THE MECHANICAL DESIGN AND SPECIFICATION OF THE TURBO-ALTERNATOR ROTOR.

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The objects of this paper are to review the conditions underlying the mechanical design of the high-speed alternator rotor, and to suggest a specification for the guidance of the purchaser's technical adviser.

The history of the development of the alternator for direct coupling to high-speed steam turbines is very similar to that of most machinery. The old construction has been applied to meet the new conditions, and, as failures have developed, improvements have been introduced to overcome them. The process is lengthy and wasteful, but essentially natural.

The outstanding feature of the earlier electrical machinery was the disregard of mechanical soundness. Reliance was placed on string and tape in an astonishing degree. Nothing has led to the development of electrical machinery from the mechanical point of view more than making alternators suitable for direct coupling to high-speed steam turbines. With the advent of peripheral speeds of over $4\frac{1}{2}$ miles per minute the unsound construction of earlier times has disappeared, and the modern turbo-alternator rotor is not surpassed as a high-class engineering production. The need for the use of material of the highest grade is obvious when it is recalled that at the periphery of a 31-in. diameter rotor running at 3,600 r.p.m. a mass of 1 lb. exerts a centrifugal force of $2\frac{1}{2}$ tons, and in the case of a 5,000-kw. alternator at 3,000 r.p.m. the total centrifugal force at the normal running speed is about 15,000 tons. The kinetic energy of such a rotor at the normal running speed is about 7,000 foot-tons—equal to the muzzle energy of 23 75-mm. field guns and sufficient to hurl 380 lb. of steel a distance of 6 miles. It is clear that to eliminate weakness of construction by the lessons of failure is not a permissible policy with such machinery. The precise nature of the duty at the various parts of the rotor must be thoroughly investigated, the material employed must be of the highest grade, and the factor of safety must have a satisfactory value.

The conditions to be faced are most severe, and the author doubts if they are sufficiently well realized by either user or maker. Even with the smaller sizes of high-speed machines that marked the progress of a few years ago, mechanical failures were not unknown, and the author is of the opinion that with the machines for large outputs at from 1,500 to 3,600 r.p.m., such as are being made to-day, mechanical failures must result unless greater attention is paid to the excellence of design and of material.

It is somewhat remarkable that the specifications drawn up by consulting engineers make little attempt

to ensure mechanical excellence, although the electrical performance is usually very fully stipulated. Occasionally one notices in specifications that the engineer has some uneasiness with regard to the mechanical performance and endeavours to safeguard his client's interests by specifying the tensile qualities of the rotor steel and by calling for an unusually high overspeed test, but one never sees stipulated the all-essential permissible ratio between the stress and the quality of the material *in the direction of the stress*. For instance, it is not sufficient to test the tensile qualities of the steel in an axial direction at the end of the shaft when the maximum stress is in the body of the rotor in a radial direction; nor is it logical to display concern in the quality of the material and disregard the complementary factor—intensity of stress. It will be shown later that an excessive overspeed test is on the one hand undesirable, and on the other does not necessarily establish mechanical soundness.

COMPARISON OF THE SOLID AND THE LAMINATED ROTOR.

Even up to a few years ago the controversy as to the relative merits of the salient-pole rotor and of the cylindrical rotor still flourished, but with the advent of large outputs at high speeds the former type has now disappeared. There still remains, however, some discussion as to the advantages of the solid as compared with the laminated rotor. From the electrical point of view, for polyphase work there is no essential difference. For a given design theoretical considerations would indicate that the maximum momentary short-circuit current would be less with the laminated than with the solid rotor, for the reason that there is a greater resistance to the flow of the rotor eddy-currents at the moment of short-circuit, on the creation of which depends the momentary loss of the stator self-induction. Since, however, the alternator with a solid rotor can quite well be designed to stand short-circuit at the terminals without damage, this property of the laminated rotor is without practical value. On the other hand, the better electrical contact that is established between the slot wedges with the solid rotor causes the rotor to constitute a more effective damping winding and so gives greater electrical stability. With an unbalanced load there is less heating with a laminated than with a solid core, but again this quality is usually of no advantage, since by exercising certain obvious precautions the solid rotor can be employed successfully with any degree of unbalanced load that occurs in practice. In a normal turbo-alternator the output is limited by the rotor heating,

and it is obviously a great advantage of the solid rotor that the ventilating channels can be arranged to secure the best cooling effect, whereas with the laminated rotor they have to be provided where mechanical considerations permit. The essential weakness of the laminated rotor is the presence of the central hole through which the shaft passes. On account of the thickness of the metal between the bottom of the slot and the inner bore being comparable in magnitude with the diameter of the bore, the maximum stress exceeds considerably the average stress, and the design becomes limited by the stress at the inner periphery. As an example, with a 3,500-kw. alternator at 3,000 r.p.m. the maximum stress on the inner fibres exceeds the average stress across the belt of metal by about 20 per cent. In addition to the stress caused by the centrifugal force, there is to be considered the stress due to the shrinkage fit of the stampings on the shaft. To ensure that the core shall remain tight on the shaft when running at the overspeed, the amount of the shrinkage must exceed the amount by which the core opens due to the centrifugal force. Thus, in the case being considered, the shaft diameter would be about 12 inches, and at the overspeed of 3,600 r.p.m. the inner bore would increase about 0.018 inch in diameter. Making allowance for normal tolerances in workmanship, the shrinkage would need to be not less than 0.02 inch to ensure that the core shall always remain tight on the shaft. This requirement for mechanical soundness gives an appreciable increase to the stress in the core discs. Just at this part of the core disc where the stress is a maximum the strength is very considerably reduced by the keyway cut into the lamination. The weakness could be avoided if instead of cutting the keyway in the disc a projecting tongue could be formed to register in a keyway in the rotor shaft. Such an arrangement is not practicable, because the tongue would prevent the inner bore of the core being machined. To give the essential good fit of the core on the shaft, the core should be bolted solidly together between stout end plates and be bored to a smooth surface, and the shaft then be ground to suit with the necessary shrinkage allowance. Apart from being desirable to ensure a good fit, it is vitally important to bore out the core in order to remove the metal that is strained and weakened by the punch. This matter will be referred to later.

The solid rotor has the great advantage that the bursting stress is always less than the stress in the tooth, and the former stress has therefore never to be considered. A comparison of the hoop stress for a laminated and a solid rotor is shown in Fig. 1 for a 3,750-kw. alternator. It will be seen that even with a much deeper tooth the maximum stress in the forged rotor is only 13,800 lb., against 34,500 lb. in the laminations.

Another matter to receive consideration when discussing the laminated rotor is the question of the material available for the core discs. Such material is prepared by rolling down hot slabs into sheets of the thickness required. The structure of the steel is influenced very considerably by the finishing temperature of rolling and by the rate of cooling. With a

large, thin sheet it is impracticable to maintain a uniform temperature throughout, and a micro-examination after rolling shows appreciable variations in the crystalline structure at different parts of the sheet. Careful annealing, by the re-crystallization that results, does much to remove such variations, but no matter what care the manufacturer may exercise, some variations in the finished sheet are unavoidable, and particularly so when thorough annealing is not permissible on account of the lowering of the tensile strength. Moreover, the fibre is determined by the direction of rolling; the effect of the final passes predominates, although the earlier passes have a slight permanent effect. Even at a given part of a given sheet the ductility is found to vary somewhat according to the direction chosen, although the tensile strength may be fairly uniform. Minute local imperfections that would be without effect on a fairly thick plate may appreciably weaken a thin plate. Since material of this kind is inherently less dependable than a high-class forging, the factor of safety should be higher than with the solid rotor, but due to the weakening effect of the central hole the designer has to be content with a lower factor of safety.

To meet the demands of the electrical engineer for steel sheets having a high tensile strength, some manufacturers have taken advantage of the additional strength given by cold working. An interesting theory of the hardening of metals by cold working has been put forward by Sir George Beilby.* By making the final passes through the rolls at a low temperature the tensile strength is considerably increased. The action is to deform the crystals and to produce amorphous material on the surfaces of the gliding planes, resulting in appreciable hardening. Up to a certain point the more the steel is reduced in area the more the strength is increased in this way, but beyond that point the steel is rapidly weakened. The effect of annealing such material is to remove largely the effects of the cold working, by the re-crystallization of the amorphous material that results. The elongation may be brought to a low value according to the extent of cold working—even to less than 1 per cent. Under a tensile test such material breaks abruptly without any appreciable stretching. With the steel sheets that are prepared in this way for rotors, the cold working is not carried so far, and there is usually an elongation of about 5 per cent with a yield point of about 35 tons per square inch and a breaking strength of about 40 tons per square inch. The effect of the cold working with large thin sheets is essentially somewhat uncertain, and such material cannot be regarded as being of a high grade. There is always the chance of the process being carried too far locally and of minute internal flaws being caused. Another danger of material worked in this way and only partly annealed is the possibility of there being what is known as "rectangular brittleness." This remarkable phenomenon is more usually encountered with pure iron and very mild steel, but the author has had experience of it

* *Philosophical Magazine*, 1904, vol. 8, p. 258; *Proceedings of the Royal Society*, vol. 79 A, p. 463; *Journal of the Institute of Metals*, 1911, vol. 6, p. 5.

occurring in some degree with steel of the kind in question. It has been discovered that if steel is worked at a temperature below 500°C ., and is then annealed at a temperature below the critical point, it is liable to become coarsely crystalline. It was shown by an interesting experiment by Professor Sauveur, and independently by Mr. H. S. Kipling, that in addition to the critical temperature there is also a critical range of deformation, and if the steel is worked either more or less the phenomenon does not occur. Dr. Stead* found that the property he designated "rectangular brittleness" is due to the material crystallizing with the cleavages at an angle of 45° to the direction of rolling and at 90° to the surface of the sheet. It is obvious that if the crystals are all similarly oriented

FIG. 1.—3,750-kw. 3,000-r.p.m. alternator with 27-in. diam. rotor. Curves showing the much lower hoop stress in a solid as compared with a laminated rotor of the same diameter. Stresses calculated at 20 per cent above normal speed.

the sheet would be readily fractured if the stress were applied at right angles to any of the three cleavages, although it remains tough in directions at 45° to the cleavage planes. Annealing at a temperature above the critical point would remove this weakness, but it would at the same time largely remove the effect of the cold working and very considerably lower the tensile strength.

