UNIVERSITY OF SOUTHAMPTON

RETRIEVAL OF COMPATIBLE COMPONENT INFORMATION FOR SYSTEM DESIGN

Volume 2 of 2 - APPENDICES

by

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APPENDIX A

DISCUSSION OF THE EXTENDED ANALOGOUS PARAMETER LIST FOR MOTORS

The following section is the definition and identification of analogies for the characteristics in the extended parameter list for motors. It would constitute Section 4.1 of Chapter 4.

Most of the electrical, physical and performance characteristics for electric motors are defined in various parts of the British Standards 4999 and 5000. When appropriate, reference to the exact part will be made. Fluid (hydraulic and pneumatic) motors do not have any standards defining their characteristics. Thus, the definitions below, are generalised to enable them to be applicable (whenever possible) to all motor types. REF. 10 is a list of all the British Standards dealing with electrical machines.

A4.1.1 Rated Power

This is the useful power produced by the motor. It is the product of the rated torque and rated speed of the motor (these are the primary output characteristics of the motor). This characteristic will be quoted for all motors, it provides a rough idea of relative size for a particular type of motor.

A4.1.2 Rated Torque

This is the value of torque the motor will produce constantly when supplied with its rated input and operating at rated speed. The motor will therefore produce its rated torque at its rated power and speed.

A4.1.2 Rated Speed

The speed at which the motor will operate continuously whilst providing rated torque thereby producing rated power. The significance of the latter two characteristics is that they are continuous values. The motor should be able to run at these two values without failing before the manufacturer's quoted lifetime or maintenance period. Thus, a motor providing a steady rotary power source, should be selected to produce these characteristics, thereby allowing the motor to work most efficiently. A motor running continuously too far below these criteria will be costly (because it will generally be oversized) and inefficient (the conversion of input power to rotary mechanical output will not be as efficient as that for the rated values).

A4.1.4 **Starting Torque**

The torque available when the motor starts. This is an important characteristic because it is the starting torque which will overcome the system inertia and cause the motor to accelerate the system up to the desired speed. It is usually higher than the rated torque because the torque required to accelerate a system (i.e. overcome its inertia) is greater than the torque required to keep a system rotating at a fixed speed (this torque only overcomes losses - any further acceleration requires a larger torque - Newton's First Law).

The starting torque is an instantaneous value and may well cause instantaneous excesses of the supply power components. Thus it is not a value which should be called upon continuously in certain types of motor because the supply may become overloaded and damaged, or the motor itself will be damaged. This is particularly true in electric motors, both ac and d.c. There are two main reasons, firstly, if the motor is stalled, the windings will not be cooled by their motion, or by the fan which is sometimes built onto the rotor shaft. The windings will therefore overheat causing their insulation to degrade, resulting in short circuits etc., ruining the windings and making the motor unserviceable.

Fluid motors can endure their stall condition but in general, the starting torque is not a great deal higher than the rated value. The starting torque may also be called the "locked rotor" or "stall" torque.

A4.1.5 **Pull Up Torque**

This is primarily a characteristic of a.c. synchronous motors. It is defined as the minimum torque produced between starting and rated torque. It is defined because in some of the smaller, single phase, induction motors, using a capacitance start, the pull up torque is less than the starting torque. This means that the full starting torque is not available to accelerate the load and if the designer assumes this to be true, when using this type of motor, the system will not perform as expected. In the case of most d.c. and fluid motors, the starting torque is higher than the rated torque and the progression between the two is fairly linear (d.c. shunt wound, most hydraulic motors). The pull-up torque would thus be equal to the rated torque for most cases.

A4.1.6 **Pull Out Torque**

This is another a.c. motor term and is used to indicate the maximum torque produced between the pull-up and rated torque. This is in fact the largest torque produced by these machines. If this torque is exceeded, the motor will rapidly stall. It can be defined as the maximum torque the motor can produce. In the general case, the motor will stall once this value is exceeded. For electric motors it is an intermittent value, since the motor cannot run continuously at this torque without damage to the winding insulation caused by heating. In fluid motors, it can be translated to the maximum pressure the motor can be run at before excessive seal damage occurs. In this case it will also be an intermittent vale. However, it is a torque that the designer may be able to utilise in certain circumstances.

A4.1.7 **Static Friction Torque**

This is the torque caused by the mechanical losses in the motor. Bearing friction and hydrodynamic friction are the primary contributors. It is difficult to ascertain exactly whether this torque includes the rotor inertia and its associated windage / churning loss. If this were true, it would probably have analogies with the breakaway pressure of fluid machines. This analogy is also discussed in Chapter 4, Section 4.2.9.

A4.1.8 **Maximum Speed**

This is the maximum speed the motor can produce. Again, it may be an intermittent value because of wear on bearings etc. However, many motors are controllable over a speed range and this value identifies the upper end of this range. It is sometimes called the no - load or free speed, this implies a motor running without a load, it is therefore unattainable in many practical cases, but, it does give an indication of the speed range. The no load speed has analogies with synchronous speed and is discussed in Chapter 4, Section 4.2.8.

A4.1.9 **Reversibility**

Knowing whether a motor is reversible or not is clearly important from a design viewpoint. Certain fluid motors may not be reversible because of porting constraints, sealing or design of flow paths. Most electric motors are reversible. This definition of reversibility should not be confused with the concept of reverse power flow, where the system may drive the motor. In some cases the motor may act as a brake. If connected to a high reduction worm and wheel gearbox, a locked drive will result - the wheel cannot drive the worm because of the high inefficiency of the large reduction being used.

A4.1.10 **Rated Input**

This is the product of the input power components. It will always be higher than the output power because there will be losses in the power conversion through the machine. It is derived from electric motors where the rated input is the product of the rated current and voltage. These are the values of current and voltage the supply provides to the motor when operating at rated torque and speed. If a power factor is also quoted with these values, the input power is the product of current voltage and power factor. In fluid machines, the rated input is the product of the fluid pressure and flow rate for rated torque speed conditions.

A4.1.11 **Efficiency**

The overall efficiency of the motor. The rated input power multiplied by this figure should yield the rated power. Efficiency will include all of the machine losses such as mechanical, electrical and fluidic losses. The product of these individual losses will result in the total motor efficiency.

A4.1.12 **Mechanical Efficiency**

This allows for the mechanical losses occurring in the motor. These will be caused by bearing friction, rotor windage, fan torque (in electric motors), churning and viscous drag losses (in fluid motors).

A4.1.13 **Electrical Efficiency**

This is primarily concerned with the electrical losses in electric motors. It simply measures the total losses in the supply power components. These will include resistive, inductive and capacitive losses which can be summarised as the power factor for the input components. Losses which may not be part of the power factor such as magnetic losses will also be included. This parameter can be renamed as supply efficiency thereby allowing an analogous parameter to be developed for fluid machines. There are flow and pressure losses in fluid machines, these are discussed in Chapter 4, Section 4.2.5.

A4.1.14 **Rated Voltage**

This is usually the supply voltage, the R.M.S. value in a.c. systems and the supply voltage in d.c. systems. The significance of supply voltage in d.c. systems is that it is one of the variables determining motor speed. In a.c. systems, the motor speed is not dependent on the supply volts. The direct analogy in fluid systems would be the supply pressure. Unfortunately, the anomaly caused by the crossed over nature of electric machines means that speed and voltage are associated in d.c. machines (rather than speed and current as would be expected.). Thus, the correct fluid analogy would be fluid flow rate which of course is not consistent with the classical analogy. The problems with this cross over effect are debated in Section 3.2.

A4.1.15 **Rated Current**

This is the current the motor draws whilst running at its other rated values. It is not the maximum or overload current sometimes stated (see start current Section A4.1.18, below). In generic terms, current is the "supply through variable" - see Chapter 2. Thus, fluid flow rate is the fluid analogy.

A4.1.16 **Power Factor**

This is a measure of the effective electrical power losses caused by the various impedances in the motor windings. The product of rated current, rated voltage and the power factor will give the electrical power used by the machine. There is no apparent equivalent in d.c. or fluid systems.

A4.1.17 **Armature Current**

This is a limiting factor in d.c. machines. The armature current cannot exceed this value for a continuous period because the windings will be damaged. The generic title for this parameter would be "rotor supply through variable".

A4.1.18 **Start Current**

This is a limiting factor in all electric motors, high staring currents are needed to help produce high starting torques. However, for efficiency, the motor windings are designed for rated operation. Thus, the start current is an intermittent maximum which is why the electric motor cannot be allowed to remain stalled. Generically, it can be termed "starting supply through variable" and is analogous to the supply pressure in fluid machines. It is here where the anomaly caused by the crossed over nature of electric machines must be remembered and the correct analogy chosen. The direct analogy for current is flow. However, it is noted that the electric machine has the cross over equating current and torque rather than current and speed. Thus, to keep the analogy consistent with fluid machines, the fluid pressure is what must be considered since it governs torque in this domain.

A4.1.19 **Stator Power**

This is the electrical power developed in the stationary windings in the motor casing. In the case of a.c. motors, this will be a rotating magnetic field, in d.c. motors, it will be the permanent poles set up by the field windings. In fluid machines, there is no analogy to "rotor power" because no fluid is used to power the rotor. Thus, the stator power would be the fluid power available from the supply.

A4.1.20 **Stator Excitation Volts**

This parameter is a component of the stator power parameter defined above. It is the supply voltage for a.c. machines, the field excitation voltage in d.c. machines and the fluid pressure in fluid machines.

A4.1.21 **Armature Resistance**

This is the resistance of the rotor winding in d.c. machines. Certain a.c. machines will also have a wound rotor, hence a value for resistance can be found. The significance of this value is mainly for starting considerations. Certain starting techniques involve the addition of series resistances to the rotor windings to limit the starting current. The resistance of the armature alone, will indicate the peak current to be expected from a given supply voltage, the size of the additional series resistance to limit this current can then be determined. Although this criterion is most important for d.c. machines, it should be available for a.c. machines as well because resistance starting techniques are also employed. REF. 6. There is no comparable mechanism to armature characteristics in fluid machines, so no analogy can be found for armature resistance in fluid systems.

A4.1.22 **Armature Inductance**

This is again mainly a d.c. motor characteristic. As a transient characteristic in d.c. systems it will effect the motor's response to supply changes. Thus, it will probably be most significant when considering the starting conditions for the motor. The d.c. rotor field windings of a.c. motors will therefore have similar considerations. Again, no analogy has been developed for this characteristic for fluid machines. Fluid inertia, is a characteristic that may have a similar affect. This is discussed more generally in Chapter 4, Section 4.2.6.

A4.1.23 **Capacitance Values**

This is purely concerned with the starting conditions for single phase a.c. motors and is unique to them. The use of capacitors in these a.c. motors must not be confused with the idea of capacitance at a "circuit" level. The capacitance value is not the capacitance of the "motor" as a circuit element, it is the value of the capacitance employed within a section of the motor windings to change the phase of the supply voltage. This produces a pseudo two phase winding, allowing a rotating field to be generated, making starting much easier. This technique is not employed in other domains, hence there are thus no analogies in d.c. or fluid systems.

Capacitance of windings and the motor as an element, for which analogies exist in all domains, is discussed in Chapter 4, Section 4.2.6.

A4.1.24 **Number of Phases**

This is an a.c. motor characteristic. The study of the analogies with the other energy domains has yielded some interesting concepts which are discussed, in full, in Chapter 4, Section 4.2.7.

A4.1.25 **Frequency**

This is a characteristic of the a.c. supply. Again, the study of this parameter along with the above characteristic is discussed, in Chapter 4, Section 4.2.7.

A4.1.26 **Rotor Poles**

The number of rotor poles in a synchronous motor determines its speed. Rotor poles also determine d.c. motor speeds but are not as significant. The analogies with fluid machines are discussed, in detail, in Chapter 4, Section 4.2.7.

A4.1.27 **Maximum Locked Rotor Time**

This parameter is a limiting factor for the allowable time an electric motor can be stalled without causing damage to the winding insulation etc.. This parameter can be said to be infinite in the case of most fluid motors as they are designed to allow continuous stall conditions.

A4.1.28 **Maximum Start Time**

Many types of electric motor take time to reach their rated values. This is mainly caused by attempts to reduce the starting current to avoid excessive heating. So, a gradual stepping up of current is employed, resulting in a much slower acceleration under load. Thus, this time period is not merely a symptom of rotor inertia which is the limiting factor for fluid machines.

A4.1.29 **Intermittent Rating Period**

This parameter is fully described in BS 4999 part 1. (REF. 10). For convenience it will be summarised here: The time for the machine to be cycled repeatedly without it reaching its full duty, under rated conditions. In other words, the number of starts and stops the machine can take before getting too hot. As with maximum start time, this may also be irrelevant in fluid machines because of their ability to endure rapid cycling without undue effects.

A4.1.30 **Maximum Intermittent Current**

This is the current drawn when a machine is cycled under intermittent rating conditions. The starting current is always much higher than the rated value and will therefore, on average, be applied more often to the windings when the machine is started, stopped and then started again. Thus, if repetitive stops and starts are employed, the machine will not have the cooling period associated with continuous operation, the mean temperature will therefore increase as the result of the effectively longer time the starting current is applied. This may mean that in certain cases, the starting current must be reduced to stop excessive heat built up.

The fluid analogy is starting pressure as mentioned above. This parameter is less significant in fluid machines although damage to seals through excessive pressure is possible.

A4.1.31 **Type of Start**

i) Direct On Line

This is the simplest starting type because the motor is switched directly into the supply. This is possible for many low and medium power electric motors but is inadvisable when high starting currents are possible. Thus an alternative starting method must be used. Most fluid motors can be started on line because, as described earlier, the full supply pressure will not damage the motor.

ii) Star / Delta Start

This is an a.c. method of reducing the staring current. No analogies in the other domains have been found as yet.

iii) Resistance Start

This is by far the most common method (other than an electronic type controller) used to reduce high starting currents. Resistances are simply placed in series with the windings and then gradually switched out as the motor accelerates to its rated values. The principle is similar in both d.c. and a.c. motors and the analogy in hydraulic systems is a straight forward one - valves. Valves produce a head loss and will thus allow a gradual increase in the staring torque of the fluid machine.

In fluid systems, this technique may not necessarily be used to protect the motor, but to protect other elements in the system from experiencing a large shock torque as the motor rapidly accelerates.

iv) Auto Transformer

This is another a.c. technique used to limit current. Again, no analogies for other domains have been developed.

A4.1.32 **K.V.A. Code**

This parameter is an a.c. motor characteristic. The full definition is given in BS. 4999 part 41 (REF. 10). This essentially deals with the starting characteristics of induction motors. It covers data such as the apparent input power the motor can absorb, with the rotor stalled, under full load conditions.

A4.1.33 **Maximum Operating Temperature**

Excessive heating can cause damage or a drop in performance of all types of motor. Thus, the maximum ambient temperature the motor can endure, whilst producing rated output is something designers must consider. (REF. 10).

A4.1.34 **Altitude**

Altitude can also affect performance of electric motors, particularly when fan cooling is used. (REF. 10).

A4.1.35 **Loads On Output Shaft**

Any axial or radial forces on the output shaft caused by the load will effect the life of the shaft bearings. Thus it is important to know the maximum permissible continuous axial and radial loads that the shaft bearings can endure for a given speed to ensure the lifetime of the motor. This parameter clearly applies to all motor types.

A4.1.36 **Rotor Mass**

The mass in kg of the rotating parts of the motor. The significance of rotor mass is that it is connected with the moment of inertia of the rotor. This is a very important characteristic because it contributes to the system inertia, which in turn, affects the starting requirements of the motor. Thus, if rotor mass is quoted, the radius of gyration must also be given. The product of mass and radius of gyration gives the moment of inertia.

A4.1.37 **(Rotor) Radius of Gyration**

This is the moment of inertia of the rotor divided by the rotor mass. The units are meters squared and it is often denoted as k^2.

A4.1.38 **Moment of Inertia of Rotor**

The moment of inertia of the rotating parts of the motor. This is an important characteristic for the reasons given above. It can often be ignored because it is small in comparison to the inertia of the load. It becomes more significant in smaller machines where the starting torques available are not proportionally as large as the bigger machines.

A4.1.39 **Moment of Inertia of Load**

The maximum moment of inertia of the load the motor is to drive. If this value is exceeded, the starting torque provided by the motor will be insufficient to accelerate the system to its operating speed.

A4.1.40 **Motor Mass**

The total mass of the motor in kg.

A4.1.41 **Frame Type**

BS 4999 part 10 has full details as to the frame types available for a.c. electric motors (REF. 10). The significance of the structuring of frame types in AC motors is that the frame code is a direct reflection of the motor casing size. Thus, given the frame code, a standard set of dimensions can be applied. When selecting a motor, a separate set of dimensional data would not have to be considered. The various sets of types of frames could be represented by a set of parametric drawings. The motor dimensions would be implied and encoded by the frame type, thus, reducing the amount of data stored. Unfortunately, not all of the d.c. motor and none of the fluid motor suppliers, follow the same structured classification for motor frames. Thus, the idea discussed above cannot be used.

The motor frame code will play an important part in the retrieval process. Each motor has its own, unique, frame code, allowing suppliers and designers to distinguish between the motors. Thus, when a retrieval is made, the exact motor on a catalogue page, can be identified.

A4.1.42 **Mounting Type**

This also is an area where no structuring has been undertaken for fluid machines but has been standardised for electrical machines. BS 4999 part 22 (REF. 10) allows both the mounting type and orientation of the motor relative to its mounts to be classified using a simple code. These classifications will also be applicable to fluid motors. This characteristic is dealt with in greater detail in Chapter 4, Section 4.3.4.

A4.1.43 **Enclosure**

This is mainly an electric motor consideration. BS 4999 part 20 (REF. 10). This will also cover data such as cooling forms (BS4999 part 21) and protection from ingress of water or foreign bodies (BS4999 part 20). This data is fairly irrelevant for fluid motors since the motors tend to be completely sealed from the surroundings.

A4.1.44 **Insulation**

This is covered by BS 2757. This is clearly not applicable to fluid machines.

A4.1.45 **Noise**

This characteristic is covered by BS 4999 part 51. It can be applied to all motor types.

A4.1.46 **Regulation**

Regulation is the speed control of fluid motors by using a valve. Thus, the electrical analogy would be to control motor speed by resistive means.

A4.1.47 **Balanced**

If a fluid motor is balanced it indicates that its action requires at least two inlet ports to the rotor. The advantage of using a balanced system is that any forces on the rotor bearings are equal and opposite thus increasing bearing life allowing better performance.

Most electric machines are balanced by the nature of the opposing magnetic fields used.

A4.1.48 **Filtration**

The working fluid in fluid machines must be kept free of contaminants because they may attack seals or scratch machined faces etc.. Analogies, if any, in electrical machines have not been investigated.

A4.1.49 **Pressure Loss on No Load**

This is the drop in pressure across the motor caused by the internal losses within the motor. This concept is discussed in Chapter 4, Section 4.2.2. Also, it is clear that there will be voltage drops in electric motors caused by internal resistance etc. A problem exists with d.c. motors because of the back e.m.f. generated. This will essentially give an internal voltage drop equivalent to the supply voltage. However, if this value could be ignored and the intrinsic losses identified, the d.c. analogy could be quantified. There will also be a loss caused by the impedance of the windings in a.c. machines.

A4.1.50 **Delivery**

This is the amount of fluid leaving a fluid machine by the outlet port. It will be slightly less than the volume of fluid entering, because there will be a certain amount of leakage within the machine, which is removed via the casing drain. These losses in fluid machines and their analogies with other devices are discussed in Chapter 4, Section 4.2.5.

APPENDIX B

DISCUSSION OF THE EXTENDED ANALOGOUS PARAMETER LISTS FOR GEARBOXES

Appendix B is Section 5.5 of Chapter 5. The philosophy behind this discussion is outlined in Chapter 5, Section 5.3. Many of the analogies are explained in in deatail in Chapter 6.

B5.5.1 **Driver Characteristics**

This is a general term prompting the designer to consider the characteristics of the driver being used. The inclusion of a gearbox is the result of some inadequacy in the driver performance. This consideration will help the user to identify characteristics which may affect the service factor for the gearbox; is the driver starting "on-line", are high starting torques present, is the driver cyclic i.e. reciprocating power source? It is recognised that a good designer would have considered these details.

The need to assemble "matched" systems would basically cause the designer to "carry over" from the preceding component all the necessary information. One aim of the research work is to ensure that the characteristics linking adjacent components are consistent making the matching process straightforward. In the case of a transformer, if it is to be used for impedance matching, then the driver impedance must be known. The possible analogy with gearboxes is discussed in Chapter 6, Section 6.2.

B5.5.2 **Input Power**

The power going into the gearbox is the product of rated speed and rated torque of the gearbox driver. The transformer is not usually rated in terms of "power" because of the effects of impedance on phase and therefore power factor. These facts are dealt with in greater detail below under Output Power. The input power of a transformer can be calculated from the rated input current and voltage at full load and the phase angle between these two values.

B5.5.3 **Rated Input Torque**

The torque the gearbox can transmit at rated speed and power without damaging the gearteeth or bearings. In Chapter 6, Section 6.3 it was concluded that torque through the gearbox was closely related to current in transformers because of the concept of "flux". Thus, the analogous parameter in transformers is the rated input current.

B5.5.4 **Starting Torque**

Starting torque is significant because the gearbox may have to accelerate a high inertia load. Thus, the starting torque is likely to be greater than the rated torque and care must be taken to remain within the overload limits of the gearbox.

Considering the transformer, the input current is a function of demand (see Appendix , Figures. 6.2 and 6.5). The primary current rises with the secondary current. The maximum value of current will be stated by the manufacture (BS 171 pt 1 [Ref. 22]). Interestingly, the larger the load (or inertia) in rotary systems, the larger the starting torque. Hence, consistency can be shown at this fundamental level.

The design of belt / chain systems calls for a similar consideration to be made, albeit indirectly. The use of speed against power tables essentially sets the torque limit for the belt or chain being selected. The greater the torque, the more robust the belt / chain. This torque limit will primarily avoid slipping in flat belts and stretching / wear in toothed belts and chains.

The term "abnormal starting loads" will distil to maximum starting torque for gearboxes. The maximum voltage for each winding is quoted in BS 171. The analogy with current has however, been developed. The high voltages affect the insulation. For any transformer, the relationship between voltage and current in a given winding can be defined. Thus, in this case maximum voltage can be used to derive maximum current in the windings. This will also affect insulation because of heating.

B5.5.5 **Rated Input Speed**

In most gearboxes, this will be a constant speed. The gearbox, by its nature, will modify this to the appropriate value. In the industrial situation, gearboxes are used to reduce the speed of electric motors which generally operate in the low thousands of R.P.M.

The electrical analogy of velocity is voltage, which in the case of the supply voltage of a transformer, is a constant. Tied into the supply voltage is the supply frequency. This characteristic was mentioned in Chapeter 6, Section 6.4, it primarily affects hysteresis and eddy current losses which are discussed in Chapter 6, Section 6.5.3.

The overall comment about these first five characteristics is that they are all motor output characteristics. This observation may seem trivial, however, many manufacturers of motor and gearboxes do not provide all of these basic characteristics. This comment reinforces the need for a consistent system of component classification and thereby will help the rapid assembly of matched systems.

B5.5.6 **Moment of Inertia at the Input Shaft**

The inertia of the rotating parts of the gearbox referred to the input shaft. This means that the output gear inertia is modelled as part of the input gear inertia by adjusting its value using the speed ratio. The mathematical manipulation is given in Appendix P.

Using the inertia / inductance analogy, the inductances in the primary and secondary windings can also be referred to the primary using the turns ratio. In many cases, the entire output side impedance can be referred to the primary winding as in the approximate equivalent circuit (Appendix J, Figure 5.5). This technique allows a complex situation to be modelled very easily. A similar method is used by gearbox manufacturers to allow the designer to derive the system starting torque when a large inertia is to be accelerated.

The inertia / inductance analogy and associated losses are discussed in Chapter 6, Section 6.5.2.

B5.5.7 **Maximum Friction Torque at Input**

This is the amount of torque "seen" at the input opposing the input torque. The term "friction" not only means bearing friction but will include "opposing torques" caused by churning and pressure drag caused by lubricants and friction between the gearteeth. These losses are discussed in Chapter 6, Section 6..2. As with inertia, the friction torque on the output side can be referred to the input side using the tooth ratio. This is shown in Appendix P.

B5.5.8 **Type of Driven Machine**

This consideration causes the designer to think about the type of load the gearbox is to drive. The type of load will affect the gearbox duty and hence service factor. The heavier the duty, the larger the service factor and hence a bigger device is selected. The significance of supply and load in the case of transformers used for impedance matching is discussed in Chapter 6, Section 6.5.1.

B5.5.9 **Rated Output Power**

This is the product of rated output torque and rated output speed. The difference between this figure and the input power (section B5.5.2) will indicate the losses in the power transmission through the gearbox and will hence yield gearbox efficiency.

