SYNOPSIS

This paper discusses the factors that need to be considered when deciding whether a flexible coupling is needed at all. It then gives general guidance on flexible coupling selection by a qualitative comparison of the various important factors relating to design, operation and maintenance. This is intended as a starting point for discussions between machine designers and coupling manufacturers, within the context that the coupling is an essential part of the whole machine system.

THE COMPARATIVE PERFORMANCE OF COULING TYPES

The previous sessions of this Conference have considered each type of coupling in details, and the purpose of this paper is to try and bring together the performance characteristics if the various types of coupling, so that they can be compared. It is hoped that this may provide some guidance on coupling selection and application.

The need for flexible couplings

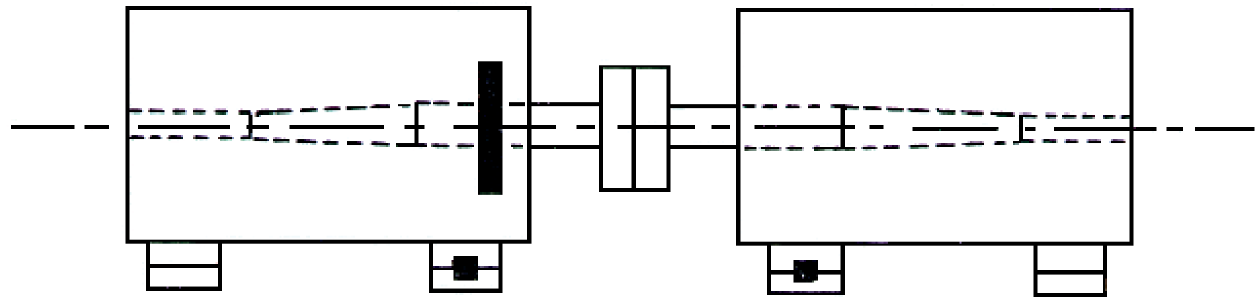
Before considering the relative performance of the various coupling types it is useful to consider the nature of the problem which they are required to solve. Essentially, of course, flexible couplings are used to allow various kinds of movements and relative misalignments to occur between the shafts of coupled machines. This situation may be a genuine technical necessity which arises because, in the nature of its operation, one machine has to move or deflect slightly relative to the other. There are, however, also some applications in which flexible couplings are used for commercial convenience. In the best cases this may be where ii is economical to use two machines which have already been designed quite separately, so that the relative movements of their shaft ends do not necessarily match in any way, and which therefore require flexibility in their method of coupling. In the worst cases, machines may be purchased and even installed separately with the hope that a flexible coupling between the two will take up any resulting inaccuracies. In all these cases an essential starting point is to know how much misalignment various couplings can tolerate under the conditions of operation, so that this can be taken into account, and so that the maximum allowable inaccuracies in alignment can be specified.

There are very few coupled machines, particularly among those which operate at high powers and speeds, that do not experience changes of temperature, with resulting relative expansions, during their operating cycles, and to obtain the optimum design arrangement the coupled machines should be considered as a complete system in conjunction with appropriate operational targets such as minimum initial cost, minimum total life cost, maximum plant availability, etc. Such a consideration might lead to a decision to use rigid couplings throughout with the whole plant designed as an integral unit, or it might lead to the use of flexible couplings only in positions where they were technically essential. The decision whether to use flexible or rigid couplings involves the consideration of several factors such as:-

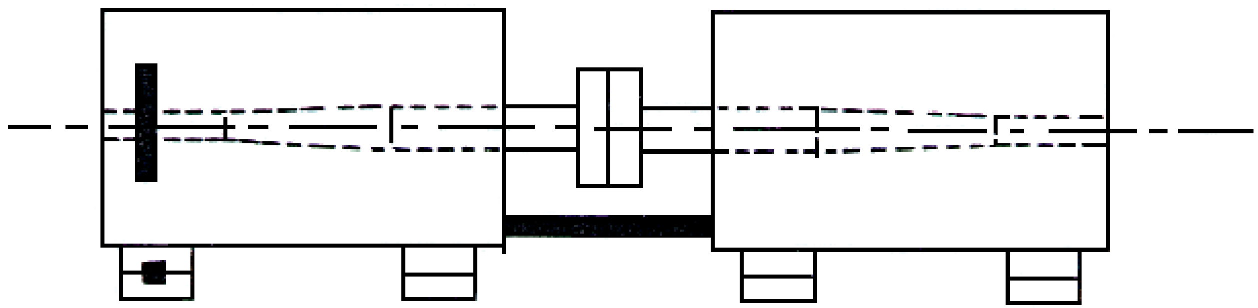
1. Since they contain no moving or deflecting parts, rigid couplings in themselves are inherently more reliable than flexible couplings.

2. From the point of view of relative movements, rigid couplings are likely to give rise to greater transient journal bearing loads and will always require more accurate alignment in multibearing systems.

