DIMENSIONAL ACCURACY OF MAIN MACHINERY

by

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PART II.

Part I of this article, which appeared in the JOURNAL, *Vol.* 1, *No.* 3 (*October,* 1947), *covered the general introduction to the subject, reasons for the need of a higher standard of dimensional accuracy, a brief analysis of design features, and a description of allowance, tolerance, and unilateral and bilateral limits.*

Part II gives specific examples of the allocation of tolerance, mentions the implications which arise from the expression of such tolerance, and describes the former and revised practices of position tolerancing.

Two systems of limits and fits are in use in this country designed to meet most needs and conforming to standard practice. Firstly, the Newall system, which was devised to meet a long-felt want many years ago, but is now almost entirely superseded by the British Standard System (B.S. 164—Limits and Fits).

Newall specifies two grades of hole and six grades of shaft, thus permitting a choice of 12 fits for any given basic size.

The British Standard specifies four grades of hole and fourteen grades of shaft, thus permitting a choice of 56 fits for any given basic size. This extensive range covers most engineering requirements and, in fact, should be restricted for use in particular design offices, in like manner to the restriction of preferred basic sizes.

With further reference to the example quoted on page 58, Part I, it is proposed to re-examine the example of bearing and shaft, using B.S. 164—Limits and Fits, to ascertain if design requirements can be met, or so nearly met, as to warrant adoption. On page 8 of B.S. 164, is a table—Guide to classes of fit. The example falls in the " clearance " range. The tolerance on " U " holes (Table 1) for 2 inches diameter is ·0014″, and this is quickly seen to be of a grade higher than that warranted by the magnitude of permissible variation on allowance. The tolerance on the next grade of hole, viz., " V " is ·0028″, which appears to be in the region of requirements. So " V " hole is tentatively recorded. Turning to Table 2, Limits for Shaft, "S" shaft has ·0021″ tolerance and ·0035″ allowance. The " T " shaft has ·0028″ tolerance and ·0056″ allowance.

From B.S. 164, *Fit VT* :—

Bearing	2·0″	+·0028 — ·000
Shaft	1·9944″	+ ·000 —·0028

Examine extreme conditions which may arise :—

(1) Largest Bearing	2·0028″
Smallest Shaft	1·9916″
Clearance	·0112″
(2) Smallest Bearing	2·000″
Largest Shaft	1·9944″
Clearance	·0056″

From B.S. 164, *Fit VS* :—

Bearing	2·0″	+·0028 −·000
Shaft	1·9965″	+·000 −·0021
(1) Largest Bearing	2·0028″	
Smallest Shaft	1·9944″	
Clearance	·0084″	
(2) Smallest Bearing	2·000″	
Largest Shaft	1·9965″	
Clearance	·0035″	

It is suggested that fit VT most nearly meets design requirements, and is so close to those requirements that the distinct advantages accruing from the use of standards should be welcomed and incorporated in the design.

Two-inch standard V and T gauges are readily obtainable and production gets quickly under way. The sizes of B.S. holes and shafts will be found commensurate with B.S. standard drills, reamers, and other tools, and the science of engineering will be scientifically embarked upon, if these principles are adopted. It should be noted that the expression of liberal tolerance does not infer that work will be on extremes of tolerance and this, in practice, will but rarely occur. It is an expression of the worst acceptable article, and where better results can be achieved without difficulty, then such should be the aim of contractors ; work which incorporates moving parts has a short life when made on the minimum metal extreme of tolerance.

For work which does not fall into what may be termed a " life or death " category, a designer may well assess probability in allocation of tolerance. When tolerances are large and quantity production is proceeding, for instance, it is extremely unlikely that the largest shaft will ever be brought to assembly with the smallest hole, or vice versa. Thus, even greater tolerance can be apportioned, without probable detriment to the assembled parts. In " Life or death " work, e.g., the fit of a cartridge in a gun, for service purposes, no such risk should be taken. It is imperative that every cartridge should fit every gun and the largest cartridge that would not insert in the smallest gun, would be an unacceptable design statement. However, to facilitate production, some such slight risk might be taken when considering bolts and holes for securing radio components to the chassis.

The Implications of Tolerance

Having discussed the allocation of tolerance in the foregoing paragraphs, it is considered desirable to draw attention to the implications arising from the expression of such tolerance, unless particular notes are recorded upon the drawing eliminating the implication or implications not desirable. This is of very great importance in considering dimensional accuracy.