The most significant transformer characteristic from suppliers' catalogues was the output rating in volt-amperes. This is the product of rated root mean square (R.M.S.) voltage and rated RMS current with a unity power factor. In this sense it is the maximum output power because the power factor will be less than or equal to one when a load is present. The output volt-amperes is a direct measure of the 'size' of the transformer and therefore gives the user a similar feel to the "power" of gearboxes. Terms such as maximum continuous power and maximum continuous volt-amperes can be considered to be the same quantity.

B5.5.10 **Peak Power**

This is an intermittent value. It is the greatest power the gearbox can transmit for short periods before severe damage is done. The damage is usually to the gearteeth. In the case of belts and chains damage to the belt / chain may occur as torque and speed limits are approached.

Transformers like all electrical machines have power limits, the intermittent power overload is given as 1.5 times the rated value in BS 171 pt. 1.

B5.5.11 Rated Output Torque

The maximum continuous torque the gearbox can transmit at the output shaft without damage to the gearbox. It is the torque available from the gearbox to drive the system. In many cases where the gearbox is used as a speed reducer, the associated increase in torque must be considered. Excessive starting torque, for instance, could damage other system elements such as couplings.

The analogies between gearboxes and transformers for this characteristic are discussed in Chpater 6, Section 6.5.6.

B5.5.12 Maximum Output Torque

The largest torque available at the output shaft. This will be an intermittent value, it could be identical to the starting torque if high inertial loads are to be accelerated.

In transformers, the load dictates the output current which then affects the input current. Certain loads will draw very high starting currents (electric motors), so, instantaneous, peak values of current must be considered in order to avoid damage to insulation etc. Depending on the direction of "step" the peak current may occur in either winding, therefore the designer must be careful to calculate the current in both windings using the turns ratio.

A similar problem can be seen to exist in gearboxes if the designer designs to a torque that the output of the gearbox can transmit but cannot be provided by the driver through the tooth ratio.

B5.5.13 Number of Outputs

Many transformers have "tappings" which are alternative terminals attached at points along the windings. If used they will clearly alter the overall turns ratio. In most cases, it is the secondary winding which is "tapped" allowing a variety of output voltages.

Several possible gearbox constructions can be proposed to mimic this effect. A "Derailleur" sprocket and chain set on a bicycle is a very clear analogy. A vari-drive gearbox does not have discrete ratios, but, the effect is also comparable. An inelegant, impractical and expensive analogous gearbox would be one where the input shaft drives a number of output shafts through a series of differing gearwheels. Whichever output shaft produced the correct torque / speed requirement could be used, the others left idling. If two outputs were used, the input torque would change. This gearbox is fictitious, but a lateral step would produce the epicyclic gearbox. The analogies between epicyclic gearboxes and transformers are discussed in Chapter 6, Section 6.5.5.

B5.5.14 **Nominal Output Speed**

This is the rated output speed. It is therefore the basic speed the gearbox has been designed to run at continuously without damage to the gearteeth, bearings etc. It is analogous to the transformer output voltage. The detail is given in Chapter 6, Section 6.5.4.

B5.5.15 **Output Speed Range**

If the gearbox has a variable ratio, the output speed will have a range. The tapping range on the winding of the transformer is the clearest analogy. The transformer voltage regulation is an indication of the output voltage change from no-load to full-load. It is caused by the secondary impedance. A discussion as to whether a single speed gearbox would experience "regulation" is given in Chapter 6, Section 6.5.4.

B5.5.16 **Backlash Measured at Output**

Manufacturing tolerances will always give a certain amount of movement of the meshing gearteeth. This movement can be measured by locking the input shaft and carefully rotating the output shaft. Any angular rotation before the teeth mesh is known as backlash. Backlash causes two main problems. Firstly, if high torques are being transmitted, excessive play will cause the teeth to impact upon each other rather than a gradual rolling transfer of power. This of course will damage the gearteeth by fatigue. Secondly, if positional control is desired, play in the gearbox will cause errors.

In belt and chain drives, provided a tension is maintained in the belt / chain, relative movement of the drive pulleys / sprockets should be minimal. There is a possibility that if large inertias are being started or cycled the belt / chain may be stretched by elastic deformation. This is a very unlikely event because the belts and chains are selected in such a way that the tensile loading does not cause significant stretching.

The analogy of mechanical backlash in an electrical transformer is discussed in Chapter 6, Section 6.5.3.

B5.5.17 **Power**

In the case of gearboxes, this is a general figure and does not refer specifically to the input or output. For an efficient gearbox, input and output powers will not be significantly different. Power in volt-amps referred to the output is the quantity generally available for transformers (Section B5.5.9)

B5.5.18 **Effective Power**

This is also called the design power. Once the service factor (see Section B5.5.27 below) has been established, the "system power" (the power the system is designed to transmit) is multiplied by the service factor to give the effective power of the device. The effective power is always greater than or equal to the system power. Under extreme service conditions, the effective power may be up to three times greater than the system power.

Service factors are also applicable to transformers and are related to ambient conditions and duty (how the transformer is used). These are mentioned in BS 171 pt. 1 and are usually derived as part of the supplier's selection procedure.

B5.5.19 **Full Load Efficiency**

The full load efficiency of a gearbox is a measure of the losses in the gearbox whilst operating at rated values. Manufacturers will attempt to minimise losses in the gearbox so that there are negligible torque losses at the rated values. The efficiency of transformers varies over the operating range, resistive losses in the coils increase with current. There is, however, a design detail which will allow the efficiency to be a maximum for rated values.

This is fully explained in Reference 6. The implications for gearboxes are discussed in Chapter 6, Section 6.5.6.

B5.5.20 **Nominal Gear Ratio**

Most 'bought out' gearboxes are of a fixed ratio. The problems associated with manufacturing gears with a whole number of teeth around a certain pitch circle diameter will cause exact ratios to often be impossible to achieve. Thus, a reduction gearbox may be quoted as having a ratio of 1.5. The actual tooth ratio may be 26/17 (= 1.53). Thus, care must be taken if final speed is very important. The term "nominal" should make the designer aware of this potential error.

Tolerances on voltage ratios (the turns ratio) for transformers are outlined in BS 171. The tolerance is less than one percent and this should be achievable because the high voltage winding will contain a lot of turns and can therefore be carefully adjusted towards a specific value. Also, because there is no direct contact between the windings there is no need for the physical constraints experienced with gearteeth.

The treatment of gear ratio(s) is discussed, in Chapter 6, Section 6.6.1.

B5.5.21 **Type of Reduction**

Most gearboxes are "speed reducing" or reduction in nature, (see Section B5.5.5). Situations where step up gears are needed will occur but the choice of "off the shelf" units is limited. Some vari-drive units allow a speed increase but the ratio is usually low.

Transformers are genuinely two-way devices being able to step up or step down. Recognition of this fact is highlighted in BS 171 where windings are not designated primary and secondary but high voltage and low voltage. The direction of 'step' is chosen by the way the windings are connected into the system. This observation is very useful and has been used to help define the gearbox ratio - see Chapter 6, Section 6.6.1.

B5.5.22 **Type of Ratio**

Most gearboxes have a fixed ratio. Gearboxes with variable ratios are available and therefore must be catered for. The various types of ratio are listed in Chapter 6, Section 4.8.1. The way these ratio types are handled is also discussed in that section. As described in Section B5.5.13, tappings on the transformer winding will give a series of ratios.

B5.5.23 **Direction of Rotation of Shafts Relative to Each Other**

Most engineering systems have a "direction" in which they operate. Rotary mechanical systems are no exception, conveyors, rolling mills, indexing tables etc. must rotate in a given direction in order to function correctly. Thus, the direction of rotation of the output shaft relative to the input shaft is important in case the driver has to be reversed in order to keep the gearbox output shaft driving the system in the correct direction.

The supply voltage and the induced output voltage in transformers are up to 180^{D} out of phase. Because the input and output windings are not connected, these opposing voltages do not counteract each other. The phase change is not important because the load regards the output as a sinusoidal voltage supply, the relationship with the input is not normally important.

B5.5.24 **Maximum Speed of Case**

This is a characteristic of epicyclic gearboxes and the analogies are discussed in Chapter 6, Section 6.5.5.

B5.5.25 **Duty Cycle**

This data affects the service factor (Section B5.5.27). The duty of a component is a measure of how it is stressed, a gearbox running at rated speed with a low torque will wear slower than a gearbox experiencing very high shock loads. A similar idea applies to transformers and was detailed in Section B5.5.18 above.

B5.5.26 **Reversing Cycle and Frequency of Reversing**

This is a consideration within the duty of the gearbox. Frequent stops, starts and reverses will stress the gearteeth more than continuous operation. Manufacturer's will often increase the service factor as the number of "cycles" increases.

By their nature transformers have a sinusoidal supply and output of normally constant frequency. Hysteresis and eddy current losses are a function of frequency, see Chapter 6, Section 6.5.2. However, the sinusoidal power flow through a transformer cannot be considered as having an affect on the life of the transformer.

B5.5.27 **Service Factor**

This is the value by which the system power is multiplied to create the design power for the component. The service factor may be built up by combining several constituent factors such as duty and reversing cycle. These factors will all be assigned using data provided by manufacturers. Typical service factors are given in Appendix F, Table 4.3. Although specified for motors, the factors are applicable to gearboxes.

Transformer manufacturers apply similar factors to cope with the duty of the transformer i.e. periods of heavy current supply will cause the transformer to heat up. If no additional cooling is applied, a larger transformer must be selected.

B5.5.28 **Friction Torque**

The rotation of the gearbox parts must be unhindered for power to be transmitted efficiently. Any undue mechanical interference will cause a force and hence a torque opposing the rotation of the gearbox. The design of modern bearings, gearteeth profiles and lubricants has helped to minimise these internal losses and makes them insignificant to the torque being transmitted. The analogies of torque losses in gearboxes with losses in transformers are discussed in Chapter 6, Section 6.5.2.

B5.5.29 **Gearbox Stiffness**

Gearbox stiffness is a parameter that is usually not considered for most above fractional horse power (F.H.P.) applications. This is because the gearboxes are heavily over designed and therefore elastic deformation is minimal. In control applications (usually below F.H.P.) gearbox stiffness is often quoted by the manufacturer.

In larger applications, the stiffness of the gearbox and its mounts should also be considered when large torques are being transmitted. Twisting of the gearbox shafts or excessive force (causing bending) on the gearteeth will cause fatigue and hence affect the service life. There will be a reactive torque on the gearbox casing which must be born by the surrounding structure.

If the stiffness of the surrounding structure is adequate, it may be appropriate to check the tensile strength of the mounting bolts, the reactive torque will tend to strain certain bolts.

In applications where positional feedback is important, twisting of the gearbox shafts can cause an error. The gearbox stiffness parameter was stimulated by the analogy with capacitance in transformer windings. This is discussed in Chapter 6, Section 6.5.7. The dielectric loss in transformers is related to the capacitive loss and is dealt with as part of the capacitance analogy.

B5.5.30 **Total Number of Shafts**

The number of shafts emanating from the gearbox. This information is virtually useless, unless accompanied by a drawing showing the relative positions of the shafts. In certain applications, the gearbox housing may be an output, thereby adding another "shaft". The treatment of this data is discussed in Appendix G.

The relative orientations of transformer terminals are fairly unimportant as they are wires which can be lead out in any direction. The position of specific terminal boxes may, be limited, but again, the wires from it can be in any direction.

The relationships between input and output shafts in belt and chain drive systems are left largely up to the designer. However, considerations such as centre distances must be made. Also, by the nature of the belt / chain drive, the input and output shafts occupy the same plane. If a right angle drive is required, the belt must be twisted, this requires longer centre distances and will fatigue the blet. The use of bevel gears etc. in gearboxes allows the gearbox shafts to be offered in a variety of orientations.

B5.5.31 **Mounting Type and Options**

Gearboxes can be integrated into systems in many ways. It is important that the gearbox can be fitted easily and firmly supported as power is transmitted. The reactive torque to the supporting structure must not be overlooked (Section B5.5.29). The various mounting types are discussed in Chapetr 6, Section 6.6.11.

The mounting considerations of transformers are less significant than gearboxes. Mounting bolt holes are provided in the transformer "body". If cooling fluids etc. are used, care must be taken to ensure they can be supplied, replaced etc. The size and weight of larger transformers should be noted in case of problems with poor supporting structures etc.

B5.5.32 **Frame Size**

This is simply the manufacturer's code to designate the gearbox. Similar codes appear for transformers. Although standard power values are specified in BS 171 pt. 1, dimensional standards do not exist.

B5.5.33 **Weight**

The mass in kg of the device.

B5.5.34 **Number of Gear Clusters / Number of Planetary Trains**

For gearboxes to have a very high reduction ratio or output shafts with differing speeds, a more complex internal construction must be used. The more complex the construction the more costly the gearbox. This idea is also true for belt and chain drives. The more complex the system, the more parts and hence greater costs.

The quality of the construction of transformers will clearly affect the final cost. The type of core material, the method of winding, quality of insulation, the number of tappings etc will all contribute to the final price.

B5.5.35 **Axial Thrust Capacity**

This is an external load acting along the gearbox shaft. It affects the bearing life because excessive loadings will cause wear. This parameter is irrelevant for transformers because it is "physical" rather than electrical.

B5.5.36 **Permissible Overhung Loads**

This could also be called the maximum radial thrust capacity. If the gearbox is to drive a belt and pulley system say, the belt tension will place a radial load on the gearbox bearings. This again may cause increased wear.

When designing belt and chain systems, the designer must select the appropriate bearings. Part of that selection process will be to consider the above two parameters.

B5.5.37 **Environment**

This is a general heading under which many of the following parameters can be grouped. It is a useful consideration because it allows the designer to contemplate all of the characteristics of the surroundings of the device. This term applies to gearboxes, transformers and belt / chains.

B5.5.38 **Cooling**

The meshing of gearteeth and smooth running of bearings and the associated potential of metal to metal contact causes lubricants to be used in gearboxes. The effects of friction and churning will cause the lubricant to heat up. If the lubricant gets too hot, it will cease to provide protection to the metal surfaces and the gearbox will wear. Hence the gearbox may have to be cooled, or the lubricant cooled by circulation. An alternative is to select a much larger gearbox which can dissipate the heat energy sufficiently fast enough to avoid overheating. This of course, is more expensive and inefficient.

Induction heating effects in the core and windings will cause transformers to have a heat dissipation problem. Manufacturers usually provide a figure for the expected temperature rise for a fully loaded transformer (full load current). Various methods can be used to calculate the temperature rise, they all rely on a knowledge of the heat capacity of the device. The result of the temperature rise calculation will be the addition of cooling or the selection of a bigger transformer. Many transformers use an insulating oil which can be recirculated to help cooling.

B5.5.39 **Thermal Rating**

This is a value, provided by manufacturers that attempts to encapsulate the relationship of power through the gearbox to the temperature rise that it produces. Heat capacity may be used as the basis or part of the calculation. A similar calculation can be performed with transformers. The thermal rating will affect the insulation class (or insulation type) which is specified by BS 171 pt. 3 (REF. 23).

B5.5.40 **Maximum Thermal Dissipation**

This is a very similar concept to the other thermal properties. It usually expresses the heat being radiated from the gearbox in Joules or the rate of dissipation in Watts. Clearly, the constraint for selection is that the device must be able to dissipate the heat energy being produced. As with the other thermal properties, additional cooling or a larger unit are the solutions.

Parameters B5.5.37-B5.5.40 are all caused by energy losses generated by inefficiencies within the devices. They are therefore a direct measurement of the total losses. It is vital that this figure is as low as possible.

B5.5.41 **Ball Bearing Type**

Gearboxes are designed by manufacturers and made available as "off the shelf" items. The result of this is that the component parts are not fully described because they are only part of the system. The bearing details will be limited to the axial and radial loads and the shaft speeds. From a "system" viewpoint this is sufficient information. A greater knowledge of the bearing type would help with misalignments, noise and limits of the shaft loadings.

In the case of belt and chain drives, bearing selection is an integral part of the design procedure. Often, extensive bearing information is provided by the belt / chain manufacturers.

There is no direct analogy with transformers for this parameter.

B5.5.42 **Gearbox Life**

This is determined by the fatigue life of the gearteeth, bearings etc. Manufacturers may quote the life in terms of continuous running at rated values or the total number of overloads to failure. In most cases, the gearbox life can be taken to be the life of the system. It is likely that other components such as motors, clutches, couplings, belts etc. will fail before the gearbox.

The transformer is a magnetic machine and has no moving parts. Life is determined by the rate insulating materials degrade. Thus, as with gearbox life under average operating conditions, this characteristic is less significant than other system constraints.

B5.5.43 **Dimensional Information**

Dimensional information is usually presented as fully annotated drawings. Not only do these drawings provide dimensional data but important spatial characteristics are conveyed. Information such as positions of shafts, positions and types of mounting and the relationships between these features can all be gleaned from a clear drawing. The problems with representing this kind of information for the user within a computer aided retrieval system are discussed in Chapter 6, Section 6.6.9 and 6.6.10.

Transformers must also be mounted and the positions of terminal boxes etc must be considered for accessibility. The problems are less significant because of the flexibility of cables.

The arrangement of shafts, mountings etc in belt / chain systems are dictated by what is being driven. Design of casings and mounting points are in the control of the designer.

B5.5.44 **Gear Tolerances**

This is a contributory factor to backlash. If the gears do not constantly mesh, there will be play between the gearteeth and greater wear will occur. Transformers have no moving parts and this kind of error does not occur. BS 171 pt. 1 does outline the tolerances on all the quoted performance characteristics and losses.

In belt / chain systems, slackness or play in the belts / links will allow the sprockets to move relative to each other.

B5.5.45 **Shaft Radial Play**

This is a function of the bearings and method by which the shafts are mounted within the bearings. The radial play is the maximum amount of movement of the gearbox shafts in a direction perpendicular to the axis of the shaft. This of course, should only be a very small distance. If the gearbox is to be connected to other system elements, these tolerances should be noted so that the coupling selected can absorb any potential misalignments.

In the case of belt / chain drive systems, the position of the bearings can be designed so as to best suit the application, this will help reduce play and wear.

B5.5.46 **Shaft End Play**

This may also be called axial play. It is the amount of movement along the line of the shaft. As with radial play, this movement should be minimal. Also, in belt and chain systems, careful bearing selection will reduce play.

Both of these physical characteristics of gearboxes cannot be applied to transformers but can be lumped in with the general ideas for tolerances touched on in Section B5.5.44.

B5.5.47 **Shaft Sizes**

This is the diameter, tolerance and keyway details of the shafts emanating from the gearbox. The shaft diameter is important so as to satisfy connection details for pulleys, bushes, couplings etc.

B5.5.48 **Diameter of Unit**

Many epicyclic gearboxes have a cylindrical casing that can be used for input / output. Thus, the diameter and length of the unit are often quoted rather than orthogonal dimensions.

B5.5.49 **Noise**

Certain gear types produce less noise because of the way the teeth mesh. Transformers hum because of the a.c. frequency. Belts and chains also create noise when running. In situations where noise may be a problem or simply an irritation, steps can be taken to select quiet gears, belts or chains or to sound proof the device casing.

B5.5.50 **Lubrication**

All rotary mechanical systems must be lubricated adequately. This reduces wear and its associated problems. Lubrication is clearly not applicable in transformers.

APPENDIX C

DISCUSSION OF THE EXTENDED ANALOGOUS PARAMETER LISTS FOR COUPLINGS

The following section defines each characteristic in the extended parameter list and attempts to assess its importance for selection. Also, any possible analogies are outlined and are detailed in Chapter 7, Section 7.5.

C7.4.1 Maximum Torque

The maximum torque the coupling can transmit. The maximum torque is an intermittent value and hence may exceed the starting (or static) torque (Section 7.4.3 below) and will be greater than the rated (or dynamic) torque (Section 7.4.2 below). It may also be called the ultimate torque of the coupling. As an intermittent value if the device is operated continuously at this value or above it, damage will be done to the component.

C7.4.2 Dynamic Torque

The torque the coupling can transmit continuously whilst running. This is the equivalent of rated torque.

C7.4.3 Static Torque

The torque the coupling can transmit continuously with the output locked. This is a continuous value and hence may be less than the starting torque which is an intermittent rating.

C7.4.4 Rated Torque

The continuously rated dynamic torque.

C7.4.5 Starting Torque

The maximum torque the coupling can transmit on starting. This is an intermittent value and hence will be greater than the static torque in most cases.

C7.4.6 **Torque Range**

The range of torques the coupling can transmit. The lower value of this range is somewhat confusing as it should be zero. A coupling driving a load that for some reason may not be dissipating energy will have no torque difference across it. The case where the lowest torque will have a value is when couplings are absorbing misalignments and forces within the coupling materials prevent the coupling from rotating freely. This is discussed in Chapter 7, Section 7.5.6.

C7.4.7 **Torque at 100 RPM**

This is a misleading characteristic. Notice, it is not torque per 100 RPM but torque at 100 RPM. It does not therefore imply that the torque at any speed can be calculated by extrapolation. It does, however, yield a value for power at 100 RPM which is in keeping with the power against speed tables (Section C7.4.27) often provided by manufacturers. Since the torque is quoted at constant speed, it is the rated torque at 100 RPM.

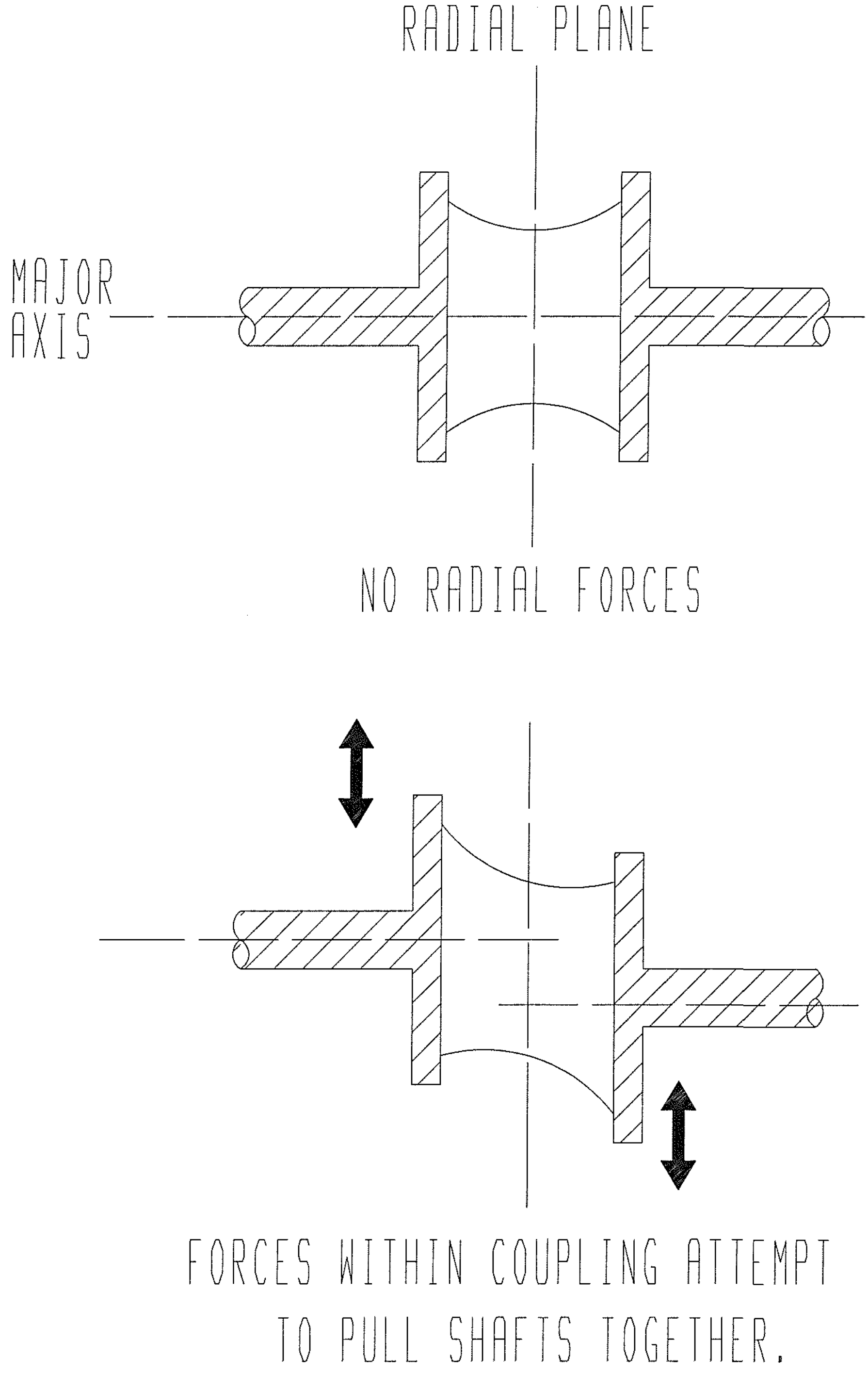
C7.4.8 **Torsional Stiffness**

Couplings must transmit torque whilst absorbing misalignments. To do this, there is usually an intermediate mechanism between the driving and driven parts of the coupling. Often, there is some compliancy associated with this mechanism e.g. rubber elements, springs etc. Thus, there will be a twist stiffness or torsional stiffness associated with the coupling. What this means is that for a given torque difference across the coupling, the coupling will have deflected by an angle, μ. The angle, μ and the torque applied do not necessarily follow a linear relationship, Ref. 25, particularly in the case of rubber element couplings. However, an indication of the twist at rated values would be sufficient (see Sections C7.4.12 and C7.4.13).