Reference has already been made to the weakening effect of punching metal sheets. If a micro-section of the sheet is made it is clearly seen that the effect of punching the metal or, in a slightly less degree, of cutting it with shears is to distort and harden the crystals. Metal that was quite ductile before punching

is found to be brittle round the punched edge. It is most important, therefore, that the core should be assembled in a jig and be bored out, since otherwise there would be brittleness and weakness at the part subjected to the greatest intensity of stress.

In place of cold working another method of producing plates of high tensile strength is to employ a high carbon content. The ductility of such steel, however, is not good, and it cannot be regarded as being suitable for the duty. An entirely satisfactory high-tensile plate could be produced using nickel-chrome steel, but a rotor constructed with such plates would be more expensive than a solid forged carbon steel rotor, and would be less satisfactory mechanically and electrically.

Another point to receive consideration with the laminated rotor is the question of the critical speed. Practically always, the running speed is above the first critical speed, and with large outputs at 3,000 r.p.m. the running speed may be above the second critical speed. If the laminations were fairly loose in the axial direction the critical speed of the shaft could be accurately determined, but for mechanical soundness it is desirable to assemble the laminations under heavy end pressure. Additional stiffness may be given to the shaft, the deflection being reduced and the critical speed raised. The stiffening effect is usually not permanent: under the combined action of vibration, stress and pressure, the minute undulations in the laminations are removed, and in the course of time the end pressure which might have been initially 200 tons is found to have disappeared. In the case of a typical 3,750-kw. alternator at 3,000 r.p.m., with a shaft diameter of 12 inches, a core diameter of 27 inches, a core length of 50 inches, and a distance between bearing centres of 134 inches, the first critical speed with the laminations absolutely solid would be about 2,000 r.p.m. and with the laminations free would be only about 1,400 r.p.m. It will be seen that with a newly constructed rotor the critical speed may have a satisfactory value, but after some running there may be a lowering of the critical speed and some vibration may result. The author has had experience of several cases of slight incurable vibration with laminated rotors that has developed after prolonged running, but in only one case was the vibration so bad that the rotor had to be reconstructed.

When high-grade forgings are available the solid forging is undoubtedly preferable for all sizes of rotors for polyphase work. For small outputs the laminated rotor can be made satisfactory, but to employ the construction for large outputs is to take unnecessary risks. It is the author's opinion that for outputs in excess of 3,000 kw. at 3,000 r.p.m. the laminated construction should be employed only when it is impossible to obtain suitable forgings.

For large-diameter rotors at medium speeds thick plates are sometimes employed instead of laminations. This construction is preferable to using thin sheets, but the weakness of the centre hole and the high hoop stress, together with the unsuitability of using rolled plate with a pronounced fibre subjected to high stress in all directions, are still present, and when the critical

* *Journal of the Iron and Steel Institute*, vol. 53, p. 180.

speed is below the running speed, trouble with it is equally if not more pronounced.

Another alternative is the construction of using plates of considerable thickness held together with through bolts with a short shaft length at either end and no central hole. The great weakness of this construction is the difficulty of making the bolts satisfactory. The deflection is due almost entirely to the elongation of the bolts, the area of the plates being large compared with the area of the bolts and the compression of the plates therefore negligible. A very slight elongation of the bolts takes all compression off the plates, and consequently at the underside of the rotor the plates are out of contact. Each time the rotor turns round, stress in addition to the initial stress is thrown on the bolts. In the case of a 1,500-r.p.m. rotor this intermittent superimposed stress is applied and removed over two million times a day. It is claimed for the construction that the variation of stress can be maintained at so low a value that in spite of the rapid variation the duty is within the capacity of the steel. It is to be remembered, however, that the calculated stress may be considerably exceeded. In the first instance, it is difficult to know exactly the initial stress, which is usually determined either by measuring the pull on the spanner tightening the nuts or by measuring the elongation of the bolts. Again, the action of tightening the nuts may impose a very considerable torsional stress. Also, the driving torque imposes a longitudinal as well as a shear stress on the bolts. Above all, a bolt cannot be regarded as being dependable for carrying a rapidly fluctuating stress. The design usually calls for the bolts to be a reamed fit in the plates, and there is thus a sudden reduction of area at the parts where the thread is cut. The added elongation when the additional load is thrown on the bolts by the action of the rotor turning round is not uniformly distributed over the whole length, but is concentrated to an extent dependent on the reduction of area at the short length between the end of the fitting part and the face of the nut. In the course of time this concentration of elongation in company with the weakening effect of cutting the threads tends to cause failure by fatigue close to the nut. With one case that came to the author's notice of a 6,000-kw. 1,500-r.p.m. alternator constructed in this way, it was found that after $2\frac{1}{2}$ years' running some of the plates at the centre of the core had moved slightly out of centre. On dismantling the rotor it was found that the opening and closing of the plates at each revolution, as described above, had caused fretting of the spigots and between the bolts and the reamed holes, and where initially there had been a good fit there was found to be a clearance on the spigot and in the bolt holes of $\frac{1}{50}$ inch.

Another disadvantage of the plate construction is apparent when the critical speed is considered. Since the bolt area in tension is small compared with the rotor weight, the deflection in the body of the core is considerable and hence the critical speed is very low. It is almost impossible to avoid some degree of synchronism of the normal speed with the harmonics of the natural period of transverse vibration. The effect

of the plates meeting, by the great area of compression they offer, prevents any appreciable swing of the rotor beyond the neutral axis. Vibration is thus damped, and such a rotor has no real critical speed, but there may be sufficient vibration to be appreciable.

Such a construction can be seriously considered only with relatively large and relatively low-speed rotors, and is only to be employed when forgings are unobtainable.

THE MAXIMUM DESIRABLE DIAMETER OF THE SOLID FORGED ROTOR.

The forged rotor having no central hole is very much stronger in its resistance to bursting than is the laminated rotor, as shown in Fig. 1. It can be shown mathematically* that a minutely small hole through the centre of a rotating cylinder doubles the stress. In addition to the gain due to the omission of the hole, there is the greater area of steel to carry the load. The capacity of the added area to carry load is considerably greater than the extra load due to the centrifugal force of the added metal, and so altogether the bursting stress is so low that it has not to be considered.

The maximum stress with the solid forged rotor is always in the tooth, but the stress never reaches a high value, the maximum rotor diameter being determined by other considerations.

With the type of winding now in universal use, of concentric coils of flat copper strip, there is a fairly clearly defined limit to the rotor diameter that may be employed for a given speed. There are to be considered the stability of the end windings exterior to the core, the pressure on the insulation between turns, and the retention of the end windings against the action of centrifugal force. In the case of a 5,000-kw. alternator at 3,000 r.p.m. with a 31-in. diameter rotor, when running at the overspeed of 20 per cent above normal the average value of the maximum pressure on the insulation is about 5,000 lb. per square inch. The desiderata of the insulation between turns are good electrical properties, capacity to stand high temperatures without deterioration, and mechanical strength. The author is very strongly of the opinion that pure, hard mica is the only material that is really suitable for the purpose. Asbestos sheet is sometimes employed, but it cannot be regarded as being a suitable material. The electrical properties of the asbestos may perhaps be made sufficiently good by careful impregnation, and it is, of course, very suitable for resisting heat, but mechanically it is not good. The space available for the rotor copper determines the rotor output and hence the alternator output, and it is obvious that the thickness of the insulation between turns must be reduced to a safe minimum. A thickness of 0.01 inch cannot be exceeded without unduly adding to the size of the machine for a given output. However carefully asbestos may be made, with such a thickness it is not possible to avoid weak spots. The copper is never absolutely flat, and any high point is very liable to crush the asbestos locally and cause short-circuiting between turns. Pure, hard mica is very good mechanically and does not yield locally in this way.

* MORLEY'S "Strength of Materials," art. 126.