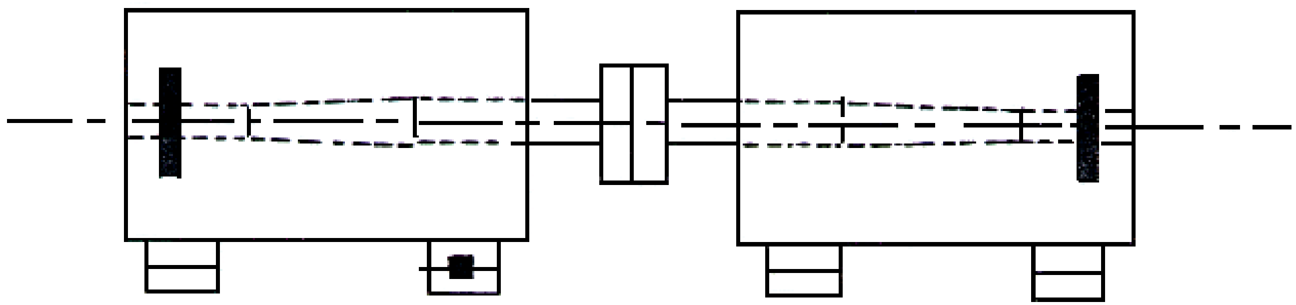
3. From the point of view of maintaining appropriate relative axial positions of shafts and machine casings, rigid couplings require careful thrust bearing location, and provision for casing expansion. Fig. 1 shows some possible arrangements with their related problems.



Axial location good.
Thrust bearing power loss may be high because it is on a large diameter shaft.



The relative axial expansion movement of the shafts and casings may be excessive.



Axial location good.
If the feet do not allow movement the thrust bearings will fail.

AXIAL EXPANSION ARRANGEMENTS WITH RIGID COUPLINGS

FIG. 1

4. If a machine system includes a number of rotors and elastic shafts, and particularly if gear drives are incorporated, it may be prone to torsional vibrations. In these circumstances the incorporation of flexible couplings provides a relatively small component which can be modified to change the torsional resonance of the system.

If, in the light of these various factors, it is decided to incorporate one or more flexible couplings in the system, the choice of an appropriate coupling then needs to be made, and guidance on this is given in the next section.

Coupling selection

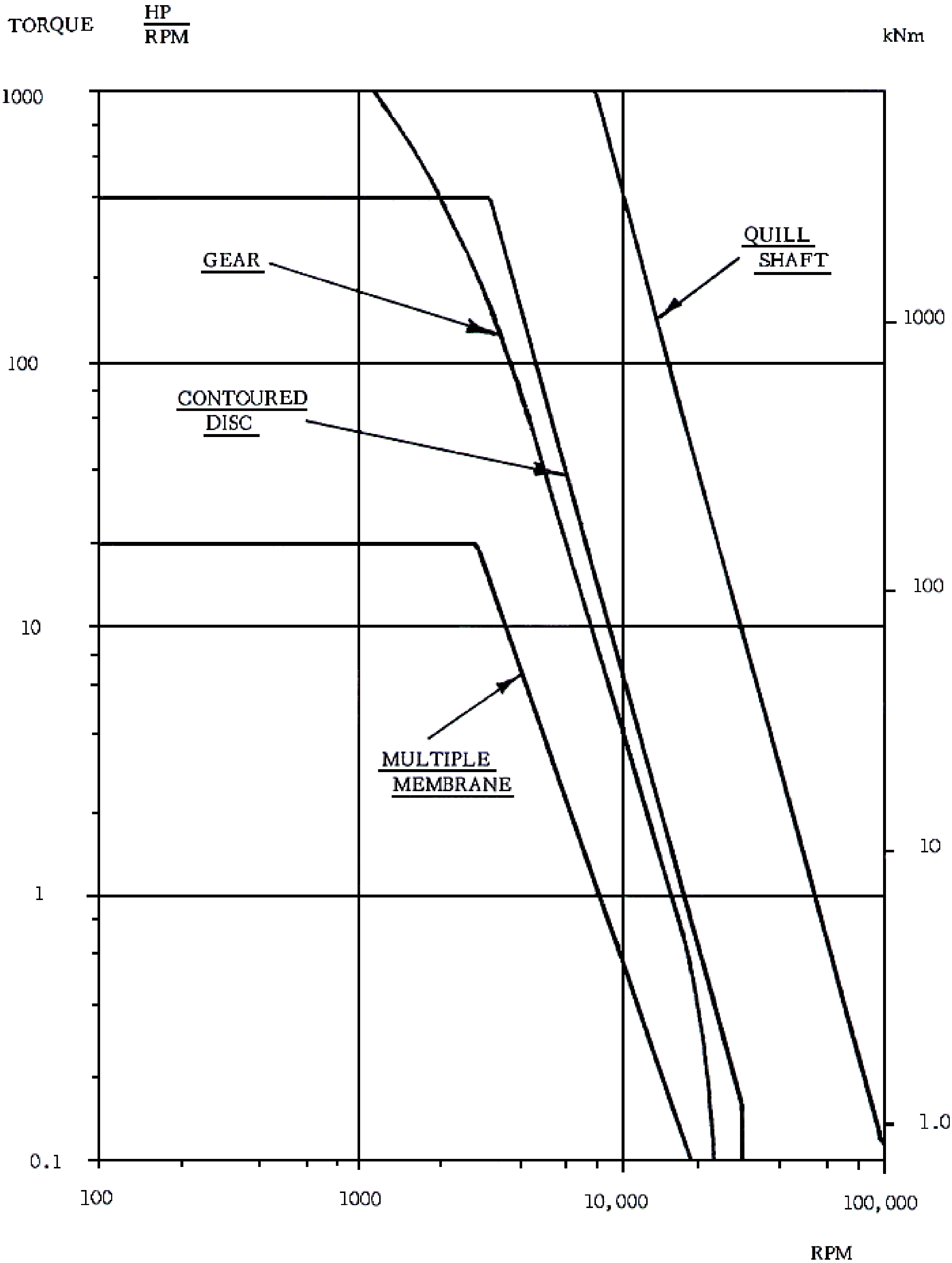
In order to select a satisfactory coupling, it is essential to regards it as a part of the whole machine system and not to view it merely as a component which couples the shaft ends together in a convenient way.

