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RETRIEVAL OF COMPATIBLE COMPONENT INFORMATION FOR SYSTEM DESIGN

Volume 1 of 2

by

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ABSTRACT

FACULTY OF ENGINEERING AND APPLIED SCIENCE

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Master of Philosophy

Retrieval of Compatible Component Information for System Design.

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Selection of individual engineering components is a time consuming and labour intensive task. Rapid access to a range of possible components is difficult as manufacturer's catalogues must be obtained and interrogated to find a suitable item. Making comparisons between components from different suppliers is difficult because the components tend to be defined in different ways. These problems are compounded as individual components are assembled into systems and the matching process is carried out.

Thus, the selection process could be made more efficient by being able to rapidly access component data. If the selection and retrieval process was structured so that data defining a particular type of component was presented in a useful and consistent way, comparisons could be made between different products. If this idea is applied not only within component groups but also between component groups, the compatibility or matching process of components would also benefit.

The focus of the research was to identify sets of characteristics that describe rotary mechanical power transmission components in a way that was familiar to designers and also suitable for a computer aided retrieval. As well as carefully studying the components and selection procedures, the novel approach of considering analogous components was used. This caused the number of potential retrieval characteristics to be increased thereby making the research very thorough. The characteristics highlighted by analogy often proved to be important for selection. By careful consideration of component function, the exhaustive lists of selection characteristics were reduced to concise lists of user friendly, discrete criteria suitable for a computer based retrieval system. Analogies helped to determine the relative importance of each characteristic throughout this process.

The concise characteristic lists were then used as the basis of the record format for a commercial database management system. A bespoke user interface with the novel feature of 'picture prompts' was written. The demonstration system, christened 'R.O.C.C.I.' was given field trials. The results of the field trials were favourable as users were able to quickly select components that met their needs.

However financing the data capture has not allowed the project to be further developed.

Contents

PREFACE

The work described in this thesis - the development of a computer aided selection system for rotary mechanical components, relates to work carried out by the author between September 1984 and August 1988. Between 1984 and 1986 the research was sponsored solely by the A.C.M.E. Directorate of SERC with the industrial collaboration of Technical Indexes Ltd. Between 1986 and 1988 the research was funded jointly by A.C.M.E. and Technical Indexes Ltd. The period of registration for the higher degree of Master of Philosophy began in September 1986.

The first two years of work consisted of developing the research techniques and philosophy and preparing component specifications suitable for a computer aided system. The second two years work saw the practical application developed and a demonstration software package created. Additional components were also studied and the research findings applied to related areas.

P. Vedamuttu
August 1999

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The research was made possible by the sponsorship of the A.C.M.E. Directorate of SERC and Technical Indexes Ltd.

I would like to dedicate this work to my parents who have given me every opportunity to progress in life.

Nomenclature.

Standard S.I. units are used. (Reference 53).

A : Surface Area

b : Rotary damping constant

b_x : Linear damping constant

C : Capacitance

E : Induced e.m.f.

F : Force

g : Gravity

H : Height

i : Direct Current

I : Alternating current

J : Moment of Inertia

k : Linear spring constant

k_t : Torsional stiffness

L : Inductance

N : The "North" pole of a magnet

n : Turns or gear ratio for a transformer or gearbox.

P : Pressure

P_{21} : Pressure difference between points 2 & 1

Q : Volumetric flow rate

R : Electrical Resistance

R_f : Fluid Resistance

rpm : Revolutions per minute

S : The "South" pole of a magnet

t : Time

T : Torque

V : Applied a.c. voltage.

v : Voltage

v_{21} : Voltage difference between points 2& 1

v_X : Velocity

v_{X21} : Relative velocity between points 2 & 1

X : Inductive reactance.

Z_X : Impedance of a.c. electrical component "x".

+ve : The positive pole of a d.c. supply.

-ve : The negative pole of a d.c. supply.

∅ : Theta, an angle in degrees.