A simple example will serve to illustrate the implications and methods of treatment.

The example comprises a stepped shaft of diameters 1″ and ½″, located in a plate with ½″ diameter hole (Fig. 1). Tolerances of ·0024″ and ·0004″ are allocated to the respective shaft diameters and ·0008″ to the hole size.

It is required to examine implications of each toleranced diameter individually and then to examine the implications of the tolerance on the assembly as a whole.

Whilst the expression of ·0024″ tolerance on the size of the diameter of this

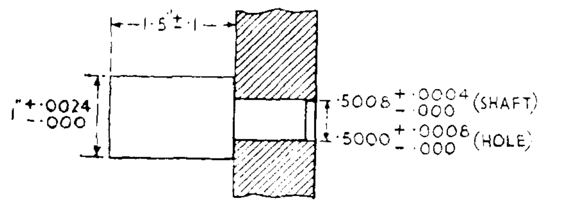


FIG. 1

shaft is primarily intended to mean that size may vary from 1·000″ diameter to 1·0024″ diameter, it also implies that this portion of the shaft may, throughout the length of the 1″ portion, be tapered by ·0024″, be out of round by ·0024″ or be bowed by ·0024″, when in the minimum metal condition of 1·000″ diameter size.

The tolerance of ± ·1″ on the 1·5″ length, primarily intended to convey that the length may vary between 1·400″ and 1·600″, also infers that either end may be out of square by ·2″, or that both ends may be out of square by ·1″ when the length is in minimum metal condition. It also implies that ends may be concave or convex by these amounts, or in any condition of departure from flatness by these amounts.

