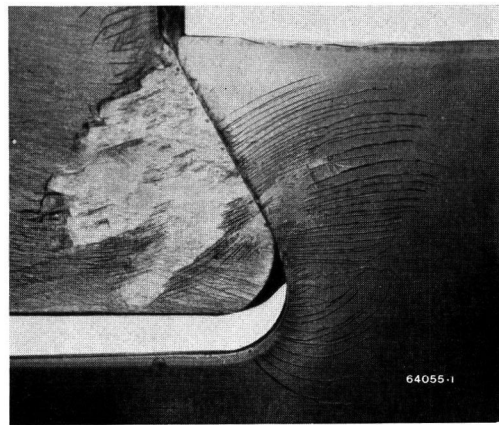


Fig. 16. — Cracks in varnish coating during pulling-out test.
Load on pole = 17,000 kg/cm.

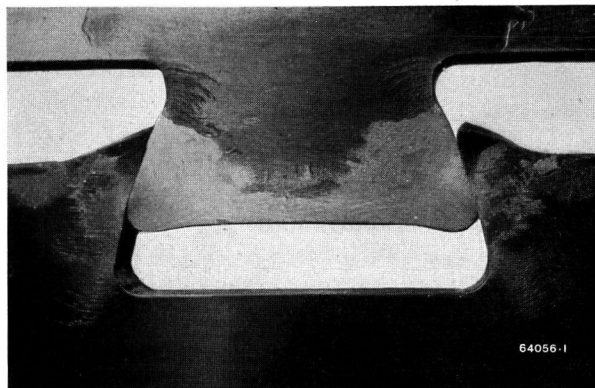


Fig. 17. — Cracks in varnish coating during pulling-out test.
Load on pole = 19,000 kg/cm.

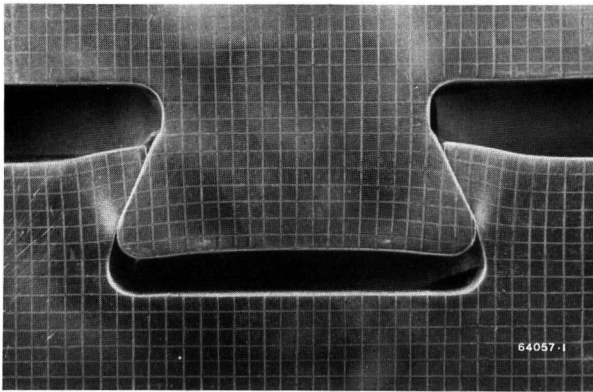


Fig. 18. — Deformation of cast steel model under load of 17,500 kg/cm.

The changes in a 5 mm grid applied to the surface of the model give a good idea of the deformations occurring during the pulling-out test. Fig. 18 shows the model after a tensile force of 17,500 kg/cm had been applied to the pole and Fig. 19 at the maximum load just before the sliding of the pole out of the groove. The illustrations portray in an exemplary manner the flexibility and resiliency of the material employed, and also show how local over-stressing is reduced until the whole cross-section participates in carrying the load.

After each increase in the test load the edges of the grooves in the spider rim and pole were carefully examined for cracks. It was impossible to find even the finest cracks in the tested parts of the model, although the tensile forces were nearly three times greater than during the overspeed run of the wheel.

4. Stress Measurements.

The stress was measured at thirteen points on the wheel and at ten points on the pole. As in the case of the photo-elastic tests the loads were kept low to avoid permanent deformation and to obtain exclusively elastic strains from which alone it is possible to compute the stresses. When the strains measured at each point in three directions with extensometers (gauge length 10 mm) were determined at $\frac{1}{n}$ of the tensile force occurring on the pole during the over-speed test the strains at the over-running speed were n times greater. These then allowed the two principal stresses, the directions, and finally the reduced comparative stress to be computed from known laws. Fig. 20 shows the

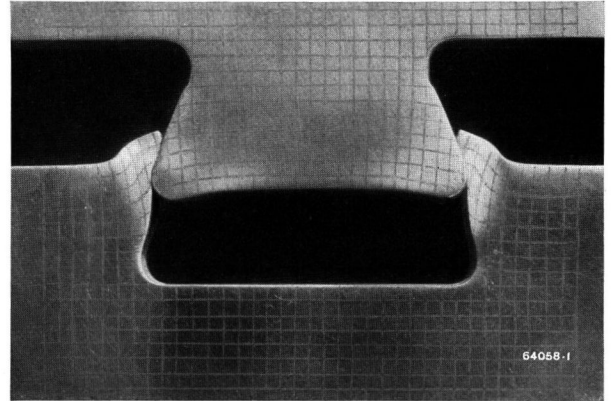


Fig. 19. — Deformation of cast steel model under load of 19,200 kg/cm.

lines of equal reduced stress obtained from the individual test points, which again indicate the zones of maximum stress. The similarity to Fig. 8 is very great, the differences arising from the different loading employed for the two tests. Here, too, the maximum stress is found to be in the corner of the groove.

One principal stress vanishes on the free edge, i. e. where no force is transmitted, and the contour stress can be determined by direct tangential measurement along the contour of the plane cast steel model. The extensometer, having a gauge length of 10 mm, naturally indicates the mean strain along this length, and when these strains are subject to great local variation, i. e. the stress peaks are very steep and narrower than the gauge length, the measured mean value will be much lower than the actual stress peak. These mean values were measured at distances of 2.5 mm; they give a curve of the mean stresses which is much flatter than that which would be obtained with the real peak stresses and are plotted around the contour of the wheel groove in Fig. 21. They attain a maximum at point 6, approximately on the bisection of the angle. The mean value measured for a gauge length of 10 mm corresponds to 30.9 kg/mm² at the tensile load on the pole during the overspeed test. The actual peak stress at this point, however, is much higher. In the pole neck the contour stresses are clearly lower than in the wheel. It must also be mentioned that as with the photo-elastic measurements, these values are likewise subject to a certain inaccuracy, viz., the proviso of a biaxial state of stress ($\sigma_3 = 0$) is only approximately fulfilled in a cast steel plate, since probably certain stresses occur perpendicular to the measured plane. It is impossible to give the

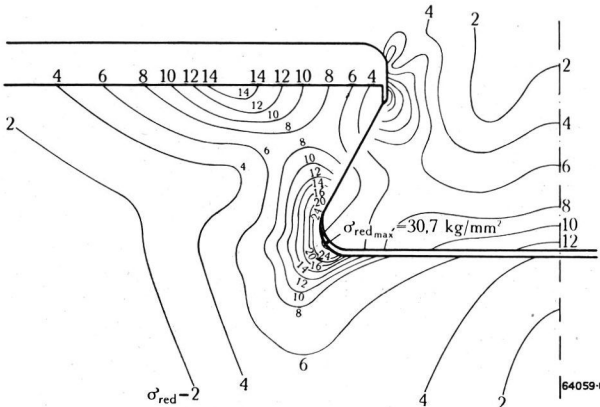


Fig. 20. — Curves of constant stress in pole and spider rim at over-speed revolutions (from tests on cast steel model).

contour stresses at points 1, 2, 3 and 21, 22, 23, because the wheel and pole bear against one another here and exert pressure.

The results of these stress measurements can be clearly presented in the form of a model in three coordinates, a view of which is shown in Fig. 22. The high stress concentration in the contour zone at the corner of the groove and the shear stresses in the tooth show up well here. It is now clear how sharp this stress peak really is: the yield point of the cast steel (27 kg/mm^2) is only exceeded within a contour zone of about 9 mm in length and a few tenths of a millimetre deep.

5. Measurement of Strains in the Inner Angular Points of the Groove during the Pulling-out Test.

The question now is: What happens at such a spot when the yield point is so greatly exceeded? It might, for instance, be anticipated that at this moment the strain would increase locally and reduce the stress by plastic deformation, i. e. that the steel

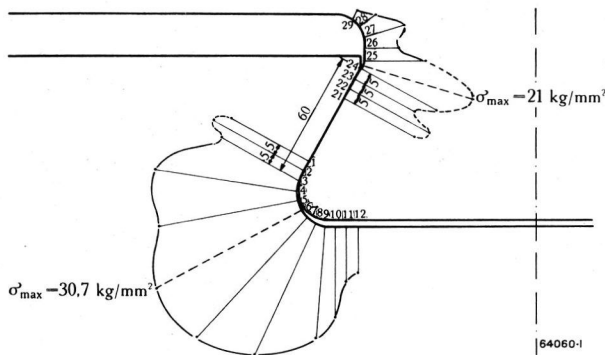


Fig. 21. — Characteristic of contour stresses at overspeed revolutions (measured on cast steel model).

would stretch at this confined point. A further pulling-out test was, therefore, made on the cast steel model and the radial displacement of the pole out of the wheel rim observed as the pole load increased. Concurrently, the extensometer was fixed to the point of maximum stress, point 6 on the corner, and the strain observed at this point during the whole of the tensile test — that is not only in the elastic portion as before — until the “yield point of the pole joint” was reached. The unloading after every load step gave the permanent set. In this way both the radial displacement of the whole pole and the local strain at point 6 in the corner along the gauge length of 10 mm were obtained at one and the same time, the latter again being a criterion for the stress concentration there. The resulting test values are

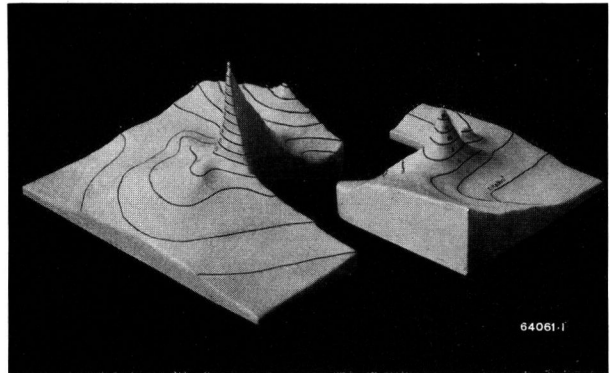


Fig. 22. — Triaxial presentation of stresses in the pole joint (on left spider groove, on right pole dovetail, in each case left half) from tests on cast steel model.

given in Fig. 23. In the left-hand part of the figure the pulling-out curve of the pole is again shown to facilitate comprehension; the yield point of the joint is found at a tensile load on the pole of $11,000 \text{ kg/cm}$. In the centre the local strain measured at point 6 (mean value of 10 mm gauge length) is plotted, while on the right the mean stress values computed from these elastic strains are shown. The real peak of the stress at this point can be obtained from the photo-elastic measurement (Fig. 9); with combined load during the overspeed test this amounts to $58,8 \text{ kg/mm}^2$ and gives the dotted straight line graph corresponding to the actual stress peak with increasing tensile force on the pole, i. e. speed.