The various interactions between the coupling and the system must therefore be considered, and these are:-

(a) The effect of the system on the coupling, such as the misalignments, torques, and speeds to be carried, and the available space and lubrication arrangements.

(b) The effect of the coupling on the system, such as the lateral and axial forces which it may generate, and the effect of the mass, stiffness and position of the coupling components, on the dynamics of the system.

The main factors which need to be considered are included in Tables 1 and 2, which give qualitative comments on the relative performance of the various types of coupling. Quillshafts have also been included in these tables because, although they are not usually classified as flexible coupling, they can be used at high powers and speeds in situations where some angular freedom is required in a drive system. The various comments in the tables have to be qualitative because in each case actual numerical values for any particular characteristic will depend upon the values of the others. For example, there is an interaction in all types of couplings between outside diameter, allowable angular freedom, and allowable axial freedom. Because of these interactions the best that can be achieved in any general advice on selection is to expose the main strengths and weaknesses of the various types. An engineer trying to make a selection can then make some assessment of which types might suit his particular machine system, and in addition will have guidance on what questions he should ask the suppliers of the various types. This should help him to work with them towards the selection and design of a coupling which is particularly suitable for his application. Co-operation with the coupling manufacturer is also important because all the different types of coupling appear, on the basis of the papers at this Conference, to still have considerable scope for further development. As each manufacturer makes an advance in the technology of his product, the relative merits of the various types of coupling may change in relation to any particular application. Typical improvements are likely to be in the materials, and methods of connecting the flexible elements of couplings which incorporate them, and higher allowable tooth sliding speeds, and therefore considerable increased allowable angular freedom, on new types of gear couplings. Fig. 2 gives an indication of the present boundaries of performance in terms of torque and rotational speed.



APPROXIMATE PERFORMANCE BOUNDARIES OF COUPLINGS FOR HIGH POWERS AND SPEEDS

FIG. 2

TYPE OF COUPLING	Axial freedom	Angular freedom	Outside diameter	Steady loads on machines	Potential dynamic loads from mass overhang	Ease of balancing	Remarks
Gear coupling	Large	Low at high powers and speeds	Low	Lateral and axial loads can be high	Low	Difficult and can drift in service	Excessive tooth wear must be avoided or high axial machine loads may occur
Multiple membrane coupling	Moderate	Moderate	High for large axial freedom	Low	Moderate for large axial freedom	Requires care and can drift if membrane bolts are undone	The tightening torque on the membrane clamping bolts is critical. Coupling must maintain integrity if membranes fail
Contoured disc coupling	Moderate	Moderate	High for large axial and angular freedom	Low	Moderate for large axial freedom	Fairly easy and usually remains consistent	The disc requires specialist manufacture. Coupling must maintain integrity if disc fails
Quillshaft and disc	Low	Moderate	High at disc end for large axial freedom	Low	High at disc end for large axial freedom Low at other end	Fairly easy and usually remains consistent	Shaft end connection needs careful design. Coupling must maintain integrity if disc fails
Quillshaft	None	Low	Very low	Lateral loads low. Axially fixed	None	Easy and never changes	Shaft end connections need careful design

COMPARATIVE PERFORMANCE CHARACTERISTICS

TABLE. 1

TYPE OF COUPLING	Noise and heat generation	Failure mode	Need for lubrication	Need for maintenance	Robustness
Gear coupling	Quiet. Cooling oil essential at high powers and speeds	Progressive wear. Vibration levels usually give warning	Pressure or flood lubrication with oil is essential at high power and speeds	Inspection and cleaning is necessary after long service	Very good
Multiple membrane coupling	Can be noisy and generate heat at very high speeds	Membranes designed for indefinite life, and would fail progressively. Bolt failure can be sudden	Not required	Inspection of membranes and bolts when convenient after long service	Good
Contoured disc coupling	Can be very noisy at very high speeds	Discs designed for infinite life because if failure starts it could be rapid	Not required	Inspection of discs when convenient after long service	Discs must be designed with adequate protection
Quillshaft and disc	Can be noisy at very high speeds	Disc and shaft designed for infinite life because if failure starts it could be rapid	Not required	Inspection of disc when convenient after long service	Disc must be designed with adequate protection
Quillshaft	Silent and cool	Shaft designed for infinite life because if failure starts it could be rapid	Not required	Inspection when convenient after long service	Very good