C7.4.9 **Radial Torsional Stiffness**

The torsional stiffness of the coupling in a plane perpendicular to the coupling's major axis. The stiffness is not really "torsional" and would best be described as "radial stiffness". Figure 7.1 clarifies this deflection. It is an important value to know because when deflected, forces will be working on the coupling attempting to move the shafts radially.

FIGURE 7.1 : RADIAL STIFFNESS WITHIN A RUBBER COUPLING.



If the coupling radial stiffness is low, care must be taken to ensure that it is adequately restrained otherwise, excessive radial force will damage it i.e. a rubber coupling will tear. If the radial stiffness is high e.g. a universal joint, the radial forces will be transferred directly to the adjacent components with the possibility of bearing damage.

C7.4.10 **Axial Torsional Stiffness**

The stiffness of the coupling along its major axis, see Figure 7.2. Many rubber couplings are weak in this direction and may be damaged if the forces in this direction are excessive.

The radial and axial stiffness, should not be confused with deflections caused by "Poisson's ratio" considerations. Although these effects will be present in rubber couplings, they are not accounted for in the general case.

C7.4.11 **Static Torsional Stiffness**

The continuous stiffness the coupling exhibits when at rest. This may be larger than the dynamic stiffness because there will be no cyclic fatigue and hence larger forces can be tolerated.

C7.4.12 **Dynamic Torsional Stiffness**

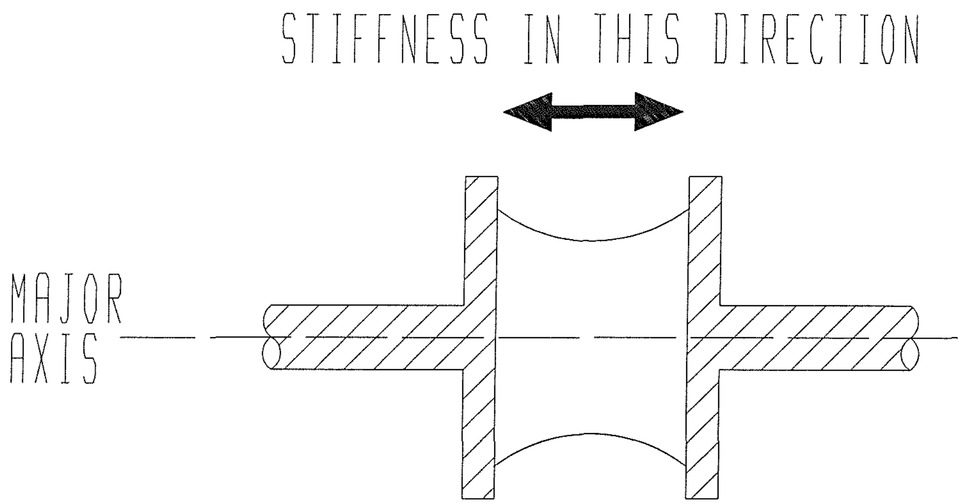
This is identical to the torsional stiffness defined in Section C7.4.8. It is worth noting that the word dynamic does however imply the stiffness of the coupling when rotating and therefore the stiffness around its rated speed and torque.

The characteristic of torsional stiffness is discussed in more detail in Section 6.5.7.

C7.4.13 **Torsional Angle at Rated Torque**

When transmitting rated torque, there will be a certain amount of twist in the coupling. The angle of this deflection is the torsion angle. Ideally, it will be small in most cases so that no motion is "lost" in winding up the coupling.

FIGURE 7.2 : AXIAL STIFFNESS WITHIN A RUBBER COUPLING.



C7.4.14 **Torsional Deflection**

The degree of twist in a coupling for a given torque. The stiffness of rubber couplings is usually non-linear, Ref 25 (Section C7.4.8 above) and hence the twist cannot be easily predicted. A graph of deflection against torque applied may be provided by the manufacturer, but in most cases, a single value of torsional deflection is quoted and will be that for the rated torque.

C7.4.15 **Static Wind-Up**

This is the twist of the coupling with no angular velocity being transmitted. It is the amount of deflection the coupling experiences under the static / starting torque described in Section C7.4.3 and C7.4.5 above. Detail as to whether the value is static or starting must be gleaned from the manufacturer's specific data.

C7.4.16 **Torsional Vibration**

This parameter covers two parameters:

C7.4.15a **Maximum Attenuating Torque;**

The allowable amplitude of an alternating torque superimposed on the rated torque of the coupling. Thus, the sum of the rated torque and the maximum attenuating torque will not exceed the maximum torque of the coupling.

C7.4.15b **Frequency of Vibration;**

The maximum allowable frequency of the alternating torque. If the frequency becomes too high the coupling may be damaged. This could be caused by excessive heating (see damping factor below) or simply because the life of the coupling is shortened by excessive cyclic stressing.

C7.4.17 **Resonance Factor**

The ratio by which the torsional vibration is magnified across the element at resonance. The resonant frequency of the system must be calculated by the designer and there is often a guide to help with this provided by manufacturers. The natural frequency of the system and the frequency of the driver can then be compared and attempts made to operate the coupling at speeds outside these coincidences.

C7.4.18 **Damping Factor**

The ratio of damping energy to elastic deformation energy for one period of oscillation. This is a measure of how much energy is lost as heat when a coupling is deflected and then relaxed. The higher the damping factor the greater the amount of damping. Note, as the damping factor approaches unity and over, the coupling will respond more slowly to torque changes, hence helping to "smooth out" fluctuations.

C7.4.19 **Backlash**

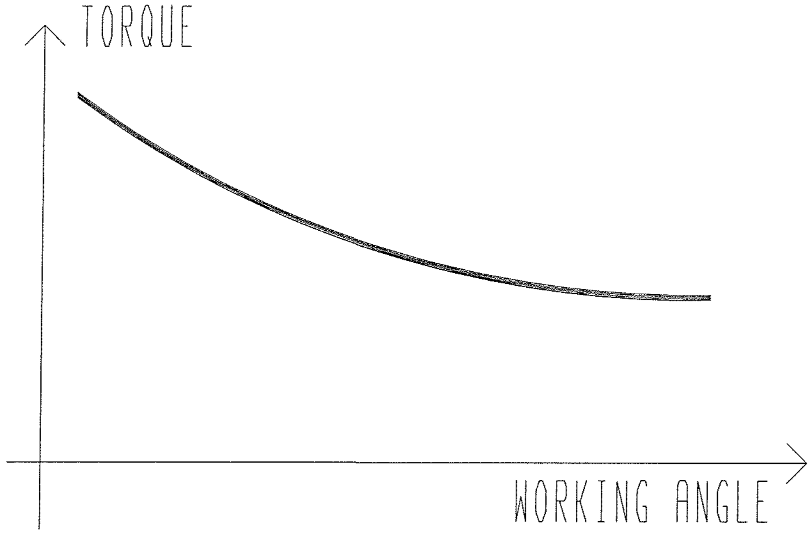
Many couplings rely on the interaction of mechanical elements on the input and output halves to transmit torque. If the input and output halves are not rigidly joined, forces are transmitted by surface to surface contact, often having an element of sliding. Thus, as with all mechanical components, manufacturing tolerances will be present. Thus, play or backlash will exist within the coupling. Backlash differs from torsional stiffness because a phase change between input and output is caused without torque being transmitted. Backlash is not recovered when the torque stops. Cycling and high torque starts could damage the coupling if the backlash is not considered.

C7.4.20 **Coupling Misalignments**

The misalignments of two shafts can be broken down into three basic types which rarely occur singularly. It is convenient for manufacturers to quote the maximum individual values of these misalignments and to allow the designer to combine them to absorb the errors in a particular application.

The maximum values for misalignments are continuously rated maxima and not intermittent values. Hence, for design purposes, the service factor will not be affected by the coupling operating at maximum misalignment. In certain cases, universal joints being the most common, the amount of torque capable of being transmitted does decrease with the degree of misalignment (Figure 7.3). How this kind of anomaly is handled is discussed in Chapter 7, Section 7.8.4.

FIGURE 7.3 : THE GENERAL RELATIONSHIP BETWEEN TORQUE AND WORKING ANGLE FOR COUPLINGS.



C7.4.21 **Axial Misalignment**

This is also known as end float, see Figure 7.4. Any movement of the shafts relative to each other along their axes will put a thrust pressure on the bearings in the coupled devices. Thermal expansion of the shafts, motion of the devices caused by vibration or movement whose centre of rotation is not about the coupling are common causes. Excessive thrust pressures on bearings not designed to accept them will cause damage and / or seizure.

C7.4.22 **Radial Misalignment**

This occurs in a direction perpendicular to the axis of the shafts (see Figure 7.5). Radial misalignment is usually caused by vibration and play in the shaft bearings. To avoid vibration being passed from one device to another, a coupling can be introduced to absorb the vibration energy.

C7.4.23 **Angular Misalignment**

The axes of the shafts are not in line but are in the same plane and intersect. The coupling must absorb this misalignment and avoid transmitting large radial forces when rotation occurs (see Figure 7.6).

C7.4.24 **Nominal Power**

This is also the rated power of the coupling. It is the product of rated torque and rated speed, it is therefore a continuously rated value.

C7.4.25 **Design Power**

The data quoted by manufacturers for power will usually be the rated or nominal value. The designer is encouraged to select on this value because it will yield an economical solution. Often because of duty, environment and influences from other system elements, a service factor must be applied to the device. The service factor is used to scale the rated power and creates the design power. This value is the rated power and creates the design power. It is then used to select the device. Hence, the designer identifies a more robust (usually physically larger) component to cope with the "stresses" put on it by the system and environment.

FIGURES 7.4 - 7.6 :

THE COMPONENTS OF MISALIGNMENTS.

FIGURE 7.4 : AXIAL MISALIGNMENT.

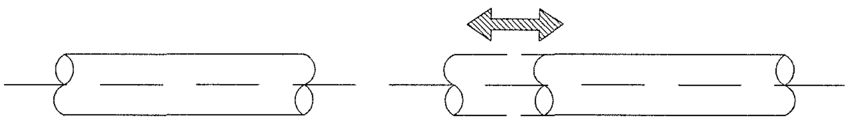


FIGURE 7.5 : RADIAL MISALGNMENT.

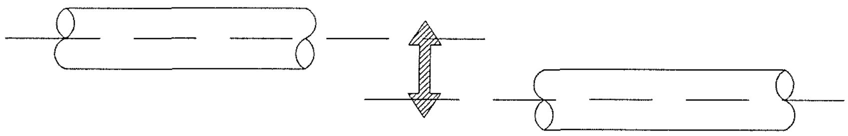
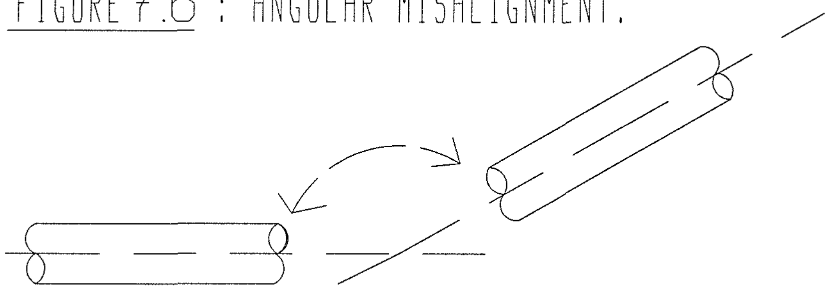


FIGURE 7.6 : ANGULAR MISALIGNMENT.



C7.4.26 **Power Per 100 RPM**

This informs the designer that power increases proportionally with speed. It must be assumed that this holds for power up to the maximum speed of the coupling. This type of parameter is dealt with in more detail below.

C7.4.27 **Power and Speed Tables**

This format is used by many manufacturers. The table is made up of rows and columns containing progressive values of speed and power. At the intersection of the rows and columns are coupling identification codes which essentially identify the "size" of the coupling.

Analysis of the values of speed and power yields a roughly constant torque for intersections where a code is repeated. Thus, rather than setting a torque limit for each coupling size, and expecting the designer to always pick the appropriate device. The manufacturer has allowed the designer to quickly identify the best coupling for the specification without having to cross check power and speed required with those available.

With reference to Section C7.4.26, it is clear that the manufacturer is suggesting that up to the maximum speed, the torque limit of the coupling is constant.

The term torque limit is assumed to be the continuously rated torque.

C7.4.28 **Maximum Power at 100 RPM**

Given the maximum power at 100 RPM, the maximum torque value can be calculated. It is assumed that these are continuously rated maxima. Much like the torque at 100 RPM (Section C7.4.7), it is doubtful whether the power can be extrapolated as power per 100 RPM. Therefore, this parameter is also fairly unhelpful to the designer.

C7.5.29 **Shaft Speed**

The rated speed that the shafts the coupling is to join run at. This will be a continuous value.

C7.4.30 **Minimum Shaft Speed**

The minimum speed the coupling is expected to transmit continuously. As a minimum speed it is fairly insignificant unless the torque associated with this speed is larger than the rated value, in which case, the coupling selected for rated values will be inappropriate.

C7.4.31 **Maximum Shaft Speed**

The maximum speed the coupling can transmit safely. This will be an intermittent value. Centrifugal forces may damage the transmission mechanisms at high speed. Also, if the coupling is "stressed", running at this speed, above the rated value, the service life will be reduced.

C7.4.32 **Velocity Ratio**

In the ideal case, the input and output speeds will be identical for any coupling absorbing misalignments. Unfortunately, in many cases, the mechanisms involved in transmitting the torque do not transmit velocity uniformly but produce a cyclic variation in velocity which frequently increases with the degree of misalignment (Ref. 21). The velocity ratio is a numerical expression of the difference in input and output speeds for any position of the input shaft. It usually contains a factor accounting for the degree of misalignment. It becomes important because cyclic variations in transmitted speed may be undesirable (resonance) particularly if the driver already has a cyclic component.

C7.4.33 **Moment of Inertia**

As a rotating mass on a shaft, a coupling will have a moment of inertia. The coupling inertia will contribute to the system inertia and hence to the starting torque required of the driver. Due to their generally cylindrical shape and relatively low mass, the inertia of most couplings will be small in comparison to the system inertia and can be ignored.

It is at this point two observations must be noted: inertia is quoted by most of the coupling manufacturers because the inertia is large compared to motor rotors and gear trains, typically, at least, two to three times larger.

Secondly, the coupling is the third element to be studied and although its inertia is large compared to its potential adjacent elements, the value is still considered small compared to the load or system inertia. However, it must be stated that the sum of the component inertias will become significant in many cases and hence they should be available because the starting torque provided by the driver could be too small if it is selected to start the load inertia alone.

C7.4.34 **Coupling Type**

This is the name given to the general design of the coupling and may also describe salient functional or performance details. The coupling types are best summed up by the T.I. listing given in Appendix K, Table 7.1. The functional/performance details can be broadly defined by the terms; rigid, torsionally rigid and torsionally flexible. A rigid coupling accepts almost no misalignments and transmits torque without flexing e.g. two flanges bolted together. A torsionally rigid coupling will accept misalignments and will not flex when loaded e.g. a universal joint. A torsionally flexible coupling will accept misalignments and will deform under load e.g. most "rubber" couplings.

C7.4.35 **Service Factor**

This is a factor used to scale the system power (the rated power the coupling is expected to transmit) to create the design power (the power the coupling is given for selection). The service factor will over-size the coupling in order to make it less stressed by overloads, extreme conditions, etc. The parameters defined below in Sections C7.4.39-42 will contribute to the service factor.

C7.4.36 **Bore Sizes / Shaft Sizes**

Couplings are usually intended to join two shafts. Thus, built into the coupling will be bushes that will accept shafts so that torque can be transmitted. There will be limits to the size of shafts the bushes can accommodate so it is important to select a coupling which will accept the shafts of the devices to be coupled.

C7.4.37 **Coupling Dimensions**

Most manufacturers will provide a dimensional drawing of the coupling. Since it is a rotating device on a shaft it is important to ensure adequate clearances with the surroundings. The rotational envelope of the coupling should be noted by the designer.

C7.4.38 **Method of Integration**

Couplings commonly use bushes to join shafts, however, other possible mounting arrangements include flanges and keyed shafts. Thus, a combination of these is possible, depending on the application.

C7.4.39 **Duty**

This is a measure of the "stress" the coupling must endure when operating. Frequent stops and starts under heavy load, shock loads, cycling and the environment will all cause the coupling to wear faster. Considering the duty causes the designer to quantify all of these parameters so that a service factor can be obtained.

C7.4.40 **Shock Loading**

This is a specific quantity within the duty of the device. Direct-on-line starting, high inertia loads, rapid braking, etc. will all cause a shock torque to be transmitted through the system and therefore across the coupling. This must not exceed the maximum torque of the coupling, otherwise it will be damaged.

C7.4.41 **Shock Factor**

An empirical value provided by manufacturers which is used to build the service factor. It causes a larger coupling to be selected to cope with the expected shock loads.

C7.4.42 **Temperature**

The ambient temperature of the environment the coupling is expected to operate in. Extremes of temperature may effect materials used in the coupling and lubricants. Rubber element couplings dissipate heat when cycled and hence adequate cooling must be available.

C7.4.43 **Pollution**

The coupling may have to withstand hazardous environments such as dust, corrosive fluids etc. Damage may be caused by greater wear or degradation of coupling materials. The ability of couplings to withstand such conditions has not been formalised. The degree of motor protection has been codified by the I.P. classifications (Ref. 10) and perhaps this could be used for couplings.

C7.4.44 **Electrically Insulating**

In certain cases, it may be important to prevent electrical or magnetic activity to pass from one device to another. If bearings are magnetized, they tend to wear faster as debris is not removed.

C7.4.45 **Service Life**

A coupling absorbing misalignments will be stressed more than an aligned coupling. Cyclic stressing and fatigue will contribute to a reduced life. The decrease in service life due to degree of misalignment will be dealt with where appropriate by the manufacturer.

C7.4.47 **Weight**

Compared to the moment of inertia of the coupling, this parameter is trivial. The weight of coupling does have to be supported statically and dynamically by the bearings in the coupled components so the acceptable radial loads in these bearings should be compared to the coupling weight.

APPENDIX D

DISCUSSION OF THE EXTENDED ANALOGOUS PARAMETER LISTS FOR CLUTCHES ET AL

As with Chapters 4,5 and 7, this section will move through the ordered, extended parameter lists, commenting on the parameters and identifying analogies. It would constitute Section 8.5 of Chapter 8.

D8.5.1 **Maximum Dynamic Torque**

This may also be called engaging torque or switchable torque. The word "dynamic" is used to indicate relative movement, or slip between the input and output sides of the component. It is a continuously rated value and thus, is not necessarily the starting torque available at the output. In the case of a slipping friction clutch, it will be the torque available at the output whilst rated power is supplied to the input. In a fluid coupling the maximum dynamic torque is its rated torque. For dampers, it is the torque at which the product of torque and speed reach the maximum continuous power rating. In torque limiters, slipping power may be zero or some value between zero and the non slip power value. If zero, the clutch has completely disengaged at the torque limit. Overrunning clutches would have this parameter set at zero or the drag torque (see Section 8.5.14 below).

D8.5.2 **Brake Torque**

This is identified to dynamic torque and highlights the problem of semantics. When a clutch is used as a brake, energy must be removed from the driving shaft. This is usually done by causing the driven shaft to slip relative to the output shaft. Slip is achieve by using friction to transmit torque and friction generates heat energy representing energy lost from the driving shaft. Thus, the brake torque is the continuous torque available opposing the motion of the driving shaft. It could be considered as a negative dynamic torque. A rotary damper with a fixed driven shaft will also have a brake torque. Fluid couplings, by the nature of the power transfer within them will produce a brake torque if the output is fixed or the output attempts to overrun the input. Overrunning clutches will have the brake torque set at their rated locked torque i.e. when the driving and driven shafts move in the same direction with the same torque.

D8.5.3 **Torque v Speed Difference Graphs / Tables**

As mentioned above, the limiting factor for slipping friction clutches is the heat that can be dissipated. The heat generated is a function of torque transmitted and the speed difference (or slip) between input and output. Thus, tables of maximum continuous torque against slip can be prepared so that heating calculations are avoided. A graph of torque against slip will give the "safe" operating region for the device. Similar limits constrain the performance of fluid couplings and rotary dampers.

In Appendic C, Section C7.4.27, the power against speed tables used in the selection of couplings simply specified the torque limit for the coupling without the designer having to calculate it. The conclusion was that the torque limit should be stated because the designer would know the limits of the torque the coupling would need to transmit from the general system data. In the case of heat generated through slip, the designer will not know how much power is to be dissipated without making additional calculations. However, these calculations must be performed if the tables and graphs are not available. It is therefore important that the maximum continuous power dissipation is known so that the power generated by friction can be compared to ensure safety.

D8.5.4 **Static Torque**

Also called running torque or torque through the clutch. The strict definition of this parameter is the torque the component transmits with no relative movement (slip) between input and output. This would therefore be the rated system torque in most conventional clutch applications. In fluid couplings and dampers, static torque does not exist because there is slip between input and output. In torque limiting clutches, the static torque will be the maximum torque before the clutch activates - it will be the limiting torque. In overrunning clutches, the static torque will be the torque transmitted when input and output more in the same direction with no slip.

D8.5.5 **Starting Torque**

The torque available through the clutch for starting. As mentioned in previous chapters, this is usually greater than the rated torque because of the need to accelerate the system inertia. Thus, care must be taken when selecting the torque limit for torque limiting clutches - a high starting torque may activate the limiter. For centrifugal friction clutches, fluid couplings and rotary dampers, slip speed and available torque are related and for a high starting torque, a high slip must be present.

D8.5.6 **Torque Range - Maximum to Minimum Torque**

This is usually associated with the controllable settings for torque limiting clutches. Rotary dampers have a torque range proportional to slip speed ranging from a very low no slip torque to the maximum continuous torque at high slip. Fluid couplings have almost no useful output torque at low input speeds. The upper torque limit for most components will be the starting torque.

D8.5.7 **Maximum Anti-Overrun Torque**

Also called maximum anti backup torque or maximum reverse torque.

This is purely concerned with "overrunning" clutches. It is the torque transmitted from input to output with no slip. Depending on the application, it may be intermittent or continuous. It is important that it is not exceeded because the overrunning mechanism will be damaged.

D8.5.8 **Torque Variation**

This will also include the parameters; amplitude of ripple and frequency.

Systems powered by prime movers with a cyclic variation (piston motors etc.) must be carefully considered for resonance at certain speeds. All system components play a part in dealing with these problems. Fluid couplings and rotary dampers will reduce the amount of cyclic variation transmitted because of the power absorption associated with slip. Clutches with flexible couplings built-in will also absorb torsional vibration. These details are discussed in Chapter 7.

D8.5.9 Dynamic Torsional Stiffness at 10 Hz

D8.5.10 Torsional Deflection

D8.5.11 Torque Wind-up

D8.5.12 Torsional Angle at Maximum Torque

D8.5.13 Static Torsional Angle

These are all characteristics of flexible element couplings which are often built into clutches. They are all discussed in detail in Chapter 5. Also, these are not really characteristics of the clutch, they are characteristics of the coupling and should therefore be separated from clutch selection or specification.

D8.5.14 Drag Torque

Also called breakaway torque or resistance to overrunning or no load torque. This is the torque that exists between output and input because of the mechanisms within the clutch. In electromagnetically actuated clutches it may be residual magnetism; in overrunning clutches it will be the friction caused by the overrunning mechanism; in fluid couplings and magnetic powder / fluid clutches, it will be the viscosity of the working fluid causing drag on the rotor. This value will be small compared to the operating torque.

D8.5.15 Life at Rated Torque

The life of the clutch running at rated torque. Fluid couplings have a very robust design and no highly stressed moving parts and hence, have an almost unlimited life provided it is run within the heating regime. Friction clutches run within slip and heating limits will also have a reasonable service life. This parameter is a little superfluous because an adequate service life at rated values is expected.

D8.5.16 **Lining Pressure at Maximum Torque**

Certain friction clutches use a hydraulic or pneumatic system to force the clutch plates together or apart. This value simply tells the designer how much pressure is needed for maximum torque transmission. It is a supply characteristic and not a primary performance characteristic.