The yield point of the cast steel (27 kg/mm^2) is attained at a tensile force on the pole of the order

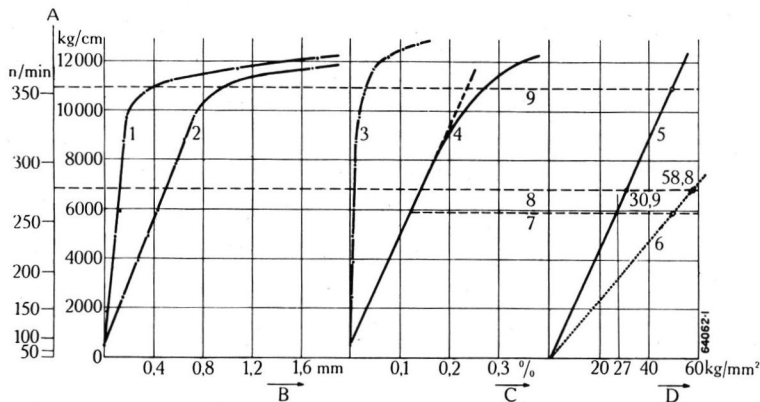


Fig. 23. — Strains and stresses at most highly stressed point of magnet wheel during pulling-out test on cast steel model.

- A. Tensile load on pole in kg/cm and speed r. p. m.
- B. Radial displacement of pole in mm.
- C. Mean strain in per cent at point 6 per gauge length of 10 mm.
- D. Contour stress at test point 6 in kg/mm².
- 1. Permanent displacement after removal of load.
- 2. Total displacement under load.
- 3. Permanent strain after removal of load.
- 4. Total strain under load.
- 5. Mean stress values during pulling-out test.
- 6. Actual stress peak during overspeed test.
- 7. Yield point of cast steel.
- 8. Stress at overspeed revolutions.
- 9. Pole begins to slide out. Yield point of pole joint.

of 3100 kg/cm, the stress peak reaching this value at about 190 r. p. m. and the mean value of the stress at approximately 260 r. p. m. The curves now clearly show that, quite surprisingly, nothing out of the ordinary happens as a result. Neither the radial displacement of the pole nor the strain at point 6 suddenly increases or shows any tendency to change; both continue to increase linearly with the pole tension. The same naturally also applies to the mean stress determined from the elastic elongation. Even when the tensile force is pushed up to the load corresponding to overspeed conditions and beyond, nothing happens until it reaches approximately 10,000 kg/cm. Not until then does any noticeable deviation from the proportionality between tensile force and strain at point 6 occur.

It is thus seen that the curves for the strains (centre) and the stresses (right) pass through and even exceed the yield point of the material without the familiar discontinuity in the curve occurring, i. e. the material does not yield. What is the explanation of this? Beyond a stress of 27 kg/mm² the fibre measured in the corner no longer takes its full share of the load. If it were free it would cede and extend greatly. It is, however, rigidly bound to the next fibre on the inside along its whole length and has to follow practically the entire movement of this fibre, which is much smaller due to the fact that the yield point

is not attained there. (The steep drop at this point will be recalled from the stress model Fig. 22.) In brief, we have the phenomenon of prevented change of form in a non-uniformly stressed section. In the case of the spider rim this results in the fibre at point 6 behaving as though its yield point were much higher, possibly at about 60 kg/mm² (with a gauge length of 10 mm). The yield point is commonly said to increase, although — as we saw — it is only nominal. If the curve of the strains at point 6 is examined closely it will be seen to deviate slightly at about 27 kg/mm² from the straight line graph which is shown dotted. This deviation of a few thousandths of 1 per cent is the only sign that an equalization of stress has taken place there. The outside

fibre is backed up and relieved by that further inwards, the latter taking over somewhat more load; a reduction of stress thus occurs which with increasing tensile load on the pole penetrates more and more into the centre, until finally the whole cross-section yields. Not until then can the pole yield point, i. e. the moment when the pole joint begins to give way, be observed. A proviso for this process is a sufficiently tough and ductile material to permit of multiaxial stressing with an adequate margin of safety. Such a material withstands amazingly high local strain without cracking under conditions of non-uniform triaxial stressing, as can be proved on a tensile test specimen, where at the point of greatest reduction of area the strain may attain well over 100% prior to rupture. Likewise the strain of the outside fibre of a bending test bar may attain 100%. In view of this characteristic of the strain in the corner it can be understood why the first cracks in the varnish (Fig. 13) did not show up at this point; this part, being only nominally overstressed, will not be strained substantially more than the neighbouring zone, subjected exclusively to elastic stress, and as a result the varnish showed no tendency to crack. On the other hand, the first cracks occurring transversally across the tooth show that the whole cross-section there began bearing uniformly earlier, notwithstanding the fact that the stress concentration in the corner was greater. The

really critical cross-section, therefore, lies on this shearing surface, already frequently referred to, at an angle of about 60° , and not in the corner.

There is a difference between the plane plate model and the actual pole in that the plate has one degree of freedom more, i. e. can become deformed perpendicular to the plane of measurement, which, however, is not possible with the 1560 mm long pole. As a result the prevention of deformation will be much more pronounced in the pole itself. It is therefore to be assumed that the entire massive pole will withstand a much higher load and show less tendency to become deformed if subjected to a pulling-out test, than 156 plates 1 cm thick arranged one alongside the other.

The maximum stress in the wheel groove under conditions of combined stress at the over-running speed was found to be 59 kg/mm^2 from the photo-elastic tests. It is true that the yield point of the joint under combined stress conditions is not known, but it can be assumed that the additional tangential stressing in the rim will not greatly affect the yield point of the pole root as determined during the stripping test. This assumption is backed up by the great qualitative similarity of the stress distribution in the two load cases to which attention has already been drawn (compare Fig. 5 with 7 and 8 with 20). If the above-mentioned stiffening of the whole pole is taken into consideration it can be safely assumed that the pole yield point will not be attained when this stress peak occurs during the overspeed test on the magnet wheel. In other words, notwithstanding this seemingly dangerous stress intensity no radial displacement of the poles is to be anticipated during the test run.

The fact that local stress concentrations in grooved design components, although considerably exceeding the yield point of the material, do not lead to permanent deformation under conditions of static stress unless the whole cross-section flows, is not only of importance here, but also fundamentally. The phenomenon is well known and made use of daily in engineering applications. For instance, such stress concentrations are encountered along the threads of all screws, as well as in the case of countless parts having a change in cross-section, grooves, etc. In all these cases no one would think of proportioning

the part to keep the local peak stress below the yield point, and as has been shown here, it is not even necessary in the case of a large magnet wheel. The effect of such stress concentrations is naturally fundamentally different under conditions of impact or alternating stresses, where a progressive weakening of the part at the point in question quickly sets in and sooner or later leads to fatigue cracking and finally fracture. Only when the kind and intensity of the stress are accurately known is it possible reliably to decide whether and to what extent a local stress can be tolerated in any given case. Generalization is impossible and particularly in the case of superposed alternating stresses might prove disastrous.

C. OVERSPEED TEST.

As a result of the knowledge of the stress conditions obtained during the foregoing tests it was decided to carry out the overspeed test on the magnet wheel in the open workshop. Apart from the pole joint, the greatest interest was taken in the behaviour of the spider rim. The latter is of two-piece construction, both halves being bolted together with a strong plate. This rim is shrunk on as a whole on a ten-arm spider. The shrinkage is so calculated that at normal speed (100 r. p. m.) the rim is still definitely shrunk on the spider, whereas at the overspeed of 280 r. p. m. it expands under the influence of the centrifugal forces until it is clear of the arms and rotates quite freely.

This movement between the ends of the spider arms and the rim, i. e. the increase in diameter of the rim was accurately measured at various speeds with the help of very soft, readily deformable pressure elements. No increase in diameter takes place up to a speed of 167 r. p. m., the shrinkage pressure remaining effective up to this point. At this speed the rim begins to leave the arms. The increase in diameter took place practically linearly to the tangential hoop stress, as was to be anticipated, within the elastic limit. Once the stressing had been removed, i. e. the rotor brought to rest, no permanent increase in diameter could be measured.

Without very complicated measuring equipment it was impossible to measure the displacement of the poles during the test run. The checks had therefore to be confined to measurement of all poles before and

after the test to see that no permanent radial displacements had taken place. These measurements corroborated the tests on the cast steel model in Fig. 23, in that in no single case did the displacement exceed 0.1 mm. What is more, upon dismantling the machine it was impossible to detect the slightest trace of deformations or impressions on the bearing surfaces of the dovetail or in the inner corners of the wheel grooves or elsewhere.

In conclusion it might be mentioned that this result is not restricted to the case exhaustively described here. Photo-elastic tests have also shown that another type of pole joint has stress concentrations of approximately the same intensity. This method of fixing was employed earlier for three large generators which were also subjected to an overspeed test before being put into service. No disadvantages or doubtful phenomena were observed in the case of these machines either.

It is thus proved that even under conditions of combined stress the yield point of the joint is far from reached, notwithstanding the fact that local stress concentrations greatly exceed the nominal yield point of the material.

The practical overspeed test on the finished machine has therefore fully corroborated the conclusions drawn from the results of the model tests.

D. SUMMARY.

The stipulation that large magnet wheels must be given a test run at an overspeed of 2.8 times the normal service speed increases the material stressing to eight times. The resulting stress concentrations may reach an extremely high intensity, particularly in the corners of the pole joints. Since these peak stresses cannot be computed sufficiently accurately, they were determined photo-elastically by the Swiss Federal Materials Testing Institute, Zurich, on a celluloid model of the pole joint to a scale of 1:2.5 under service stress conditions. It was found that the contour stress in the corner reaches an intensity of 59 kg/mm²

over a small zone, thus greatly exceeding the yield point of the material.

Since this result is based on material behaving absolutely elastically, extensive tests were conducted on cast steel models of the pole joint in the Company's materials testing laboratory. Special attention was paid to the plastic zone to determine how the pole joint behaves in general and at the critical points when the peak stress exceeds the yield point. The load tests proved a well marked yield point of the joint to exist when the pole begins moving outwards. This yield point of the joint is 62% higher than the tensile load occurring on the poles during the test run. Even when the pole was entirely pulled out of the groove there were no signs of cracking of the material. A varnish coating on the model indicated the position and character of the deformations under conditions of rising load. The first and greatest deformations occur on a shearing zone plane (at about 60° to the radius) directly across the teeth of the wheel. Numerous extensometer measurements permitted the principal and reduced combined stresses to be determined, thus enabling a triaxial presentation of the stress across a plane section of the pole joint to be built up.