COMPARATIVE ENVIRONMENTAL FACTORS

TABLE. 2

DISCUSSION SUMMARY by B.J. Woodley

On the subject of vibration of shaft systems involving flexible couplings, a delegate pointed out how difficult if often is to come to the correct practical conclusions, even if what appear to be the right theoretical calculations are carried out. As an example a case was quoted of a gas turbine driven turbogenerator set which, when installed, suffered fro a very high vibration level. A vibration analysis showed that there were two critical speeds in the running range, but at full speed where it would normally operate there was no critical speed. It was therefore concluded, from a theoretical point of view, that there was no critical speed problem and many other hypothetical and far fetched causes of failure were suggested. However, it was noticed that the vibration levels went up with speed and also that the machine was not dynamically balanced. These facts suggested that although theory said it would not vibrate badly at full speed, the out of balance was causing the shaft to vibrate so much as it went through its critical speed that the shaft continued to vibrate badly at full speed. The complete shaft system was therefore very carefully dynamically balanced and since then it has operated satisfactorily.

One delegate asked whether the method of assessing critical speeds in the appendix of paper E1 assumed that the oil films of the journal bearings were infinitely stiff and that the pedestals were infinitely rigid as he noted that for rigid rotors the critical speeds can drop to one half or one third of such a value when such flexibilities are allowed for by sophisticated calculation methods. The author agreed that such flexibility effects had been neglected in order to develop a simple technique which can be used in coupling selection and he agreed that once the coupling had been selected it is desirable to use more complicated methods to get an accurate assessment of critical speed. It was, however, pointed out that the method of critical speed assessment given in Figure 2 of paper E1 estimates the change of critical speed caused by the addition of a flexible coupling to the shaft system. Therefore, if the critical speed has been calculated originally taking into account the effect of all the various system flexibilities, the method given in Figure 2 can be used to give the effect of the flexible coupling on that calculated critical speed.

From the vibration point of view, it was emphasised that a major advantage of gear couplings is not only their low weight, but also their short overhung length, particularly with reversed hub designs. It was also noted that because of the rigidity of bearing pedestals, particularly on high speed machines, non-contact probes measuring the shaft motion are a better check on unbalances than seismic probes mounted on the bearing housings. It was pointed out that using the latter type of probe virtually no vibration may be measured in the vertical direction, although there may be more vibration in the horizontal direction. Another delegate pointed out the importance of considering the number of balancing planes when using the German balancing standard VDI 2060. If there is only one correction plane, e.g. with a disc shaped rotor, then all the unbalance is corrected in that plane, but with an axially longer rotor with two correction planes then the required total balance correction is halved and then added to each correction plane. A reply to this point by the author of paper E1 is included in the next section.

Some discussion also took place on the methods of gear coupling spacer centring namely, torque centring and tip centring. One speaker noted that about half of the high speed gear couplings in service use tooth tip piloting and considered that high torque centring required a high coefficient of friction at the teeth contacts which is not advantageous from many other design viewpoints. Another delegate argued that torque centring was caused solely by the torque force acting on the flanks of the teeth and did not involve tooth friction and therefore tooth tip centring was only useful if the torque is not applied at speed. It was also pointed out that centrifugal strains at high speeds give clearances in the original tip piloting and that under torque, torque centring had the major effect because the flanks of the teeth are stronger than the tooth tip piloting projections.

The use of performance envelopes for initial flexible coupling selection was considered to be useful by several delegates and there were suggestions that some of the latest designs of couplings offer improved performance in terms of torque capacity at a given speed compared with the data in Figure 2 of paper E2. A user suggested a new method of plotting performance envelopes for flexible couplings with limits related to typical design criteria and it was agreed that such techniques could be of much practical use in the future. Written contributions to the discussion relating to both these areas are included in the next section.

One delegate noted that on 500 MW generating sets the shafts are solidly coupled and, although there is 1/8 inch distortion on the total length, the bearings operate satisfactorily without any signs of distress. Another speaker noted that quill shafts have high torque and speed capacity and suggested that more should be said about such devices.

AUTHOR'S COMMENT: J.S. WOODCOCK (Michael Neale and Associates Ltd., England)

Because of language difficulties during the discussion the comments on VDI 2060 were not fully understood at the time. The point was made by Mr. Brueggen and he was referring to the paragraph at the top of the table shown in Figure ED1 which is reproduced from an English translation of VDI 2060.

This paragraph has not been ignored in Paper E1, but is, in a sense, central to the arguments presented. The allowable values of unbalance (ew) listed in the table are effectively the 'total' allowable unbalance for a rotor assembly and the statement above the table simply defines the way in which this unbalance should be apportioned along the rotor assembly, depending upon the number of balance correction planes used. Throughout Paper E1 the very simple model of a massless shaft carrying a central unbalanced mass has been used and, consequently, for a rigid shaft the allowable unbalance for the central mass would be that value given in the table for the appropriate class of machine. This would result in each bearing experiencing an unbalance force equivalent to half of the total unbalance. If two balance correction planes were used, half of the value given in the table would be permissible for each correction plane so that, again, each bearing would experience half of the total unbalance. Though not specifically stated in Paper E1, this point is recognized and, on the basis of the forces experienced by the bearings, the paper goes on to suggest ways in which the rigid rotor balancing standard VDI 2060 might be modified to cover the balancing of flexible rotors and their couplings.