6.5.17 **Speed of Rotation**

For conventional clutches this could also be called static speed. This may seem like a contradiction but it implies that the input and output have no relative movement. Therefore clutches which are rated under slip conditions i.e. fluid couplings, could not be indexed under the term static speed. However, speed of rotation would allow the output speed to be used.

D8.5.18 **Speed Difference**

The difference in speed between the input shaft and output shaft can be used to calculate the slip between input and output. Slip can occur under many conditions, but the slip at rated torque is usually the most significant because the continuous heat dissipation can be calculated. This will enable the correct size of clutch to be selected.

D8.5.19 **Engagement Speed**

This is primarily a centrifugal clutch parameter, it is the speed at which the clutch begins to transmit torque. This is a similar mechanism to a fluid coupling running up to speed before full torque is available at the output. In both cases, the clutch will allow an a.c. motor to reach a speed where the pull out torque (see Chapter 3 Section) is available to accelerate the system.

D8.5.20 **Permissible Overrunning Speed**

Also called maximum continuous overrunning speed or speed at overrun. This characteristic relates specifically to overrunning clutches. When overrunning, the driver and driven shafts move relative to each other. This parameter usually limits the speed of the overrunning part of the clutch so that it is not damaged by centrifugal forces.

D8.5.21 **Reversibility**

The ability of the drive to be reversed. This is possible in most designs of friction clutches and rotary dampers. Fluid couplings and overrunning devices will not transmit torque if the direction of rotation of the input shaft is reversed.

D8.5.22 **Speed Regulation**

With fluid couplings, it is possible to adjust the amount of slip at rated input speed by controlling the amount of fluid within the device. This essentially means that the rated output speed can be controlled whilst maintaining a constant output torque. Slip in friction clutches can also be controlled by changing the plate pressure or by introducing oil between the plates. In all cases where slip is controlled, care must be taken to evaluate the heat generated and the ability of the component to dissipate it.

D8.5.23 **Maximum Input Speed**

Most rotating devices have a maximum speed limit quoted to avoid damage by centrifugal forces i.e. overrunning clutches. In components where slip occurs, the input speed must be limited at the 100% slip condition to avoid excessive heating. The maximum input speed will generally be a continuously rated maximum rather than an intermittent rating.

D8.5.24 **Maximum Speed and Minimum Speed**

The operating range of the component. The maximum speed of the clutch will be broadly the same as the maximum input speed defined above, but may be an intermittent value in some cases. Therefore, careful consideration of the values quoted by the manufacturer at these limits should be made.

D8.5.25 **Power Through Clutch**

This is the power the clutch can transmit at rated torque and speed, it is the rated power of the device and not necessarily the power the clutch can dissipate whilst slipping. Thus, in friction clutches and torque limiters, it is the static (no slip power). In fluid couplings, it will be the power available at the output shaft at rated input power.

For rotary dampers, it will be the maximum continuous power dissipation at maximum continuous input speed.

D8.5.26 **Slipping Power**

This may also be called the maximum slipping power. A related quantity is the maximum time at 100% slip before failure. Whenever a speed "loss" occurs across an in-line device, energy must be lost. In clutches, slip causes an energy loss as heat is generated by simple friction or fluid shear. The rate at which this heat is generated is the power lost by the device. The slipping power will have a particular significance, depending on the application. A fluid coupling running at rated conditions will have a slip of only a few percent. This slight slip gives rise to a slight heating of the working fluid. However, if the working fluid is not adequately cooled by recirculation or by the casing being able to dissipate the energy, the fluid temperature will rise and the associated problem will ensue, i.e. change in viscosity, boiling etc. Under full slip conditions, i.e. the output is locked, all the input power will be "dissipated" into the working fluid. Thus, the slipping power will be very much greater and if not removed quickly, the clutch will rapidly overheat and fail. If the cooling arrangements are set up for rated conditions rather than full slip, they will clearly be inadequate for a prolonged period of full slip. Hence, the use of the safe time at 100% slip parameter. This enables periods of full slip to be endured, intermittently. Friction and torque limiting clutches also transmit power under slipping conditions. These are usually intermittent ratings dealing with start up, or overload conditions. Rotary dampers, are by their nature, continuously rated for 100% slip.

Thus, it would be best to treat slipping power as 100% slipping power which will be an intermittent maximum value for all devices except rotary dampers.

D8.5.27 **Heat / Power - Speed Difference Tables / Graphs**

Many manufacturers provide graphs or tables of heat/power generated against the difference between input and output speeds. This enables the user to locate regions of safe operation for the chosen device for the particular application. However, the general observation is that the amount of heat generated increases with slip and although it is not a perfectly linear relationship, it may be easier to quote the slipping power at 100% slip and allow the user to interpolate the intermediate values. Thus, the tables / graphs can be replaced by a constant, the definition of which is given in Section 8.5.26.

D8.5.28 **Engagement Work**

This is the energy in Joules created by the slipping process as a centrifugal clutch accelerates its load. This energy and the run up time (the time taken to fully accelerate the load) would give the average power generated while engaging. This is essentially the 100% slip power and hence similar calculations can be made for all other times clutches during the run up period.

D8.5.29 **Friction Work per Engagement**

Other related parameters are; friction per cycle, friction work per second, cycles per hour, slips per hour, engagements per hour. All of these characteristics help to calculate the amount of heat the component must dissipate whilst slipping. Again, the maximum power at 100% slip and the period for which this can be endured will summarise all of the aforementioned characteristics.

D8.5.30 **Heat Capacity**

Also called thermal capacity and thermal rating. Given the devices heat capacity, the ambient temperature, the period of slip, the input power and the input/output speed difference; the temperature rise within the clutch can be calculated. If this rise in temperature is excessive, modifications, such as cooling or selecting a larger clutch can be considered. The heat rise calculations allow the designer to check the performance of the clutch at conditions other than those anticipated by the manufacturer. If a heat against speed difference graph or table is provided, the calculations are unnecessary.

D8.5.31 **Input Power**

This is the power available on the input side of the clutch. It will be the system power and in many cases the output of the prime mover. Thus, in general terms, the clutch will be pre-selected using this parameter as a primary characteristic.

D8.5.32 **Actuation Power**

The related quantities are electrical power, maximum hydraulic pressure, actuation voltage, actuation current, coil inductance and coil resistance. The characteristic; actuation power does not apply to permanently on-line components such as fluid couplings and rotary dampers.

The above parameters refer to additional components such as solenoids and hydraulic actuators used to move the clutch mechanisms. The ratings of these actuators will be very small in comparison to the clutch power and they will be achievable in most circumstances.

D8.5.33 **Type of Actuation**

Most clutches are used to engage and disengage a power supply. Thus, some form of control is needed to activate the clutch mechanism. Friction clutches usually engage and disengage by sliding one plate up to the other. This can be done manually (using a fork), electromagnetically, pneumatically etc. (see Appendix L Figure 8.4). Continuously engaged devices such as fluid couplings, rotary dampers and some torque limiters will have "type of action" deemed as continuous.

D8.5.34 **Type of Reset**

This pertains mainly to torque limiting clutches which have been activated i.e. the torque limit has been reached. Depending on the application, the reset method may be manual, automatic, semi-automatic, timed, remote etc. (see Appendix L Figure 8.5) shows the many reset alternatives. Most conventional clutches are designed so that the actuation mechanism is spring loaded requiring an external actuation force in only one direction. In these cases, the type of reset will be "automatic".

D8.5.35 **Moment of Inertia of Clutch Parts**

At its simplest level, the clutch parts consist of two rotating elements attached to the input and output shafts. These elements are fairly robust; fluid coupling rotors, multi-plate friction clutches etc. Thus these rotating masses should be added to the appropriate part of the system inertia and will therefore affect the starting characteristics and response of the system. Generally, the output side inertia should be available as it will contribute to the system inertia. The input inertia is less significant because, on starting, with the output disengaged, the input inertia will be small compared with the system inertia which has to be accelerated.

D8.5.36 **Response Time**

The time taken for a clutch to move from the fully disengaged to the fully engaged condition. Response time will be defined as the time taken for the device to actuate rather than reset because the reset position is usually the default position. Thus, in a continuously engaged clutch, response time will be that for the plates to move apart. In a continuously disengaged clutch, it will be the time for the plates to move together. In torque limiting clutches, it will be the time for the clutch to respond at the torque limit and not the time taken for it to reset. Fluid couplings have a run up time dependent on the driver speed and the amount of oil filling. A mean response can be calculated for these devices using rated values. Rotary dampers are continuously engaged and thus, their response time can be considered instantaneous.

Clutches activated by an external source can have their response time shortened by over-excitation. This essentially is the use of maximum intermittent values to start the actuation and then reducing these parameters to their rated values once actuation is complete. This would require the use of an additional controller etc. for the actuation mechanisms and cannot be dealt with here. The over-excited response time will therefore not be considered as a primary retrieval characteristic. Also, as it is only a slight improvement over the "rated" response, the additional costs would make only very special applications worthwhile.

D8.5.37 **Mechanical Time Constant**

This is a measure of the speed of response of the mechanical parts of the clutch. Fiction and momentum will affect the sliding of a clutch plate along a shaft. Fluid couplings, by their nature, have a high inherent damping, thus, their rate of response is very slow. Hence their use in systems where vibration is present because the vibration is attenuated.

D8.5.38 **Switching Time**

This is related to the electrical time constant and therefore the time for the clutch to respond to electrical control signals. A magnetic powder fluid clutch will depend solely on the magnetic field used to magnetize the working particles. Hysteresis, inductance etc. within the fluid will cause the system response too slow.

D8.5.39 **Coil Build up Time / Decay Time**

If a solenoid is used to activate a clutch, induction and hysteresis will oppose the build up/decay of the solenoid magnetic field. This time, added to the mechanical response time as well as the switching time, will give the overall response time of the clutch.

D8.5.40 **Current Time Constant**

Inductance, capacitance, hysteresis etc. within the electrical circuit will resist the current build up thereby causing the magnetic flux not to reach full power instantaneously.

It should be noted that the response of most of the electrically activated clutches is in the milli-second region and hence need not be considered for most conventional applications.

D8.5.41 **Dimensions**

Manufacturers usually provide a dimensional drawing. Most of these devices tend to be cylindrical in nature as they rotate on shafts. The drawings will provide details such as keyways and mounting arrangements. A full discussion of which dimensions would be most useful for selection is given below in Chapter 8, Section 8.7.14.

D8.5.42 **Mounting Type**

As rotary mechanical devices, these components will be expected to couple all possible input/output fixings i.e. shafts, flanges, pulleys etc. Thus, the mountings at each end of the component may not be identical because the adjacent system elements may not have identical mounting arrangements. The way this problem is handled for retrieval purposes is explained in Section.

D8.5.43 **Flow Volume (Fluid "Charge")**

As previously mentioned, the torque speed characteristics of fluid couplings and rotary dampers can be adjusted by changing the characteristics of the working fluid. One of these characteristics is the amount of fluid present to transmit / dissipate power. Reducing the flow volume causes the fluid coupling to run with more slip and therefore less output speed. Reducing the working fluid in dampers is less common but would reduce the torque caused by fluid shear. In both cases, increased heating of the fluid will occur, particularly if the rated power dissipation is maintained. Friction clutches can be made to run with more slip, whilst reducing heating by running them "wet". This is an oil filling, lubricating and cooling the friction plates. The observation about this parameter is that it need not be mentioned as it simply extends the major performance characteristics.

D8.5.44 **Environment**

This will include data such as ambient temperature, pollution, noise etc. The major consideration is ambient temperature because additional cooling may be required if the temperature difference between the device and the surroundings is insufficient to allow an adequate heat exchange rate. If the ambient temperature is high, extra cooling such as fans, recirculation, radiators etc may be needed, all of which cause additional cost. Dry plate friction clutches must also be protected from ingress of contaminants, otherwise the friction surfaces may be rendered useless.

D8.5.45 **Insulation Class**

Electrically controlled clutches will require wiring, switches, slip rings etc. which will all have to satisfy the standard insulation types specified in British Standards .

D8.5.46 **Cooling**

Heat dissipation and therefore cooling are critical parameters when slip occurs in these devices. There are no standardized methods of cooling for clutches. Cooling must be achieved by the addition of fans (reduces ambient temperature) or by recirculation of the working fluid. It should be possible to select a component on rated values without considering cooling because, by definition, the rated characteristics are within the natural cooling parameters of the device.

D8.5.47 **Lubrication**

Lubrication must be available in all mechanisms where metal to metal contact occurs. Thus, the sprags, balls, cans etc. in overrunning clutches must all be lubricated. In wet running friction clutches, sufficient lubricant must be present for the slipping and cooling the friction surfaces. Fluid couplings and rotary dampers may well use the working fluid (oil) to self-lubricate.

D8.4.48 **Behaviour of the Friction Material**

Parameters such as wear, thermal properties, friction coefficient etc. will also be covered by this characteristic. These parameters are largely out of the control of the system designer, they define the performance of the device and can only be selected at the device design stage which is out of the scope of "off the shelf" component selection. For completeness, wear rate will affect the life of the clutch and therefore the specification of rated parameters. The thermal behaviour of the friction material will usually be lumped with that of the surrounding clutch parts and it is this lumped value that is used for selection. The friction coefficient will allow the closing force for a given transmitted torque to be calculated.

D8.5.49 **Radial and Axial Stiffness**

A related parameter is relative damping. Many clutches have flexible couplings built into them to help to absorb misalignments. The characteristics of flexible couplings are fully described in Chapter 7.

D8.5.50 **Misalignments**

As a device installed in-line between components, the ability to absorb radial, axial and end-float misalignments is important because of imperfection. These characteristics are described fully in Chapter 7. The ability to cope with misalignments is an important practical consideration for designers. However, it must be given secondary status to the ability of the clutch to perform the required power transmission functions.

D8.5.51 **Angle of Indexing, Angle of Rotation**

An overrunning clutch connected to an alternating rotary mechanical source will produce intermittent motion. The angle of index will depend on the frequency of oscillation.

Some rotary dampers have a limited rotation and therefore cannot be over-rotated because damage will occur if excessive torque is used.

D8.5.52 **Indexes per Minute**

This places a limit on the frequency of oscillation of the supply. Very rapid oscillation may damage the internal mechanisms of the device. If the backlash within the clutch is large, small indexes may become very inaccurate.

D8.5.53 **Weight**

Weight is less significant than inertia although it will effect the radial loads on the end bearings of the adjacent components.

D8.5.54 **Air Gap Size**

The size of the air gap between the friction surfaces in plate clutches. It is more significant in electromagnetic clutches where one plate is pulled onto another by forces exerted from the stationary plate.

Appendix E

Table 3.1.

Figure 3.2

Figure 3.3

Figure 3.4

Figure 3.5

Figure 3.6

Figure 3.7

Figure 3.8

Figure 3.9

Figure 3.10

Figure 3.11

Figure 3.12

Figure 3.13

Figure 3.14.

TABLE 3.1 : COMMON SOURCES OF ROTARY MECHANICAL POWER.

1) Internal Combustion Engine.
2) Steam Engine.
3) Gas Turbine.
4) AC Electric.
5) DC Electric
6) Hydraulic.
7) Pneumatic.
8) Clockwork.
9) Linear Mechanical.
10) Windpower.

FIGURE 3.2 : AN EXAMPLE OF DATA EXTRACTED FROM A COMPANY BROCHURE.

COMPANY NAME : Stephan Drives
T.I. MICROFILM NUMBER : 0396 - 2750
PARAMETERS USED (and comments) :

3 and Single Phase AC Motors

Enclosure and compliancy with DIn and IEC stands
Insulation class , Mounting types ,
Explosion classes - flameproof motors
Winding limiting temperature
IEC motor size
Supply volts , frequency
Balancing of rotating parts
Rated power
Rated speed
Rated torque
Efficiency
Starting torque (% of rated)
Load torque
Starting time
Rotor inertia

Nominal current
Starting current (% of rated)
Power factor

GRAPHICAL INFORMATION (and comments) :
Power v ambient temp / altitiude tables
Dimensions and parametric drawing
Supply frequency v power and speed tables

FIGURE 3.3 : AN EXAMPLE OF DATA EXTRACTED FROM A COMPANY BROCHURE.

COMPANY NAME : GEC Large Machines
T.I. MICROFILM NUMBER : 3690 - 5272
PARAMETERS USED (and comments) :

AC Motors

Standards ; degree of protection , cooling , mounting , dimensions , insulation.
Frame sizes, speeds , lubrication and bearings.
Intermittent ratings
Locked rotor KVA (given by √3 times rated volts x stator line current)
Locked rotor (starting) torque
Pull up torque
Overspeed (1.2 x rated speed)
Method of Start
Voltage for DOL start
Start torque
Start current
Starts per hour
Rated speed
Rated current at rated volts
Efficiency
Power factor
Rated torque
Pull out torque
Max locked rotor time
Max start time
Max inertia of load
Inertia of rotor
Noise
Number of poles

GRAPHICAL INFORMATION (and comments) :

Table for poles , power , frame size - relates size to power and rated speed.
Speed v Vibration table.
Speed v axial / radial forces graph.

FIGURE 3.4 : AN EXAMPLE OF DATA EXTRACTED FROM A COMPANY BROCHURE.

COMPANY NAME : Mawdsleys
T.I. MICROFILM NUMBER : 3683 - 4380
PARAMETERS USED (and comments) :

AC/DC Motors

Power
Starting torque
Pull in (up) torque
Pull out torque
Dimensions
No of poles
Frame size
Rated Speed
Supply ratings
Base speed
Max speed
Efficiency
Armature resistance / inductance
Field power

Enclosure
Mountings
Ambient temperature
Shock loads
Insulation

GRAPHICAL INFORMATION (and comments) :

Power v speed curves

FIGURE 3.5 : AN EXAMPLE OF DATA EXTRACTED FROM A COMPANY BROCHURE.

COMPANY NAME : SEW Eurodrive

T.I. MICROFILM NUMBER : 0402 - 0002

PARAMETERS USED (and comments) :

"VARIATOR"

Type of unit	Ordering details
Frame type	Output power
Mounting type	Output speed max / min
Power	Output torque max / min
Speed range	Overhung loads
Speed ratio	Axial force
Torque range	Duty
	Supply details
	Ambient temp
	Stop / start frequency
	Inertia of load

GRAPHICAL INFORMATION (and comments) :

FIGURE 3.6 : AN EXAMPLE OF DATA EXTRACTED FROM A COMPANY BROCHURE.

COMPANY NAME : Oilger Towler
T.I. MICROFILM NUMBER : 0379 - 5189
PARAMETERS USED (and comments) :

Axial piston hydraulic motor

Dispacement - fixed or variable (ml per rev)
Speed - max , min , rated
Pressure - continuous , maximum
Breakaway pressure at no load
Torque at rated pressure and max displacement
HP at rated speed, pressure and max displacement
Inertia of rotor
Weight
Dimensions
Type of control
Type of mounting
Type of fluid

GRAPHICAL INFORMATION (and comments) :

Torque v pressure - linear
HP v speed for fixed pressures - linear
Input delivery v speed
Efficiency v pressure at various speeds

FIGURE 3.7 : AN EXAMPLE OF DATA EXTRACTED FROM A COMPANY BROCHURE.

COMPANY NAME : Bosch, R-Hydraulics
T.I. MICROFILM NUMBER : 0397 - 0002
PARAMETERS USED (and comments) :

Gearmotors

Direction of rotation - Reversibility Y / N
Speed range
Mounting positions
Max starting pressure. Peak pressure when switching on.
Max discharge pressure for continuous running
Displacement
Max speed at continuous pressure
Min speed
Fluid working temp range
Weight
Input power
Dimensions / mountings
Leakage line pressure
Shaft loadings

$$\text{Efficiency} = \frac{\text{Power out}}{\text{Power in}} = \frac{\text{kW}}{p \times Q}$$

GRAPHICAL INFORMATION (and comments) :

Flow v speed for various displacements at set pressures - linear
Noise v pressure
Power out v speed at set pressures - linear
Torque v speed at set pressures - linear
Torque v speed at set pressures and flows with efficiency plotted
Flow v power at set preesures - linear

FIGURE 3.8 : AN EXAMPLE OF DATA EXTRACTED FROM A COMPANY BROCHURE.

COMPANY NAME : ARO Corporation

I.T. MICROFILM NUMBER : 3868 - 5048

PARAMETERS USED (and comments) :

Vane Motor

Speed range

Torque range

Free speed

Load speed at max HP

Weight

Mountings

Dimensions

Inlet air pressure

GRAPHICAL INFORMATION (and comments) :

FIGURE 3.9 : INPUT CHARACTERISTICS FOR AC MOTORS.

Rated input

Mechanical efficiency

Electrical efficiency

Rated colts

Rated current

Start current

Capacitance values

Number of phases

Frequency of supply

Number of poles (rotor and Stator)

Maximum locked rotor time

Maximum start time

Intermittent rating period

Type of start (direct on line / resistance / star-delta)

KVA code

FIGURE 3.10: OUTPUT CHARACTERISTICS FOR AC MOTORS.

Rated power

Rated torque

Rated speed

Synchronous speed

Slip speed

% slip

Locked rotor torque

Pull up torque

Pull out torque

Static friction torque

Ripple torque

Breakaway torque

Reversibility

FIGURE 3.11 : PHYSICAL CHARACTERISTICS OF AC MOTORS.

Dimensioned drawings

Frame type to BS

Enclosure

Insulation class

Acceptable loads on output shaft

Mounting details

Rotor moment of inertia

Rotor mass

Motor mass

Noise

FIGURE 3.12 : OTHER CHARACTERISTICS OF AC MOTORS.

Manufacturer/supplier name

FIGURE 3.13 pt 1 : INCOMPLETE PARAMETER LISTS.

AC MOTORS	DC MOTORS	HYDRAULIC MOTORS	PNEUMATIC MOTORS
Rated Power	Rated Power	Rated Power	
Rated Torque	Rated Torque	Rated Torque	
Rated Speed	Rated Speed	Rated Speed	
Locked Rotor Torque	Starting Torque		
Pull Up Torque			
Pull Out Torque			
Static Friction T\que	Static Friction T\qu		
		Maximum Speed	Maximum Speed
		Minimum Speed	Minimum Speed
		No Load Speed	Free Speed
Reversibility	Reversibility	Reversibility	Reversibility
cont...	cont...	cont...	cont...

FIGURE 3.13 pt 2 : INCOMPLETE PARAMETER LISTS FOR MOTORS.

AC MOTORS	DC MOTORS	HYDRAULIC MOTORS	PNEUMATIC MOTORS
	Rated Input Power		
	Efficiency		
Mechanical Eff\cy			
Electrical Eff\cy			
Rated Volts	Rated Volts		
Rated Current	Rated Current		
	Armature Current		
Start Current			
	Field Power		
	F\ld Excitation V\ts		
	Armature Resistance		
cont...	cont...	cont...	cont...

FIGURE 3.13 pt 3 : INCOMPLETE PARAMETER LISTS FOR MOTORS.

AC MOTORS	DC MOTORS	HYDRAULIC MOTORS	PNEUMATIC MOTORS
	Armature Inductance		
Capacitance Values			
Number of Phases			
Frequency of Supply			
Stator Poles			
Rotor Poles			
Mx L\ked Rotor Time			
Max Start Time	Initial Acceleration		
Int\nt R\ng Period	Int\nt R\ng Period		
		Max Int\nt Pressure	Max In\nt Pressure
D-O-L Start	D-O-L Start	D-O-L Start	D-O-L Start
cont...	cont...	cont...	cont...

FIGURE 3.13 pt 4 : INCOMPLETE PARAMETER LISTS FOR MOTORS.

AC MOTORS	DC MOTORS	HYDRAULIC MOTORS	PNEUMATIC MOTORS
Star - Delta Start			
Resistance Start			
Auto Transformer S/t			
KVA Code			
		Max Operating Temp\	
Bearing Load - Axial	Bearing Load - Axial	Bearing Load - Axial	Bearing Load - Axial
Bearing Load - Rad\l	Bearing Load - Rad\l	Bearing Load - Rad\l	Bearing Load - Rad\l
Rotor Mass	Armature Mass	Rotor Mass	Rotor Mass
Rotor Inertia	Armature Inertia		
Inertia of Load			
Motor Mass	Motor Mass	Motor Mass	Motor Mass
cont...	cont...	cont...	cont...

FIGURE 3.13 pt 5 : INCOMPLETE PARAMETER LISTS FOR MOTORS.

AC MOTORS	DC MOTORS	HYDRAULIC MOTORS	PNEUMATIC MOTORS
Frame Type	Frame Type	Frame Type	Frame Type
Mount Type	Mount Type	Mount Type	Frame Type
Enclosure	Enclosure		
Insualtion	Insulation		
Noise			
			Regulation
		Balanced	
		Filtration	Filtration
			Pres\ L\ss on No L\d
		Delivery	Delivery

FIGURE 3.14 pt 1 : COMPLETE, ORDERED PARAMETER LISTS FOR MOTORS.