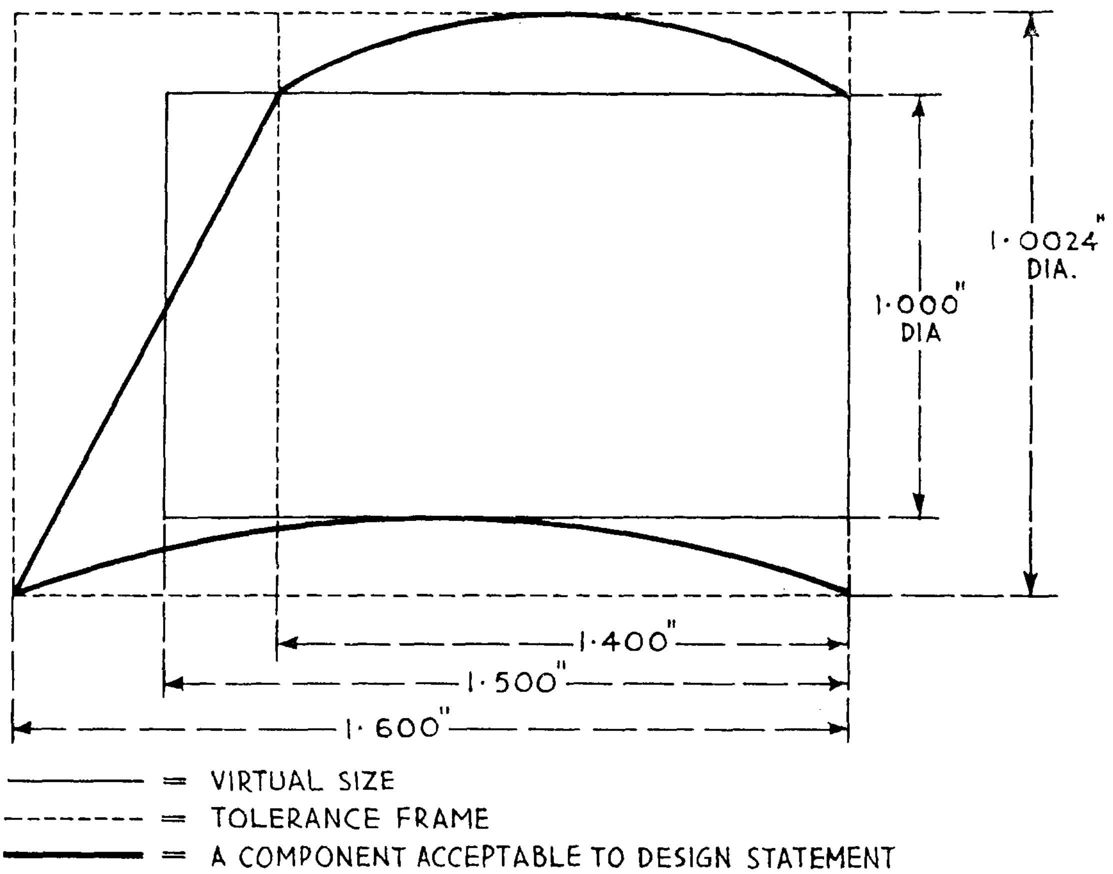


FIG. 2

The foregoing statements can be diagrammatically represented by a tolerance frame, Fig. 2. The virtual size and shape of the component is a cylinder of 1″ diameter and $1\frac{1}{2}$″ length. The tolerance frame of the component becomes two cylinders, one of 1·0024″ diameter × 1·4 length and the other of 1·0024″ diameter × 1·6 length. Now any component falling between the virtual size and shape and the tolerance frame size and shape is an acceptable component, unless additional limitations are recorded upon the drawing.

Implications of Tolerance on Assembly

Continuing the lines of thought enumerated for one toleranced diameter, it will be seen that ·5008″ $\begin{smallmatrix}+ & \cdot 004 \\ - & \cdot 000\end{smallmatrix}$ diameter of shaft and ·500″ $\begin{smallmatrix}+ & \cdot 0008 \\ - & \cdot 000\end{smallmatrix}$ diameter of hole might be accepted in similar conditions. In addition to this, the shaft of two diameters may have its diameters eccentric to each other by an amount equal to the sum of the tolerance of each diameter. (Fig. 3.)

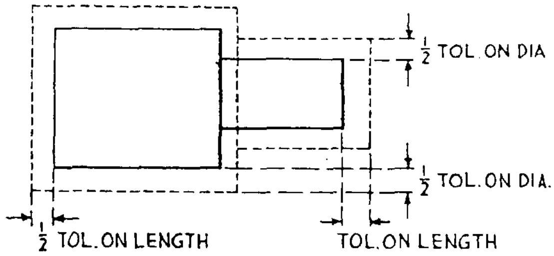


FIG. 3

It should be noted that the various aspects referred to, i.e., taper, bow, eccentricity, etc., may not be an accumulation of full tolerance on each feature, but may be accumulative to a total amount of tolerance on the whole feature.

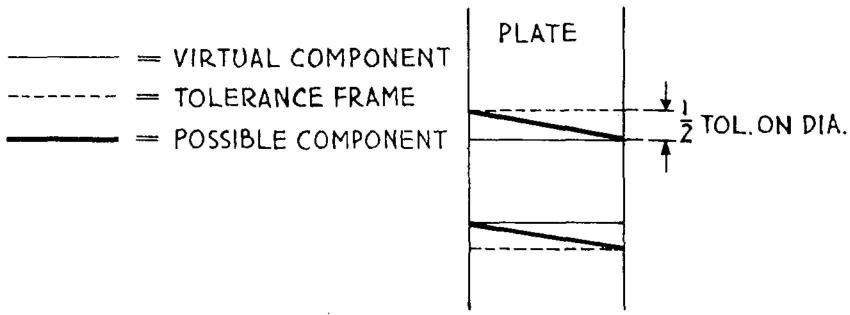


FIG. 4

The tolerance on hole diameter may permit such a condition as illustrated in Fig. 4.

Thus, the assembly, unless noted to the contrary, may be as drawn in Fig. 5, with each error exaggerated in extent for purposes of clarity.

It is not suggested that these errors are likely to arise or that they will be serious in extent if they do arise. It is, however, pointed out, that if it is essential that they should not arise, then design steps must be taken to safeguard against such of those errors, the magnitude of which, by reason of the feature tolerance, will be detrimental to the assembly or function of the work. Such steps can best be taken in note form. Turbine rotor journals provided a practical example where at least some of the errors referred to would be detrimental to the final product. Here, a wide tolerance on journal diameter would appear to be quite permissible in view of the large clearance allowance between journal and bearing. This large tolerance would infer that the journal may be out of round,