At the point of maximum contour stress, strain measurements were made to follow the behaviour of the contour fibres with increasing tensile load on the pole. It was found that, although the yield point was exceeded, the material behaved more or less elastically well beyond the load occurring during the test run, as far as the yield point of the pole, due to prevention of deformation. Neither dangerous strains nor cracks, therefore, occur, but the phenomenon of an apparent increase in the yield point under conditions of multiple stress is observed.

The overspeed test run to which the wheel was subjected, together with the careful checks for strains and displacements, fully corroborated the conclusions arrived at from the tests, viz., that notwithstanding the enormous forces and stresses involved no deformation occurred.

(MS 639)

H. Zschokke. (E. G. W.)

An Infinitely Variable Regulating Transformer without Contacts:

THE PLUNGER-COIL VOLTAGE REGULATOR.

Decimal Index 621.314.214

Starting with a short survey of the possibilities of voltage regulation in distributing systems, this article proceeds to describe the construction and application of the plunger-coil voltage regulator recently developed by Brown Boveri. Features of the apparatus are the absence of contacts subject to wear and its suitability for outdoor erection. Apart from the standard types for 380 V, currents between 10 and 400 A, and a range of regulation of 20%, many special types for other voltages and purposes are available. This simple and sturdy regulating transformer is provided with automatic control gear.

REGULATION of the secondary voltage at points of consumption is not only often a necessity in rural districts with long secondary supply lines, but in densely populated areas there are also consumers requiring a constant supply voltage.

across the turns of a coil, the insulation being removed in the path of the contact, as with a sliding resistance. For heavier currents the contact, usually a roller, moves along the bare turns of a winding. In this latter case infinitely variable regulation is even

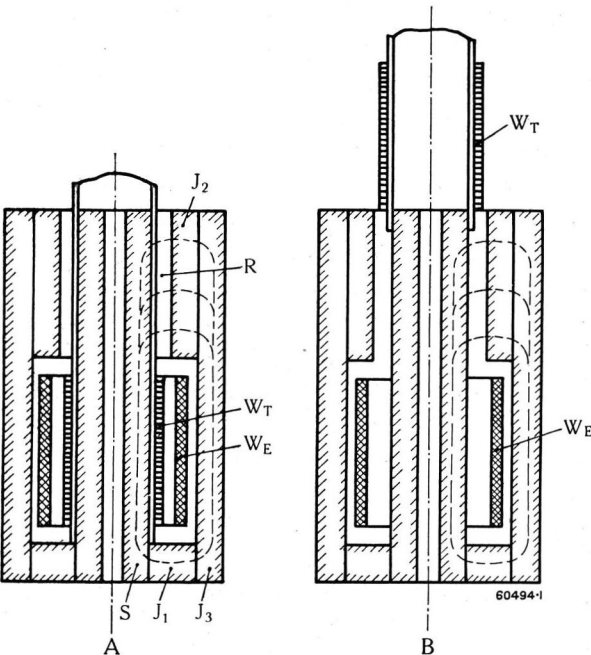


Fig. 1. — Fundamental design of a single-phase plunger-coil voltage regulator.

The coil W_T moves in and out of magnetic linkage.

- A. End position, inserted.
- B. End position, withdrawn.
- J_1 . Lower yoke.
- J_2 . Upper yoke.
- J_3 . Return magnetic circuit.
- R. Annular gap.
- S. Radially laminated core.
- W_E . Energizing coil.
- W_T . Plunger-coil.

Experience has proved that tapped auto-transformers with an automatic on-load tap-changing switch are too expensive for small outputs. Moreover, operation is often so frequent that the switch is subject to rapid wear. Up to a certain amperage regulating transformers are obtainable with which a contact slides

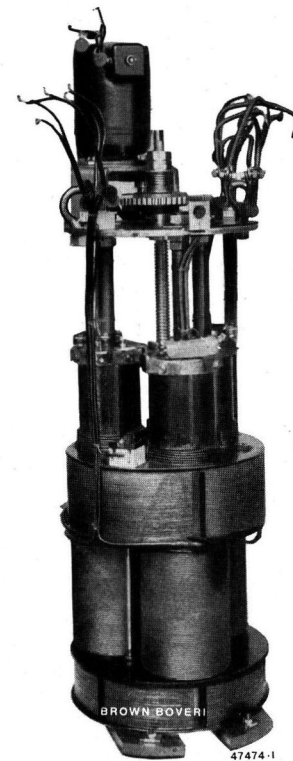
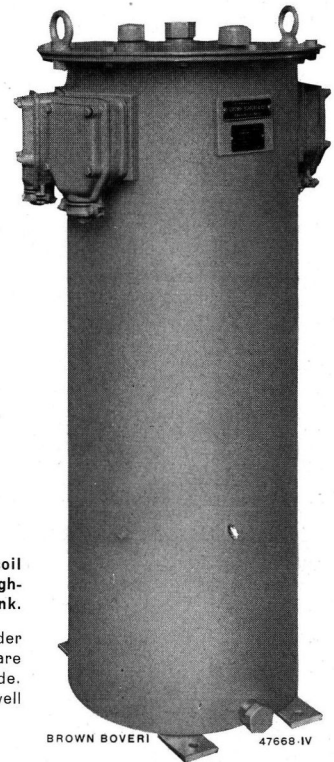


Fig. 2. — Active parts of a three-phase plunger-coil voltage regulator for a 50 A through-current.

The plunger-coils of the three phases are here shown in an intermediate position. Note the simplicity of construction of this voltage regulator.

Fig. 3. — Three-phase plunger-coil voltage regulator for a 50 A through-current in weather-proof oil-tank.

The control motor is located under the tank cover. The terminals are lodged in cable boxes on the side. This type lends itself particularly well to pole-mounting.



possible, though of course the moving connection of the contact cuts through the magnetic flux in a gap across the core or in a longitudinal slot in a rotary core. The necessary moving contacts of these regulating transformers are, from their very nature, subject to a certain amount of wear, especially if, due

to sensitive adjustment of the control relay, the regulating mechanism functions frequently. A regulator having no contacts is therefore to be preferred, particularly for out-of-the-way sites. Brown Boveri have developed a simple and sturdy apparatus of this class, the plunger-coil voltage regulator type TQ, described in the following notes.

winding (Fig. 1). In the end position A the coil W_T is inside the core and energizing coil W_E , as in an ordinary transformer. Thus both windings link with the entire magnetic flux and the maximum voltage is induced in the coil W_T . In position B, however, the coil W_T is completely withdrawn from the core and no longer links with the magnetic flux, so that no voltage is induced.

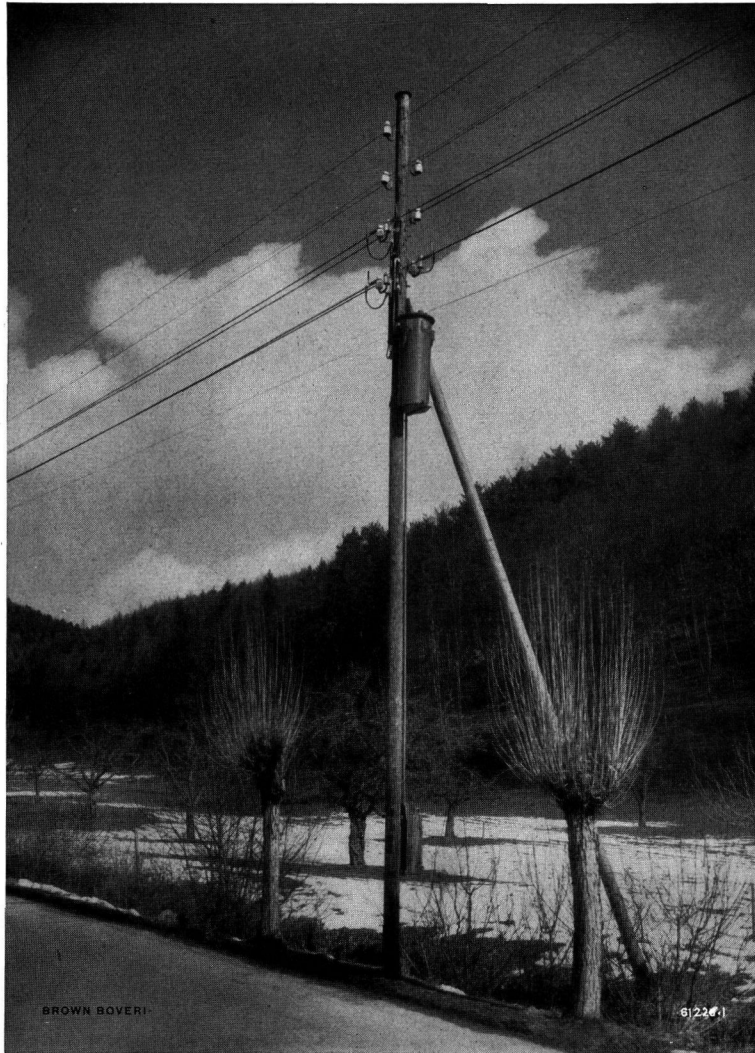


Fig. 4. — Plunger-coil voltage regulator mounted on pole.

The control relay is mounted lower down on the pole at a convenient height. In the tube between the regulator and relay box there are only the control wires (normal control voltage 190 V, tapped off the energizing coil).

In the case of the conventional induction regulator and the sliding coil transformer, which may be included in the same category, one wound iron core is moved in relation to another, whereby the change in magnetic linkage produces a change in the voltage in one winding. Where the plunger-coil voltage regulator is concerned, however, only a *winding* is moved in relation to a core and yoke system carrying an energizing

Between the two end positions the coil moves in an annular gap in the core. As it travels up and down, the influence of the magnetic flux and consequently the induced voltage varies constantly. Because the coil W_T is so to speak plunged into the iron core, it is styled "plunger-coil", and the complete transformer "plunger-coil voltage regulator". As a rule the plunger and energizing coils are auto-transformer-

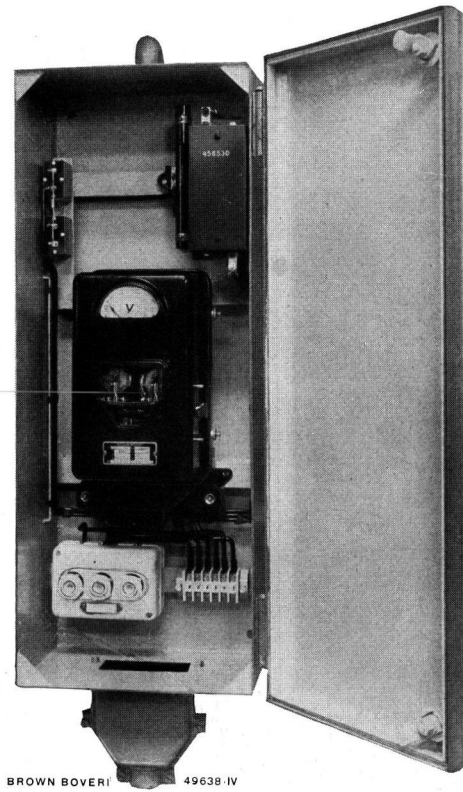


Fig. 5. — Control apparatus in weather-proof box.