AC MOTORS	DC MOTORS	HYDRAULIC MOTORS	PNEUMATIC MOTORS
Rated Power	Rated Power	Rated Power	Rated Power
Rated Torque	Rated Torque	Rated Torque	Rated Torque
Rated Speed	Rated Speed	Rated Speed	Rated Speed
Locked Rotor Torque	Starting Torque	Starting Torque	Starting Torque
Pull Up Torque	Pull Up Torque	Pull Up Torque	Pull Up Torque
Pull Out Torque	Pull Out Torque	Pull Out Torque	Pull Out Torque
Static Friction T\que	Static Friction T\qu	Breakaway Pressure	Breakaway Pressure
Synchronous Speed	Maximum Speed	Maximum Speed	Maximum Speed
Pull Out Speed	Minimum Speed	Minimum Speed	Minimum Speed
No Load Speed	No Load Speed	No Load Speed	Free Speed
Reversibility	Reversibility	Reversibility	Reversibility
cont...	cont...	cont...	cont...

FIGURE 3.14 pt 2 : COMPLETE, ORDERED PARAMETER LISTS FOR MOTORS.

AC MOTORS	DC MOTORS	HYDRAULIC MOTORS	PNEUMATIC MOTORS
Rated Input Power	Rated Input Power	Fluid Power	Fluid Power
Efficiency	Efficiency	Efficiency	Efficiency
Mechanical Eff\cy	Mechanical Losses	Mechanical Losses	Mechanical Losses
Electrical Eff\cy	Electrical Losses	Fluid Losses	Fluid Losses
Rated Volts	Rated Volts	Rated Pressure	Rated Pressure
Rated Current	Rated Current	Rated Flow	Rated Flow
Armature Current	Armature Current		
Start Current	Start Current	Starting Pressure	Starting Pressure
Stator Power	Field Power	Fluid Power	Fluid Power
St\r Excitation V\ts	F\ld Excitation V\ts		
Armature Resistance	Armature Resistance		
cont...	cont...	cont...	cont...

FIGURE 3.14 pt 3 : COMPLETE, ORDERED PARAMETER LISTS FOR MOTORS.

AC MOTORS	DC MOTORS	HYDRAULIC MOTORS	PNEUMATIC MOTORS
Armature Inductance	Armature Inductance		
Capacitance Values			
Number of Phases		Function of Porting	Function of Porting
Frequency of Supply		F\n of Speed & P\ng	F\n of Speed & P\ng
Stator Poles	Stator Poles	F\n of Inlets	F\n of Inlets
Rotor Poles	Rotor Poles	& Outlets	& Outlets
Mx L\ked Rotor Time	Mx L\ked Rotor Time	Mx L\ked Rotor Time	Mx L\ked otor Time
Max Start Time	Initial Acceleration	Max Start Time	Max Start Time
Int\nt R\ng Period	Int\nt R\ng Period	Int\nt R\tng Period	Int\nt R\tng Period
Max Int\nt Current	Max Int\nt Current	Max Int\nt Pressure	Max In\nt Pressure
D-O-L Start	D-O-L Start	D-O-L Start	D-O-L Start
cont...	cont...	cont...	cont...

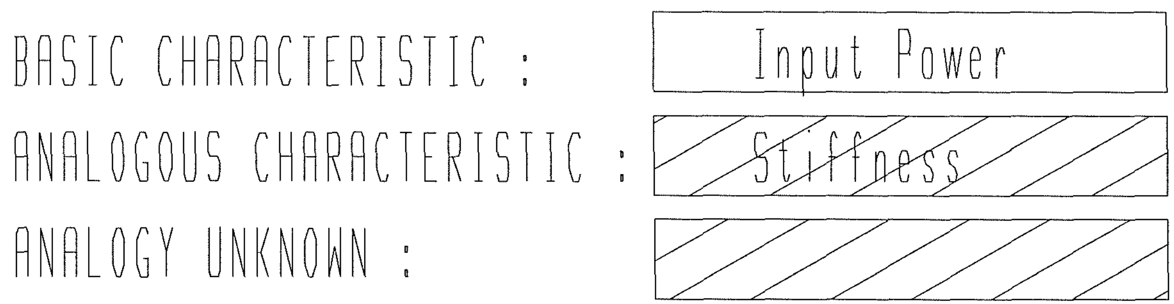
FIGURE 3.14 pt 4 : COMPLETE, ORDERED PARAMETER LISTS FOR MOTORS.

AC MOTORS	DC MOTORS	HYDRAULIC MOTORS	PNEUMATIC MOTORS
Star - Delta Start			
Resistance Start	Resistance Start	Valves	Valves
Auto Transformer S/t			
KVA Code	Cont\ Stall Input	Cont\ Stall Input	Cont\ Stall Input
Max Operating Temp/	Max Operating Temp\	Max Operating Temp\	Max Operating Temp\
Bearing Load - Axial	Bearing Load - Axial	Bearing Load - Axial	Bearing Load - Axial
Bearing Load - Rad\l	Bearing Load - Rad\l	Bearing Load - Rad\l	Bearing Load - Rad\l
Rotor Mass	Armature Mass	Rotor Mass	Rotor Mass
Rotor Inertia	Armature Inertia	Rotor Inertia	Rotor Inertia
Inertia of Load	Inertia of Load	Inertia of Load	Inertia of Load
Motor Mass	Motor Mass	Motor Mass	Motor Mass
cont...	cont...	cont...	cont...

FIGURE 3.14 pt 5 : COMPLETE, ORDERED PARAMETER LISTS FOR MOTORS.

AC MOTORS	DC MOTORS	HYDRAULIC MOTORS	PNEUMATIC MOTORS
Frame Type	Frame Type	Frame Type	Frame Type
Mount Type	Mount Type	Mount Type	Frame Type
Enclosure	Enclosure		
Insualtion	Insulation		
Noise	Noise	Noise	Noise
			Regulation
		Balanced	
		Filtration	Filtration
		Pres\ Loss on No L\d	Pres\ Loss on No L\d
		Delivery	Delivery

<u>FIGURE 3.14 KEY :</u>



Appendix F

Figure 4.2.

Table 4.1.

Figure 4.11

Figure 4.12.

Table 4.3.

Figure 4.14

Figure 4.15

Figure 4.16

Figure 4.17

FIGURE 4.2 : Typical Torque / Slip Curve.

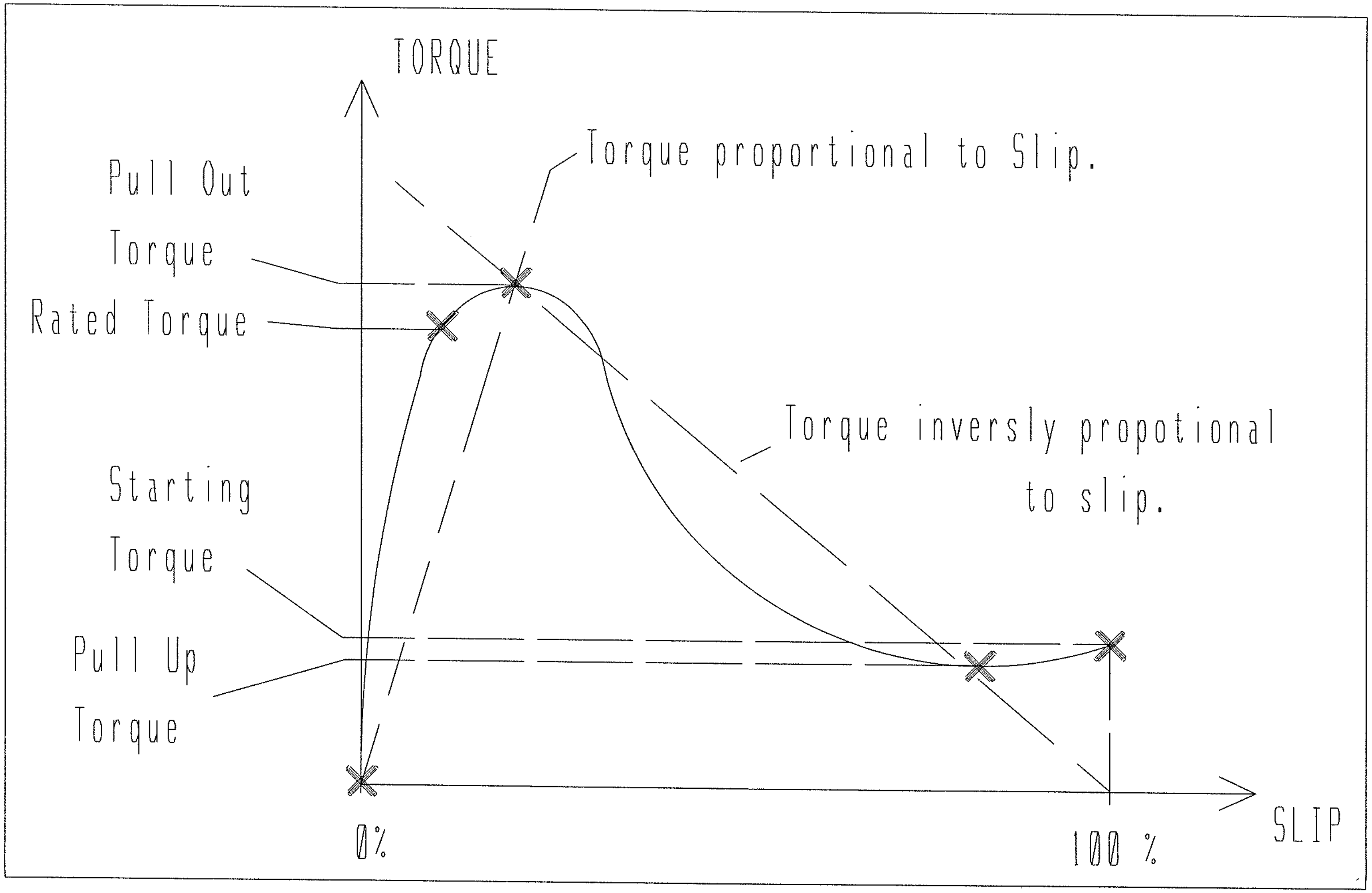


TABLE 4.1 : CONNECTIONS BETWEEN THE WORKING PARTS OF A PISTON MOTOR.

MOTOR PART	PINTLE	PISTONS	SWASHPLATE	DRIVESHAFT
PINTLE	▨			
PISTONS	✓	▨		
SWASHPLATE	X	✓	▨	
DRIVESHAFT	✓	X	✓	▨

KEY :

✓	MECHANICAL LINK CAN BE BROKEN.
▨	MECHANICAL LINK CANNOT BE BROKEN.
X	NO MECHANICAL LINK.
	DUPLICATION.

FIGURE 4.11 : Common Motor Mounting Types.

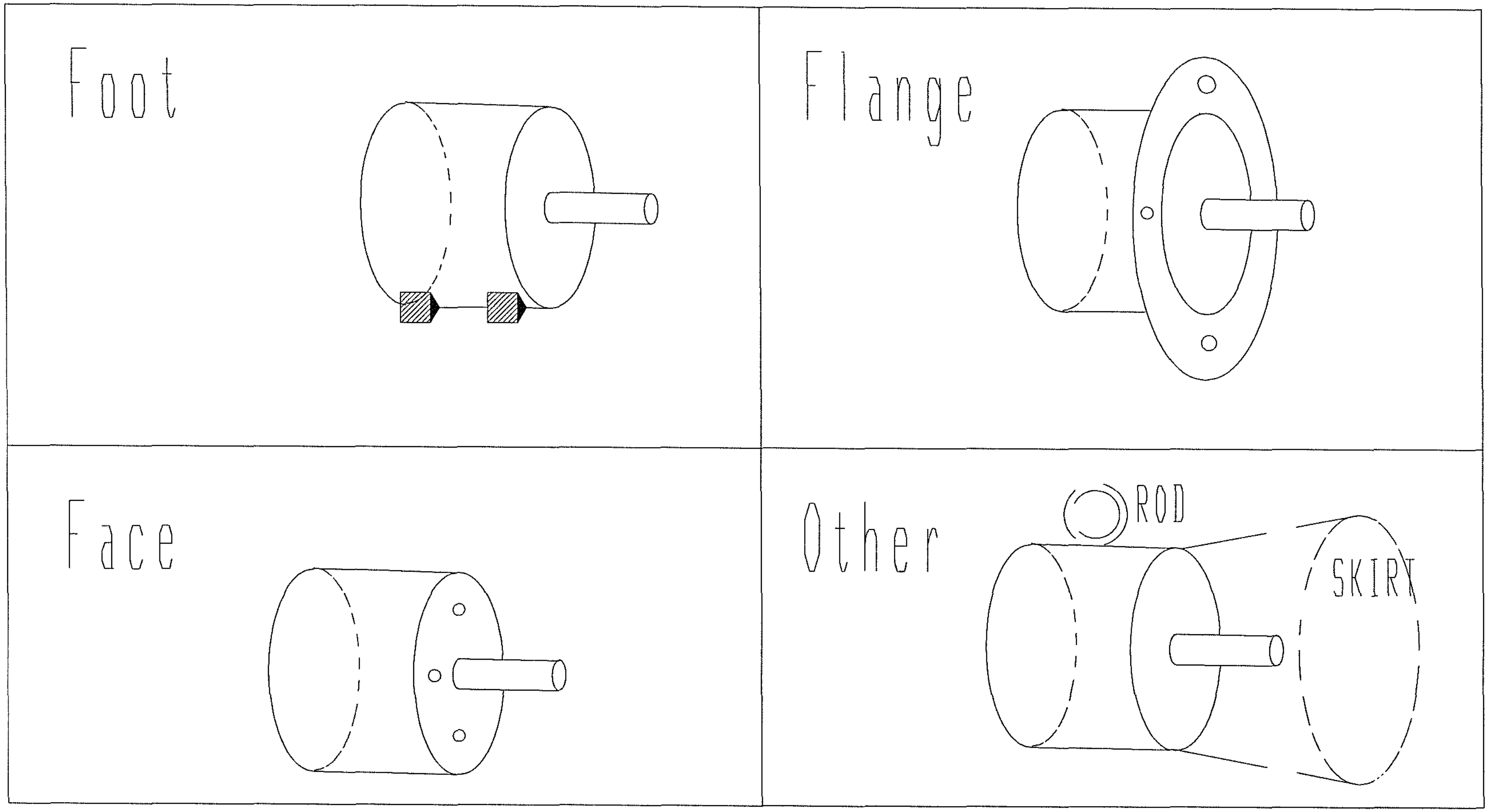


FIGURE 4.12 : Extract From BS 4999 part 7.

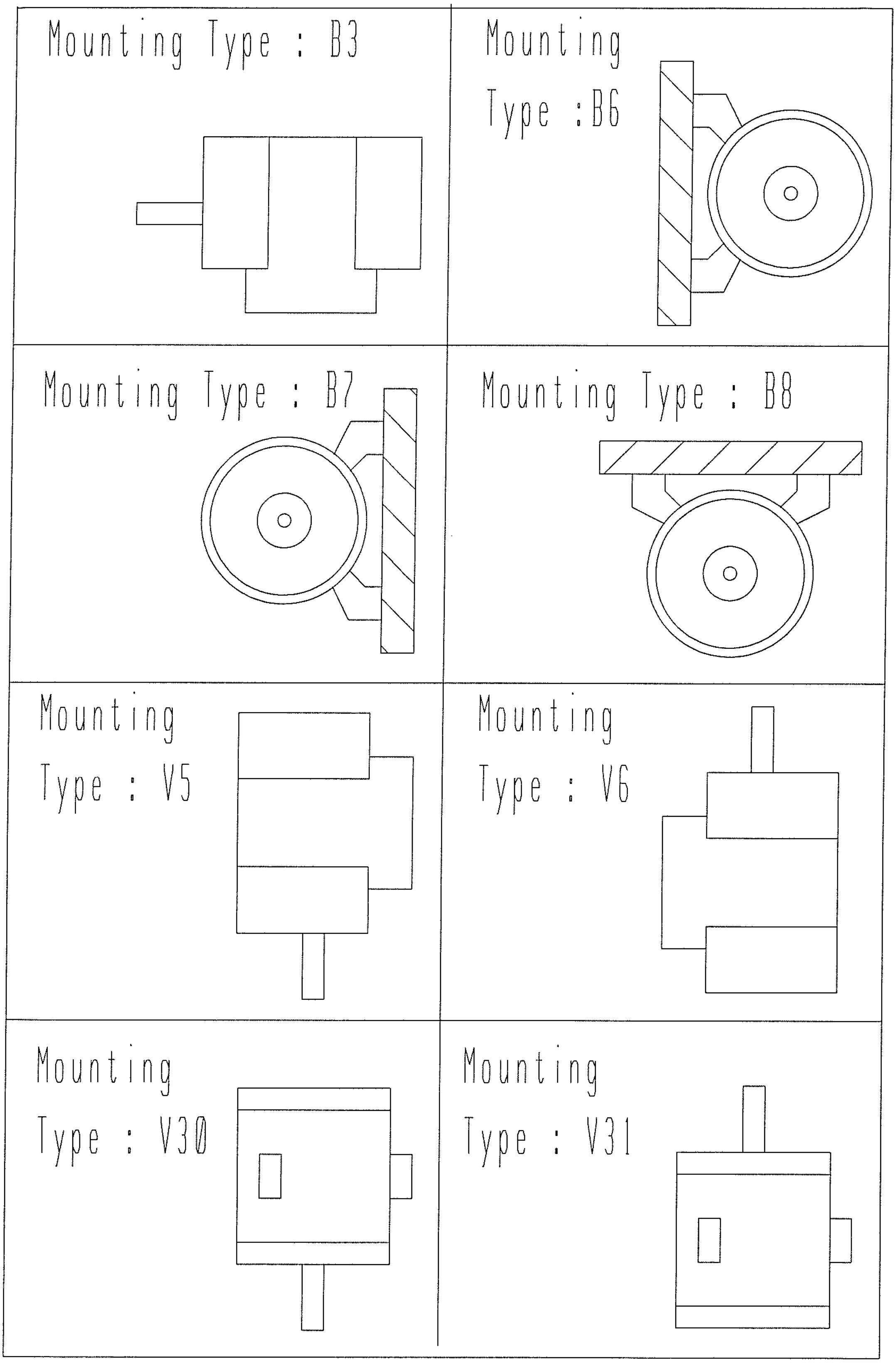


TABLE 4.3 : <u>Typical Service Factors for Motors.</u>

TYPE of DRIVEN Machine	DUTY : Hours per Day.					
	Soft Start			Heavy Start		
	10 and under	Over 10 to 18	Over 18	10 and under	Over 10 to 18	Over 18
LIGHT DUTY : Blowers, Fans, Centrifugal Compressors and Pumps.	1.0	1.1	1.2	1.3	1.4	1.5
MEDIUM DUTY : Machine tools, Belt Conveyors, Printing Machinery.	1.2	1.3	1.4	1.5	1.6	1.7
HEAVY DUTY : Bucket Elevators, Reciprocating Machinery.	1.5	1.6	1.7	1.8	1.9	2.0

FIGURE 4.14 : EXAMPLE OF INDEXED MOTOR.

Rated Power = 4.0 kW	
Rated Torque = 13 Nm	
Rated Speed = 2880 RPM	
Starting Torque = 26 Nm	
Maximum Speed = 2880 RPM	
Minimum Speed = 2448 RPM	
Maximum Length = 410 mm	
Maximum Height = 279 mm	
Maximum Width = 304 mm	
Mount Type = F	
Motor Type = A	
Company Name = Broook Crompton Parkinson	
Pull Up Torque =	
Pull Out Torque =	
Motor Frame Code = HD112M	
Microfilm Number = 3673 - 3740	

FIGURE 4.15 : EXAMPLE OF INDEXED MOTOR.

Rated Power = 4.2 kW

Rated Torque = 44 Nm

Rated Speed = 900 RPM

Starting Torque =

Maximum Speed = 3500 RPM

Minimum Speed = RPM

Maximum Length = 749 mm

Maximum Height = 430 mm

Maximum Width = 480 mm

Mount Type = F

Motor Type = D

Company Name = Mawdsley\s

Pull Up Torque = 3500 Nm

Pull Out Torque =

Motor Frame Code = 180 Q

Microfilm Number = 3675 - 5080

FIGURE 4.16 : EXAMPLE OF INDEXED MOTOR.

Rated Power = 5.0 kW
Rated Torque =
Rated Speed = 1000 RPM
Starting Torque =
Maximum Speed = 3000 RPM
Minimum Speed = 500 RPM
Maximum Length = 160 mm
Maximum Height = 147 mm
Maximum Width = 118 mm
Mount Type = L
Motor Type = H
Company Name = Sundstrand Hydratec
Pull Up Torque =
Pull Out Torque =
Motor Frame Code = GM32.7
Microfilm Number = 388 - 0590

FIGURE 4.17 : EXAMPLE OF INDEXED MOTOR.

Rated Power = 5.0 kW

Rated Torque = 118 Nm

Rated Speed = 400 RPM

Starting Torque = 128 Nm

Maximum Speed = 850 RPM

Minimum Speed =

Maximum Length = 495 mm

Maximum Height = 288 mm

Maximum Width = 288 mm

Mount Type = L

Motor Type = P

Company Name = Atlas Copco

Pull Up Torque =

Pull Out Torque =

Motor Frame Code = MHZ 42 G51 P1

Microfilm Number = 3885 - 0002

APPENDIX G

The Shaft Configuration Code Algorithm

A great deal of time has been spent by various undergraduate project students (REFs 12, 13, 14) to try to find a simple, unambiguous way of defining the position of shafts on a gearbox. The results although valid would have proved far from simple to implement because the problem is complex and perhaps the students attempted to do too much. Certain solutions resulted in the generation of a twenty character code. This is obviously unacceptable because it would be impractical to expect the user to derive it and if the code was generated for the user by asking a series of questions, the number of questions would be excessive (around ten). It was decided early on in these studies, to abandon any of the previous ideas and attempt to start afresh and find a simple, elegant and original way of classifying the shaft positions on a gearbox. The way this problem was approached was to firstly understand exactly what was needed.

When trying to define the position of any object, the problem is to set a datum so that the position of the object can be defined relative to it. In the case of a gearbox, the datum must be part of the gearbox and thus all other references are made relative to this. This is where the problem lies. Firstly, when the database is created the compiler must create the code using the manufacturer's catalogue. This will be a photograph or more often, a dimensioned drawing. The datum must be located and then the classification made. This may be difficult and time consuming if the compiler has to perform a transformation to change the orientation of the gearbox to establish a datum. For instance, if the datum is the input shaft and some sort of classification is used defining shafts clockwise relative to it, and the input shaft is at the "back" of the gearbox, the compiler must re-draw the gearbox in its preferred orientation. A more serious problem occurs if the gearbox is epicyclic and therefore may have two inputs. In this case further classifications are needed to define which shaft takes precedence and how the other input is to be handled.

Another consideration is attempting to define the gearbox in an unambiguous way. It is vital that the same gearbox, viewed from two different angles, is not classified differently. This can be avoided by having a very rigorous classification system. This only leads to the problems mentioned above and is illustrated in the student projects. If such classification systems were employed, the user would also be inconvenienced by having to follow a very precise method for defining the shafts. It is important that the system allows the user to define the gearbox shafts in the way the user "sees" it and not the way the system would prefer it. Thus the system must be able to decipher and re-configure information.

The next step is to identify what is needed to specify the layout of the gearbox. Dealing firstly with the shafts, the number of shafts must be known. The shafts must be identified as input or output. The relationships between the shafts must be defined. This would mean which input drove which output their ratio and finally their position on the gearbox.

The method and position of the mounting is also something that must be considered and the relationship between the mounting and the shafts may be important. It should be noted that a similar problem occurred when the mounting type and orientation was defined for motors. The conclusion was that simply specifying the type of mount was sufficient, the numerous permutations of mount types and orientations were an unnecessary consideration. This was because motors usually only have one shaft, and in the majority of cases the relative position of the shaft and mounting is obvious. For instance, face, flange and skirt mounting types are all used to let the motor bolt onto another device so that the motor shaft can power it. It is therefore clear that these three mount types are located on the shaft face of the motor. The relationship between mounting and particular shafts is not so well defined even in the simplest gearboxes. It is therefore important to consider if it is possible to include data relating mounting type to the positions of the input and output shafts.

From these studies it is now possible to accurately define what is needed from the gearbox shaft configuration information :

i. The number of shafts.
ii The number of inputs and outputs.
iii Which input drives which output.
iv The ratios for the above relationships.
v. The positions of the shafts relative to each other.
vi The mounting arrangement.

In addition to these constraints, two simplifying assumptions can be made. By the nature of their construction, most gearboxes have shafts which are always in the same plane. Within that plane, the shafts are separated by angles whose lowest common denominator is ninety degrees. For a gearbox to fall outside both of these criteria, it would have to have a skew, offset shaft. Figures G1.1 - G1.3 show some common gearboxes, Figure G1.4 shows a skew offset shaft - no real 'boxes' of this type are known to the research.