ρ : Density

ω : Angular Velocity

ω_{21} : Angular velocity between points 2 & 1

CHAPTER 1

INTRODUCTION

1.1 The Present Situation

The design of any engineering system, however complex or trivial will always have required a great deal of thought and the use of previous experience. The process will be time consuming especially if the initial conceptual design must be done. Having established the how the system should work. The system can be designed and the relationships between components described. With the components identified, their characteristics can be quantified given their tasks within the system. Care must be taken to select components that will perform correctly. Components that are not appropriate will affect the overall system performance. An example may be a conveyer belt system. The system may be described as follows ; an electric motor as the prime mover, a flexible coupling to absorb misalignments and shock load, a reduction gearbox to moderate the motor speed and a chain to drive the conveyer rollers via sprocket wheels. Then adding the constraint that items of mass 20 kg must be moved at 1 m/sec, conclusions as to the motor power, type of flexible coupling, gearbox ratio, type of chain drive and sprocket sizes can be made. The system can then be optimised to ensure that components with these characteristics are compatible and will assemble together in such a way as to meet the specification. It is with this process that the thrust of this thesis is aimed.

Looking at engineering systems in general, the designer, to save time and money, will attempt to utilise 'off the shelf' (or 'bought out') components. In electronic design, for instance, circuits are built up from standard components, special, "one off" components are very rare. Chips for televisions, videos, CD's may be specially designed items but are then produced in thousands.

Thus, having designed the system and assigned values to the parameters which describe the functionality of the individual components. The system must be built by obtaining the specified components. There are several problems associated with this process, some of which good design will account for, others, may be simply a matter of luck. The good designer will have realised that the idealised system that has been developed will have to be flexible to meet the imperfections of real components. Thus, each parameter for each component will have some tolerance associated with it. As components are identified that meet the specification, the tolerances associated with the parameters will absorb any slight mismatches, enabling the system performance requirements to be met. If the system does not perform to its specification because of a characteristic mismatch, the design will be questioned. However, it was not bad design that ruined the system but the inability to find the components having the best characteristics. It follows that access to a large database of components, that could be interrogated by the designer, would greatly help in the process of systems design. Real systems could be quickly assembled on paper and analysed. (REF. 1).

1.2 The Need for Rapid Access.

The current methods of selecting real components are slow, clumsy and at times short sighted. Selection is often carried out by an individual who, will always be limited by their own knowledge and experience. This will have its drawbacks because the ability to broaden these boundaries will be inhibited. For instance, when selecting a component that has been used before and has performed well, there will be a tendency to return to that catalogue or manufacturer. While this is laudable, it is short sighted if other manufacturers' components are ignored. Overlooking components or manufacturers through lack of information about them is also to be avoided. Clearly, the designer cannot be expected to hold every catalogue of every manufacturer but, being able to multi-source without personal bias is important and is sometimes a requirement. Two further problems a designer faces are that repetitious calculations must often be performed with no guarantee that the effort will be rewarded with a suitable component. A direct comparison of components is difficult because manufacturers present their information in different ways. These differences may range from choice of units to nomenclature. Manufacturers are becoming aware of the need to make selection a quick and easy process for designers and are beginning to produce computer aided selection for their products (REF 28). These systems tend to promote the manufacturers' components and are essentially computerised catalogues. Thus, from a selection view point, the designer has not gained the breadth that the ideal retrieval should provide.

In the market area that could be called 'component information for engineering designers' there is one system that meets the above requirements in a simple but effective way. Technical Indexes (TI) Limited are an information vendor. TI market microfilm library of manufacturers' catalogues and provides a set of indexes which will allow a user to identify products and manufacturers in various ways (i.e. by product type, trade names, company names) and then locate the appropriate microfilm and frame within the film. The index provides a very circumspect identification of a product. Time must still be spent identifying the component needed and there is always the chance that although the company produces a particular type of component, the characteristics required may not be available. However, the advantage of the TI system is that it allows access to a large component database which is regularly updated. The disadvantages are that it is only as good as the manufacturers' catalogues and the same amount of time is spent making a selection as would otherwise be achieved had a catalogue been at hand.

The growth in CAD has established an environment where previously time consuming tasks are becoming almost instantaneous, draughting packages allow drawings to be changed quickly without having to be re-traced. Finite element / difference packages allow designs to be tested and modified before components are manufactured, simulation packages allow the complex architecture of integrated circuits to be designed and tested fully before the chips are made. Similar packages are available for engineering systems to be assembled and it is in this area that component selection will play a significant role and where the research work presented in this thesis will have an important contribution.