The lever on the right of the control relay adjusts the regulated voltage indicated directly by the voltmeter. The box also contains the radio interference suppressor for the relay and, where such is provided, the compounding resistance.

connected, since a limited range of regulation, e. g. 20%, is usually quite adequate.

It might be supposed that the rather wide annular gap in the iron core would result in an excessive no-load current. This, however, is avoided by the choice of a radial core with a high upper yoke. In this way the flux density in the annular gap is decreased to a small fraction of that in the core. The fact that with a radial core the passage of the flux from the limb to the yoke takes place absolutely uniformly around the whole circumference, is particularly useful here. In conjunction with a suitable reinforcement of the upper yoke the effective cross-section in the gap can easily be brought to ten times the section of the limb.

For three-phase operation the three cores are preferably arranged in triangular formation and fitted in round yoke plates (Fig. 2). At the bottom the cores are connected to the yoke with only a thin layer of insulation between, so that there is no gap of any importance here. The upper yoke plate is also carried by the bottom plate through the intermediary of a

system of supports, since the annular gap must leave a free passage at the top for the plunger-coils. The latter are wound on bakelized paper cylinders attached to a spider which moves up and down a robust, threaded spindle like a travelling nut.

Strong metal tubes projecting above the three cores carry the upper bearing of the spindle, the other end of which rests on the upper yoke plate. The spindle is located in the centre of the triangularly arranged

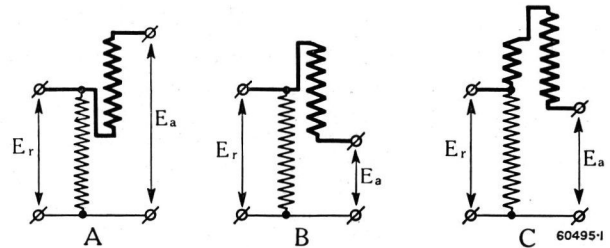


Fig. 6. — Auto-transformer connections of the plunger-coil voltage regulator.

- A. Booster connection
e. g. $E_a = 454 \dots 380 \text{ V}$, $E_r = 380 \text{ V}$.
- B. Buck connection
e. g. $E_a = 306 \dots 380 \text{ V}$, $E_r = 380 \text{ V}$.
- C. Buck connection with auxiliary winding.
e. g. $E_a = 343 \dots 417 \text{ V}$, $E_r = 380 \text{ V}$.
- E_a . Unregulated, incoming voltage.
- E_r . Regulated voltage.

cores and is rotated by a small motor through change-speed gear. Immediately before the end positions a simple limit switch disconnects the control motor. A sliding contact is dispensed with, connection to the plunger-coil being made by very flexible copper cables. The whole is housed in a cylindrical tank filled with oil and provided with cable boxes for connecting the unregulated and regulated voltages and the control wires. As the control motor is also lodged in the oil tank, the transformer lends itself very well to outdoor erection. The whole construction is robust and requires no attendance. The smaller types can be mounted on poles as shown in Fig. 4.

The control relay type Q (Fig. 5), of proved design, is in this case housed in its own weatherproof box at a convenient height low down on the pole (Fig. 4). The value of the regulated voltage can thus be readily adjusted and if necessary altered, since a setting device is provided.

The plunger-coil voltage regulators type TQ are normally built for 380 V and with a regulation range of 74 V. According to the rated through-current, transformers for 10, 25, 50, 100, 200 and 400 A have been developed. As three-phase regulators, however, they are available for all voltages between 190 and 500 V, and as single-phase for 110—500 V,

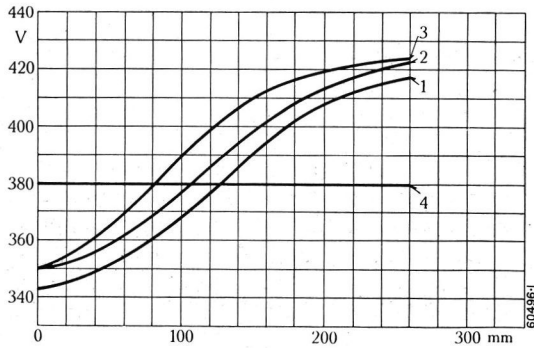


Fig. 7. — Relation between unregulated voltage and path of regulating coil in a plunger-coil voltage regulator for a 25 A through-current.

Regulated voltage = 380 V (connection C).

- 1. Unregulated voltage at no-load.
- 2. Unregulated voltage at 25 A load and p.f. = 1.
- 3. Unregulated voltage at 25 A load and p.f. = 0.7.
- 4. Regulated voltage (380 V).

Abscissæ. Path of regulating coil.
Ordinates. Unregulated and regulated voltages.

the rated current and the range of regulation remaining unchanged.

The plunger-coil can be connected to the energizing winding in boost or buck or through the intermediary of a small auxiliary winding. The following table gives an example of the ranges of regulation that can be obtained for a voltage of 380 V.

The control relay can be adjusted to a sensitiveness of ± 0.2 to $\pm 3\%$. With normal supply conditions $\pm 1\%$ is generally adequate. If a small current transformer and a compounding resistance is installed it is possible to regulate the secondary voltage in

Range of regulation No.	Position of regulating winding	Range of regulation V	Unregulated voltage with reference to a 380 V regulated supply V
Three-phase regulator			
1	withdrawn	0	380
	inserted	- 74	306
2*	withdrawn	+ 18.5	398.5
	inserted	- 55.5	324.5
3	withdrawn	- 37	417
	inserted	- 37	343
4*	withdrawn	+ 55.5	435.5
	inserted	- 18.5	361.5
5	withdrawn	+ 74	454
	inserted	0	380

* Only for 50-400 A regulator types.

relation to the current, instead of to a constant value. Fig. 7 shows the path of the plunger-coil in relation to the unregulated voltage.

It will be noticed that over the whole regulating range the voltage varies sufficiently uniformly for practical purposes. For special applications, as e.g. laboratories, or for darkening equipment in cinemas and lecture rooms, the plunger-coil voltage regulator can also be provided with independent windings for regulation from zero.

(MS 586)

A. Meyerhans. (M. S.)

INCLINED HOIST WITH AUTOMATIC DIRECT-CURRENT DRIVE IN A QUARRY.

Decimal Index 625.52-83

A Ward-Leonard-controlled d. c. drive with automatic control gear proved to be particularly economic for an inclined hoist in a quarry. It is shown in the following notes how the severe service requirements were met with the minimum of attendance.

IN a large porphyritic stone quarry the blocks of stone are loaded on to an inclined hoist at the lower terminal station, conveyed to the upper station, and from there transported to the processing plant. The electric drive of the winch was required to be as economical and reliable as possible. This dual requirement is best met with an *automatic drive* which allows the man attending to the arrival and dispatch of the wagons at the upper station — the so-called hanger-on — also

to operate the hoist. Once the hanger-on has set the hoist in motion by depressing a push-button, he can turn his attention to his main job, i. e. the supervision of the incoming and outgoing wagons. The operation of the automatic hoist, therefore, requires no more workmen than is otherwise required for discharging and loading the trucks, that is, one man each at the upper and lower stations. Moreover, with such a drive each winding cycle is accomplished entirely without human intervention. If the equipment is set to the shortest cycle the plant is utilized to a maximum with the minimum of attendance. This is of special value under heavy duty conditions.

The Type of Driving System

was determined mainly by two requirements.

Firstly, the trucks must come to rest at the end of the trip within ± 10 mm of the level of the wagon track, independently of the load, to ensure the wagons running on and off easily. Secondly, for winding men the trucks must be able to run at half speed. A *Ward-Leonard-controlled d. c. motor drive* permits these requirements to be met. Smooth starting, retardation, and stopping diminishes the whipping of the cables to a minimum. As a result, wear and tear on both the cables and the entire machinery of the hoist is considerably reduced. The speed characteristic of the wind can be readily adapted to the prevailing service conditions by suitable adjustment of the resistances and control switch. A readily adjustable compounding equipment ensures the trucks running into the stations at a very low speed practically independent of the load, while a rapidly, but smoothly operating mechanical brake finally brings them to rest with great accuracy.

The hoist is double-acting (Fig. 1), the two trucks travelling over two superposed tracks inclined at an angle of 53° . To avoid the difference between the level of the wagon tracks and the trucks being too great — as might easily be the case under fluctuating load conditions if the freely suspended trucks were to be held by a mechanical brake at the end of the wind — the trucks are designed in two parts as will be clear from Fig. 2. The lower part, the truck proper, carries at its upper end the compensating device for the double traction rope; the upper part, the loading platform, carries three wagons one behind the other. At the end of each wind, the loading platform of the lower truck rests on supports in the lower station, whereas the truck itself continues to move downwards a few inches further. Thereupon, the two trucks, that in the upper station with its loading platform and that in the lower station without it, freely suspended from their cables, are held by the mechanical brake. The cables of both trucks consequently remain taut, thus avoiding whipping when restarting. To enable the truck to take the loading platform up without shock, the speed is also kept very low at the beginning of the wind.