With the problem well defined, possible methods of classification could be investigated. Also, the possibility of part solutions could be investigated.

A method that had not been tried by any of the previous researchers, was to attempt to describe the shafts vectorially. The reasoning behind this was that firstly the relationships between the shafts would be held within the position vectors. Secondly, the positional data could be easily extracted and manipulated and thirdly, there may have been a possibility of including direction of rotation and magnitude of torque for each shaft. The direction of the vector (towards the origin or away from the origin) could be taken to indicate direction of rotation by the right hand grip screw rule. The magnitude of the vector could represent the torque or speed for the shaft. Furthermore, because of the simplifying assumptions made, the relationships between the shafts can be investigated using simple vector manipulations such as dot and cross products.

The advantage of a vectorial system is that, provided the i, j, k relationships are obeyed, the x, y, z axes can be laid onto the gearbox representation in any convenient orientation. If the gearbox is imagined as a cube, with the x, y, z axes at its centre and cutting the cube faces at right angles, the negative x, y and z axes will then cut the opposing faces at right angles. Thus, each face will be "labelled" by an axis. From the simplifying assumptions, any shafts on the gearbox face will always lie on one of the axes. Thus, to simply indicate if a shaft was present on a gearbox face, a unit vector from the origin in the direction of the face with the shaft would be a sufficient indication of a shaft position relative to the other axes and hence, shafts. Simple vector analysis could then be used to derive the relationships between shafts. Dot products will identify shafts on the same face, shafts on opposing faces and shafts at right angles. A cross product using any two shafts found to be at right angles from a dot product test will indicate on what line the third shafts must lie in order for there to be three shafts mutually at right angles.

FIGURES G1.1 to G1.4: SCHEMATIC DRAWINGS OF THREE COMMON GEARBOX CONFIGURATIONS AND A SKEW GEARBOX.

FIGURE G1.1.

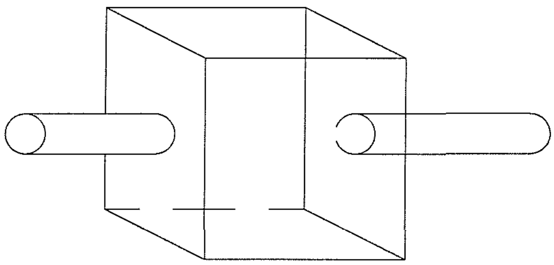


FIGURE G1.2.

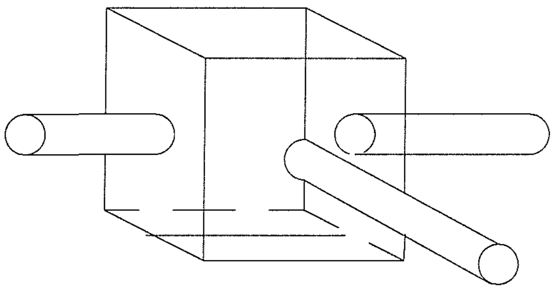


FIGURE G1.3.

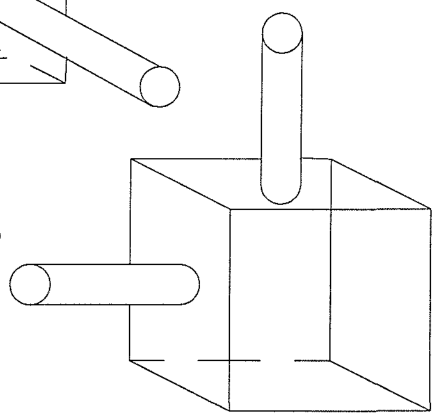
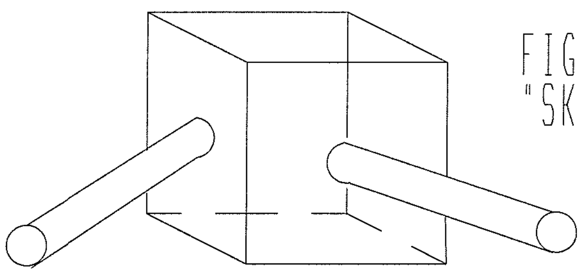


FIGURE G1.4.
"SKEW" GEARBOX.



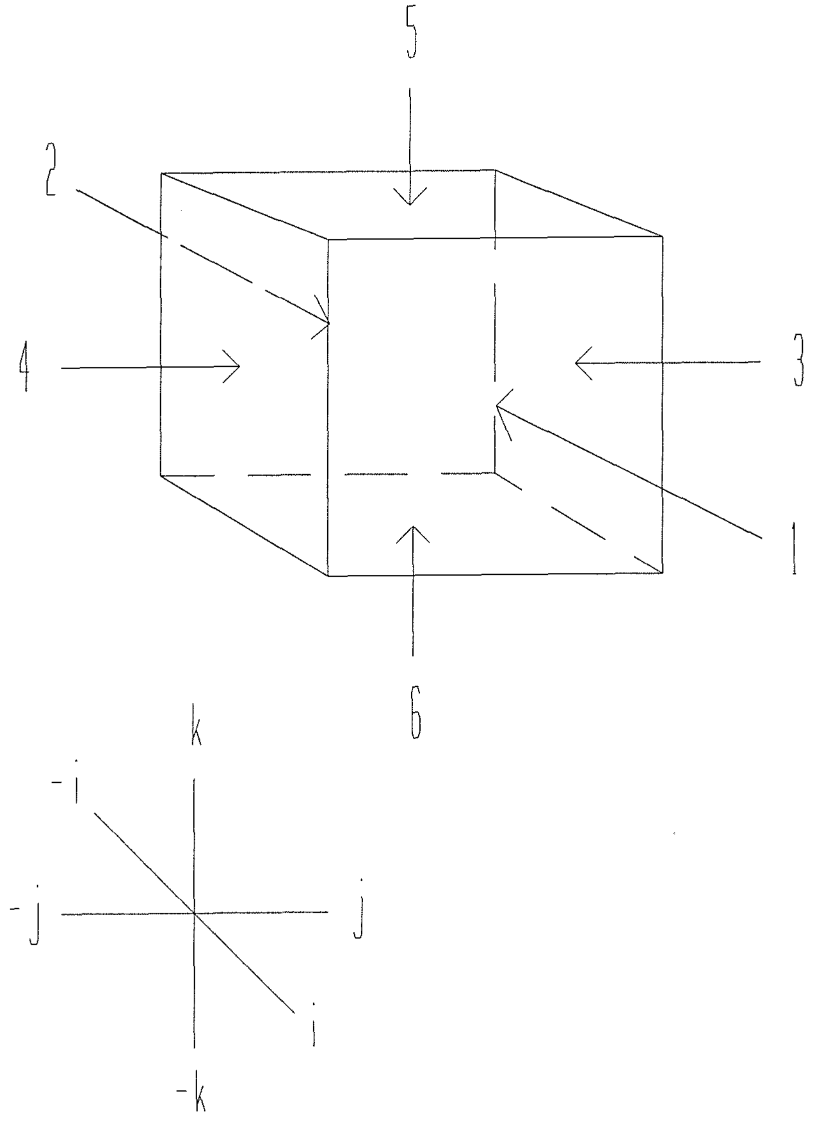
The previous simplifying assumptions making all shafts enter the gearbox at right angles to a face, allows a very simple and effective method to be proposed for the definition of the relative positions of the shafts. The relationships between the shafts and not their relative positions could be used to describe the relative positions. For instance, two shafts in line is an unambiguous description. It is simpler than attempting to say the dot product of their vectors is minus one!. Two shafts at right angles is also unambiguous. Three shafts, two in-line and two right angle pairs can only mean one configuration, a "T-shape". Adding the further possible feature of three shafts at right angles, a descriptive system for classifying the relative positions of the gearbox shafts without the need to establish datums can be developed. The shaft relationships must be quantified as follows :

(i) The total number of shafts.
(ii) The number of right angles between the shafts.
(iii) The number of sets of three shafts mutually at right angles.
(iv) The number of shafts in-line including those involved in the other features.

The above quantities are simple to understand and easy to collect. They can be combined into a four digit number. The generation of the shaft configuration code can be done very simply using around twenty lines of very simple Basic programming.

The code is derived by firstly imagining the gearbox to be a cube whose faces are numbered one to six as in Figure G2. This figure can be made available to the user. The user can then imagine this cube superimposed on the gearbox layout required. The user can then be prompted to indicate the number of shafts located on each face of the imaginary cube. These inputs can be stored in an array of dimension six, the subscript of each element corresponding to a cube face. The values in the array elements will be integers greater than or equal to one if there are shafts passing through a particular face, or zero, if there are no shafts passing through the cube face. A gearbox shaft corresponds to any input or output shaft entering or leaving the gearbox. A continuous shaft through the gearbox would be counted as two shafts because it passes through two faces. Shaft mounted gearboxes are assumed to have the through shaft present.

FIGURE G2 : THE IMAGINARY GEARBOX USED TO DEFINE THE SHAFT POSITIONS FOR THE CODING ALGORITHM.



Since the relative positions of the imaginary gearbox faces are known to the software and the design gearbox shafts have been located using this imaginary reference. The relationships between the design gearbox shafts can now be investigated.

With reference to Figure G2, the sets of three shafts mutually at right angle is given in Table G1. Rather than list all of the permutations shown below, it should be possible to derive the set of combinations to be tested mathematically. The proof should be fairly simple. However, because of the small number of permutations involved, it was quicker to simply write out and eliminate repetitions by hand.

Using the information in Table G1, an algorithm to detect sets of three shafts at right angles can be created. The listing is given at the end of this Appendix. The algorithm will end with the number of sets of three orthogonal shafts contained in the variable "B".

The number of shafts in-line can also be determined very simply by listing the permutations, again, small numbers make this quicker and easier by hand, rather than mathematically. Thus from Figure G2, Table G2 can be generated. From Table G2 an algorithm can be created to test the appropriate array elements. The program listing is given at the end of this Appendix. The algorithm will end with the number of shafts in-line stored in variable "C".

With reference to Figure G2 and Table G3, an algorithm can be created to test the array for pairs of orthogonal shafts. The listing is given at the end of this Appendix. The algorithm will end with the number of right angles stored in the variable "D".

As with the other two algorithms, it should be possible to derive the combinations to be tested mathematically. However the small number of tests make it simpler to generate the combinations by hand.

The three configuration code elements, B, C and D can now be combined along with the variable holding the total number of shafts ("A" say) into a four figure code of the form: ABCD.

TABLE G1 : THE POSSIBLE COMBINATIONS OF THREE SHAFTS MUTUALLY AT RIGHT ANGLES.

1)	E(1) ,	E(3) ,	E(5)	SETS OF COMBINATIONS TO BE TESTED
2)	E(1) ,	E(3) ,	E(6)	
3)	E(1) ,	E(4) ,	E(5)	
4)	E(1) ,	E(4) ,	E(6)	
5)	E(2) ,	E(3) ,	E(5)	
6)	E(2) ,	E(3) ,	E(6)	
7)	E(2) ,	E(4) ,	E(5)	
8)	E(2) ,	E(4) ,	E(6)	
9)	E(3) ,	E(1) ,	E(5)	REPETITIONS OF FIRST EIGHT SETS
10)	E(3) ,	E(1) ,	E(6)	
11)	E(3) ,	E(2) ,	E(5)	
12)	E(3) ,	E(2) ,	E(6)	
13)	E(4) ,	E(1) ,	E(5)	
14)	E(4) ,	E(1) ,	E(6)	
15)	E(4) ,	E(2) ,	E(5)	
16)	E(4) ,	E(2) ,	E(6)	
17)	E(5) ,	E(1) ,	E(3)	
18)	E(5) ,	E(1) ,	E(4)	
19)	E(5) ,	E(2) ,	E(3)	
20)	E(5) ,	E(2) ,	E(4)	
21)	E(6) ,	E(1) ,	E(3)	
22)	E(6) ,	E(1) ,	E(4)	
23)	E(6) ,	E(2) ,	E(3)	
24)	E(6) ,	E(2) ,	E(3)	

TABLE G2 : THE POSSIBLE COMBINATIONS OF TWO SHAFTS INLINE.

1)	E(1) , E(2)	SETS OF COMBINATIONS TO BE TESTED
2)	E(2) , E(1)	
3)	E(3) , E(4)	SETS OF COMBINATIONS TO BE TESTED
4)	E(4) , E(3)	
5)	E(5) , E(6)	SETS OF COMBINATIONS TO BE TESTED
6)	E(6) , E(5)	

Combinations (2) , (4) and (6) are repetitions of (1) , (3) and (5).

TABLE G3 : THE POSSIBLE COMBINATIONS OF TWO SHAFTS MUTUALLY AT RIGHT ANGLES.

1)	E(1) ,	E(3)
2)	E(1) ,	E(4)
3)	E(1) ,	E(5)
4)	E(1) ,	E(6)
5)	E(2) ,	E(3)
6)	E(2) ,	E(4)
7)	E(2) ,	E(5)
8)	E(2) ,	E(6)
9)	E(3) ,	E(1)
10)	E(3) ,	E(2)
11)	E(3) ,	E(5)
12)	E(3) ,	E(6)
13)	E(4) ,	E(1)
14)	E(4) ,	E(2)
15)	E(4) ,	E(5)
16)	E(4) ,	E(6)
17)	E(5) ,	E(1)
18)	E(5) ,	E(2)
19)	E(5) ,	E(3)
20)	E(5) ,	E(4)
21)	E(6) ,	E(1)
22)	E(6) ,	E(2)
23)	E(6) ,	E(3)
24)	E(6) ,	E(4)

SETS OF COMBINATIONS TO BE TESTED

With reference to the seven configuration requirements above, the code developed so far meets only two of the constraints - 1 and 6. Number 1 is trivial and number 6 is one of the more complex problems. The necessity of developing the code so that the additional constraints are met is clear. However, although far from complete, the configuration code has achieved far more than previous solutions. It provides a simple and efficient method for defining the positional relationships between the gearbox shafts. The major inadequacy is that the position of the mounting is not included in the specification. The connections between the inputs and outputs and their associated ratios are also not included in the code.

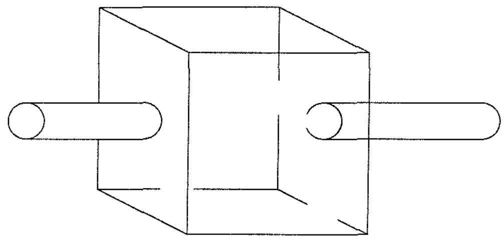
The final observation about the configuration code is that it deals very successfully with all of the common gearbox configurations and will also cope with any unconventional arrangements - a gearbox with more than three shafts or with all shafts on the same face. This ability to cope with oddities gives the coding technique a great advantage over a system that simply attempts to list all possible gearbox configurations and to categorise them. As seen with BS 4999 part 22 (REF. 10), the number of options would become excessive, requiring the user to scan a series of diagrams until the best match was found. The user would probably find looking through a series of very similar diagrams tedious and also, if an uncommon configuration was required, it may not be listed.

Figures G3.1-G3.4 show schematic representations of common gearbox shaft layouts and the configuration codes to describe them.

From the field trials (see Chapter 9 for more detail), the users were very happy with the configuration code generation routines. They were helpful and certainly a step in the right direction given the complexity of the problem. The code could be perhaps further developed to include the relative position of the baseplate. This would then be a very thorough specification of the spatial characteristics of the gearbox. It must be noted that these relationships are only one "characteristic" in the gearbox specification. A sense of priority must be maintained and it was felt that the configuration code thus far developed, was sufficient for the needs and purposes of research.

FIGURES G3.1 - G3.4: SCHEMATIC DRAWINGS OF FOUR COMMON GEARBOX LAYOUTS AND THEIR ASSOCIATED SHAFT CONFIGURATION CODES.

FIGURE G3.1
Configuration
Code = 2020



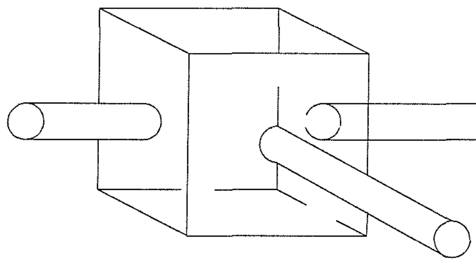
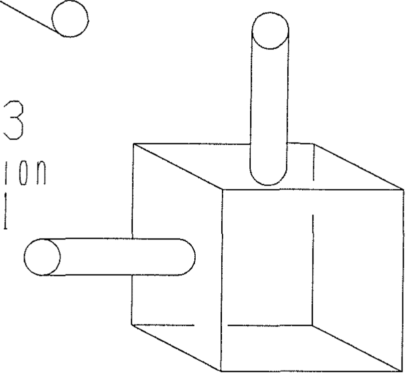


FIGURE G3.2
Cofiguration
Code = 3022

FIGURE G3.3
Configuration
Code = 2001



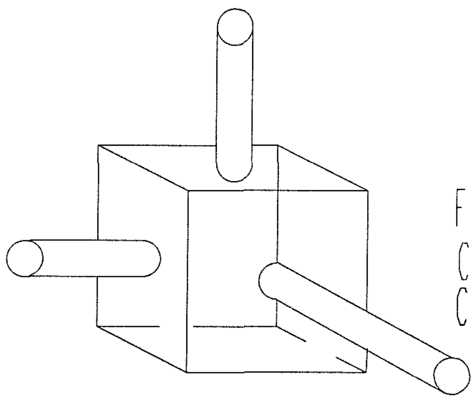


FIGURE G3.4
Configuration
Code = 3103

Program Listings.

The following program listing have been created in the Basic programming language and are used in the retrieval program.

****

LISTING 1.

```
B = 0

FOR I = 1 TO 2

        FOR J = 3 TO 4

                FOR K = 5 TO 6

                IF E(I) > 1 AND E(J) > 1 AND E(K) > 1
                                                        THEN B = B + 1

                NEXT K

        NEXT J

NEXT I
```

****

LISTING 2

```
C = 0

FOR I = 1 TO 5 STEP 2

IF E(I) .  1 AND E(1+1) > 1 THEN C = E(I) + E(I+1)

NEXT I
```

****

****

LISTING 3

```
D= 0
FOR I = 1 TO 4
IF I > 3 THEN K = 5 ELSE K = 3
        FOR J = K TO 6
        IF E(I) >=1 AND E(J) >=1 THEN D = D + 1
        NEXT J
NEXT I
```

****

APPENDIX H

SELECTION OF BELTS AND PULLEYS USING A RELATIONAL DATABASE.

For a computerized retrieval of belts and pulleys, data must be handled as two types of records. The belt and pulley records would not need to be stored in the same database. The 'Superfile 16' database management system allows an applications program to "open" as many databases as required. It also can handle a single database with a mixed record type. As long as the mixed records had common, linking fields, the relational qualities of Superfile could be exploited. In the case of toothed belts and pulleys, it would be the "pitch" that related a belt record to a pulley record.

The applications program, although performing simple retrieval tasks would have to process and manipulate a great deal of user information. The program would firstly have to derive any user inputs not given i.e. given a centre distance and a "gear" ratio, a possible belt length can be calculated. Once all the necessary information has been derived, a series of possible system solutions could be retrieved. These solutions could then be optimized and the best system presented to the user.

The research does allow the algorithms required to perform such an optimization to be defined. However, the additional programming required, particularly for the optimisation processes, is out of the scope of this project. It should also be noted that such work is being undertaken by the University of Bath who are specifically looking at the optimization of sub-systems.

Appendix I

Table 5.2

Table 5.3

Table 5.5

TABLE 5.2 : THE TECHNICAL INDEXES GEARBOX CLASSIFICATION SYSTEM.

STANDARD NON-MOTORIZED :

Spur :
- shaft mounting
- face flange mounting
- foot mounting
- universal mounting

Helical :
- shaft mounting
- face flange mounting
- foot mounting
- universal mounting

Bevel :
- shaft mounting
- face flange mounting
- foot mounting
- universal mounting

Worm :
- shaft mounting
- face flange mounting
- foot mounting
- universal mounting

Other tooth form
Unspecified tooth form
Combination

STANDARD MOTORIZED FHP :
STANDARD MOTORIZED ABOVE FHP :
SELECTIVE RATIO :
INFINITELY VARIABLE :
OPEN GEAR TRAIN ASSEMBLIES :

TABLE 5.3 : THE TECHNICAL INDEXES BELT AND CHAIN CLASSIFICATION SYSTEM.

BELTING :

- Circular
- Single V
- Multiple V
- Single wedge
- Multiple wedge
- Flat
- Toothed
- Linked

PULLEYS :

- Fixed ;
  - single groove
  - multi-groove
  - toothed
  - flat
- Variable;
  - pre-set
  - spring controlled
  - externally controlled

CHAIN :

- Single drive
- Multiple drive
- Conveyor

SPROCKETS :

TENSIONERS :

TABLE 5.5 pt 1 : COMPLETE PARAMETER LISTS FOR GEARBOX ANALOGIES.

GEARBOXES	TRANSFORMERS	BELTS/CHAINS
Driver Characteristics	Supply Characteristics	Driver Characteristics
Rated Input Power	Rated Primary Power	Rated Input Power
Rated Input Torque	Rated Primary Current	Rated Input Torque
Maximum Input Torque	Maximum Primary Current	Maximum Input Torque
Rated Input Speed	Rated Primary Voltage	Rated Input Speed
Abnormal Starting Loads	Maximum Spike Input	Abnormal Starting Loads
	Rated Frequency	
Gearbox Inertia	Primary O/S Inductance	Sub System Inertia
Gearbox Inertia	Mx Pr\ry L\kge Ind\, Sec O/S	Sub System Inertia
Gearbox Inertia / Friction	Imp\ Volts at Rated Current	Sub System Inertia
Gearbox Inertia	React\ Volts at R\d Current	Sub System Inertia
cont ...	cont ...	cont ...

TABLE 5.5 pt 2 : COMPLETE PARAMETER LISTS FOR GEARBOX ANALOGIES.

GEARBOXES	TRANSFORMERS	BELTS/CHAINS
Max Fric\ Torque at Input	Res\nce V\ts at Rated Cur\nt	Max Fric\ Torque at Input
Gearbox Inertia	S/C Impedance of Primary	System Inertia
Type of Driven Machine	Load Characteristics	Output Characteristics
Rated Output Power	Output Rating (KVA)	Rated Output Power
Maximum Output Torque	Maximum Output Current	Maximum Output Torque
Number Of Outputs	Output Tapping Details	Number of Outputs
Nominal Output Speed	Output Volts	Output Speed
Output Speed Range	Output Tapping Range	Output Speed Range
Speed Loss Across Gearbox	Voltage Regulation	Speed Loss Across System
Backlash Measured at Output		Belt Tension
Rated Power	Max Continuous VA	System Power
cont ...	cont ...	cont ...

TABLE 5.5 pt 3 : COMPLETE PARAMETER LISTS FOR GEARBOX ANALOGIES.

GEARBOXES	TRANSFORMERS	BELTS/CHAINS
Peak Power	Max Continuous VA	Maximum Power
Effective Power	VA x Service Factor	Rated Power x Serv\ Fact\
Efficiency	No Load + Full Load Losses	Output Power / Input Power
Nominal Gear Ratio	Turns Ratio	Sprocket Ratio
Type of Reduction	Step Up / Down	Type of Reduction
Fixed Ratio	Single Tapping	One Ratio
Selectible Ratio	Multi - Tapping	Multi - Ratio
Variable Ratio	Autotransformer	Infinitely Variable Ratio
Eppicyclic	"Multi - Loads"	Several Output Loads
Relative Shaft Rotation	Input/Output Phase Diff\	Relative Shaft Rotation
Maximum Intermittent Torque	Load Rejection	Maximum Overload Torque
cont ...	cont ...	cont ...

TABLE 5.5 pt 4 : COMPLETE PARAMETER LISTS FOR GEARBOX ANALOGIES.

GEARBOXES	TRANSFORMERS	BELTS/CHAINS
Maximum Speed Of Case		
Duty Cycle	Duty Cycle	Operating Time and Condit\s
Frequency of Reversing		Starts, Stops, Reversals.
Mechanical Service Factor	Service Factor	Service Factor
Speed Related Losses	Resistive Voltage Drop	Windage / Friction
Inertial Speed Loss	Reactive Voltage Drop	Backlash / Belt Tension
Speed Losses Across \box	Imp\nce V\ at Rated Current	Speed Losses Across System
Stiffness Through \Box	Max Prim\ to Sec\ Capac\nce	Stiffness Through \Box
	Dielectric Loss	
Power Loss Across \Box	Load Loss	Power Loss Across \Box
	No of Phases	
cont ...	cont ...	cont ...

TABLE 5.5 pt 5 : COMPLETE PARAMETER LISTS FOR GEARBOX ANALOGIES.