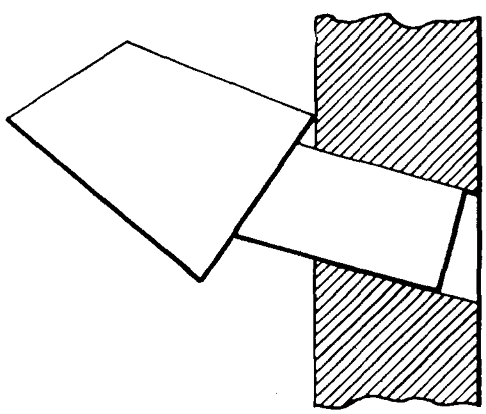


FIG. 5

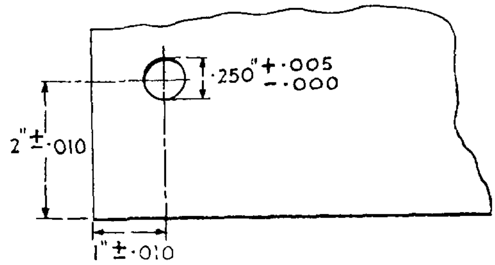


Fig. 6

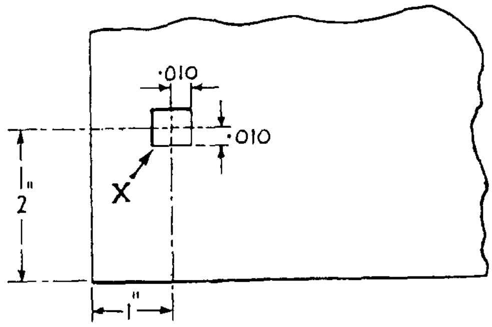


Fig. 7

taper, bowed, or irregular to any shape or form within the tolerance band on virtual diameter.

Example :—

Turbine journal

9·980″ diameter $+ \cdot 000$
$ - \cdot 005$

—to parallel and straight within ·0005″.

—to be circular within ·001″.

—The two journals to be concentric within ·002″.

The dimensional accuracy of the journal will be assured if such conditions be expressed by designers, as clear guidance to manufacturers and inspectors. If the above conditions are obligatory, then limits of size would give such an indication.

Effect of Inspection Gauge Tolerance

The effect of inspection gauge tolerance must be recorded in a section on drawing practice, but will be better understood after reading that section devoted to gauges later in these notes.

Where " Inspection " gauges are to be used as the criterion for acceptance of work, then it must be borne in mind by designers when apportioning tolerance to work, that the use of such gauges may permit up to a 10% increase in the latitude of such tolerance. This will often mean, where tolerances are critical, that work tolerance must be restricted by 10%. This rarely occurs except on strictly interchangeable work to fine tolerances.

Position Tolerances

A theory evolved by S. Parker, Esq., of the Inspectorate of Naval Ordnance, which has received the favourable consideration of the Inter-Services Committee on Dimensioning and Tolerancing Engineering Drawings, is to apportion tolerance for position of features rather than to apportion tolerance to linear dimensions affixing the centre or face of such features. Past endeavours to control position tolerance resulted in multitudinous and often conflicting dimensions. The achievement of assembly, interchangeability, and proper function of a component has been realised only by the ability of the craftsman. The true realisation of planned dimensional accuracy can only be achieved by completed design information in respect of position. This method of position tolerancing is to relate the feature to be positioned with a functional or tooling datum, and to denote tolerance as an amount on position in relation to that datum.

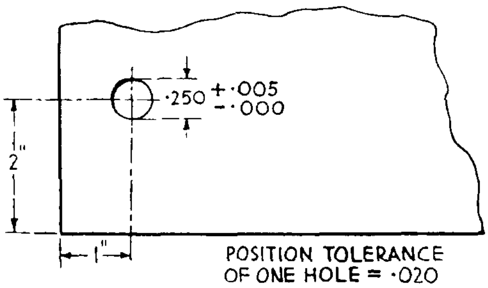


FIG. 8

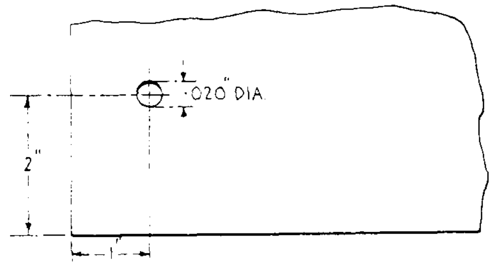


FIG. 9

Former practice is illustrated by the example in Fig. 6 and the tolerance frame arising from it in Fig. 7.

Note that the centre of the hole may lie anywhere within the tolerance square of ·020″. This ·010″ amount of tolerance has probably been derived from a bolt to enter the hole in the position shown by the centre, and under the hole size by ·010″.

Now examine the point X. This is within the tolerance frame. It has departed from its true position by $\sqrt{\cdot010^2 + \cdot010^2} = \cdot014''$. Thus, if designed for entry of maximum bolt into minimum hole positioned within ·010″, this method of dimensioning leads to failure.