The *push-button control* of the winch drive is so designed that the hoist cannot be set in motion from the upper station until the hanger-on at the lower station has given the "ready" signal, i. e. until an appropriate lamp has lit up at the upper station. Official regulations prevent the hoist being operated

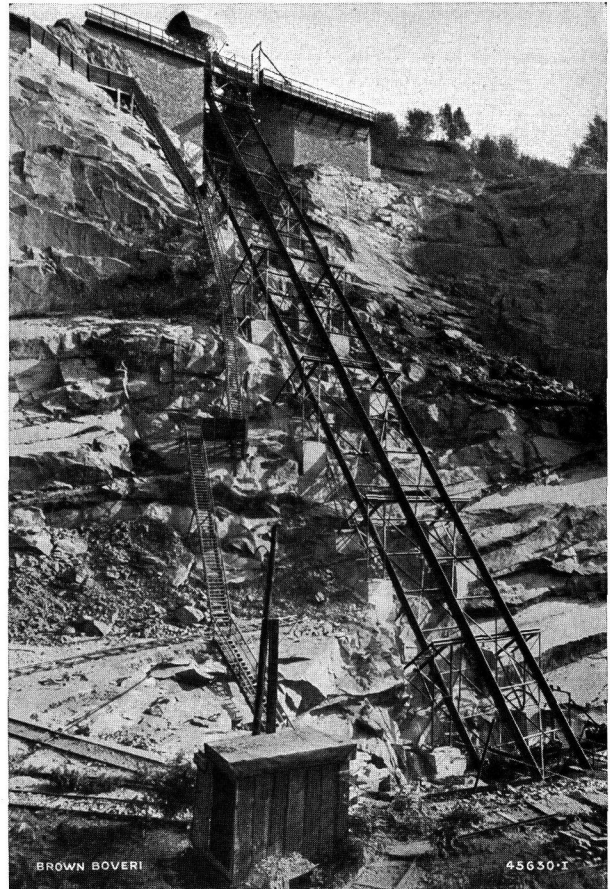


Fig. 1. — Double-acting inclined hoist in a porphyritic stone quarry. It is designed to wind a mean useful load of 2600 kg at 2.4 m/s on two superposed tracks inclined at an angle of 53° and having a length of 67 m (ultimately 90 m).

from the upper station during winds with men. In this case a picked man controls it from the machine house behind the headgear (Fig. 3). For this purpose a controller in the upper station is first switched over to the "men winding" position, and automatically reduces the speed to half of that for material. As soon as the hanger-on at the upper station has given the "ready" signal to the attendant in the winch room the motor can be started. In addition to the automatic control gear, *manual control gear* is also provided, which can likewise be operated from the winch room for certain operating conditions and in case of emergency.

A switch cubicle in the winch room embodies the majority of the apparatus for automatic and manual control, together with the pilot lamps necessary for the supervision of operation. To change over from the normal automatic to manual operation a multi-pole change-over switch must be thrown over. This switches all of the control circuits over to manual operation and cuts out all of the automatic apparatus. It is only

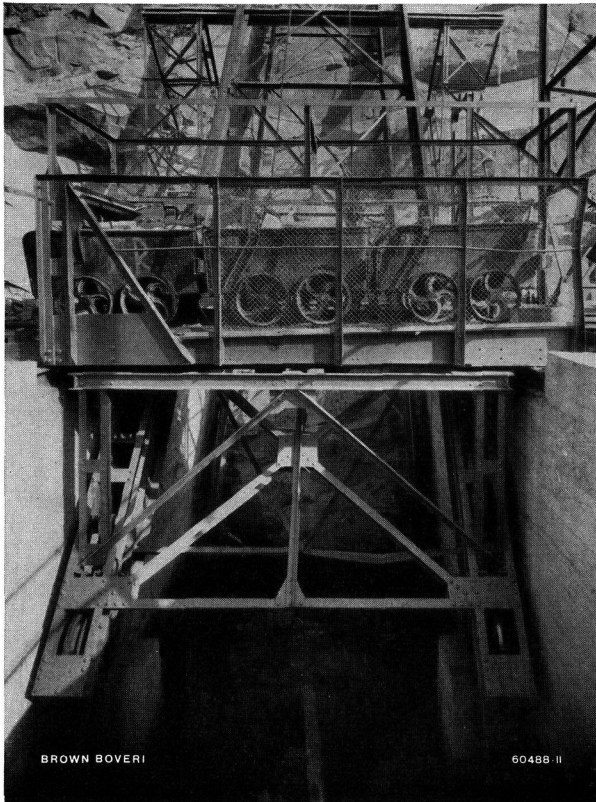


Fig. 2. — Lower terminal station of inclined hoist with two-part truck in end position.

in this position of the change-over switch that its crank can be removed and fitted to the spindle of the controller for manual control. In this way faulty control is out of the question. Moreover, both stations contain boxes of pilot lamps and push-buttons which serve for the supervision of operation and control of the winch, as well as for the release of the emergency brake when necessary.

Great importance was attached to the design of the most vital control unit, i.e. the *cycle switch* (Fig. 4), which determines — together with switches on the depth indicator and on the track — the exact winding times. Each individual cam on this control switch can be shaped and adjusted to ensure the winding cycle

being carried out under the most favourable conditions. The oil-immersed drive, together with the special design of the contacts, makes for maximum reliability and long life. Overhauls are only necessary at lengthy intervals.

A Brown Boveri *electro-hydraulic thruster* (Fig. 5) is employed to release the blocking brake. When the limit switch cuts out the motor of the thruster at the end of a wind the blocking brake is applied in the shortest possible time by the brake weight without shocks or oscillations. The damping action of the thruster is adjusted so as to give the most favourable braking characteristic. Due to its smooth operation the thruster plays a big part in the satisfactory accomplishment of the winding cycle and saves wear and tear on the plant.

It is extremely important that the plant should be brought to rest as quickly as possible whenever disturbances occur during automatic operation. The *quick-acting free-fall safety brake* (Fig. 6), developed and employed by Brown Boveri for large winders for many years past, is particularly suitable for this purpose. This is a pure weight brake with the drop-weight suspended from a conical plate and supported on rollers held closed by a single-phase retaining coil. Upon operation of any one of the numerous safety devices — which protect against all conceivable sources of danger — the circuit of the retaining coil is inter-



Fig. 3. — Upper terminal station of hoist and machine house with winch.
The machine house contains the necessary equipment for automatic and manual control.

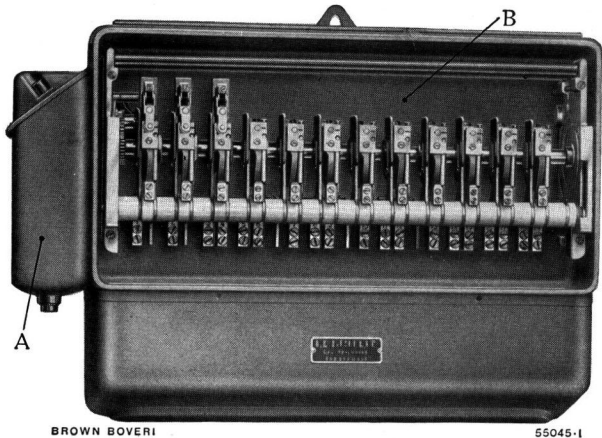


Fig. 4. — View of interior of cycle switch after removal of door of casing.

This is the brain of the automatic control and ensures the correct sequence of the different switching operations. This robust switch greatly simplifies the control and enhances reliability. In the present case the cycle switch operates the blocking brake, the field switches of the winder motor and control generator, and the starting contactors.

A. Light-metal oil tank of auxiliary motor.

B. Cam-operated hammer contacts. These contacts have only to carry the control current of the starting contactors and are therefore subject to very little wear. The cams are in two parts and can be adjusted within wide limits to enable the beginning and duration of the making of contact to be varied. Moreover, the ratio of the cycle drum drive can also be altered. It is therefore possible to adjust every point on the cycle to obtain optimum conditions.

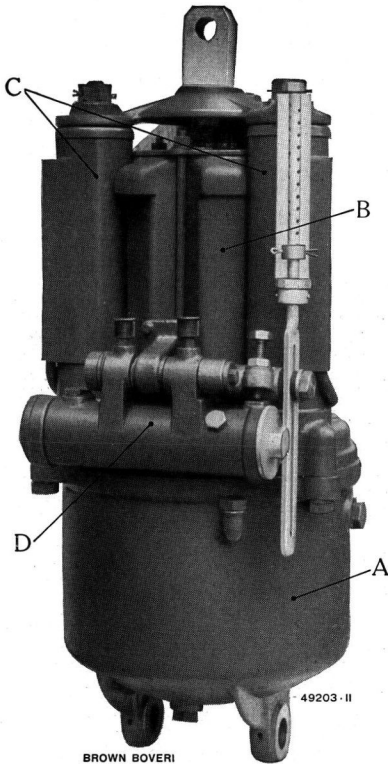


Fig. 5. — Electro-hydraulic thruster with adjustable damping.

The blocking brake is applied by the counter-weight very rapidly and, due to the damping device incorporated in the thruster, very smoothly (see description of method of operation of thruster in Brown Boveri Rev. 1943, pp. 317/318).

A. Oil tank. B. Motor. C. Cylinders. D. Damping device.

be directly connected in its circuit. In case of emergency, therefore, the safety brake trips out without any loss of time.

Due to the excellent results obtained with the foregoing hoist the Company obtained a further order for an automatic equipment for a second inclined hoist for the same quarry. In the case of this second hoist not only is the travelling speed practically twice that of the first, but two intermediate stations are provided in addition to the two terminal stations. Material can likewise be conveyed from these to the upper station. This

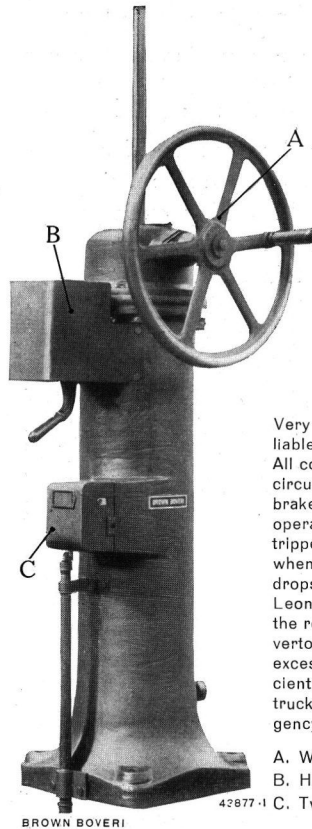


Fig. 6. — Brown Boveri quick-acting free-fall safety brake for a maximum weight of 1000 kg.

Very robust construction, absolutely reliable, rapid, and virtually shock-free action. All contacts of the safety devices lie in the circuit of the holding coil mounted on the brake. In the present case the safety brake operates as soon as the main switch is tripped (over-current or no-voltage), further when the voltage in the auxiliary systems drops too low or the current in the Ward-Leonard circuit rises too high; also when the revolutions of the Ward-Leonard converter or the travelling speed increase excessively, the retardation is not sufficient just before the end of the wind, the trucks overrun the stations, or the emergency brake push-buttons are operated.

A. Winch.

B. Holding coil with manual tripping lever.

C. Two-pole auxiliary switch.

hoist with its more exacting requirements (intermediate stations, higher speed) has also proved entirely satisfactory.

Specification of Electrical Part of Inclined Hoist.

Mean useful load	2600 kg
Travelling speed when winding material	2.4 m/s
Travelling speed when winding men	1.2 m/s
Present length of track	67 m
Ultimate length of track	90 m
Diameter of two cable drums	1.6 m

Drive by d. c. motor rated 95 kW, 1460 r. p. m., 410 V, through two-stage gearing.