Fig. 2 shows a cross-section through one of the rotor coils of such an alternator, and there is drawn to scale on top of it a block of steel equal in weight to the centrifugal force exerted by the coil. When it is considered that the length of each outer coil exterior to the core is about 4 feet, it will be appreciated that the difficulty of preventing any side-slipping of the coils, however well they may be supported, may become a limiting factor of speed. The illustration does not exactly reproduce the running conditions, since the centrifugal force is not uniform throughout the coil as is the pressure due to the weight, but varies from zero at the bottom to a maximum at the top. The sketch nevertheless serves to convey a much better idea of the duty than a mere statement of figures can do. As a structure such a coil is not well adapted to carry a great load without distortion—particularly having regard to the high-frequency vibration that must be present in some

FIG. 2.—5,000-kw. 3,000-r.p.m. alternator with 31-in. diam. rotor. At the overspeed of 20 per cent above normal the maximum pressure on the insulation between the rotor turns is 5,000 lb. per sq. in. In the case of a coil 1 in. wide it is equal to supporting a block of steel 11 ft. square of the same length as the coil.

degree, however perfect the balance. Moreover, the tendency to move due to expansion which may not take place uniformly with all the turns at either end of the core, and to the low coefficient of friction between the polished surfaces of the copper and the mica, adds to the difficulties.

The stability of the coils is much improved as the breadth of the strip is increased. The practice of most makers seems to be to employ rather a large number of coils with a relatively narrow section of copper—six or seven coils per pole being common. A far better arrangement is to use fewer coils of greater copper breadth. Also, reducing the ratio of wound slots to slot pitches is not only better mechanically, but is more effective electrically, as shown by Dr. S. P. Smith.* The limitation in this direction apart from electrical considerations is the section of strip that can be handled conveniently.

Under steady compression between flat surfaces mica can stand great loading before failing. In service, however, it is to be remembered that the conditions are not nearly so ideal. There is always some degree

of high-frequency vibration present and the surface of the copper is never quite flat. For these reasons a big margin of safety is necessary.

Whereas the limitations due to end-coil stability and insulation pressure are indefinite, as will be seen later the limitation due to the difficulty of retaining the rotor end-windings is quite definite. Having regard to these circumstances, the author believes that for a normal speed of 3,000 r.p.m. a 32-in diameter rotor will come to be regarded, if not as the maximum permissible size, at least as the maximum desirable size. The peripheral speed of such a rotor is 25,120 feet per minute. For a given peripheral speed the centrifugal force of a given mass is inversely proportional to the

FIG. 3.—Relationship between running speed and maximum desirable rotor diameter.

radius. With the larger diameters, however, it will be found that with an increase of diameter there is roughly a corresponding increase in the weight of the various parts, and for speeds below 3,000 r.p.m. the desirable peripheral speed is actually reduced somewhat as the diameter is increased. In Fig. 3 is shown the relationship between the angular speed and the maximum desirable rotor diameter.

THE CHOICE OF ROTOR STEEL.

The obvious requirements of the rotor steel are that the permeability should be as high as possible and that mechanically the material should be suitable for the duty. The exciting energy required to drive the flux through the rotor varies considerably according to the relative proportions of the design and the performance of the machine. In an average case, however, it may be of the order of 15 to 20 per cent of the full-load excitation. It is important, therefore, that the permeability should be as high as possible consistent with

* *Journal I.E.E.*, 1911, vol. 47, p. 562.

satisfying the mechanical requirements. As already mentioned, the maximum stress in the rotor is not great, and the designer has apparently a wide choice of steels. The choice, however, becomes rapidly narrowed down when it is stipulated that in addition to having the necessary strength the steel must have capacity to withstand shock and fatigue, be tough, ductile, of the kind that is least likely to have flaws, the best suited to use for a large forging, and above all be dependable in the highest degree. Beyond doubt, the steel that best fulfils these requirements is an oil-treated carbon steel. Such a steel, containing about 0.38 per cent carbon, after oil treatment has an ultimate strength of 38 tons per square inch, a yield point of 22 tons, an elongation of 22 per cent on 2 inches, and a contraction of area of 40 per cent. It is of the class of steel that is used for such purposes as railway axles, and has to resist shock and fatigue and be dependable above everything, and is therefore admirably suitable for alternator rotors. The permeability of this steel is quite good, as shown in Fig. 4.

FIG. 4.—Magnetization curve of medium carbon steel (oil-hardened) for turbo-alternator rotors.

It is usual to express the factor of safety in terms of the ultimate strength—which is not quite logical, since it is the yield of the material that marks the limit of safety and not the actual fracture. The ratio between the yield point and the ultimate strength differs considerably with different classes of steel, and varies a little even with any particular steel according to the nature of the heat treatment. For example, with a mild carbon steel in the annealed state the yield point may be about 50 per cent of the ultimate strength, and after oil hardening may be increased to 60 per cent. With a high-tensile nickel-chrome steel the ratio may be as high as 85 per cent, and with cold-drawn steel wire as high as 95 per cent. For this reason the author is of the opinion that the factor of safety should always be expressed in terms of the yield point. It would be better still to express it in terms of the elastic limit, but the difficulty is that the elastic limit cannot be determined in a tensile test and can really only be found by a long series of tests under reversals of stress. The yield point is defined

as being the stress at which there is a complete yield of the material that is observable to the eye. The true elastic limit is reached at some lower stress, when incipient but not measurable permanent set is caused. In stipulating the factor of safety the nature of the duty has to be considered, and the value that would be safe for a steady load would be quite unsafe for a rapidly fluctuating load. As we have seen, the maximum stress in the solid forged rotor is in the teeth. The stress is unidirectional and constant, and a factor of safety of 2 to 1 on the yield point is to be regarded as satisfactory with a dependable steel. With this factor of safety the maximum permissible tooth stress with the quality of steel suggested is 11 tons per square inch. With a well-proportioned rotor having a diameter of 31 inches—which, as explained, is approaching the largest diameter that it is desirable to employ for a normal speed of 3,000 r.p.m. with the type of rotor now being made—the maximum tooth stress at 20 per cent overspeed is about $9\frac{1}{2}$ tons per square inch. It will be seen, therefore, that the quality of steel suggested well meets the requirements. It might be thought that, if a higher stress could be allowed, a narrower tooth could be provided and there would be more room for rotor copper and consequently a greater output. The question of the tooth density, however, has to be considered. There must be some saturation in an alternator to give it magnetic stability. There are cases on record of alternators that would not operate satisfactorily due to insufficient saturation, the excitation being overwhelmed by the armature reaction at changes of load. With a turbo-alternator the necessary saturation must be provided in the rotor teeth—that being the only part of the machine where a high density can be arranged or is permissible. On the other hand, if the saturation is carried too far, the loss in the teeth becomes excessive. With a normal 5,000-kw. alternator at 3,000 r.p.m. the exciting energy required for the rotor teeth is of the order of 10 per cent of the full-load excitation. With a badly proportioned rotor the loss may be as high as 20 per cent. In general, it will be found that with a stress of about 10 tons per square inch the corresponding tooth density is as high as it is desirable to employ, and the oil-hardened carbon steel therefore satisfies the electrical as well as the mechanical requirements.

It is sometimes suggested to employ an alloy steel for the rotor, but there is very little gain, if any, in doing so with the objects immediately in view, and on the other hand there are pronounced disadvantages. For instance, if an alloy steel were employed with the object of being able to use a smaller tooth to give increased winding space, it would be found in most cases that the increased magnetization loss, due not only to the higher tooth density, but also to the lower permeability, would more than offset the gain due to the increased copper space. There is the objection to using an alloy steel that for a large rotor forging it is inherently not so reliable, there being a greater probability of internal flaws being present. A hole should always be bored through the forging when an alloy steel is used, and the bore should be carefully examined to see that there are no internal flaws. As already

explained, the effect of the hole is more than to double the hoop stress, so that the body of such a rotor actually would be weaker than if a carbon steel were used.

An interesting case came to the author's notice a few years ago of a rotor forging breaking just where the shaft merged into the body of the core. The steel used was $3\frac{1}{2}$ per cent nickel having a yield point of 29 tons and an ultimate strength of 46.5 tons per square inch, and an elongation on 2 inches of 22 per cent. It was found that a huge internal cavity had been caused by the operation of forging, as shown in Fig. 5. The corresponding part at the other end of the rotor was cut open, and although there was no cavity the steel was in a highly stressed and weakened condition. Insufficient care in forging might cause weakness in this way with any steel, but it is much more probable with an alloy steel than with a carbon steel. In another case that came to the author's notice a high carbon steel was employed having the composition of 0.61 carbon, 0.59 manganese, and 0.23 silicon, and the tensile properties of ultimate strength 49 tons per square inch, yield point 22 tons to the square inch, elongation on 2 inches 18 per cent, and reduction of area 32

FIG. 5.—Forging defect in high-speed rotor.

per cent. After being in service for $3\frac{1}{2}$ years the forging failed with a fatigue flaw in the neck of the journal. A micro-examination showed that the steel had not been properly heat treated and had a large grain size. Such a steel, and more particularly in the absence of correct heat treatment, is not at all suitable for a high-speed rotor.