Table 1 Balancing quality grades and groups of rigid rotors
For rigid rotors with two correction planes, half the respective recommended value for each plane is generally taken, while the whole recommended value is taken for disc-type rigid rotors.
Directive: Fitting parts are included in the recommended values, where applicable.

Quality grades	*ew** mm/s	Rotors or machines examples
(none)	(>1600)	crankshaft drives† of rigidly mounted slow-running marine diesel engines with uneven number of cylinders
Q 1600	1600	crankshaft drives of rigidly mounted large 2-stroke engines
Q 630	630	crankshaft drives of rigidly mounted 4-stroke engines crankshaft drives of resiliently mounted marine diesel engines
Q 250	250	crankshaft drives of rigidly mounted high-speed 4-cylinder diesel engines
Q 100	100	crankshaft drives of rigidly mounted high-speed diesel engines with six or more cylinders complete motor car, truck, locomotive engines ‡
Q 40	40	car wheels, wheel rims, wheel sets, drive shafts crankshaft drives of resiliently mounted high-speed 4-stroke engines with six or more cylinders crankshaft drives of car, truck, locomotive engines
Q 16	16	high-stress drive shafts parts of crushing and agricultural machinery parts of crankshaft drives of car, truck, locomotive engines high-stress crankshaft drives of engines with six or more cylinders
Q 6,3	6,3	parts of process-plant machinery, centrifuge drums, fans, flywheels, centrifugal pumps, general machine and machine-tool parts, ordinary electric-motor armatures, high-stress parts of crankshaft drives
Q 2,5	2,5	rotors for fluid drives, gas and steam turbines, turbofans, turbogenerators, machine-tool drives, high-stress medium and large electric-motor armatures, small electric-motor armatures, pumps with turbine drives
Q 1 fine balancing	1	tape-recorder and record-player drives, grinding-machine drives, high-stress small electric-motor armatures
Q 0,4 extra-fine balancing	0,4	Armatures for precision grinding machines, shafts and discs; gyroscopes

*$w = n\,2\,\pi / 60 \sim n/10$ with w in $_{s}{}^{-1}$ and n in rev/min
† 'Crankshaft drives' comprises the crankshaft, flywheel, coupling, belt pulley, vibration dampers, rotating piston parts etc. (see Section 3.5).
‡ In complete motors, the mass of the rotor is understood to be the sum of the parts belonging to the crankshaft drive.

TABLE REPRODUCED FROM VDI 2060

(English Translation by Peter Peregrinus Ltd.)

FIG. ED1

M.M. CALISTRAT (Koppers Company Inc., U.S.A.)

In Paper E2 the maximum performance obtainable from various types of coupling is given in Figure 2. It would be helpful to know what sources of data were used for drawing these graphs as they do not fit very well with my experience. In Figure ED2 which is based on the original Figure 2 of Paper E2, the dotted line indicates the capability of high performance gear coupling. In can be seen that these couplings can transmit more power than contoured disc couplings.

I would also suggest that in the analysis of thrust bearing positions shown in Figure 1 of Paper E2, double ended machines should also be considered as these often cause problems.

B. VOWLES (Flexibox Ltd., England)

Figure 2 in Papers A and E2 give a rather misleading impression of the performance boundary for Multiple Membrane Couplings of the continuous ring form type. A more realistic boundary for this type of coupling is the dashed line shown on Figure ED2. This represents the approximate limitation for the available standard designs of multiple membrane couplings and many have been, and are being supplied for operation up to and even, in the case of specials, beyond this boundary.

W.J. STOUT (Imperial Chemical Industries Ltd., England)

For some time now we have believed that performance envelopes could be useful in coupling selection when drawn graphically and the overall performance charts for flexible couplings presented as Figure 2 of Papers A and E2 have provided us with a starting point.