NEW BROWN BOVERI STARTERS FOR MARINE-TYPE DIRECT-CURRENT MOTORS.

Decimal Index 621.316.717:629.12

Whereas alternating-current is finding ever-increasing application for driving the auxiliaries of large ships, direct-current will probably be retained for small vessels. In consequence Brown Boveri have developed new starters for direct-current auxiliary drives to meet modern requirements. These are briefly described in the following notes.

IN the development of the new marine starters particular importance was attached to obtaining a robust and reliable design incorporating all of the apparatus necessary for the control and protection of a motor, and which would enable any special requirements customers might specify to be readily embodied. In view of the wide power range for which such starters are needed it was not possible to provide the same type of apparatus throughout, face-plate starters being

All of the foregoing apparatus are mounted on a common iron frame which can be readily removed from the case for inspection purposes. This feature also renders the resistances at the back of the case readily accessible.

To start up the motor the handle on the door of the case simply has to be rotated slowly into the running position, where the starting lever becomes latched. It is impossible for the starter to be left inadvertently on the starting notches, since the switch lever is automatically returned to the "off" position by a spring if the handle is released on one of the starting notches.

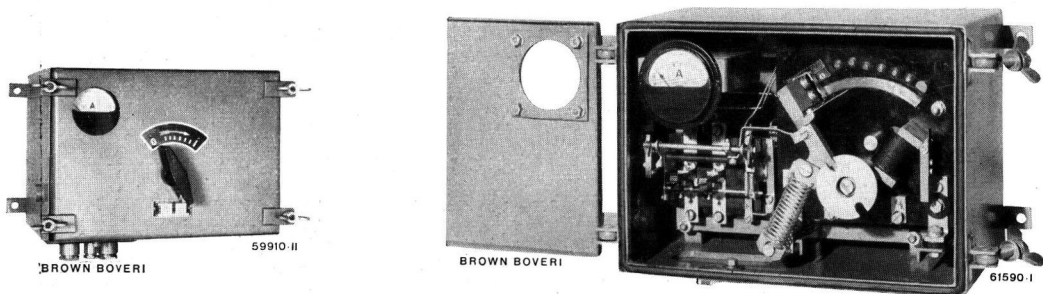


Fig. 1. — Small marine starter type T 4, of compact design, with robust totally-enclosed case for bulkhead mounting.

All marine starters have over-current and low-voltage releases.

more suitable for low and cam-operated starters for high currents. Moreover, a small, simplified type proved necessary for very low motor powers.

Small Starters Type T 4 for Motor Powers up to 5 H. P. under Easy, and 2.5 H. P. under Arduous, Starting Conditions.

These starters are totally-enclosed in a sheet-steel case designed for bulkhead mounting, with door at front and watertight cable glands directed downwards. The case contains:

The starting switch proper for single-pole switching of the motor and gradually short-circuiting the starting resistance.

A device to latch the lever of the starting switch in the running position, together with the retaining coil, which also acts as low-voltage release.

The starting resistances mounted on the back of the contact plate of the starting switch.

A two-pole over-current relay with thermal releases which acts on the retaining coil of the starting switch.

Upon the occurrence of an excessive overload the over-current relay causes an auxiliary contact to interrupt the retaining coil circuit, thus unlatching the starting lever to enable the switch to trip out. The starter is then immediately ready for a fresh start. The retaining coil also trips the switch in a similar manner when the supply voltage drops below a definite value.

To stop the motor the handle must be turned back to the "off" position, thus mechanically unlatching the starting lever.

These small starters can be supplied in the following special designs:—

With ammeter.

With pilot lamp.

For remote release.

With built-in two-pole isolating switch for making the interior of the case dead for inspection purposes and with an interlock which prevents the starter door being opened except when the isolating switch is open.

There is no special design of these small starters for speed regulation by field weakening, since this is rarely called for. In such cases a separate field regulator should be provided or a type Q starter employed.

Starters Type Q with and without Speed Regulating Notches for Motor Ratings up to 75 A.

These starters have hose-proof, ventilated cases of sheet steel for bulkhead mounting, with door at front and watertight cable glands on the underside. The

the front of the case turned step by step into the running position. Depressing the push-button switches in the contactor and keeps it closed during the entire starting operation. The running position is attained after the starting resistances have been gradually short-circuited. A mechanical interlock prevents the starting lever or handwheel being turned backwards on the starting notches, i. e. the motor being switched out over these notches. Should a faulty start involve re-starting the motor before the starting operation is completed, all that is necessary is to let go the push-

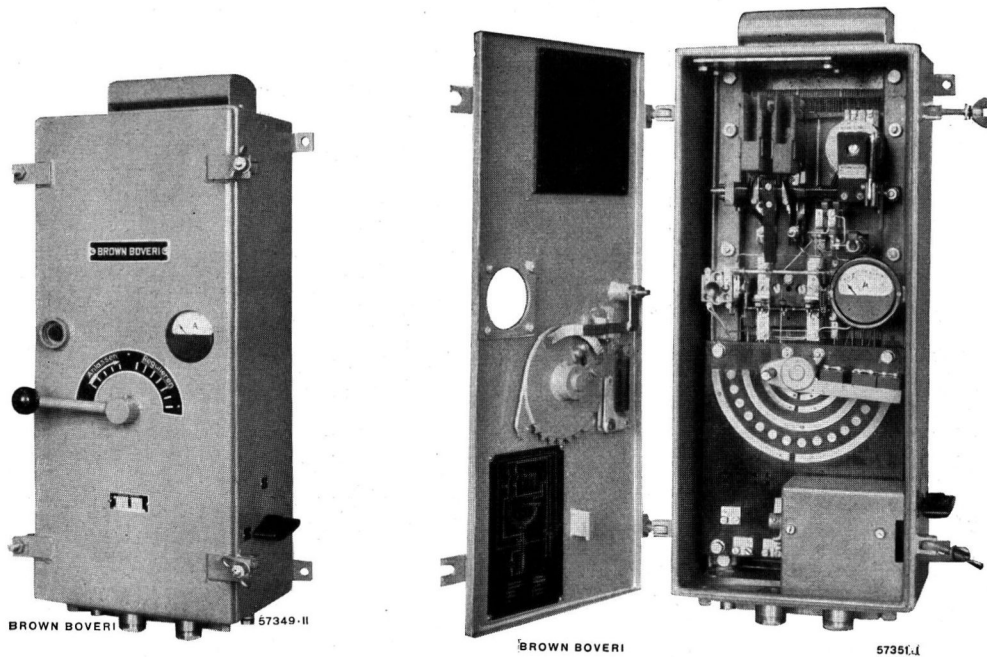


Fig. 2. — Marine face-plate starter type Q, hose-proof design, for currents up to 75 A.

Two-pole disconnection. Isolating switch under special casing in lower right-hand corner. Field-weakening notches for speed regulation. All the marine starters have apparatus and resistances mounted on a common iron frame, which can be readily removed for inspection.

whole of the equipment inside is also mounted on a frame which can be readily withdrawn from the case. In the basic design of the starter this comprises:—

The starting switch for cutting out the starting resistances step by step.

The starting resistances lodged in the back of the case.

A two-pole contactor with powerful blow-out coil for switching in and out the motor and also acting as low-voltage release.

An over-current relay with two thermal releases and an auxiliary contact for interrupting the contactor coil circuit when the motor is excessively overloaded.

A push-button switch for closing the contactor.

To start the motor the push-button merely has to be depressed and the starting lever or handwheel on

button, whereupon the contactor cuts out the motor. The starting lever must then be returned to the starting position for the fresh start. The push-button switch, which has to be actuated during the starting operation and is not short-circuited by an auxiliary contact until the running position is reached, prevents inadvertent operation on the starting notches and thus destruction of the starting resistances.

Should the motor be excessively overloaded in service the over-current relay functions and the contactor cuts out the motor. Likewise, should the supply voltage drop below a certain value the contactor opens and stops the motor.

The motor is normally stopped by returning the starting lever or handwheel to the "off" position. Immediately the running position is left the contactor cuts out the motor.

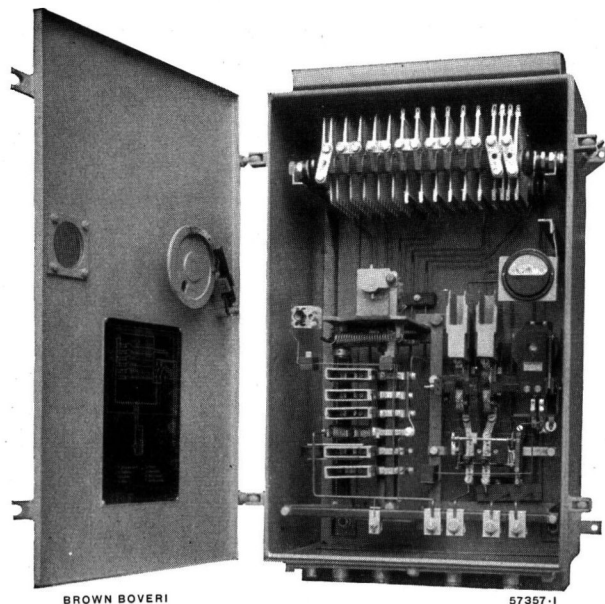


Fig. 3. — Marine starter type R with cam-operated switch units for currents up to 400 A.

Two-pole disconnection. Such starters can also be supplied with series regulation for currents up to 150 A.

As will be clear from the foregoing, faulty operation of the starters is out of the question. Since all power operations are carried out by the contactor the starting contacts are subject to very little wear and maintenance is thus reduced to a minimum.

The type Q starters can be supplied in the following special designs:—

With ammeter.

With pilot lamp.

With enclosed isolating switch.

For remote release.

With interlock only allowing the door of the case to be opened when the isolating switch is open. As will be seen from Fig. 2 this isolating switch is fitted in the right-hand lower corner of the starter case and therefore does not increase its dimensions. The isolating switch is operated by a handle on the side of the case. If desired an additional interlock can be provided which prevents the isolating switch being closed when the door is open.

When speed regulation is required a number of field weakening notches are provided on the starting switch.

Starters Type R with and without Speed Regulation for Motor Ratings up to 400 A.