The use of improper steels of this kind and bad mechanical design—with which not even a good steel could stand—have led to a number of failures, which have given rise to the feeling, amongst those not familiar with the conditions, that the solid forged rotor is less reliable than the built-up design. For the onerous conditions of high-speed work, it is not good enough to use a block of any kind of steel in any state of heat treatment and to machine it without regard to fundamental mechanical requirements. There is no doubt that provided the right kind of steel is used, correctly forged and correctly heat treated, and that the mechanical design is sound, the solid forged rotor is by far the best construction, and the author has never heard of a single failure when these conditions have been observed. The only justification for the built-up construction is for small rotors and when high-grade forgings are not obtainable, and for single-phase work.

The forging for the turbo-alternator with its radial

stress presented a new condition to the steel maker; and, not because there was any real difficulty with it but because it was a new requirement that they had not investigated, some makers would not undertake the work. With ordinary care in forging there is no difficulty in giving the required radial test. The tensile strength radially is very much the same as in the longitudinal direction, but the elongation is usually lower. Three tests should be made from each forging—one from the shaft end, one tangentially from the core, and one radially from the core, as shown in Fig. 6. With an

Part		Yield Point, Tons per sq. in.	Ultimate Tons per sq. in.	Elongation on 2 in.
Longitudinal, A	..	22.5	38.5	28 per cent
Tangential, B	..	22.5	38	27 "
Radial, C	..	22	37.5	22 "

FIG. 6.—Tests from an oil-hardened carbon-steel rotor.

oil-hardened carbon steel containing about 0.38 carbon the results shown in the table in Fig. 6 may be expected. The figures given are actual results; the makers' guarantees with regard to elongation, and particularly in the radial direction, would be somewhat lower.

LIMIT OF FORGING SIZE.

There are limits to the size of forging that a steel manufacturer can undertake. There is the inherent limitation due to the rate of cooling of the interior of a large mass of metal. The size of the crystals and hence the properties of the steel are profoundly influenced by the rate at which it cools. If the mass of steel were unduly great, the cooling of the interior would be so slow that the mechanical quality in consequence could not be made to conform to the necessary requirement. With the forgings for turbo-alternators of the sizes so far considered the problem does not arise, and the limitations are the manufacturer's facilities for casting an ingot sufficiently large and the capacity of his forging plant. Nothing definite, therefore, can be laid down as to the largest size of rotor that can be produced in a single piece, but it is in the neighbourhood of 50-in. diameter. That is to say, for all outputs so far contemplated at 3,600, 3,000, and 2,400 r.p.m. a single forging can be obtained, and also for all except very large outputs at 1,800 and 1,500 r.p.m. For very large outputs at 1,800 r.p.m. and lower speeds a single forging may not be satisfactory. In such cases the soundest construction is to forge a hollow cylinder with a shaft extension attached to each end, or better

still carried on a through shaft as shown in Fig. 7. Forging the cylinder on a mandril does work on the steel and gives it its best qualities on the inner periphery where the hoop stress is a maximum. Moreover, in quenching, the oil has access to the bore and the maximum benefits of the heat treatment are conferred at the part of maximum hoop stress. With such large rotors there is no difficulty in so proportioning the belt of metal between the bottom of the slots and the bore

elastically to exert its reactive force without any local over-stress being caused. If the design is such that there is a tendency for some part of relatively small area to resist the general deflection, that part may be overstressed and failure may result. An example of such a case is given in Fig. 8. The beam with the fin projection and the load can be so proportioned that before the beam has deflected sufficiently to give the necessary reaction to support the load the crystals at

FIG. 7.—Rotor for a large 1,000-r.p.m. turbo-alternator.

that the hoop stress is kept down to a satisfactory value. For very large rotors this construction therefore constitutes an excellent arrangement.

THE SHAFT DESIGN.

In determining the design of the shaft it is most important that the proportions should be such that the curve of deflection is smooth and free from any sudden change. The forces due to the rotor weight cause

FIG. 8.—Showing how a beam can be weakened by adding to its section.

the steel to yield elastically until the reacting force balances the weight. So long as the duty on the individual crystal is within its capacity the structure is safe, but if that limit is passed the crystal deforms by slip taking place along the cleavage planes. The excessive burden falls on to the neighbouring crystals, and they in turn fail, and so on until fracture is caused. For the structure to be safe, therefore, not only must the area of the steel be adequate for the load to be carried, but the proportions must be such that it is free to yield

the bottom of the rib are overstressed. Failure, therefore, would result with such a construction, whereas the beam without the fin would carry the same load satisfactorily. Where there is an abrupt increase of diameter in a shaft, the conditions are somewhat but not exactly similar. Just where the increase of diameter takes place the shaft portion is under stress and is yielding elastically. The flow of stress into the larger diameter is somewhat gradual, and the face of the metal that is rising at right angles from the smaller diameter to the larger diameter carries little or no stress, and there is therefore in intimate contact steel that is elongating

FIG. 9.—A sudden increase in diameter may be regarded as adding metal to a gradual increase, and is analogous to the weakening effect of adding a thin rib to a beam as shown in Fig. 8.

and compressing elastically and steel without stress, and great weakness is thus caused. A gradual change of diameter represents a normal design, and a sudden change is to be regarded as adding metal to the normal design as shown shaded in Fig. 9. The added metal tends to prevent the normal elastic deformation and has inadequate strength for the purpose. In such a case the tendency to fail is often increased by the crystals at the junction of the two diameters being torn and weakened by the action of the cutting tool.

In Fig. 21 is shown a case of failure that came to the author's notice a few years ago. It will be noticed that there is a very sudden increase of diameter between

the bearing journal and the large oil baffle that is formed solid with the shaft. After 11 months' running the shaft broke in the position shown. Not only was the design thus most unsound, but the material was unsuitable, being of a high-carbon steel that had not been heat treated and having a very coarse grain size.

Abrupt changes in the curve of deflection must therefore be carefully avoided. The shaft should be gradually stepped to avoid any sudden increase of diameter, and wherever a change of section occurs a carefully formed radius should be provided. Gauges to check the accuracy of the radii in important places, such as in the neck of the journals, should form part of every shop inspector's equipment. In addition to being carefully formed, the radii should be given a fine finish and good polish. It has been shown by Sir George Beilby* that the effect of polishing is that the superficial molecules become mobile and form a thin viscous layer with surface tension analogous to that of liquids. After a very short period the layer loses its mobility and solidifies into a vitreous amorphous state, forming a very strong cementing material. On the other hand, if the surface is left rough the surface crystals are torn and weakened, and the conditions are favourable for fatigue to develop.

The need for avoiding abrupt change of section applies to all parts of the rotor, and at the junction between the teeth and the body of the rotor ample radii should be provided.

ALTERNATING STRESS.

The maximum reverse stress in the rotor is in the neck of the bearing journal at the part marked "A" in Fig. 7. The weight of the rotor puts the figure at the top of the shaft into compression and at the bottom of the shaft into tension. As the shaft turns round, the stress changes from tension to compression. In the case of a 3,000-r.p.m. alternator the number of reversals per day is 4,320,000; that is to say, the number of reversals daily is equal to what investigators into the subject of alternating stress have regarded as a complete endurance test.

Failure from fatigue is really the same as failure under a simple tensile test, in so far as it is the result of the overstressed crystals yielding plastically by slip along the cleavage planes. The essential difference is that failure by fatigue is relatively very slow and is localized, not being accompanied by any general yield of the material.

The investigations of recent years have thrown some light on the reason for steel failing under an alternating stress that could have been carried indefinitely if applied constantly. Ewing, Humphrey, and Stanton have discovered that the failure generally starts in the ferrite grains, which consist of iron in its crystalline and therefore softest state. The stability of the ferrite is much less than that of the aggregate, so that stress insufficient to strain permanently the aggregate would be more than sufficient to break down the ferrite grains if the stress had access to them. Sir George Beilby † dis-

covered by studying the vibrations of cold-drawn reeds before and after annealing, that although the crystalline phase is more elastic than the amorphous phase, yet the amplitude of vibration is greater in the latter state. It is reasonable to expect, therefore, that when an aggregate composed of two phases is subjected to rapidly alternating stress, the harder element with its greater amplitude will cause the softer element with its less amplitude to carry an undue proportion of the stress.

Under a simple tensile test, as the load is increased the first crystals to fail are those that are the worst circumstanced with regard to size and the orientation of the gliding planes. These crystals deform plastically, and the stress is therefore to a certain extent removed and becomes more distributed over the aggregate. Such a point marks the true elastic limit, and the stress has to be increased appreciably before there is the general deformation of the crystals as a whole that is observable to the eye, known as the yield point.

That a steel crystal has strength after permanent deformation is explained by Sir George Beilby* to be due to minute particles being torn away from the contact surfaces, of a lower order of magnitude than any crystal unit, that form an amorphous cement stronger than the crystals themselves. The action is analogous to what happens when the surface of metal is polished as explained above. With any normal metal it is found that fracture always takes place along the cleavage planes of the crystals themselves, and not by the failure of the amorphous cementing material that holds them together. The presence of this amorphous material produced by the rubbing along the gliding planes may quite well, therefore, have the effect of making the crystal stronger after yielding than it was before. This explanation agrees with the fact that under a tensile test a greater stress is required to rupture the material than was necessary to cause the yield.