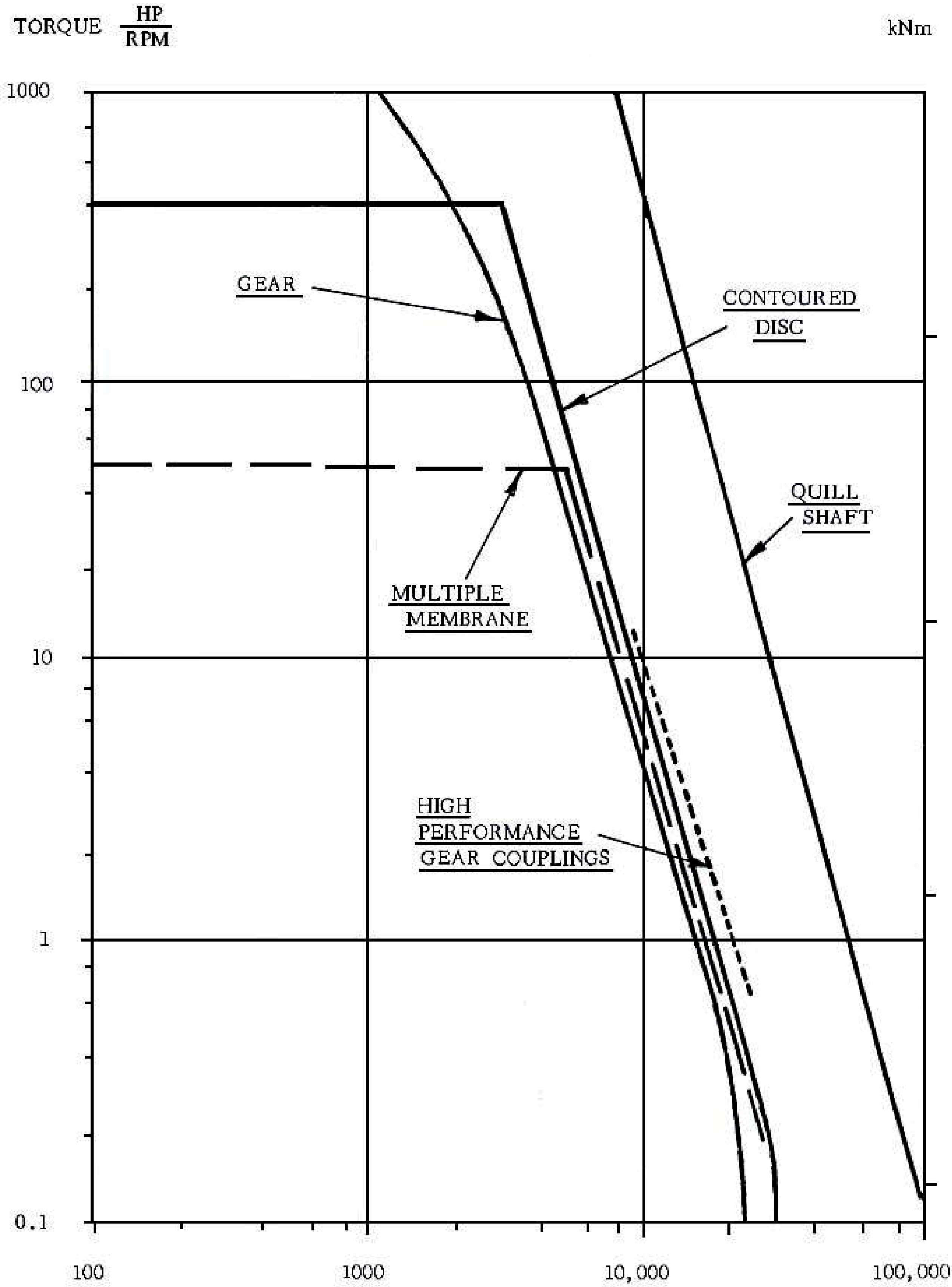
We use power times speed (K x N) and speed (N) as axes on our chart. We use K x N for two reasons. Firstly, it is a single parameter we have used for some time to assess the “size” of a coupling duty and, secondly, we believe it enables a better picture to emerge from the performance chart without altering any mathematical relationships.

Figure ED3 shows out first attempt, with elastomer insert couplings, which, although outside the scope of the conference, do present an easy example for demonstration purposes.

The left hand side is bounded by the manufacturer’s maximum permitted torque, which is of little interest to use since we rarely run couplings this slowly. The upper boundary is an empirical self imposed limit on our old parameter, K x N, for the coupling type. The right hand side is bounded by a self imposed speed limit at 3000 rpm (we only use these couplings on electric motors) and by a diagonal line which is empirically determined, but we believe it represents a limit to the centripetal acceleration the runner inserts are able to withstand:

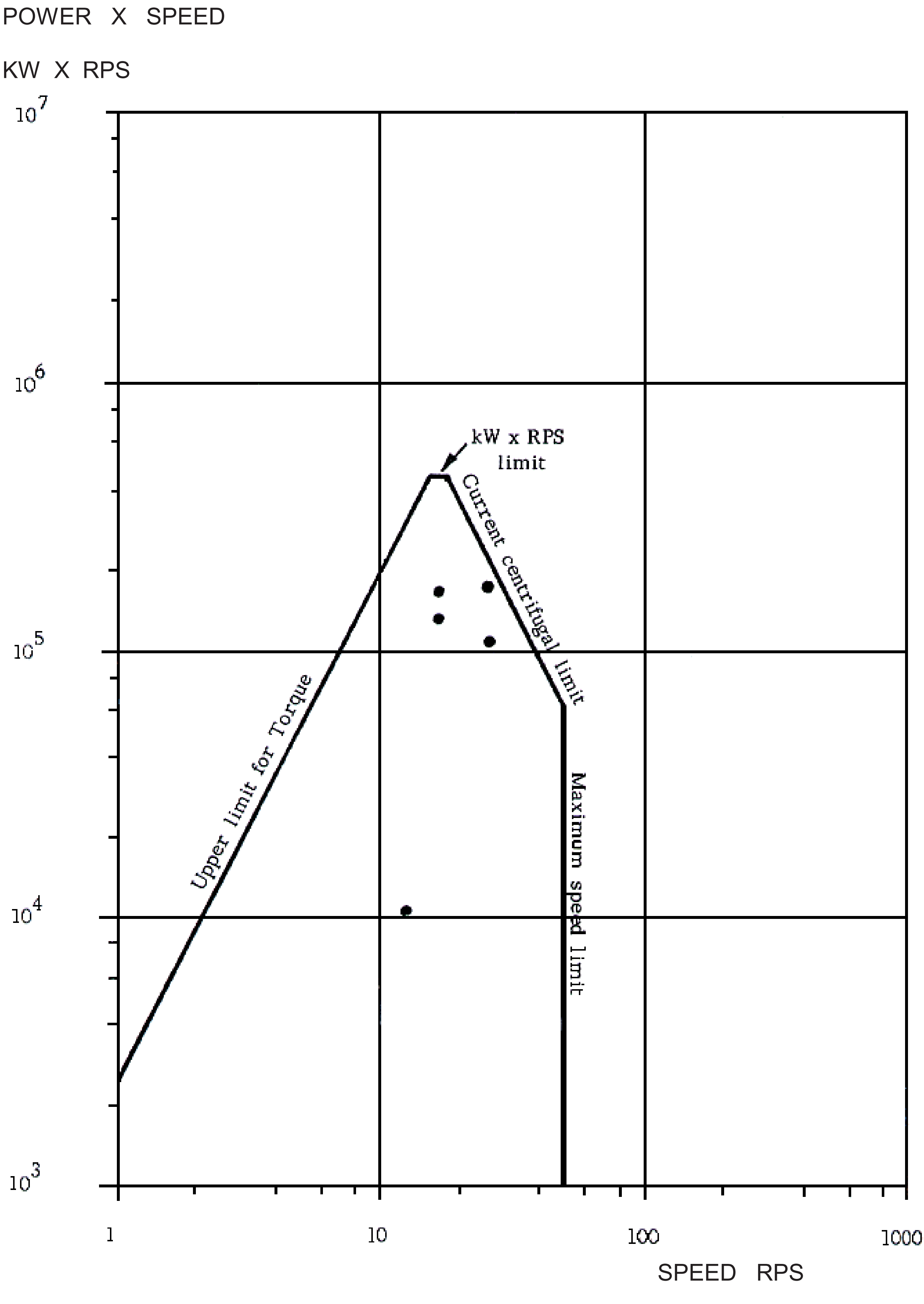
Viz: w^2r = const for centripetal acceleration