As regards construction and mode of operation of these starters the same remarks apply as for the type Q starters with the exception of the starting switch for which, in view of the higher current, a cam-operated controller is employed instead of a face-plate switch. This controller has switch units with hammer contacts operated by a vertical camshaft, and readily removable arc chambers. It is operated through bevel gearing by a switch lever or handwheel arranged on the door of

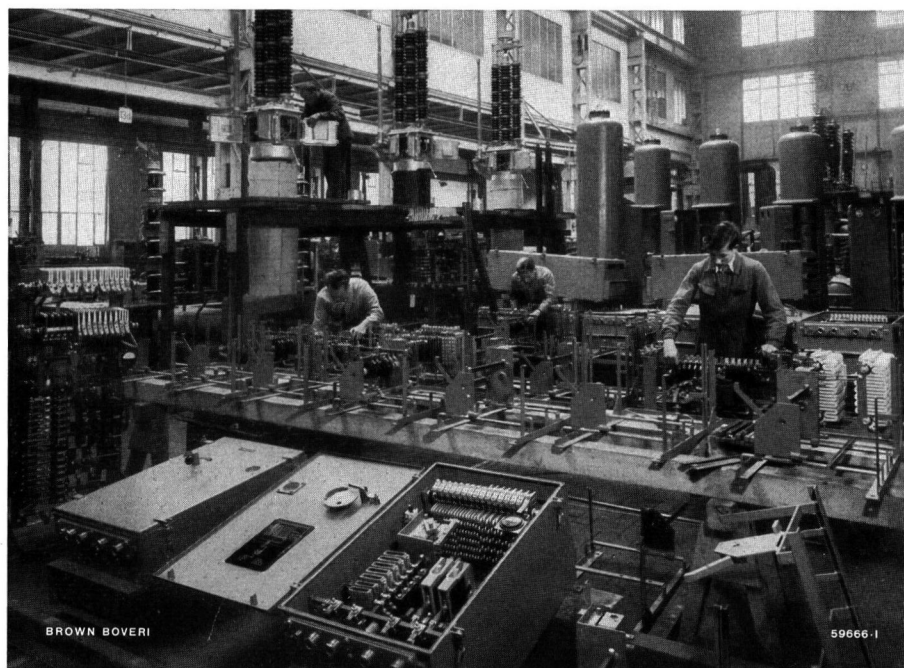


Fig. 4. — Batch of Brown Boveri marine starters under manufacture for stock without resistance elements and coils to permit of supply at short notice.

the case. The starting resistances are arranged here in the upper part of the case instead of at the rear, and are thus readily accessible from the front.

These starters can be supplied in the same special designs as the type Q starters. In cases where speed regulation is necessary a number of switch units for field regulation is added to the controller.

Apart from the foregoing starters

Reversing Starters with Series Regulation, Type R, for Motor Ratings up to 150 A

have been developed for motors driving Diesel barring gear, capstans, etc. These reversing starters differ from the type R starters in that a special switch unit is provided on the shaft of the controller instead of the

push-button switch. In contradistinction to the other starters the starting notches here are employed for temporary operation at reduced speed.

These new starters meet all the requirements of marine service (operation in inclined positions and under conditions of vibration, special insulation, rust-proof finish) and conform to the specifications of the classification societies. Over 250 have been supplied since the end of 1942. They have met with a good reception and proved extremely satisfactory.

In conclusion, it might further be mentioned that Brown Boveri have a special range of apparatus for plants such as are met with on warships, where small dimensions and low weight are of paramount importance.

(MS 575)

Th. Geiger. (E. G. W.)

BRIEF BUT INTERESTING

Centrifugal Blower for Yeast Production.

Decimal Index 621.515:663.132

FOR the cultivation of yeast fungi, the fermenting vat has to be intensively ventilated with fresh air in order to thoroughly mix the basic material. It is essential that the supply of fresh air should be continuous, i. e. not pulsating, and entirely free from oil. For this reason centrifugal blowers, which completely meet these require-

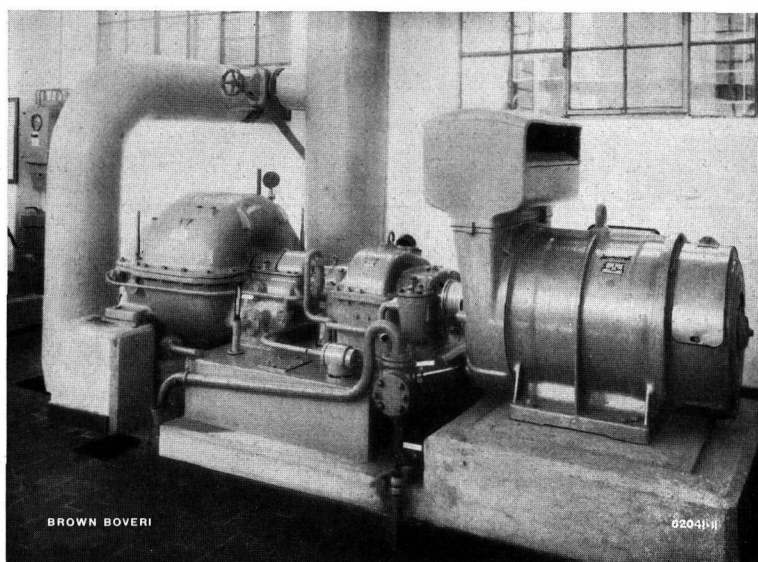
ments, are practically exclusively employed for the production of yeast from the most varied basic materials.

Brown Boveri have already supplied several such blower plants.

The accompanying illustration shows a set of this type. The two-stage blower delivers 7000 m³ of air per hour at a pressure of 1.7 kg/cm² abs. The set in question has been in day and night service for nearly twelve months.

(MS 610)

G. Krapf. (E. G. W.)



Motor-driven centrifugal blower set supplying fresh air for the production of edible yeasts from sulphite lye in fermenting vats at Attisholz Cellulose Works (Switzerland). Air intake 7000 m³/h, air delivery pressure 1.7 kg/cm² abs.

Excellent Operating Experience with the 2D₀2 Locomotives of the French National Railways E 501 Series.

Decimal Index 621.335.2 (44)

IN the early twenties the then "Cie du Chemin de fer de Paris à Orléans" (P. O.) raised a loan in Switzerland, the greater part of which was intended to cover the cost of the electrification of the Paris-Vierzon-Brive line. Just before the opening of subscriptions one of the co-founders and at the time chairman of the board of governors of the Company, Dr. W. Boveri, succeeded in having incorporated in the conditions of the loan a clause to the effect that two express locomotives were to be ordered entirely in Switzerland. As a result, Brown Boveri, together with the Swiss Locomotive and Machine Works, Winterthur, were entrusted with the construction of two express locomotives with wheel arrangement 2D₀2. These were to bear the service numbers E 501 and E 502 and to be designed for direct current at 1500 V. The one-hour rating was 4000 H.P. and the maximum running speed 130 km/h (Fig.1).¹ Dr. Boveri considered these

locomotives to have very good business prospects on account of their excellent design, and gave special instructions that all work was to be carried out with the greatest care. Unfortunately, he was not to live to see the completion and commissioning of the locomotives, the chief feature of which was the Brown Boveri individual axle drive arranged on either side. His expectations, however,

were fully corroborated, since the two locomotives were so satisfactory from the very beginning that they not only formed a prototype for the standard express locomotives of the amalgamated "P. O." and "Midi" Companies in the "Région Sud-Ouest" of the French National Railways, but were adopted by the former "Chemin de Fer de l'Etat" (now "Région de l'Ouest" of the French National Railways) as express locomotives for the electrification of the Paris-Le Mans line. To-day, about one hundred locomotives of this type are in service in these two districts of the French National Railways, which, apart from the

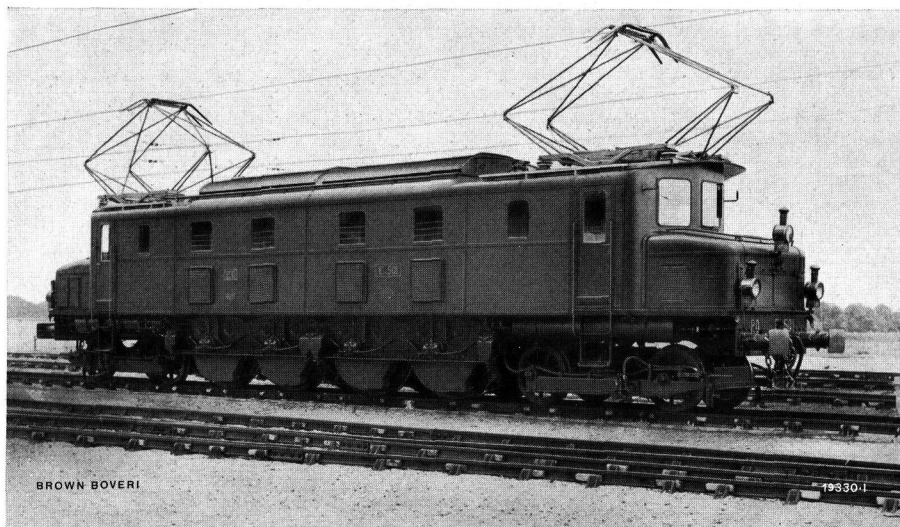


Fig. 1. — Express locomotive, wheel arrangement 2D₀2, series 501 of the former "Cie du Chemin de fer de Paris à Orléans" (now "Région Sud-Ouest" of the French National Railways), for 1500 V d. c., one-hour rating 4000 H. P., maximum speed 130 km/h. (Year of construction 1925.)

first two locomotives, were all built by the Brown Boveri affiliated firm, Compagnie Electro-Mécanique, Paris, or their sub-contractors, the Cie Fives-Lille. It is only in external appearance and in details of the electrical equipment and mechanical part that any changes have been incorporated in the locomotives built in France with the passing of the years (Figs. 2 and 3). In general, however, they are identical with the first two locomotives designed and built by Brown Boveri in Switzerland.

¹ See F. Gubler, Brown Boveri Rev. 1927, p. 197.

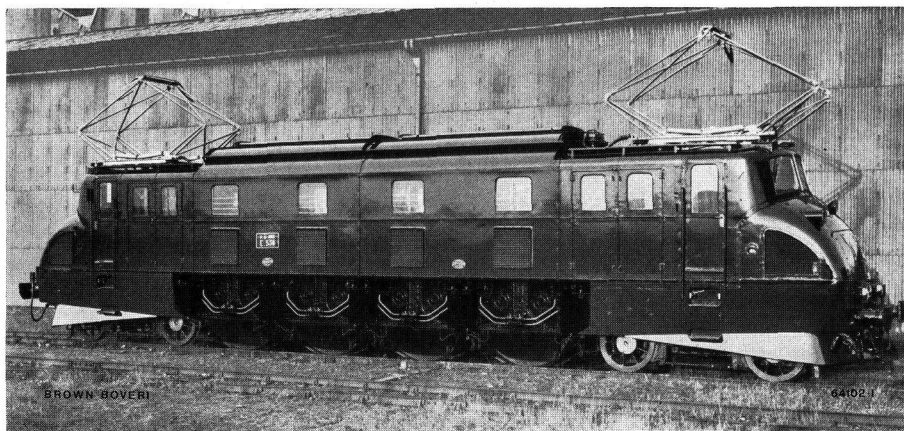


Fig. 2. — Express locomotive, wheel arrangement 2D₀2, series 528 of the former "Cie du Chemin de fer P. O. -Midi" (now "Région Sud-Ouest" of the French National Railways), for 1500 V d. c., one-hour rating 4200 H. P., maximum speed 130 km/h.