The conditions necessary for failure by fatigue are that the real elastic limit of the steel is exceeded but that the yield point is not reached. If the elastic limit were not reached, the metal would not fail however many reversals may be made; and if, on the other hand, the yield point were reached the failure would take place by ductile elongation before fatigue developed. After the first local yielding of the worst circumstanced crystals, if the load is kept constant nothing further happens and the steel would carry the load indefinitely. If, however, the stress is quickly reversed, the slip of the crystals will be reversed, the movement being considered to be facilitated by the newly produced amorphous material being in a temporarily mobile state. At each reversal there is corresponding slip along the gliding planes, and more of the amorphous material is produced until the crystal is exhausted and failure results. This process then begins in other crystals and spreads until complete failure is caused.

It was thought by some of the earlier investigators into the subject of reverse stress † that the liability to fail increased with the speed of reversal. It is now

* *Proceedings of the Royal Society*, vol. 72, p. 218.

† *Journal of the Institute of Metals*, 1911, vol. 6, p. 5.

* *Journal of the Institute of Metals*, 1911, vol. 6, p. 5.

OSBORNE REYNOLDS and J. H. SMITH: *Philosophical Transactions of the Royal Society*, 1902.

generally agreed that these investigators were misled by the peculiarities of the type of testing machine they employed. It is fairly certain that failure by fatigue is independent of the speed of the reversals unless the speed be very low or very high. It is well known* that after plastic deformation if steel is allowed to rest the strength increases. This quality, however, is without value in shafts, since the period of rest required is a matter of days unless hastened by the action of heat. On the other hand, if the speed is very high the time that the load is applied may be insufficient for yield of the material to take place. The time required for a crystal to deform is appreciable whatever stress may be applied, and it is to be expected that if the stress is applied for a very brief period only there may be insufficient time for the steel to deform plastically, although the magnitude of the stress may

FIG. 10.—Endurance tests of oil-hardened carbon steel with 2,000 alternations of stress per minute.

be sufficient to cause rupture if applied steadily. Dr. B. Hopkinson has shown† that steel wire having a yield point of 17.8 tons per square inch and an ultimate strength of 28.5 tons per square inch can carry a load of 33½ tons if it is applied for about only 1/1,000th part of a second. He has also shown‡ that with reversals at the rate of 7,000 per minute the resistance of metals to reversals of stress is appreciably greater than at low speeds. The speed at which this property becomes of value has not been clearly determined. Dr. Stanton§ states that with tests he made at speeds of from 100 to 2,200 reversals per minute there was no evidence that the rate of reversal had any effect on the resistance to failure. It is probable that at the normal running speed of the high-speed alternator the alternations of stress have a maximum effect, since

* MUIR: *Philosophical Transactions of the Royal Society*, 1902, vol. 193 A.

† *Proceedings of the Royal Society*, vol. 74, p. 498.

‡ *Ibid.*, vol. 86 A.

§ *National Physical Laboratory Report*, 1910.

the speed is not sufficiently high for it to have any ameliorating effect, and on the other hand the great speed tends to bring unforeseen load into play. It becomes necessary, therefore, to give very careful attention to the design in this respect.

When making tests under alternating stress the procedure is to apply a load that it is known will produce failure and to note the number of alternations before breakdown results. The test is repeated with a similar specimen under a reduced load, and so on until the load is found under which the specimen will not break however many alternations may be made. If the results are plotted the curve is parabolic, as shown in Fig. 10, the horizontal asymptote showing the stress with which an indefinite number of reversals would not produce fracture.

The fatigue failure is so insidious in its action, and so many unforeseen factors may be present to contribute to it, that a big margin of safety is desirable. For instance, the calculated stress may be much exceeded by a slight error of alignment, and the possibility of minute irregularities or flaws must be considered, which for any other condition of loading would be without practical effect. Moreover, the torsional stress which is a contributing factor may have its normal value much exceeded by a short-circuit. Further, in expressing the factor of safety in terms of the yield point it is to be remembered that the true elastic limit, which, as explained, determines fatigue failure, is considerably lower than the yield point. It has been shown by Ewing and Humphrey,* by observing the creation of slip bands under alternations of stress, that the true elastic limit of a carbon steel may be only 52 per cent of the yield point. Taking all these factors into consideration, it is the author's opinion that the combined calculated stress in the neck of the journal should not exceed 7½ per cent of the yield point of the steel. With the high bearing duties of to-day—journal speeds of 6,500 feet per minute being quite satisfactory—there is no difficulty in keeping within this factor of safety.

HEAT TREATMENT.

The rate at which steel cools has a profound effect on its structure and physical properties. The effect of cooling steel rapidly is much debated by metallurgists, and there is no general agreement as to the precise effect. It has been established that steel has the characteristic of allotropy and its properties vary according to the allotropic form in which it exists. As with most substances, allotropy in steel is a function of temperature with normal cooling, as is shown by the recalcence points in the time-temperature curve. The change, however, from one allotropic form to another requires time, and rapid cooling as is brought about by quenching in oil or water considerably influences the degree of the change. Under a tensile test the performance of steel is apparently little affected by the crystal size,† but the resistance to shock and

* *Philosophical Transactions of the Royal Society*, vol. 200 A, p. 241; see also STANTON and BAIRSTOW: *Proceedings of the Institution of Civil Engineers*, 1905-6.

† SHARPY: *Revue de Metallurgie Memoires*, 1910, vol. 7, p. 656.

the capacity to withstand reversals of stress are directly influenced. The greater the temperature to which steel is heated the longer it is maintained at high temperature, and the more slowly it is cooled the greater is the crystal size and the less dependable is the steel. A medium carbon steel of the kind suggested for the rotor has its yield point increased about 30 per cent by oil hardening. It is seen, therefore, that the correct oil treatment of steel very considerably improves its tensile strength, and also puts the steel generally into better condition and makes it more reliable. For these reasons it is very desirable that all alternator

FIG. 11.—Showing the variation in the centrifugal force of the rotor coils in the circumferential direction of a 5,000-kw. 3,000-r.p.m. alternator with 31-in. diam. rotor (calculated at 20 per cent above normal speed).

rotor forgings should be oil treated. The approximate treatment of the steel in question would be to immerse in oil at about 800°C. and temper at about 550°C., the treatment varying according to the carbon content. The heat treatment should be carried out after the forging has been rough turned. Some distortion of the forging is unavoidable in the oven, and the rough-turned dimensions should be somewhat greater than the finished sizes. In an average case the allowance on the diameter should not be less than 0.25 inch.

ROTOR END BELLS.

As already indicated, the most highly stressed part of the rotor is in the end bells retaining the rotor end windings, and the design of the bells, therefore, re-

quires the most careful attention. In the case of a 5,000-kw. alternator at 3,000 r.p.m. with a 31-in. diameter rotor, at the overspeed of 20 per cent above normal the centrifugal force exerted by the windings at each end of the core is about 820 tons. If to this is added the centrifugal force of the bell itself, the total centrifugal force at each end is about 2,000 tons. For such enormous forces it is obvious that the material employed must be very high grade.

In the earlier days of the turbo-alternator, steel binding wire was frequently used. The material itself is satisfactory, since it is reliable and has very great strength—an ultimate strength of over 100 tons per square inch being usual, with the yield point not much lower. Two great disadvantages of using binding wire are the difficulty of maintaining a good balance and the necessity for completely re-winding the band if the rotor windings have to be examined. For these and other reasons wire banding is not used on modern turbo-alternators.

In considering the choice of material for the end bells the designer naturally has preference for non-magnetic material. With the end bell registering on the rotor core, if it is magnetic it offers a direct leakage path of considerable area from pole to pole, and has an appreciable effect in reducing the useful rotor flux.* The leakage has a constant value with a given diameter and excitation, and the percentage value is therefore less the longer the rotor. In the case of an average 5,000-kw. 3,000-r.p.m. alternator at full load the leakage is of the order of 10 per cent of the excitation energy. Before steel bells were tried it was thought that there might be heating in them due to leakage flux from the stator, but experience shows that such is not the case, the reason being that any such flux rotates synchronously with the bell. It was thought also that for solid bells of any material such a large mass of metal close to the end of the stator core might cause undue heating due to eddy currents induced by the action of the flux swinging from tooth to tooth. Such, however, is found not to be the case, or if there is any such action it is not important in degree.