The revised practice of position tolerance is illustrated in Fig. 8 and the tolerance frame in Fig. 9.

Fig. 8 means that the centre of the hole may lie anywhere within a circle of ·020″ diameter scribed round its nominal position. A form of words sometimes used and quite acceptable is " to be within ·010″ of theoretical centre."

The statement of position tolerance in respect of the datum faces lends itself very well to jigging and subsequent gauging. The jig drawing will show the drill bush with a position tolerance of 20% of the work tolerance, which in our example is ·004″ from the same datum faces.

Inspection will be by receiver type gauge where a pin of minimum hole size minus position tolerance must enter the hole when the plate is located in the gauge by its datum faces. (Fig. 10.)

It should be noted that this, a customary type of gauge, would reject a component with a hole centre at point X (Fig. 7), although by former design statement the hole centre at X was permissible. Conversely, a component, or gauge, dimensioned and toleranced by the positional method would be accepted with a centre X if measured by direct measurement instead of fixed type gauge, unless measurement at 45° is included. Tool inspectors should note this point and ensure that such measurement is carried out.

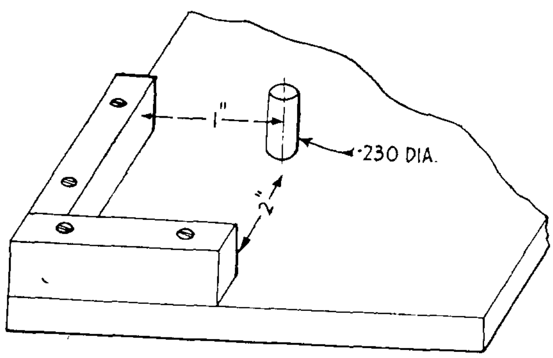
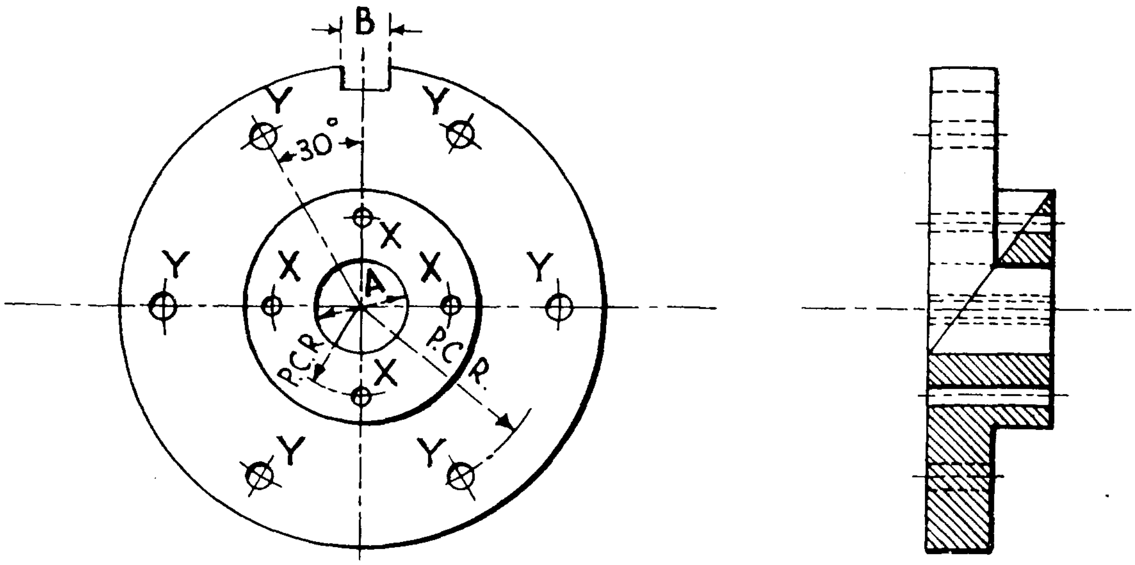


FIG. 10

From the foregoing simple example, it will be seen that in cases of more complicated dimensioning and tolerancing, the position theory greatly facilitates design and manufacture. A case in point is a ring or series of rings of holes in flanges, related to a datum, spigot diameter or diameters.