GEARBOXES	TRANSFORMERS	BELTS/CHAINS
	Number of Phases	
	Phase Displacement	
Total Number of Shafts	Number of I/O Tappings	Total Number of Shafts
Number of Output Shafts	Number of Output Tappings	Number of Output shafts
Mounting Type	Mounting Type	
Frame Size	Frame Size	
Weight	Weight	
Number of Gear Clusters		Number of Lay Shafts
Number of Planetary Trains		
Axial Thrust Capacity		Axial Thrust Capacity
Permissible Overhung Loads		Permissible Overhung Loads
cont ...	cont ...	cont ...

TABLE 5.5 pt 6 : COMPLETE PARAMETER LISTS FOR GEARBOX ANALOGIES.

GEARBOXES	TRANSFORMERS	BELTS/CHAINS
Environment	Environment	Environment
Thermal Rating	Max Temp Rise at Full Load	Thermal Service Factor
Max Thermal Dissipation	Allowable Temp Rise	" " "
	Actual Temp Rise	" " "
	Insulation Class	" " "
	Insulation Levels	" " "
Mass of Recirculating Oil	Mass Of Cooling Oil	
Bearing Type		Design Bearings
Gearbox Life	Transformer Life	System Life
Dimensions	Dimensions	Design Dimensions
Maximum Backlash	Hysteresis Effects	Belt Tension
cont ...	cont ...	cont ...

TABLE 5.5 pt 7 : COMPLETE PARAMETER LISTS FOR GEARBOX ANALOGIES.

GEARBOXES	TRANSFORMERS	BELTS/CHAINS
Gear Tolerances		
Shaft Radial PLay		Design Choice
Shaft End Play		Design Choice
Shaft Dimensions		Design Choice

KEY :

BASIC CHARACTERISTIC :	POWER
ANALOGOUS CHARACTERISTIC :	INERTIA
ANALOGY UNKNOWN :	

Appendix J

Figure 6.1

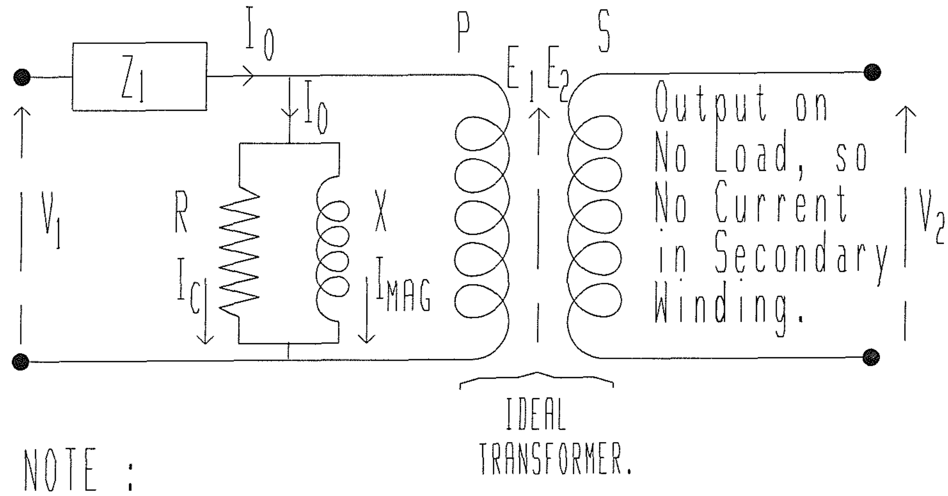
Figure 6.2

Figure 6.3

Figure 6.4

Figure 6.5

FIGURE 6.1 : EQUIVALENT CIRCUIT AND PHASOR DIAGRAM FOR A TRANSFORMER ON NO LOAD.



NOTE :

Z_1 is the primary coil impedance.
I_0 is small, so $I_0 Z_1$ is also small and therefore negligible.

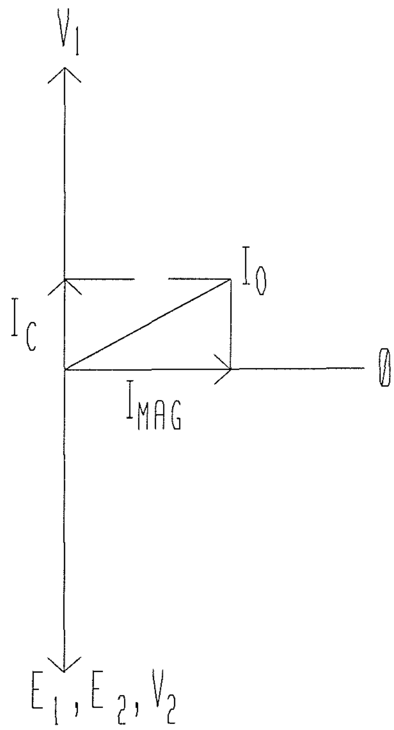


FIGURE 6.2: EQUVALENT CIRCUIT AND PHASOR DIAGRAM FOR A TRANSFORMER ON FULL LOAD.

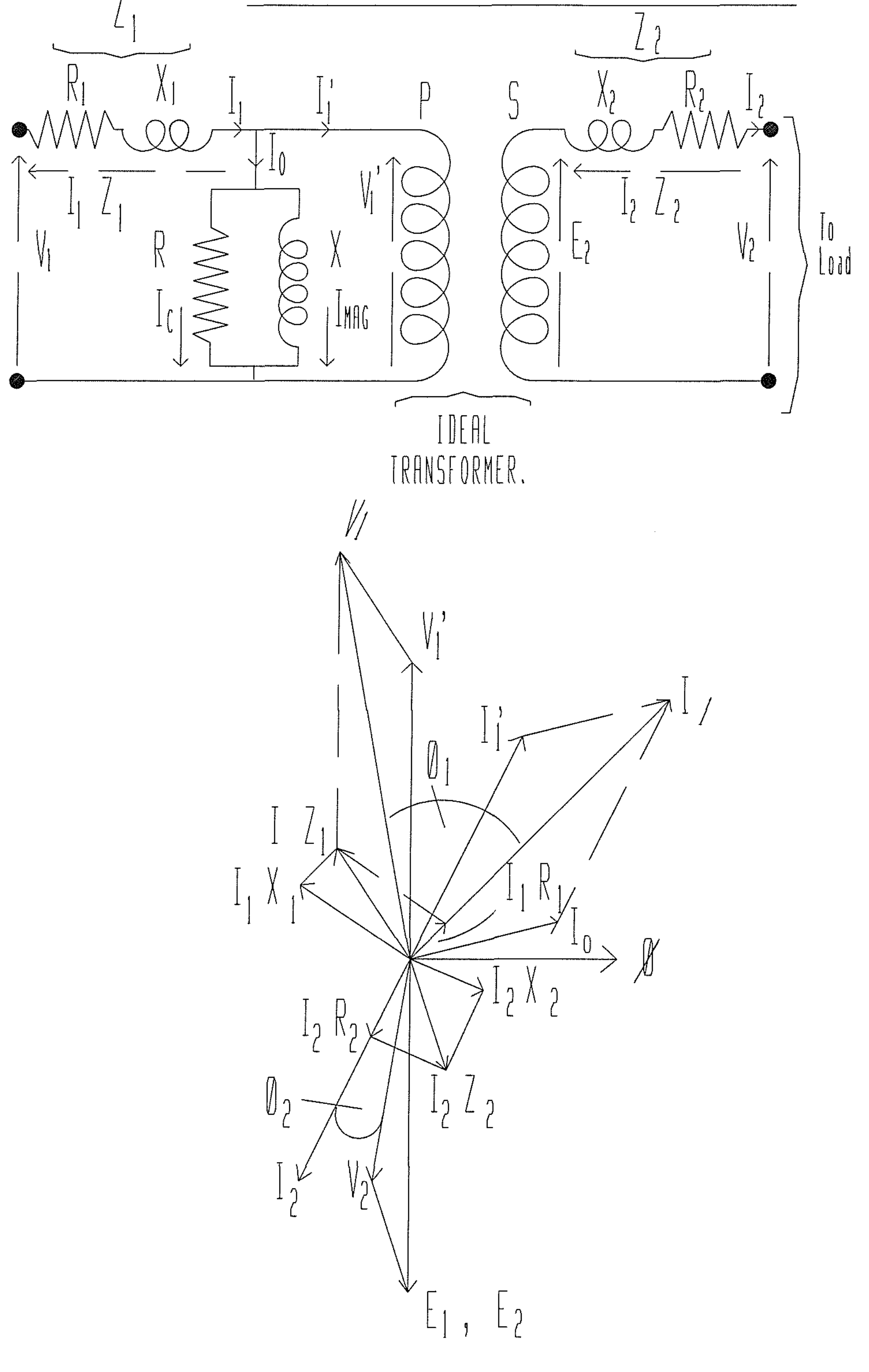
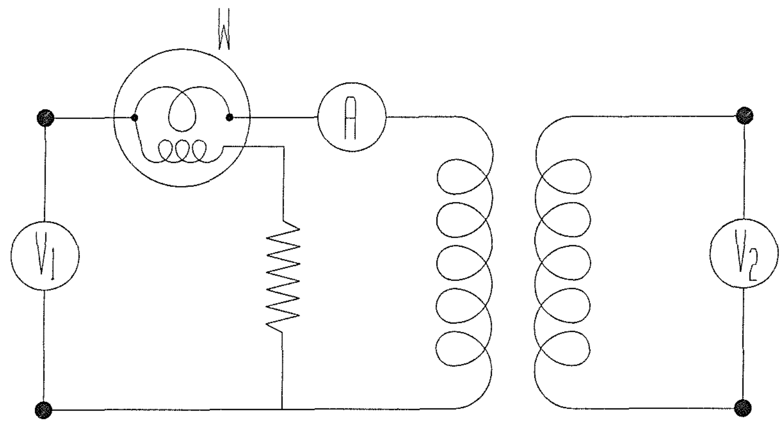
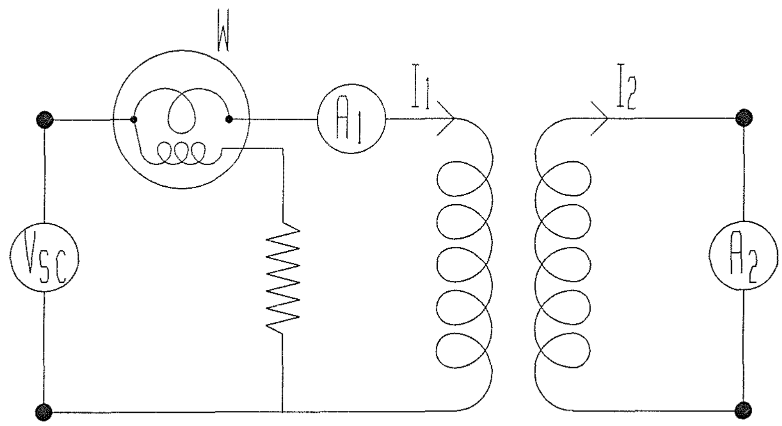


FIGURE 6.3 : THE OPEN CIRCUIT TEST FOR A TRANSFORMER.



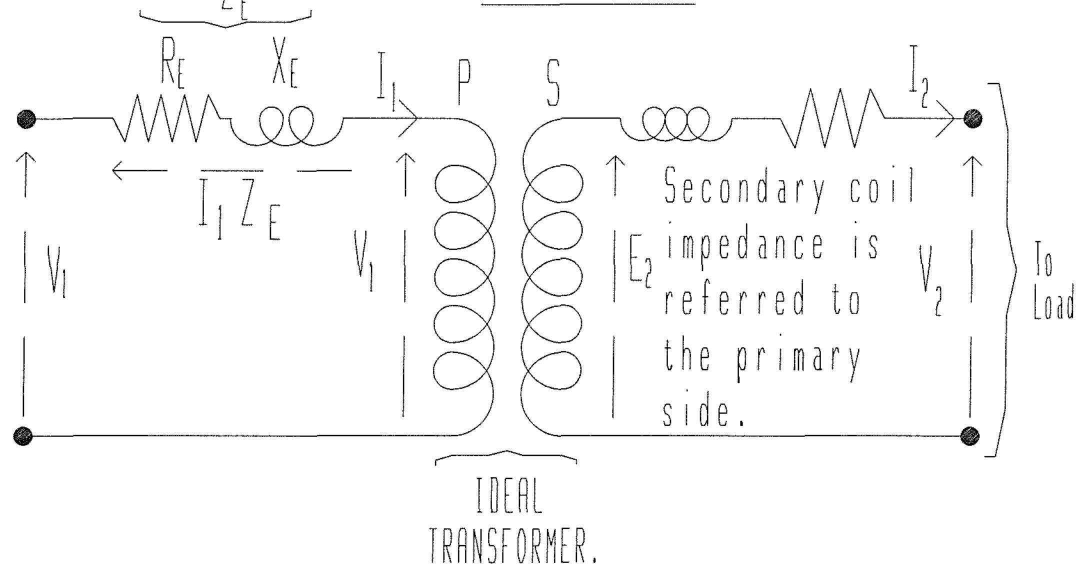
The Wattmeter gives the CORE LOSSES in the transformer. The ratio V1/V2 gives the turns ratio.

FIGURE 6.4 : THE SHORT CIRCUIT TEST FOR A TRANSFORMER.



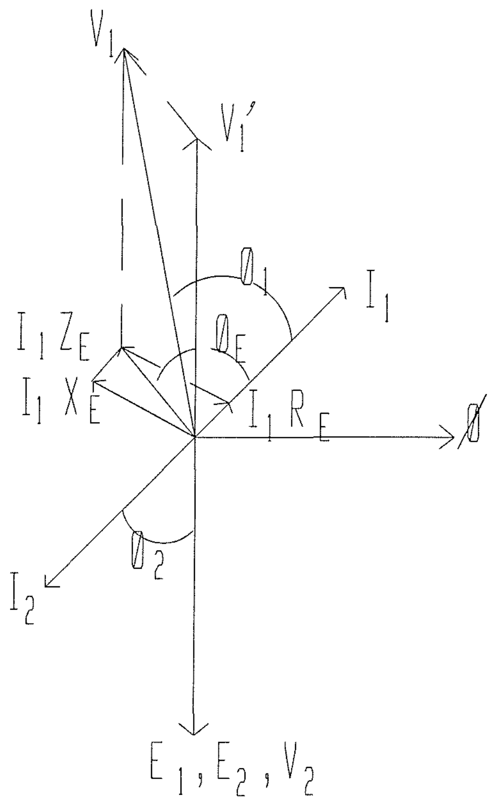
The Wattmeter gives the COPPER LOSSES in the windings.

FIGURE 6.5 : FULLY REFERRED EQUIVALENT CIRCUIT AND PHASOR DIAGRAM FOR A TRANSFORMER ON FULL LOAD.



$$Re = R1 + R2(V1/V2)^2, \quad Xe = X1 + X2(V1/V2)^2$$

$$Ze = (Re^2 + Xe^2)^{1/2}; \; Re = Ze \cos Øe \; ; \; Xe = Ze \sin Øe$$



Appendix K

Table 7.1

Table 7.2

Table 7.3

TABLE 7.1 : THE TECHNICAL INDEXES LTD. CATAGORIZATION OF COUPLINGS.

RIGID :

- Spline , Jaw
- Sleeve
- Flange

FLEXIBLE :

- Sleeve
- Spacer
- Oldham slider
- Gear
- Metal grid
- Chain
- Spring
- Bellows
- Cone
- Bush & pin
- Pin
- Spider
- Rubber block,bush,segment
- Flexible ring,element,disc
- Laminated metal disc,plate
- Universal joints

SPECIAL :

- Clutch
- Anti - backlash
- Hydraulic, Fluid
- Other types

TABLE 7.2 : THE SELECTION PROCEDURE FOR COUPLINGS AS SUGGESTED BY "SEED".

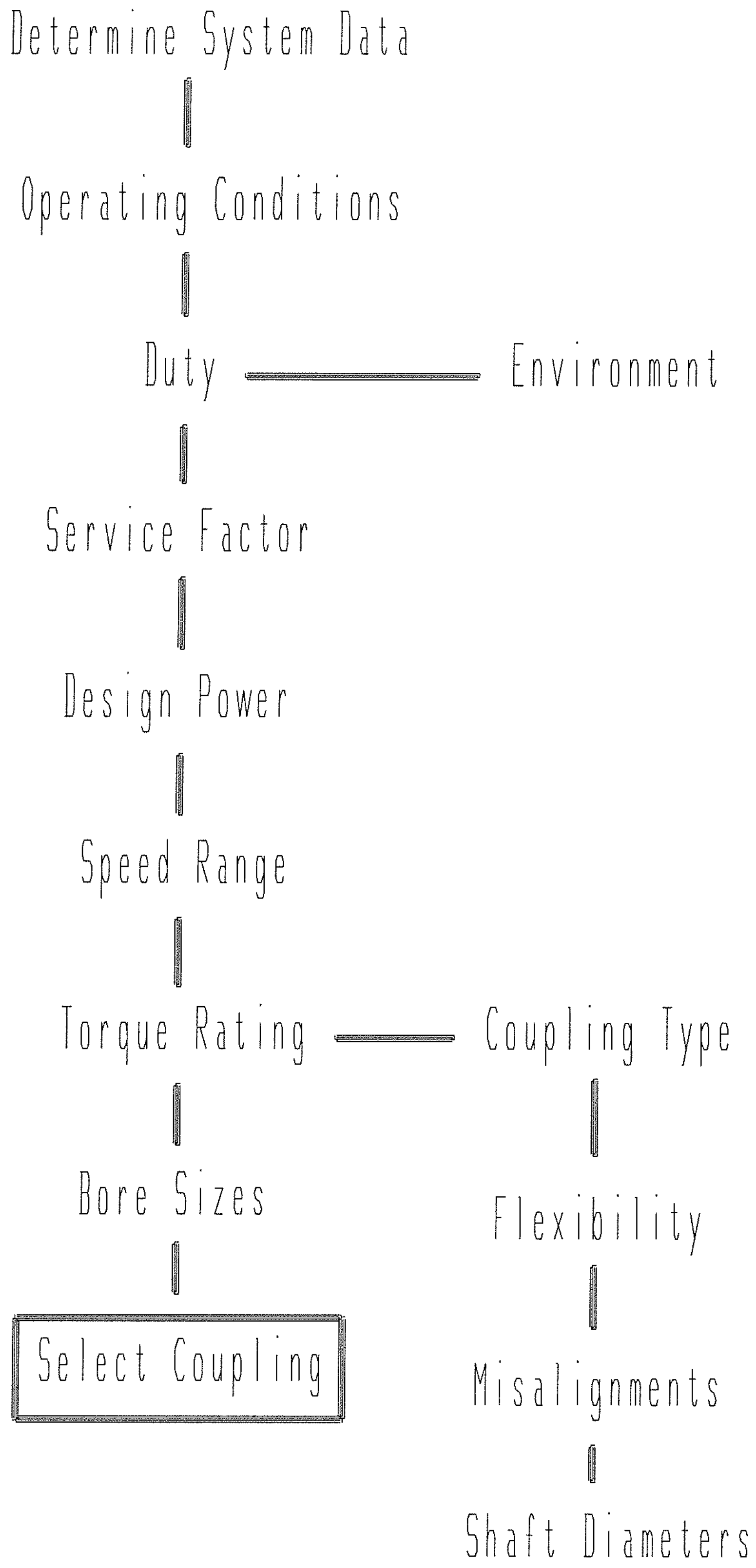


TABLE 7.3 pt 1 : THE EXTENSIVE CHARACTERISTIC LISTS FOR COUPLINGS.

TORQUE BASED CHARACTERISTICS :

Rated (Nominal) Torque
Maximum (Peak / Limit) Torque
Alternating Torque
Torque at 100 rpm
Dynamic Torque
Static Torque

SPEED BASED CHARACTERISTICS :

Rated Speed
Maximum Speed
Minimum Speed
Output Speed
Input Speed
Velocity Ratio
Relative Speed
Backlash
Moment of Inertia
Speed verses Power Table

TABLE 7.3 pt 2 : THE EXTENSIVE CHARACTERISTIC LISTS FOR COUPLINGS.

TORSIONAL CHARACTERISTICS :

Torsional Deflrction

Static Windup

Torsion angle at rated torque

Dynamic Torsional Stiffness

Radial Torsional Stiffness (Lateral Spring Rate)

Axial Torsional Stiffness (Axial Spring Rate)

Resonance Factor (Dynamic Magnifier at Resonance)

Damping Coefficient

Maximum Attenuating Torque

Freqency of Vibration

POWER BASED CHARACTERISTICS :

Nominal (Rated) Power

Design Power

Maximum Kw at 100 rpm

Rating in Kw per rpm

Speed verses Power Table

TABLE 7.3 pt 3 : THE EXTENSIVE CHARACTERISTIC LISTS FOR COUPLINGS.

MISALIGNMENTS :

- Angular Misalignment
- Axial (Parallel) Misalignment
- End Float
- Torsional Flexibility

DUTY :

- Running Hours per Day
- Service Life
- Shock Loading
- Shock Factor
- Electrically Insulating
- Lubrication
- Vibration Absorbing

ENVIRONMENT :

- Temperature
- Pollution

TABLE 7.3 pt 4 : THE EXTENSIVE CHARACTERISTIC LISTS FOR COUPLINGS.

DIMENSIONS :

Diameter of Shafts to be Coupled
Nominal (Rough) Bore
Minimum Bore
Maximum Bore
Weight
Dimensioned Drawing of Coupling

Appendix L

Table 8.1

Figure 8.2

Figure 8.4

Figure 8.5

Figure 8.6

Figure 8.7

TABLE 8.1 : THE COMPLETE ORDERED CHARACTERISTIC LISTS FOR CLUTCHES et al

PART 1.

" CLUTCHES ".	FLUID COUPLINGS.	ROTARY DAMPERS.
Power Ouput	Kw -> Input Speed Table	
Power of Driver	Maximum Power	
Permisible Power Loss	Max Power Absorption	
Heat Dissipation	Starting Power Capacity	Power Dissipation
	Rated Power	
Friction Work		
Transmitted Torque		
Moment of Inertia	Moment of Inertia	
cont ...	cont ...	cont ...

TABLE 8.1 : THE COMPLETE ORDERED CHARACTERISTIC LISTS FOR CLUTCHES et al

PART 2.

" CLUTCHES ".	FLUID COUPLINGS.	ROTARY DAMPERS.
Engagements		
Usage Per Day		
Torque at Engagement		
Speed Range	Speed Rgulation	
Maximum Speed		
	Maximum Input Speed	Maximum Input Speed
	Input Speed at Full Load	
	Output Speed at Full L\d	
cont ...	cont ...	cont ...

TABLE 8.1 : THE COMPLETE ORDERED CHARACTERISTIC LISTS FOR CLUTCHES et al

PART 3.

" CLUTCHES ".	FLUID COUPLINGS.	ROTARY DAMPERS.
Ambient Temperature		
Lubrication		
Relative Damping		Damping Rate
Axial Stiffness		
Radial Stiffness		
Torque Range		Torque Range
Torque Variation		
Torque Frequency		
cont ...	cont ...	cont ...

TABLE 8.1 : THE COMPLETE ORDERED CHARACTERISTIC LISTS FOR CLUTCHES et al

PART 4.

" CLUTCHES ".	FLUID COUPLINGS.	ROTARY DAMPERS.
Dyn\c Torsional Stiffness		
Starting Torque	Starting Torque	
Misalignments		
Slip with Constant Torq\		
Slip with Reduced Torq\		
Slipping Power	Max Time at 100% Slip	Max Slipping Power
Relative Speed of Shafts		
Maximum Torque		Maximum Torque
cont ...	cont ...	cont ...

TABLE 8.1 : THE COMPLETE ORDERED CHARACTERISTIC LISTS FOR CLUTCHES et al

PART 5.

" CLUTCHES ".	FLUID COUPLINGS.	ROTARY DAMPERS.
Standard Torque		
Dynamic Torque		
Static Torque		
Max Reverse Torque		
Drag Torque		
Method of Actuation	Method of Actuation	
Run Up Time		
Max Acceleration Time	Time to Accelerate	
cont ...	cont ...	cont ...

TABLE 8.1 : THE COMPLETE ORDERED CHARACTERISTIC LISTS FOR CLUTCHES et al

PART 6.

" CLUTCHES ".	FLUID COUPLINGS.	ROTARY DAMPERS.
	Starting Time	
Transmissible Torque		
Switchable Torque		
No Load Torque		
Slipping Load Torque		
Max Rated Br\kaway Torq		
Torque Before Slipping		
Torque at Overrun		
cont ...	cont ...	cont ...

TABLE 8.1 : THE COMPLETE ORDERED CHARACTERISTIC LISTS FOR CLUTCHES et al

PART 7.

" CLUTCHES ".	FLUID COUPLINGS.	ROTARY DAMPERS.
Speed at Overrun		
Engagement Speed		
Torsion Angle at Max Torq		
Energy per Engagement		
Actuation Force		
Reversibility		
Lubrication		
Weight		
cont ...	cont ...	cont ...

TABLE 8.1 : THE COMPLETE ORDERED CHARACTERISTIC LISTS FOR CLUTCHES et al

PART 8.