The magnetic end bells appear to have one rather remarkable effect in causing great heating in some cases at the ends of the stator core when running on short-circuit. In one case that came to the author's notice of a 5,000-kw. 1,500-r.p.m. alternator with steel end bells, when running on short-circuit the stator end plates and the laminations in the immediate neighbourhood became very hot; embedded thermo-couples placed about 2 inches from the end of the core showed a rise of 150 degrees F. Thermometers placed against the back of the core also showed a high temperature rise in the immediate vicinity of the end plates and a low temperature elsewhere. On full load the end plates were quite cool, and the temperatures shown by the thermometers and thermo-couples were quite normal. In two cases of 5,000-kw. alternators at 3,000 r.p.m. there was similar heating at the ends of the stator core in a still more marked degree. Embedded thermo-couples placed $\frac{1}{2}$ inch from the end of the core showed a rise of 250 degrees F. after an

* R. G. JAKEMAN, *Electrician*, 1915, vol. 75, p. 765.

hour's run on short-circuit. The temperature rise of the cast-iron end plates was about 120 degrees F., showing that the greatest heating was in the laminations themselves.

The author suggests that this phenomenon is probably due to the steel end bells. When a short-circuit test

FIG. 12.—Stator leakage into steel end bell on short-circuit.

is made with an alternator, the excitation is increased until the current reaches its full-load value. The stator current lags almost 90° , and the reaction is in phase with the rotor flux and in opposition to it. Most of the rotor excitation is required to overcome the

FIG. 13.—5,000-kw. 3,000-r.p.m. alternator (calculations made at 20 per cent above normal speed).

stator reaction. The leakage into the end bells weakens the rotor magnetism at the ends of the core, and quite conceivably the stator reaction may overcome it and there may be an appreciable flow of reverse flux between the ends of the stator and the end bells. The reaction

between these fluxes flowing in different directions at the ends of the stator core may quite well cause some of the reverse flux to flow obliquely through the plates

FIG. 14.—3,000-kw. 3,000-r.p.m. alternator (calculations made at 20 per cent above normal speed).

and so cause heavy eddy-current loss. Probably also there is a small leakage of flux into the stator end plates. The effect is indicated roughly in Fig. 12. Since the effect, whatever the explanation may be,

FIG. 15.—2,000-kw. 3,000-r.p.m. alternator (calculations made at 20 per cent above normal speed).

does not occur under any running condition, it is of little importance. It may, however, quite upset the method some makers have of forecasting the temperature rise on load from light-load tests—with which one

factor is the temperature rise on short-circuit. Moreover, it may cause the designer to attribute to eddy-current loss in the copper a loss that is due to something else altogether.

Of the non-ferrous metals available for making end bells the best is a high-grade manganese-bronze. Good material of this kind has an ultimate strength of about 44 tons per square inch, a yield point of 22 tons per square inch, and an elongation of 20 per cent. The real elastic limit of the material as shown by its behaviour under reversals of stress is lower with regard to the yield point than with a good steel. For this reason the author is of opinion that the maximum stress should not exceed 45 per cent of the yield point. In Figs. 13, 14, 15, 16 and 17 are shown the average load

FIG. 16.—1,500-kw. 3,000-r.p.m. alternator (calculations made at 20 per cent above normal speed).

per axial linear inch that the end bell has to carry due to the centrifugal force of the end coils for rotors for 3,000 r.p.m. of 31-in., 27-in., 25-in., 22-in. and 20-in. diameter respectively, calculated at the overspeed of 20 per cent above normal. The bottom curve in each case shows for various thicknesses the load that the manganese bronze can carry in addition to itself with a stress equal to about 45 per cent of the yield point—i.e. 10 tons per square inch. With a 31-in. diameter rotor it will be seen that the manganese bronze can only just carry itself when its thickness is varied from zero up to about 0.7 inch. If the thickness is further increased, the stress is increased and exceeds the safe value for the material. With the 27-in. diameter rotor the manganese bronze has a maximum load carrying value when the thickness is about 1.5 inch, but it can then carry only about 8.5 tons per linear inch out of about 66 tons that have to be carried. With the 25-in. diameter rotor with a bell thickness of 1.7

inch the bronze can carry only about 22 tons per inch, against 48 tons to be carried. For the 22-in. diameter rotor the bronze bell is satisfactory if it is given a thickness of 1.25 inch, and for a rotor of 20-in. diameter it is satisfactory when 0.9 inch thick. These diagrams thus show clearly the extent to which the material may be used—i.e. that it is satisfactory for 3,000-r.p.m. rotors up to about 22-in. diameter.

Figs. 15, 16, and 17 show that oil-treated carbon steel can be used satisfactorily for the end bells of rotors up to about 25-in. diameter at 3,000 r.p.m., but not for any larger diameters. The curves for the carbon and for the nickel-chrome steel are based on working at 50 per cent of the yield point of the material, the carbon steel having a yield point of 22 tons per square inch and the nickel-chrome steel 40 tons per

FIG. 17.—750-kw. 3,000-r.p.m. alternator (calculations made at 20 per cent above normal speed).

square inch. Even for the smaller diameters, however, there are advantages in favour of using nickel-chrome steel. For instance, for the 25-in. diameter rotor (Fig. 15) it will be seen that a thickness of 1.43 inch is necessary for carbon steel, whereas with the nickel-chrome steel 0.6 inch would be sufficient. The magnetic leakage with the nickel-chrome steel would be much less and the flow of the ventilating air into the air gap would be less restricted. For rotors more than 25 inches in diameter nickel-chrome steel should always be used. Nickel-chrome steel containing about 3.5 per cent nickel, 0.5 per cent chromium, and 0.25 per cent carbon has an ultimate strength of about 50 tons per square inch, a yield point of 40 tons, and an elongation of 18 per cent on a 2-in. test-piece, and a reduction of area of 50 per cent.

Of the non-magnetic steels 25 per cent nickel is the only kind of which the use has been suggested for end bells. This material, having the approximate

composition of 25 per cent nickel, 4 per cent chromium, and 0.36 per cent carbon, has the ultimate strength of about 42 tons per square inch and an elongation of 40 per cent on 4 inches. The observed yield point is about 25 tons per square inch, but the real elastic limit is very low. With a careful test the extensometer

FIG. 18.—Minimum thickness of end drum required for different rotor diameters of 3,000-r.p.m. rotors (calculations made at 20 per cent above normal speed).

shows a slight permanent set with a loading as low as 15 tons per square inch, and of course the real elastic limit must be at a still lower stress. The steel is not difficult to machine and can be drawn satisfactorily into wire. It is difficult to forge, however, there being a marked tendency for the red-hot billet to crack under the hammer. It is annealed by quenching in water at about 1,450° F. The steel is not magnetic, for the reason that at ordinary temperatures it exists in the same allotropic form as does iron heated above its highest recalcence point—i.e. about 900° C. If the non-magnetic steel is immersed in liquid air it is changed into the same allotropic form as is iron cooled below about 760° C., and it becomes magnetic. 25 per cent nickel steel is quite unsuitable for high duty, and manganese bronze, with its higher yield point and more reliable qualities, should always be used in preference.

Hadfield's manganese steel is quite unsuitable for end bells, having a very low yield point and having the remarkable quality that, although really soft, deformation makes it extraordinarily hard and it cannot be machined. Another non-magnetic steel also unsuitable is that due to Carpenter, Hadfield, and Longmuir, containing 19.91 per cent nickel, 0.41 per cent carbon, and 0.96 per cent manganese.

Magnetization appears to be due to the arrangement of the atoms in the molecule, or of the molecules themselves, and is not a property peculiar to the atoms of those elements that are magnetic. There is the effective proof furnished by Heusler* that an alloy containing only non-magnetic elements may be

* "Über die Ferromagnetischen Eigenschaften von Legierungen Unmagnetischer Metalle," Marburg, 1904.

magnetic. He produced such an alloy composed of 60 per cent copper, 26 per cent manganese, and 14 per cent aluminium, all of which elements are non-magnetic. On the other hand, alloys containing a large percentage of iron may be non-magnetic—such as 25 per cent nickel having about 70 per cent of iron, and Hadfield's Austenitic manganese steel containing about 88 per

FIG. 19.—Showing the variation in centrifugal force of the rotor coils in the axial direction of a 5,000-kw. 3,000-r.p.m. alternator with 31-in. diam. rotor (calculated at 20 per cent above normal speed).

cent of iron. Heusler's magnetic alloy undergoes allotropic change when quenched at 400° C. and becomes non-magnetic.* Hadfield's manganese steel undergoes allotropic change when it is deformed and becomes magnetic.† The non-magnetic steel of Carpenter, Hadfield, and Longmuir behaves similarly.‡

* Proceedings of the Royal Society, vol. 77 A, p. 256; see also vol. 76 A, p. 271.

† Journal of the Iron and Steel Institute, vol. 89, p. 132.

‡ Seventh Report of the Alloys Research Committee, Institution of Mechanical Engineers, 1905, p. 949.

Whereas oil treatment is desirable with carbon steel, with nickel-chrome it is indispensable. As with the rotor, the end-bell forgings should be rough turned and then be heat treated.