4 HOLES X TO BE ·250″ $^{+\cdot002}_{-\cdot000}$ DIA. POSITION TOLERANCE = ·002″ TO DATUM DIA. A.

6 HOLES Y TO BE ·500″ $^{+\cdot005}_{-\cdot000}$ DIA. POSITION TOLERANCE = ·002″ TO DATUM B

KEYWAY B POSITION TOLERANCE TO DATUM DIA. A = ·005″

FIG. 11

In this case, former practice involved tolerance on P.C.D. and on spacing of holes. When such holes had to be further related to one or more datums for rotational position, it became even more complicated. Is it not known that the matter was frequently dismissed with a statement of 12 holes " equispaced " ? The onus for assembly or interchange was thrown on the craftsman. To ensure dimensional accuracy, position tolerance of such holes must be shown upon the drawing by the designer, who is in a position to calculate what such tolerance should be.

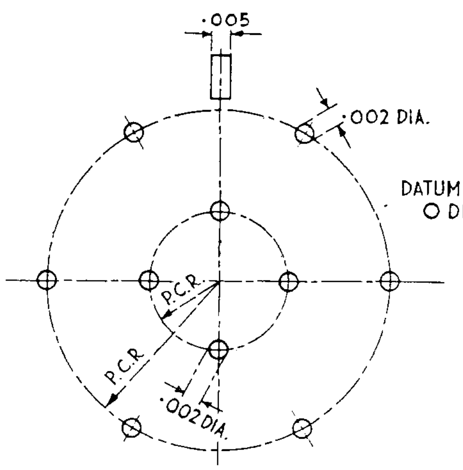


FIG. 12

A little more complicated example on the lines above discussed is shown in Fig. 11. The position tolerance frame for Fig. 11 is shown in Fig. 12, and the full tolerance frame indicating the maximum variations permissible from tolerance on size and position is illustrated in Fig. 13.

The diagrams illustrate, and consideration of fixed gauging practice will prove, that the position tolerance for female features (holes and keyways) is operative in the maximum metal condition, i.e., minimum size, and for male features (pins, keys, etc.)

is operative in the minimum metal condition.

If a feature is finished to the minimum metal condition, an additional positional tolerance becomes permissible equal to the tolerance on the size of the feature. This is quite logical and in accordance with requirements for interchangeability, assembly, and function.

The fixed gauge must be made with features of a size capable of assembly with a component on an extreme limit of size. Thus, when the other limit of size is reached on the component, that additional amount of tolerance becomes available for position.

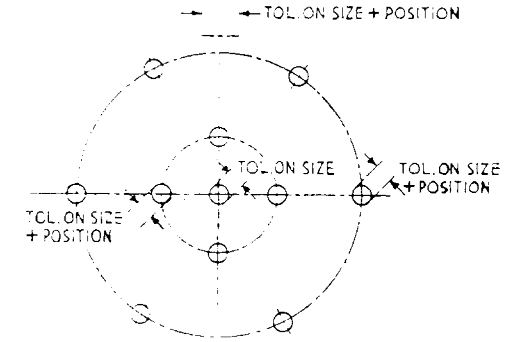


FIG. 13

FAILURE OF MAIN CIRCULATOR IMPELLER THRUST BEARINGS

Reports have been received from Light Fleet Carriers of failures of the ball thrust races on the main circulator gearwheels. It was significant from these reports that all failures occurred shortly after the ships had entered tropical waters, the inference being that the bearings were failing due to overloading by reason of the increased main circulator speeds required. This, however, could not be proved either in theory or practice.

In May, 1947, a report was received from another ship that water was being found in the lubricating oil of the main circulators, and that the thrust bearings were showing signs of watermarking. This information opened up another avenue of approach to the problem and it was decided that the ball races were failing due to corrosion of the surfaces of the balls and tracks. It has been decided to replace the ball thrusts with thrust pads of the Michell type with a larger factor of safety and which will be less affected by contaminated oil.

The contamination of the lubricating oil is evidently caused by ineffective sealing of the pinion shaft where it enters the gear case. This sealing is done by a " Git Seal " which is placed close to the lower carbon gland of the turbine and thus perishes rapidly due to the high temperatures to which it is exposed. Water is thus able to run down the spindle into the gear case.

These arrangements are being investigated and a more suitable seal, perhaps of the labyrinth or screw thread type, will be introduced. In this connection it is interesting to compare the similar trouble which is being experienced with Brotherhood's turbo-dynamos, and the complete freedom from lubricating oil contamination in the case of Allen's turbo-driven machinery fitted with the Allen McCleod device.