Alternatively N^2D = constant



APPROXIMATE PERFORMANCE BOUNDARIES OF COUPLINGS FOR HIGH POWERS AND SPEEDS

FIG. ED2



PERFORMANCE BOUNDARIES FOR ELASTOMER INSERT COUPLINGS

FIG. ED3

Generally, however, Torque $T \propto D^2$ for a family of couplings

hence $N^2T^{1/2} = \text{constant}$

or $N^4T = \text{constant}$

hence $TN^2 \times N^2$

or $KN \times N^2 = \text{constant}$

This is the mathematical form of the boundary line on our chart.

Figure ED4 shows a similar diagram for gear tooth couplings. The left hand side is the torque limit at low speed and is not our usual limitation. The upper boundary represents, as before, the K x N limit which is empirically determined.

However, if K x N or $TN^2 = \text{constant}$

$T \propto D^2$ as above

$D^2 N^2 = \text{constant}$

This is the mathematical form for constant hoop stress which seems to be a logical sort of limitation in a high speed coupling. Similarly, our limit on centripetal acceleration produces a boundary for oil retention type couplings by the same process used to fix the centripetal limit in Figure ED3. The right hand side of the chart is very important to us and there are several boundaries depending on properties of materials of construction. These lines have been decided on bitter experience.

Finally, we amused ourselves a little with the "Conti-Barbaran" criterion.

$$\varepsilon \propto \frac{K}{PN^3 D^2}$$

For any one coupling, P and D are fixed;