" CLUTCHES ".	FLUID COUPLINGS.	ROTARY DAMPERS.
	Flow Charge (Vol) of Oil	
		Viscosity of Filling
Friction Work per Hour		
Friction Coefficient		
Nominal Current		
Maximum Current		
Maximum Volts		
Resistance		
cont ...	cont ...	cont ...

TABLE 8.1 : THE COMPLETE ORDERED CHARACTERISTIC LISTS FOR CLUTCHES et al

PART 9.

" CLUTCHES ".	FLUID COUPLINGS.	ROTARY DAMPERS.
Fluid Pressure Dimensions Angle of Index	Dimensions	Dimensions Angle of Index

FIGURE 8.2: TYPES OF CLUTCH.

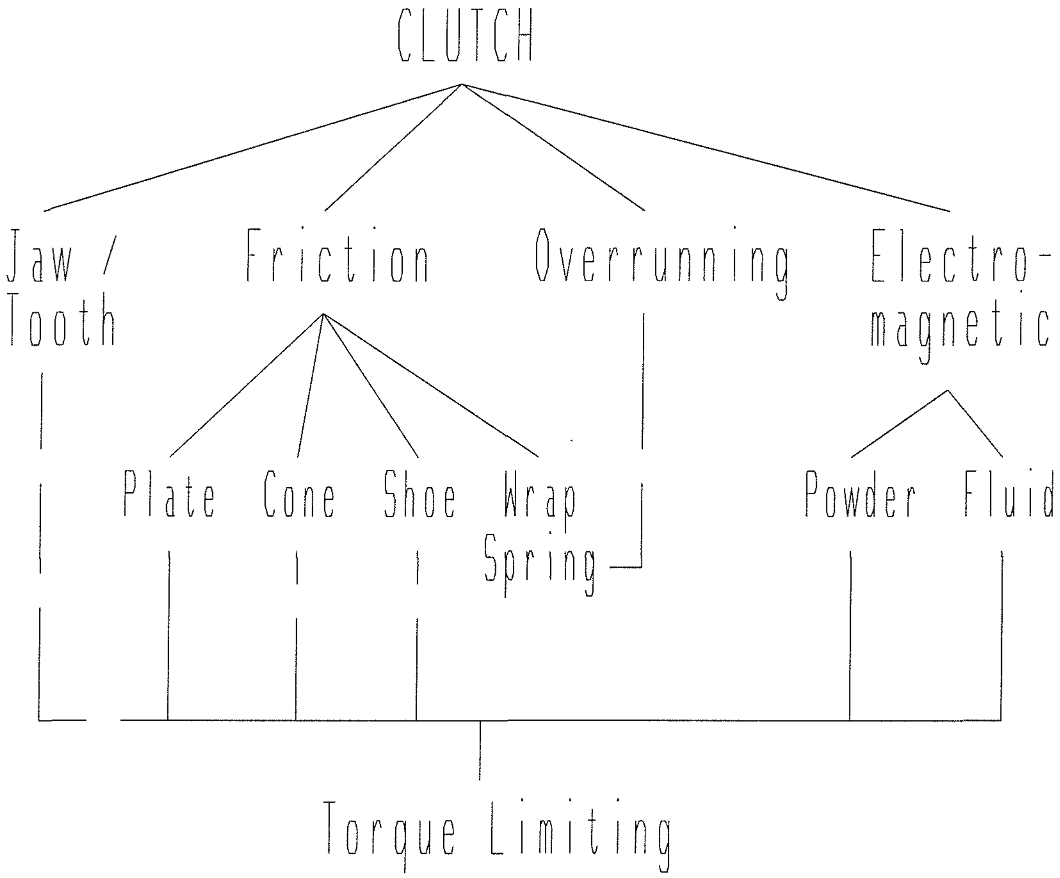


FIGURE 8.4: COMMON METHODS OF CLUTCH ACTUATION

CLUTCH IS ENGAGED MECHANICALLY BY:

1) Mechanical input

2) Electromagnetic input

3) Hydraulic input

4) Pneumatic input

CLUTCH IS ENGAGED AUTOMATICALLY BY:

5) Input shaft speed

8) Relative input/output speed

THE CLUTCH IS PERMANENTLY ENCAGED:

7) "Normally engaged"

8) Engaged up to torque limit

FIGURE 8.5: COMMON METHODS OF RESETTING A CLUTCH

CLUTCH IS RESET MECHANICALLY BY:

1) Mechanical input

2) Electromagnetic input

3) Hydraulic input

4) Pneumatic input

CLUTCH IS RESET AUTOMATICALLY BY:

5) Relative input/output speed

8) Torque falling below limit

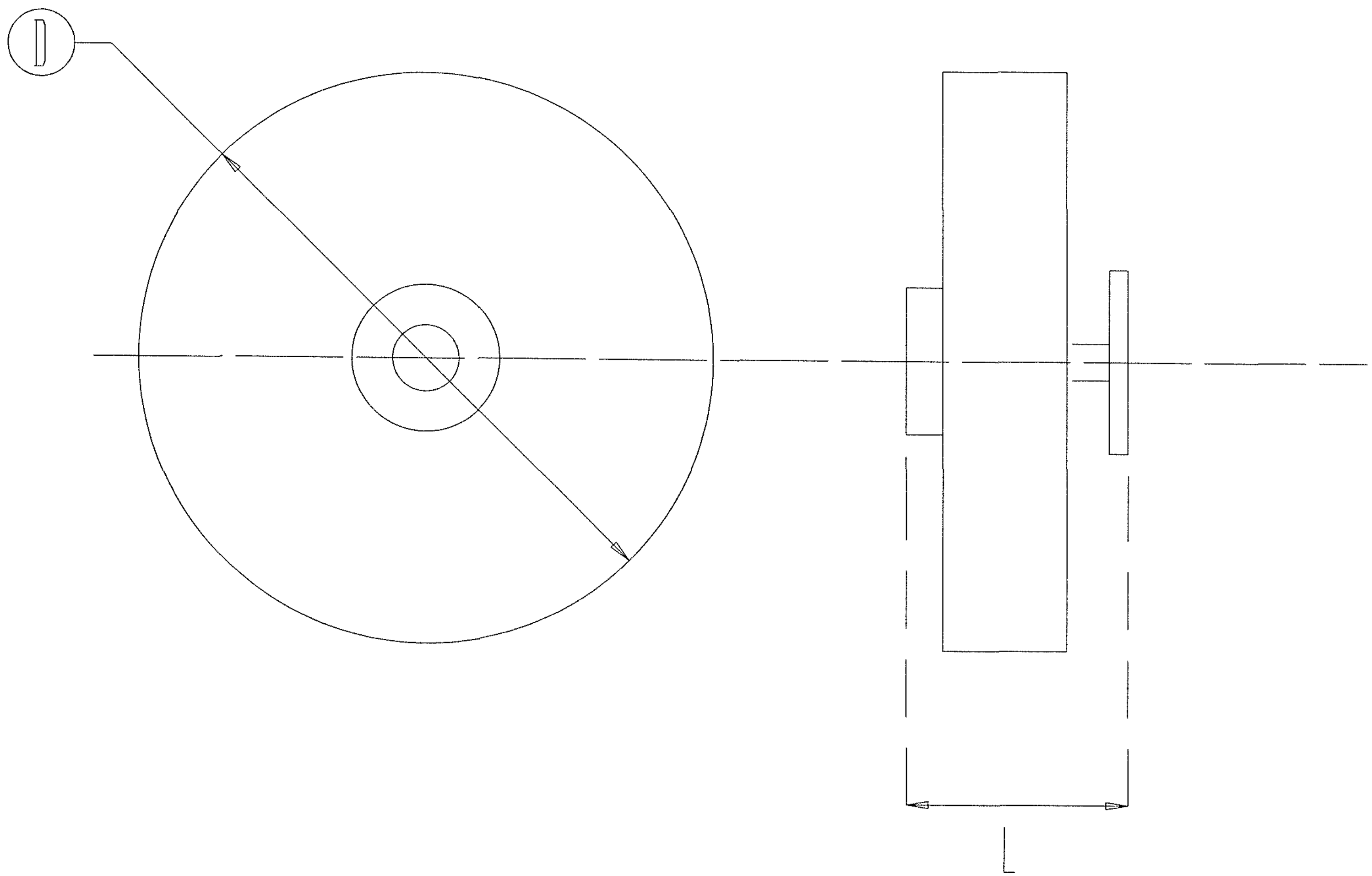
THE CLUTCH IS PERMANENTLY DISENGAGED:

7) "Normally disengaged"

FIGURE 8.6 : COMMON TYPES OF CLUTCH.

1) JAW (Tooth)
2) PLATE
3) CONE
4) CENTRIFUGAL
5) WRAP SPRING
6) OVERRUNNING (Backstopping /Indexing)
7) TORQUE LIMITING
8) FLUID COUPLING
9) ELECTROMAGNETIC POWDER/FLUID
10) EDDY CURRENT / HYSTERESIS

FIGURE 8.7 : DIAMETER AND LENGTH FOR CLUTCHES et al.



Appendix M

Estimation of Data Capture Times and Data Volumes.

Data Capture - the extraction of the relevant selection parameters from manufacturer's catalogues.

Technical Indexes Ltd (T.I.) estimated that it would take 5-7 man-years to index, check and enter data for 50,000 motors. This estimate can be confirmed by some rough calculations performed below:

GIVEN: 2 hours to decipher a catalogue i.e. find data, perform unit changes, derive expressions etc.

There are 167 motor suppliers, so time to decipher catalogues:

2 x 167 = **334** Man hours (1)

An average catalogue contains 300 units. The time taken to extract data, once the catalogue is deciphered, should, on average be around 10 minutes. Thus, the time to extract all the motor data would be:

(300 x 10 x 167)/60 = **8350** Man hours (2)

Total Man Hours = (1) + (2) = **8684** Man Hours.

1 man year = ((52x5) - holidays) x 7 hours. (Holidays = 30 days.)

1 man year = 1610 man hours.

Thus, man years to index database for motors would be: 8350 / 1610 = 5.2 man years.

Given that the extracted data must be checked and entered, the T.I. estimation of 5-7 man years appears accurate.

ESTIMATION OF THE DATA VOLUMES - How Big will the Databases Be?

In order to provide a comprehensive specification for the software, it was important to estimate the average size of the component databases in case the Database Management System chosen had any limits on the size of database it could handle. The estimation given below was performed early in the project using the motor database. It could be performed with a little more confidence once the record size had been decided. It was felt that the size of a motor record would be slightly above average. This estimate was justified by the sizes of the gearbox and coupling records, the gearbox was one parameter bigger, the coupling five parameters smaller.

Once a small scale database was set up at Southampton using Superfile, record sizes could be extrapolated to give an accurate prediction of data volumes.

(1) **Record Size**

The maximum record size will be 20 items (fields).

The data types will be a mixture of alphanumeric and numeric characters. There will be 10 - 15 numeric fields and 5 - 8 alpha fields per record. The average field size will be around 10 bytes. Thus, the estimated record size would be 200 bytes.

NOTE: Superfile handles all data as alphanumeric characters.
AND: From the trial database, 957 records used 112384 bytes. This is around 120 bytes per record. Thus, for calculation, average record size = 120 bytes.

(2) **Number of Suppliers**

The number of suppliers per component group was put at 179 by T.I.

(3) **Average Number of Components per Supplier**

This was based on a sample of 30 motor suppliers from the electrical, hydraulic and pneumatic fields, the average was around 200 motors per supplier.

(4) Average Data File Size

Average data-file size = Record size x Records per supplier x Number of suppliers

= 120 x 200 x 179 = 4.3 Mega bytes.

If company data is included in the database, 179 extra records must be included (one for each supplier). The size of these records containing name, address, telephone number etc., will be of the same order as the motor records. This is supported by the record size of a demonstration database supplied with the Superfile system. This is in fact a name and address type database and has a record size of about 150 bytes. Thus the company data would be an additional (179 * 150) bytes = .03 Mbytes. This is a very small increase in database size.

(5) Average Size of Full Database

A full Superfile database consists of a data-file and two index files. These index files are of similar size to the data file. So, the full database = 4.3 * 3 = 12.9 Mbytes.

Thus, it would be safe to say that the full data volume for a component database will be under 15 Mega bytes.

APPENDIX N

Part 1:

THE QUESTIONNAIRE USED TO IDENTIFY POTENTIAL END USERS OF ROCCI

COMPUTER-BASED SYSTEM FOR THE SELECTION OF POWER TRANSMISSION COMPONENTS

Please complete and return in the envelope provided to Marketing Department, Technical Indexes Limited, Bracknell.

1. Are you or any of your colleagues who use the "ti" microfile system actively involved in the selection of power transmission components? : Yes No

If "**Yes**" go to Question **2.**
If "**No**" please complete only Question 7.

2. What power range do your selections normally fall into:

a. 0 to 1 hp
b. 1 to 10 hp
c. over 10 hp

3. How frequently are these components selected:
(Average No. of selections per week)

AC Motors	DC Motors
Hydraulic Motors	Pneumatic Motors
Gearboxes	Couplings

4. Which suppliers do you use most frequently for the supply of these power transmission components :

Component	**Suppliers**
Electric Motors	1...
	2...
	3...
Hydraulic Motors	1...
	2...
	3...
Pneumatic Motors	1...
	2...
	3...
Gearboxes	1...
	2...
	3...
Couplings	1...
	2...
	3...

5. If a computer-based system was available to assist in the selection of power transmission components as an adjunct to your microfile system would you be interested in testing such a system? :

Yes No

If "**Yes**", please answer Questions **6** and 7.

If "**No**", please only answer Question 7.

6. Do you have an IBM compatible PC with a hard disk storage system in your department?

Yes.. No

(please give model and disk size)

If **Yes** : **(a)** does it have graphics facilities?

Yes No

(b) does it have a colour monitor?

Yes No

7. Company

Name of person completing this questionnaire :

....

Company address :

....

....

....

....

....

Telephone Number : ....

Date :...

APPENDIX N

Part 2 :

FIELD TRIAL GUIDE NOTES FOR ROCCI DEMONSTRATIONS

Name of Researcher :

....

....

Date : ....

Organisation :

....

....

Division :

....

....

Address :

....

....

....

....

Tel. No. : ....

Name(s) : **Job Title**(s) :

of person(s) interviewed :

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....	....
....	....

Before Demonstrating System :

1. Get respondent(s) to list on separate sheet, the key parameters, characteristics they would use in selecting the various power transmission components.

2. Is there a specialist or specialist department in your organisation who is responsible for the selection of power transmission components?

Yes No

If **Yes**, please specify :

....

....

....

3. Which of these statements matches the way in which you usually select the power transmission components specified? (you may tick more than one):

"When selecting any of these components, we usually start with a short list of suppliers we have used before."

"When selecting motors, we have usually already decided that it will be either an electric, hydraulic or pneumatic motor."

"When selecting gearboxes, we have usually already decided the type of gearbox spur, helical, epycyclic etc.)"

"When selecting any of these components, physical characteristics (size, fixing, weight) are usually the most important constraints."

If you have ticked none of the above, does this statement more closely match the way in which you usually select this type of component?

"When selecting any of these components, we usually have an open mind about supplier, type or physical characteristics and are concerned only about the performance specification."

Now show system and ask these questions :

4. How useful did you find the help screens? :

(show examples - get comments)

....
....

....
....

....
....

....
....

5. (Show lists of parameters) Please tick those parameters that you would expect to use in the selection of these components.

6. (Show list of parameters with explanation for inclusion) Please tick any parameters, other than those you ticked on the first list, that you would now use.

7. In terms of the efficiency of selection of these components, what improvements would the database offer you?

(Prompt for greater control, easier selection, wider selection, selections that more accurately fit requirements).

....
....

....
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8. Which of the following statements matches your opinion of the database. (You may tick more than one).

"It appears to be much better and more helpful than a directory covering similar components."

"It would make selecting these components easier than using suppliers catalogues."

"It appears that, if complete, it would be able to do all that the 'ti' microfile does and more."

If you tick this, please expand.

....
....

....
....

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....

9. What price (assuming a rental per annum) would you expect to have to pay for such a database on, say, motors (assuming that it covered all motors from all suppliers)? :

(Try not to prompt).

£ ....

10. Was there any one aspect of the database, as shown to you, which you considered most unsatisfactory? (Excluding the completeness of its content in terms of components and suppliers).

....

....

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POWER TRANSMISSION COMPONENT SELECTION.

Please list below the parameters/characteristics that you would most commonly use in selecting the power transmission components shown: (if you can place them in order of importance, please do so).

Example: Motors:

1. Preferred manufacturer
2. Motive power (electric, hydraulic, pneumatic)
3. a.c. / d.c.

Q1. Motors

1.
2.
3.
4.
5.
6.
7.
8.

Q2. Gearboxes

1.
2.
3.
4.
5.
6.
7.
8.

Q3. Couplings

1.
2.
3.
4.
5.
6.
7.
8.

Appendix O

LIST OF IMPROVEMENTS TO THE ROCCI SYSTEM.

9.6.1. Error Checks

i) Check the constraint for the correct data type i.e. Alphanumeric or numeric.

ii) If "/AND/" is used, detect and check it is correct and valid for that parameter and search type.

iii) If "/AND/" is used, check that the second search type and constraint are valid for the parameter.

9.6.2 Format Changes

i) Change the option "EXIT" in the components, search and view menus to "EXIT to DOS". That makes it clear that this option would cause the user to leave ROCCI.

ii) In the retrieval menu, do not use the "escape" key for help, use the "F1" key, this is more conventional. "Escape" normally takes the user back to a higher menu.

iii) Make the summary of search types dynamic. The types of search will vary as the prompts change from alpha to numeric data. If the search types available list changed with the prompt the user would be helped greatly by always picking the correct search type. For instance the "@" search is irrelevant for a numeric search criteria. If used, the search will fail. This also relates to the improvement of the points 9.6.1 ii) and iii) because if the search type is also made data sensitive, this error would not occur.

iv) If the return key is pressed at the beginning of a previously entered search criteria, the information is rubbed out. This currently done by overprinting with a fixed length of blank spaces. It would be better to overprint with a string of blanks as long as the search criteria. This would avoid the inadvertent erasing of data from the instructions and search type summary.

v) The format for defining ranges, although successful, has been queried. The format "=[x-y]" has been suggested as a possible replacement. Any format is possible because it must all be "post processed" into a form acceptable by the DBM. Thus, this format will be the result of user preference.

vi) The option to use imperial units should be available. The database are set up by using SI units. The user could chose a unit option on entering ROCCI. The retrieval menu could use these units but when a search is performed the units can be changed to SI and the database searched. Retrieved records could be converted to the user preferred units when displayed.

9.6.3 **Computational Changes**

i) Currently, when matching records are counted, they are not stored. Thus, when the user selects the view option, the database is searched again and the records displayed as they are retrieved. However, the retrieved records are still not saved. There is no hard copy option either. The problem associated with saving the retrieved data is basically, how many records will there be to be stored? A sensible user will probably attempt to refine the search to under 15 possibilities. This process will be done through the count menu using the abort feature when the count becomes excessive. It would be inappropriate to attempt to save the retrieved records at the count stage for the following reasons. There would have to be some limit as to the number of records stored because of the available working memory of the machine. If the records were sent to a disk file, this would have a size limit but would also slow down the count considerably. If a limit was set through the software, when reached, the count would have to be interrupted in order to inform the user of this problem and present the possible options. At the moment, the count simply records a match, nothing is done with the retrieved data. This makes the counting process as fast as possible. This is an important fact for a user attempting to refine a search. Processing data during the count will slow it down. Although the rate of retrieval will always be better that a manual search, if it is done by a computer, the user expects "instantaneous" results. If data was captured at the count stage and the number of hits deemed to be too large by the user. The search criteria could be tightened and the search repeated on the already filtered data. This would allow rapid optimisation towards a few very good matches.

From a practical point of view, the best time to save the retrieved data is at the view stage. Since the user will spend time looking at the records on the screen, the software could be busy saving it. Again the problem of data storage occurs but if coupled with a hard copy option, the user will have the chance to save the records.

ii) Many of the help screens contain list of manufactures, component types etc. These lists are part of the help screen software and since this has been compiled the entries cannot be changed. If a user sets up a database of preferred suppliers, this company names list must be edited. The current system prohibits this. The best solution to this problem would be to store any lists in ASCII files which could be read by the help screen file when run. These files could easily be edited by the user using a word processing package.

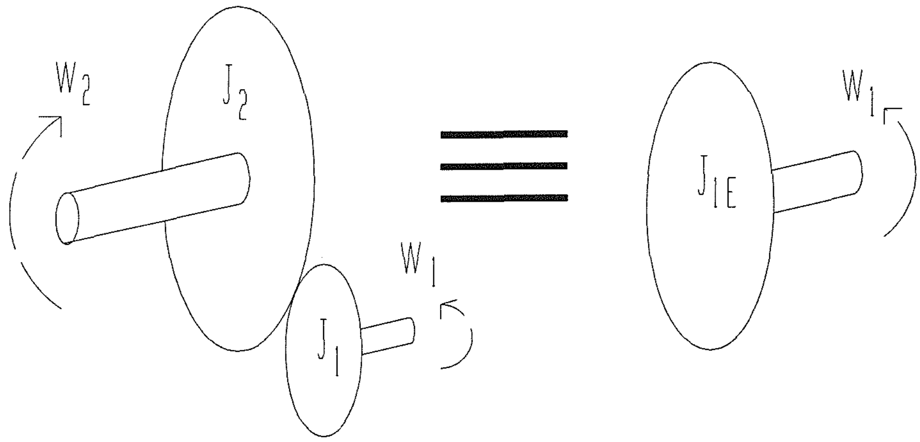
iii) The shaft configuration code help screen is designed to generate the shaft configuration code by prompting the user to answer a series of simple question. Once the help screen is selected, the user is forced to complete the various prompts. This is of course necessary to generate the code, however, the option to abandon the help screen should be offered.

9.6.4 System Changes

i) Use the relational database facilities to link CAD compatible drawings of components to a CAD package. This would allow General Assembly drawings of systems to be created.

APPENDIX P

REFERRED INERTIA.



REQUIRED :

The inertia "seen" at the primary caused by the output and secondary inertias.
The total stored energy (E) is constant.

Thus : $E = 1/2 \, (J_1 w_1^2 + J_2 w_2^2)$

Know : $w_1 / w_2 = n$ therefore : $w_2^2 = w_1^2 \, n^2$

$$E = 1/2 \, (J_1 w_1^2 + J_2 \, w_1^2 n^2)$$

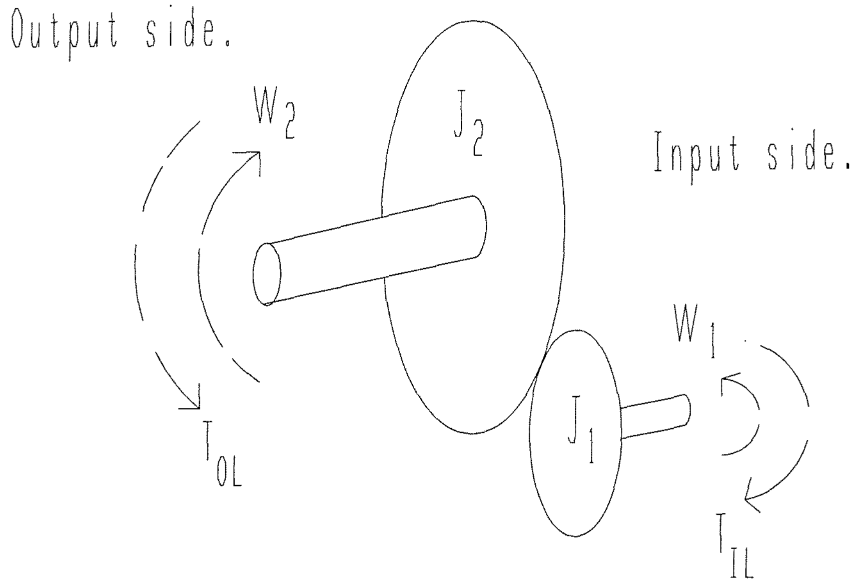
$$= 1/2 \, w_1^2 \, (J_1 + n^2 J_2)$$

Since we know the primary inertia is J_1 , the secondary inertia referred to the primary speed must be : $n^2 J_2$.

The inertia referred to primary = $J_{1E} = (J_1 + n^2 J_2)$

APPENDIX Q

REFERRED TORQUE.



NOTE : T_{IL} = torque loss on input side.
T_{OL} = torque loss on output side.

GIVEN : $T_1 w_1 = T_2 w_2$ so : $w_1 / w_2 = T_2 / T_1 = n$

therefore : $T_2 = n\, T_1$ .... (1)

LET : Tx be the additional torque supplied at the input to overcome the output lost torque.

From (1) , we know : $Tx * n = T_{OL}$
therefore : $Tx = T_{OL} / n$

Thus total loss referred to input $= T_{IL} + Tx$
$= T_{IL} + T_{OL} / n$