(Year of construction 1937.)

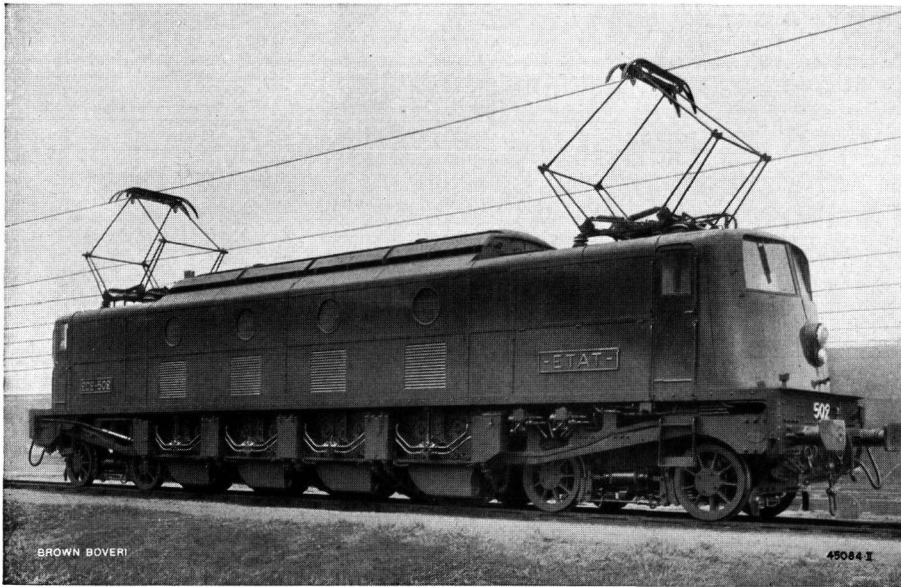


Fig. 3. — Express locomotive, wheel arrangement 2 D₀ 2, series E 501 of the former "Chemin de fer de l'Etat" (now "Région de l'Ouest" of the French National Railways) for 1500 V d. c., one-hour rating 4200 H. P., maximum speed 130 km/h. (Year of construction 1937.)

These excellent operating results have induced the French National Railways to order thirty-five further locomotives of the same type, i.e. with the Brown Boveri individual axle drive, for the extension of electrical operation in the "Région Sud-Est" (formerly P. L. M.). It is true that technical developments in the interim have led to simpler driving arrangements, with sprung discs taking up the relative

From a list of defects which occurred on fifty locomotives on the system of the "Région Sud-Ouest" of the French National Railways between 1928 and 1943 the following is of interest:—

Twenty-two defects occurred on 200 traction motors up to March, 1943, viz.:—

- (a) Motor armatures: Due to ingress of snow 6 defects
- (b) Field coil connections:
Due to faulty workmanship in connection with certain parts, only on locomotives Nos. 528—37 13 defects
- (c) Leading-out cables 1 defect
- (d) Brush-holders 2 defects

This represents only 0.1 defect per 1,000,000 locomotive-km and motor.

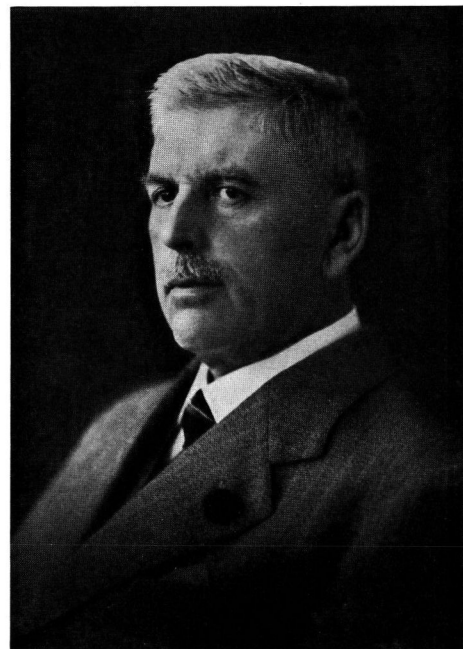
Very few defects occurred on the fifty apparatus equipments, some being due to poor quality insulation during the war and to the ingress of snow. Referred to kilometres run by the locomotives the following are the figures:—

Loco. Nos. 503—527	0.65 defect per 1,000,000 loco./km
" " 528—537	0.11 " " 1,000,000 " "
" " 538—545	2.15 defects " 1,000,000 " "

The locomotives were given a general overhaul as soon as the tyres of the driving wheels had worn down to the maximum admissible value, i.e. after each locomotive had covered approximately 170,000 km. Certain locomotives were run for distances up to 265,000 km before being given a general overhaul.

The two locomotives E 501 and E 502 manufactured in Switzerland had covered between them 3,850,000 km up to January, 1942. This corresponds to fifty times round the world for each locomotive. The forty-two further locomotives Nos. 503—545 had covered an aggregate of approximately 50,000,000 km by the same date, i.e. on the average about 1,200,000 km each, while the twenty-three locomotives of the "Région de l'Ouest" (Fig. 3) which were supplied later, already have 17.4 million km to their credit. These figures speak for themselves. As far as can be gathered no trouble whatever has been experienced with the individual axle drives.

movements of the sprung frame and the unsprung sets of wheels², but experience in France proves once again that, notwithstanding its complexity, the Brown Boveri



Dr. ing. h. c. Jakob Buchli
1876—1945

The inventor of the Brown Boveri individual axle drive.

individual drive was and still is an ingenious arrangement. This may well be emphasized in honour of its recently deceased inventor, *Dr. ing. h. c. Jakob Buchli*, one of the most original and versatile designers the Swiss railway machine building industry has known to date.³

(MS 643)

K. Sachs. (E. G. W.)

¹ Built during war.

² H. Rüegg, Brown Boveri Rev. 1940, p. 203.

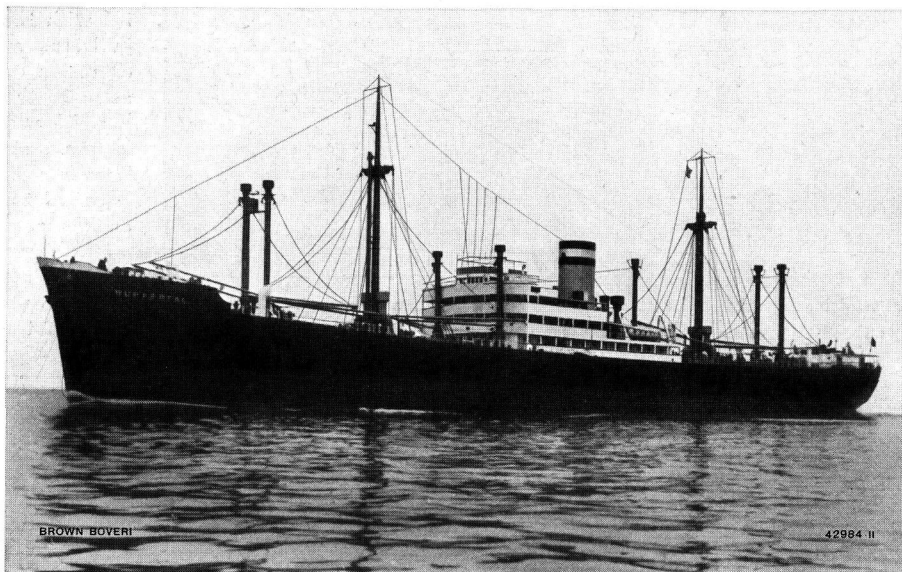
³ See Schweiz. Bauzeitung 1945, I, p. 246.

Seven Years' Experience with Alternating Current Transmission.

Decimal Index 629.123.4-833.6

THE September 1943 number of "The Motor Ship" contains a short article under the above heading on the 6800 S.H.P. cargo vessel "Wuppertal" of the Hamburg-America Line, which happened to be in Australia upon the outbreak of hostilities. According to this publication, there has been an entire absence of trouble with the electrical installation (which was supplied by Brown,

Moreover, a number of recent reports in the marine press corroborate the view that high-speed Diesel engines, either with geared or electrical transmission, will find ever-increasing application in future. In the issue of "The Motor Ship" already referred to, it is also stated that alternating-current electric drive with generators operating in parallel has been used in the U. S. A. during the war and that the results are most gratifying. As is well known, Brown Boveri were the first to propose the alternating-current Diesel-electric drive with generators operating in parallel, while the first practical



The cargo vessel "Wuppertal" of the Hamburg-America Line with Brown Boveri Diesel-electric a.c. drive.

Notwithstanding the fact that the propelling power is split up between three Diesel-alternator sets operating in parallel, control operations are extremely easy to perform and supervise, so that any engineer can quickly learn how to get the vessel under weigh and stop and reverse it. According to the marine press the propelling machinery has given every satisfaction during the seven years it has been in service. All of the auxiliaries below deck, together with the lighting and heating systems, are connected to the a.c. auxiliary system which is supplied by the main Diesel generators when the vessel is under weigh.

Boveri & Cie., A.-G., Mannheim), apart from damage to the propelling motor caused by flooding. This, of course, was not a mishap associated with the propulsion system, and the motor was dried out and replaced in service. The "Wuppertal", which has been renamed "Noesanivi", has been operating with continued success and efficiency in charge of a Dutch crew since 1940.

This report refutes the allegation sometimes made that Diesel-electric drive is too complicated for cargo vessels and involves a particularly skilled crew. If such were the case the ship could not have been taken over so quickly after the repairs, and kept in regular service by a foreign crew. In any case the number of engineers on the vessel is not greater than on others of the same size, tonnage, and speed with direct Diesel drive.¹

details of such ship's propelling plants, based on exhaustive tests were published as early as 1932². Following on this preliminary work the propelling machinery for the "Wuppertal" was built. The advantages of this type of marine drive are set out in the Brown Boveri Review for September/October, 1942 (special marine number) in the article entitled "The Diesel-electric Three-phase Propulsion of High-power Ships".

(MS 573)

Th. Egg. (E. G. W.)

¹ See page 84 of the "Shipbuilding and Shipping Record" of the 27th January, 1944.

² E. Klingelfuss: "Brown Boveri Diesel-electric Ship Propulsion with Alternating Current". Brown Boveri Rev. 1932, pp. 148-154.