As an alternative to the magnetic and non-magnetic end bells, there may be considered the question of a composite end bell. The centrifugal force of the coils varies in the axial direction, being least at the core end and a maximum at the other end. In Fig. 19 is given a curve showing this variation in the case of a 26-in. diameter rotor for 3,000 r.p.m. with six concentric rotor coils. It will be seen that the load on the end bell is appreciably lower near the stator than the average value. For a 25-in. diameter rotor, for

FIG. 20.—Composite end bell for high-speed rotor.

which a manganese-bronze bell could not be used, it would be permissible to employ a composite bell of steel with a relatively narrow band of manganese bronze next to the core. Such a bell is shown in Fig. 20. The saving in magnetization loss with such a construction as compared with the all-steel bell may be of the order of 50 per cent. The author's view is that the all-steel nickel-chrome end bell represents the preferable construction, and that the composite bell should be employed only in exceptional circumstances. With the composite bell it is important to remember that the elasticity of manganese bronze is about 20,000,000 lb. (inch units) against 29,500,000 lb. for steel; that is to say, for a given stress the manganese bronze stretches (elastically) $47\frac{1}{2}$ per cent more

FIG. 21.—Solid forged rotor with defective shaft design.

than steel. It is important for this reason that the manganese-bronze ring should be registered on the steel ring, and not under it, so that it is free to open. If the bronze were registered under the steel, the projecting lip would prevent the bronze opening sufficiently (elastically) to carry its share of the load, and the lip might be overstressed. The correct construction is shown in Fig. 20. The author has heard of failures due to this obvious precaution not being taken.

The usual design of end bell is a cylinder registering on the rotor core at one end and on a disc shrunk on to the shaft at the other. In the case of a 5,000-kw. 3,000-r.p.m. alternator the cylinder opens about 0.03 inch in diameter. It is not possible to give the cylinder

sufficient initial tension by press fit to prevent it opening in this way. The tendency of the coil loading (see Fig. 11) is to make the cylinder become elliptical, but since there is nothing to centre it definitely it may move bodily out of centre by the amount it opens radially. The vibration with some alternators that cannot be corrected completely is probably due to this cause in some cases. The author is very strongly in favour of making the cylinder and the end disc down to the shaft in a single piece as shown in Fig. 23; the bell is thus definitely centred and eccentricity is prevented. An additional advantage of this design is that it is not necessary to register the bell on the rotor core and an appreciable gap can be provided between the core and the bell, so reducing the magnetic leakage. The short length of coil between the core and the bell is supported by extending the slot wedge and by regis-

FIG. 22.

tering the end of the wedge under the bell. Another advantage that may be obtained with this construction is to make the seating on the shaft larger in diameter than the slip-rings, so that the end bells may be removed for examination or repair of the windings without interfering with the slip-rings or the connections to them. The construction also has the advantage of very considerably strengthening the cylinder at the end where it is most heavily loaded. In Fig. 18 is shown the average thickness of end bell required for rotors of various diameters for manganese bronze, carbon steel, and nickel-chrome steel.

VENTILATION.

The various systems of ventilation that have been tried at various times have now settled down roughly to two distinct types—axial and radial. The advantages of the axial system are that the omission of the radial ducts shortens the machine and that the rotor may be separately and positively ventilated. The disadvantage of this system is that there is an appreciable difference of temperature between the two ends of the machine. The temperature difference may be

reduced somewhat by arranging the flow of air through the rotor and stator in opposite directions. With the radial system the temperature rise can be kept almost uniform from end to end, but there are the disadvantages that most useful material is removed from the rotor in forming the radial ducts and that the rotor ventilation is not positive. The flow of air through the rotor is induced only by the natural fanning effect of the radial ducts, and the flow is opposed by the pressure in the gap set up by the fans. With large machines the air velocity through the stator has to be made high for the necessary volume of air to

ately measured. By varying the rotor fan the volume of air flowing through the rotor can be regulated to give the exact volume required. The omission of the rotor axial ducts gives the designer liberty to work to the tooth density he prefers rather than that to which he may be forced. There must be some difference of temperature between the two ends of the rotor, but the volume of air can be made sufficient to keep the hotter end within the desired limit. By suitably disposing the stator ducts the temperature throughout can be kept substantially uniform. In addition, when necessary, axial ducts may be arranged in the stator

FIG. 23.—Turbo-alternator with axially-ventilated rotor and radially-ventilated stator.

flow, and pressure equal to 6 inches or 7 inches of water has to be set up by the fans. At the ends of the rotor such a pressure quite overcomes the rotor fanning effect, and the discharge from the rotor ducts must be prevented if not actually reversed. It is probably only towards the centre of the core, where the air-gap pressure is reduced, that there is any appreciable flow of air from the rotor.

The author considers the ideal system to be to have radial ventilation for the stator with independent axial ventilation for the rotor as shown in Fig. 23. The air for the rotor and stator is kept quite separate and the volume and temperature rise of it can be separ-

and the discharge from these into the various radial ducts may be separately varied to equalize further the temperature rise. The omission of the radial ducts reduces somewhat the noise of the alternator and also the windage loss.

Fans.—It is quite usual to see, with an otherwise well-designed rotor, fans that are anything but robust in construction. Sheet steel about 1/16 inch thick for the blades and about 1/8 inch thick or less for the runners is frequently employed, and apparently that the fans will be satisfactory is taken for granted and is not verified by calculation. Several cases have come to the author's notice of such fans collapsing

and striking the stator end windings, with complete breakdown of the alternator. For silent running, the more blades the better, but on the other hand the more blades the greater is the load on the runners. It is unnecessary to postulate that safety must come first, and the permissible number of blades must be determined by the maximum thickness that it is desirable to employ for the runners. By running alternators with and without the fans the author has determined the effect they have on the noise. The difference, though appreciable, is not great. The blades should be given such a thickness that collapse is impossible. For the smaller sizes oil-hardened boiler plate is satisfactory for the blades and runners, but for the larger sizes oil-treated nickel-chrome steel plates should be used.

The loss due to the fans in large high-speed machines is of the order of 20 per cent of the total windage. It is important that the efficiency should be kept as high as possible, since the loss not only affects the alternator efficiency but also adds to the heating. The entering angle of the blades should be the same as the direction of the flow of air into the fan, so that there is no shock. The leaving angle affects the pressure and the direction of the air more than the efficiency. It is usual to make the blade quite straight, but the pressure is increased if the blade is curved in the direction of rotation. The blades and runners should be given a good smooth finish and be coated with a thin hard varnish. With such fans, on account of the high peripheral speed as compared with the air velocity through the fan, the direction of the air leaving the fan is almost tangential. The air, therefore, does not flow axially from the fan to the gap, but is carried round with a high velocity. This action causes some churning and loss, but on the other hand the stator end connections are well cooled. Probably the resistance to entering the air-gap is reduced by the motion, since the air in the gap is being carried round violently by the rotor, but if there are stator axial ducts the flow into them may be less easy.

The author has tried the effect of providing a row of standing blades over the fans to make the air flow purely radially into the end chamber. The air pressure was increased about 15 per cent; the volume of air flowing through the machine was increased correspondingly, and it was therefore decided that there was no gain or detriment in preventing the circular movement of the air in the end chambers.

Noise.—With very high peripheral speeds more noise must be expected than with the lower speeds of a few years ago. A good deal of the noise from an alternator is transmitted through the air leaving it, and the station can be made much quieter if the discharge air is led away through a duct. With a new station there is little difficulty in accommodating the discharge duct in the foundation, and for large high-speed sets it is very desirable to adopt this arrangement.

BALANCE.

Freedom from vibration is an essential mechanical requirement apart from being desirable for more obvious

reasons. With marked vibration, wear of rubbing parts may take place at an extraordinary rate and the soundest construction be weakened in the course of time. A press fit that would be perfectly solid in normal circumstances may become free in the presence of vibration, and relative movement of the parts may take place. Movement of the end windings is more probable, and the insulation under great pressure is very likely to suffer wear. Every component part should be so designed that change of balance is safeguarded against. It is desirable that the degree of balancing should not be left to the judgment of the erecting engineer, but should be put on a positive basis by using a vibration indicator. The grooves for the balance weights should be provided in the disc supporting the bell cylinder; the practice of attaching balance weights to the outer fan runners is most unsound. Much time is saved in balancing, and the final result is often better, if the rotor is balanced statically before running. Although the correct division of the balance weights between the two ends of the rotor cannot be determined statically, the total weight required is determined and adjustment of it only is required when making the running balance. The sensitiveness of the knife edges is much increased by resting the shaft on ball or roller races.

OVERSPEED TEST.

The stresses and pressures for a given rotor increase with the square of the speed. The increase of stress and pressure factors at overspeeds are shown in Fig. 22 based on unity at the normal speed. For proving mechanical soundness the overspeed test fails because it is fatigue failure that is most to be feared, and an increase of speed does not prove capacity to withstand fatigue. The overspeed does directly test the soundness of the rotor teeth and of the end bells, but even its value in this way is limited, because in the event of there being a flaw, due to the ductile quality of the steel, it is more probable that failure will result after prolonged running than at the moment. A great disadvantage is that the overspeed imposes on the insulation an undue stress that is harmful. The turbine emergency governor should be set to operate at 10 per cent overspeed, and since some allowance to make sure of the governor operating is necessary it is desirable that the set should be capable of standing an overspeed of 20 per cent. At this overspeed the pressure and the centrifugal force are 44 per cent greater than at the normal speed. The author holds the view that all parts of the rotor should be designed as if it were to run at 20 per cent overspeed continuously. At the normal speed the stresses and pressures are 30.5 per cent less, so that the factors of safety in the teeth and end bells are then very satisfactory.