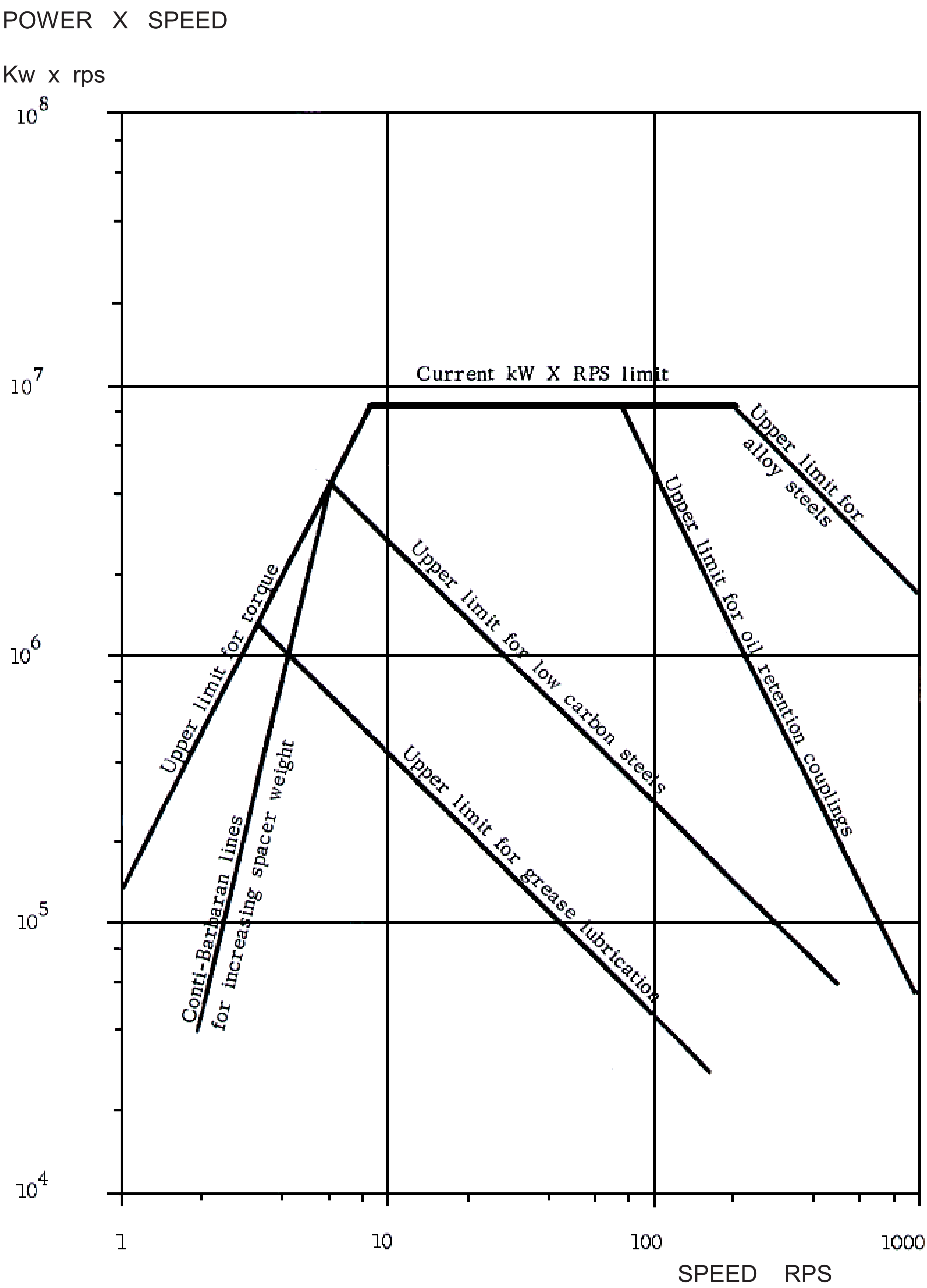
hence $\varepsilon \propto \dfrac{K}{N^3}$ for one coupling

The mathematical form of this line is drawn on the chart for any one coupling operating with ε at its critical values. The coupling in question can only satisfactorily operate above this line by appropriate choice of power and speed. The position of the critical line can be changed by changing the spacer weight or tooth PCD. The limits permitted for power and speed can also have an effect. The value of the critical ε can be changed (hence the boundary becomes a different line) by choice of material of construction (strength to weight ration, nitrided teeth etc.).

It is obvious that much more investigation is necessary before the chart is of great practical use, but we think there are definite possibilities.

AUTHOR'S REPLY: M.J. NEALE (Michael Neale and Associates Ltd., England)

I am grateful to Mr. Calistrat and Mr. Vowles for their suggested improvements to the coupling selection diagram. This was originally prepared by collecting together all the available published information on coupling performance from technical papers and



PERFORMANCE BOUNDARIES FOR
GEAR TOOTH COUPLINGS

FIG. ED4

manufacturers' component catalogues, and then drawing the boundary lines around it. The clarity of diagrams of this kind invariably stimulates manufacturers to supply more accurate data and this is perhaps one of their advantages. The new performance line for multiple membrane couplings suggested by Mr. Vowles is quite a major change compared with the data which I had available when this diagram was originally drawn about two or three years ago. This presumably reflects the considerable developments that there have been on multiple membrane couplings since then, particularly in relation to higher precision and improved balancing standards.

Mr. Stout has suggested some alternative methods of drawing performance diagrams which he has developed to suit his particular area of coupling application. There are many ways in which this kind of information can be presented, but my own experience based on preparing diagrams of this kind for a variety of engineering components is that the following simple rules should be followed:-

1. Consider carefully the exact function or functions of the component.

2. Plot these functions graphically in relation to component size and operating speed on logarithmic axes, to enable a wide spread of numerical values to be included on one diagram and to get straight lines wherever possible.

3. Consider what are the actual physical limits which determine the boundaries and check how the slope of the lines corresponds to these limits.

This is largely what Mr. Stout has done and there will be various forms that the diagrams can take depending on which variables are put on the axes and which are taken as constants for the various lines in turn will affect the shape of the boundary lines drawn round the whole performance atlas. When it is done well the diagram should appear almost obvious.

Based on these criteria I am not sure that my presentation is ideal and it is, in any case, a boundary diagram only, while Mr. Stout's is really a suggested basis for a family of performance curves.

For flexible couplings I would suggest that the function of the component is to provide a connection between shaft ends capable of transmitting a driving torque and allowing lateral misalignment and axial displacement. The diagram should therefore, ideally display a relationship between:-

Torque
Lateral misalignment
Axial displacement
Size
Rotational speed.

All these variables are easy to visualize physically which helps the user in interpreting the meaning of the diagram and in this respect I do not support the use of Power x Speed as a variable. The diagrams are, however, a good start in the right direction.

Mr. Calistrat has pointed out that double ended machines are more difficult to design in terms of thrust bearing location and this is certainly true. The general principles in terms of sliding and fixed feet etc. are still the same as indicated in Figure a of Paper E2, however, and have been successfully applied to the design of steam turbine driven turboalternators, which in larger sizes can consist of up to three double ended machines connected in series.