In summary, high overspeeds (1) do not test the most important requirement—capacity to withstand fatigue, (2) do not necessarily detect the presence of incipient flaws in the highly-stressed parts, and (3) do undoubtedly tend to damage the insulation. An overspeed of 20 per cent is essential, but anything higher is undesirable.

In conclusion, the author begs to express his obligation to Mr. Douglas Vickers and Mr. W. Clark, directors of Messrs. Vickers, Ltd., for permission to read this paper. Also, he takes the opportunity of acknowledging his indebtedness to his colleague, Mr. J. P. Nicholson, who conducted the various mechanical tests referred to in the paper.

APPENDIX.

SPECIFICATION OF SOLID FORGED ROTOR.

Rotor forging.—The rotor forging is to be of high-grade carbon steel annealed after forging and oil treated after rough turning. The forging is to be preferably in a single piece, but if too large for a single forging a hollow cylinder is to be forged on a mandril and is to be shrunk on to a through shaft. The steel is to be made and the forging to be done by an approved maker. Test-pieces are to be cut from (a) the end of the shaft, (b) the periphery of the rotor in a tangential direction, and (c) the end of the core in a radial direction. A tensile test and a bend test are to be made in each case. Test certificates are to be furnished by an approved testing house, giving the yield point, the ultimate strength, the elongation and the contraction of area; or tests are to be made in the presence of the engineer's representative at his option. The stresses are to be calculated at the speed of 20 per cent above normal, and are not to exceed 50 per cent of the yield point of the material. The maximum combined torsional and bending stress in the neck of the journal is not to exceed $7\frac{1}{2}$ per cent of the yield point of the material. Change of section throughout the rotor is to be made gradual and ample radii are to be provided. The rotor is to be given a smooth finish, and in addition the shaft portion is to be finished to a good polish. All keyways are to be carefully cut and radii are to be formed at the ends and at the bottom.

End bells.—The end bells are to be made of high-grade nickel-chrome steel annealed after forging and oil treated after rough turning. The steel is to be made by an approved maker. A test-piece is to be cut from the bell in a tangential direction, and a tensile and bend test are to be made. Test certificates are to be furnished by an approved testing house, or the tests are to be made in the presence of the engineer's representative at his option. In addition, a test-piece is to be notched and broken by impact, and the fracture is to be examined by the steel-maker and he is to certify that the steel is in satisfactory condition. The stress is to be calculated at 20 per cent above normal speed and is not to exceed 50 per cent of the yield point of the material. The maximum and not the average stress is to be taken—i.e. the stress should be calculated on the inner periphery and at the point of maximum coil loading. The end bell is to be given a fine finish inside and out. If in a single piece, the end bell is to be made a good press fit on the shaft and is to be positively driven and is to be retained

in position effectively against axial movement. If the cylinder is separate from the supporting disc, it is to be made a good press fit on the rotor core and on the disc. Both disc and cylinder are to be positively driven, and both are to be retained in position effectively against axial movement. With either construction means are to be provided for readily withdrawing the bells to give complete access to the end windings for examination or for repair.

Rotor windings.—The slot insulation is to be of micanite reinforced with suitable non-hygroscopic material, and the total insulation thickness is not to be less than 0.06 inch. The insulation between turns is to be pure hard mica not less than 0.01 inch thick. If radial ducts are used a sheet-steel slot lining to support the insulation must be used. If ventilating ducts are cut at the bottom or sides of the slot, sheet-steel support for the insulation must be provided. The end windings are to be insulated from the end bells by micanite not less than 0.05 inch thick reinforced by suitable non-hygroscopic material. Where the end-coil connections are close together suitable insulation is to be provided between to prevent contact. The end windings are to be effectively wedged to prevent movement. The coils are to be firmly pressed into the slots with a pressure at least equal to the maximum pressure they exert at 20 per cent overspeed. The rotor coils are preferably to be few in number and relatively wide so that the end windings have a maximum of stability.

Slip-rings.—The slip-rings are to be machined from carbon steel forgings containing not less than 0.4 per cent and not more than 0.6 per cent carbon. They are to be insulated with pure hard mica, and are to be shrunk directly on to the shaft or on to a forged steel sleeve shrunk or pressed on to the shaft. The rings are to be heated electrically for shrinking, and not by fire or gas, so that there is no chance of scale being formed that might impair the fit. The relative sizes of the end bells and of the slip-rings should preferably be such that the bells can be completely removed and the windings fully exposed for examination or repair without disturbing the rings.

Slip-ring connections.—The connections from the windings to the slip-rings are to be of ample size and are to be built up of copper laminations not more than 0.01 inch thick. The connectors are to be heavily insulated throughout with moulded or flexible micanite and are to be securely held against the action of centrifugal force throughout their length. Preferably the connectors are to be so disposed that the end bells can be removed without having to disturb them.

Slot wedges.—The slot wedges are to be of steel or gunmetal, and the maximum shear stress and bending stress is not to exceed 35 per cent of the yield point of the material.

Fans.—The fan blades are to be of substantial thickness so that bending and collapse is impossible. Oil-hardened boiler plate or oil-treated nickel-chrome steel plate is to be used for the fan blades and for the runners. The maximum bending stress in the blades, the shear stress in the blade fixing, and the maximum hoop stress in the runners must not exceed 40 per cent

of the yield point of the material calculated at the overspeed of 20 per cent above normal.

**SCHEDULE OF PARTICULARS TO BE FURNISHED
BY THE MANUFACTURER.**

Rotor steel made by :

Rotor forged by :

Tensile properties of rotor steel :—

(a) From end of shaft. Yield point :
Ultimate strength :
Elongation :
Reduction of area :
Bending test :

(b) Tangential test from rotor periphery.
Yield point :
Ultimate strength :
Elongation :
Reduction of area :
Bending test :

(c) Radial test from rotor core.
Yield point :
Ultimate strength :
Elongation :
Reduction of area :
Bending test :

Maximum stress in the rotor teeth :

Maximum combined torsional and bending stress in journal :

Maker of steel for end bells :

End bells forged by :

Tensile properties of end bell material :—

Tangential test. Yield point :
Ultimate strength :
Elongation :
Reduction of area :
Bending test :

Maximum stress in end bell :

Material used for slot wedges :

Tensile properties of slot wedge material :—

Yield point :
Ultimate strength :
Elongation :

Maximum shear stress in slot wedge :

Maximum bending stress in slot wedge :

Material of fan runners :

Tensile properties of material :—

Yield point :
Ultimate strength :
Elongation :
Reduction of area :
Bending test :

Material of fan blades :

Tensile properties of material :—

Yield point :
Ultimate strength :
Elongation :
Reduction of area :
Bending test :

Maximum stress in fan runners :

Maximum shear stress in the blade fixing :

Maximum bending stress in the blade :

First critical speed :

DISCUSSION BEFORE THE INSTITUTION, 21 MARCH, 1918.

Mr. A. B. Field : The development of the alternating-current turbo-generator during the last few years has been rapid, and the requirements of the immediate future give promise of demanding an equally rapid further advance in methods of construction for very large units. The problem was originally, let us say 13 years ago, largely an electrical one, but now that individual units reach 50,000 kw. output, and with a very possible demand nearer to 100,000 kw. per set in the early future, the mechanical aspects largely predominate. We ought then to be far more ready to accept electrical imperfections (excluding insulation weakness) than to tolerate poor mechanical features. It is therefore gratifying to find the author dealing with the subject so largely from the mechanical point of view. Reading the paper through, one cannot but feel that the author has sub-consciously had in mind chiefly 3,000-r.p.m. machines, although references have been made to other speeds; and this would largely explain some apparent incongruities and irrelevancies. The immediate outlook for 3,000-r.p.m. machines may reach some 10,000 kw. or so per unit; but for the coming really large units, the lower speeds of 1,500, 1,000, and 750 r.p.m. will be necessary. For 3,000 r.p.m. I am entirely in accord with the author as to the preferable, one might almost say necessary,

use of a solid forging. Indeed, I want to endorse so much in this paper that I must in the time available confine myself entirely to points of contention. Many thoroughly satisfactory machines for speeds between 1,000 and 1,800 r.p.m. have been built in the past, using laminations assembled on a shaft, but nowadays machines of such ratings—which are essentially small machines—would, except in special cases, be built for 3,000 or 3,600 r.p.m., and our interest in laminated rotors becomes largely historical. Coming then to the larger and lower-speed machines, our choice of construction lies between the use of a single forging; a multiple-part forging, the components being still large; forged discs; and rolled plates. In each of the last three cases we might have a through shaft, or we might not. The construction which is most suitable will vary with the rating. It will also depend upon steel-market conditions and upon facilities in the country in question. With the exceptional metallurgical facilities at the beck and call of the author, a construction will frequently become desirable which would be questionable elsewhere. I need not lay further stress upon the point. Those who have been dependent upon buying large forgings in the market, merely upon a purchasing specification and inspection basis, will appreciate the point at once. The author criticizes