The need to be able to rapidly access component information will become apparent when data for real components is fed into simulation packages. If the system is found to perform badly, the designer must attempt to adjust the performance of the system elements. This is easily done by replacing poorly performing elements with similar elements having more suitable characteristics. The system can then be retested using the simulation package. Provided the design was a realistic one, by this process a system meeting the design specification can be thoroughly tested "on paper" before any major capital commitment is made. In the case of a catalogue based search, the designer may be returned to the original situation of not knowing where to find the necessary component information. If component data were available on line, new characteristics could be presented to the simulation package very quickly indeed and the design rapidly optimised. The method by which the rapid access is achieved is of fundamental importance, particularly as the information retrieved would cover many types of components performing the same task. With a fast retrieval system, all these options can be put into the simulation and the best real system developed. The retrieval system should allow the designer to thoroughly define the required component, so that upon finding a suitable match, all that must be done is to order the component. The number of parameters used to define the component will be a trade off between what is the perfect specification and what is possible from the practicabilities of data capture. However, as computer science is also growing to accommodate the needs of data processing, the hardware to capture and store vast amounts of data is now available. Optical disks are capable of storing hundreds of Mega bytes of data. Although more expensive than the ubiquitous floppy disk, their market price continues to drop as they become more widely used. Data base management systems (DBMS) are already powerful enough to deal with very large databases and retrieve data from thousands of records in seconds. Thus technically, it is possible to set up very large databases of components, the databases could be created easily and then interrogated rapidly. Also, if set up, there is a large potential market.

1.3 **Scope.**

The project has been described so far in general terms, the reason being that all engineering components could be handled by computer eventually. However, for the purposes of research, some constraints or direction must be placed on the area of work so that the components studied will be compatible. Also, the commercial potential of the project would be helped if a group of related products was available. Thus, after consultation with Technical Indexes Ltd. [TI], the area of rotary mechanical power transmission was chosen. This area has a large potential market, contains a large number of engineering components and is well documented by the TI microfilm system. Rotary mechanical components can be used to develop and test principles which could then be used to build a methodology through which any engineering components could be analysed and defined in a manner suitable for a computer aided retrieval. It is the building of this methodology and the results gained from it that will be discussed in this thesis along with the analysis of component data.

The final result of the research is a computer aided retrieval system using parameters identified by the project work as the basis of the retrieval. With regards to a computer driven retrieval, it should be noted that the amount of information stored should be a minimum to enable records to be handled quickly and to allow data capture to be efficient. Once data is captured, the power of current database management systems can be exploited. One record in thousands can be found in seconds. Searches can be made using any of the predefined characteristics, tolerances can be introduced, multi-sourcing becomes easy, direct comparisons between component types can be made and manufacturers can be compared.

Time wasted by well qualified personnel working through catalogues and performing repetitive calculations, which may prove fruitless, will be avoided. All the repetitive checking will be done by the computer.

CHAPTER 2

THE USE OF ANALOGIES

2.1 The Analogy Matrix.

Analogy is a technique widely used in all aspects of life. Its primary function is to re-express an idea in a fashion that will help a person previously baffled by a concept to understand it. The re-expression takes the form of a different frame of reference. The classic example is "...you wouldn't do that at home !.". For scientists and engineers, analogy is a powerful tool for dealing with analytical problems where the physical significance of the analysis is difficult to envisage. Direct current is often modelled in the class room as water in pipes. It is easy to visualize water flowing from high pressure to low pressure along the pipe. These ideas can then be used to describe the movement of electrons in a wire, a phenomenon many will never "see". This is not a trivial example, the major building blocks of d.c. electrical systems can be modelled by representing them as the more "physical" water equivalents. It is also important to note that these analogies are not only true in the intuitive, physical sense so far described. They are also equivalent in the analytical sense - the equations governing their behaviour are also analogous. Consider the analogy between a capacitor on charge and a hydraulic accumulator filling. The simple analysis of the accumulator is performed in Figure 2.1. What is the physical meaning of this simple differential equation ?. It "says" that the rate of flow into the accumulator is proportional to the rate of change in head within the accumulator.

It is at this point where a little more thought must be applied. Simple physics tells us that the flow will cease when the head in the accumulator is equal to the reservoir head. Also, we know that the flow is caused by the pressure or head difference between the two fluid masses. Clearly, the maximum flow will be when the pressure difference is greatest - when the accumulator is empty and about to fill. The flow will reduce and finally cease as the accumulator head reaches that of the reservoir. Taking the simple principles already intuitively defined above that flow and current, pressure head and voltage are equivalents and then considering the charging cycle of a d.c. capacitor, an immediate analogy can be

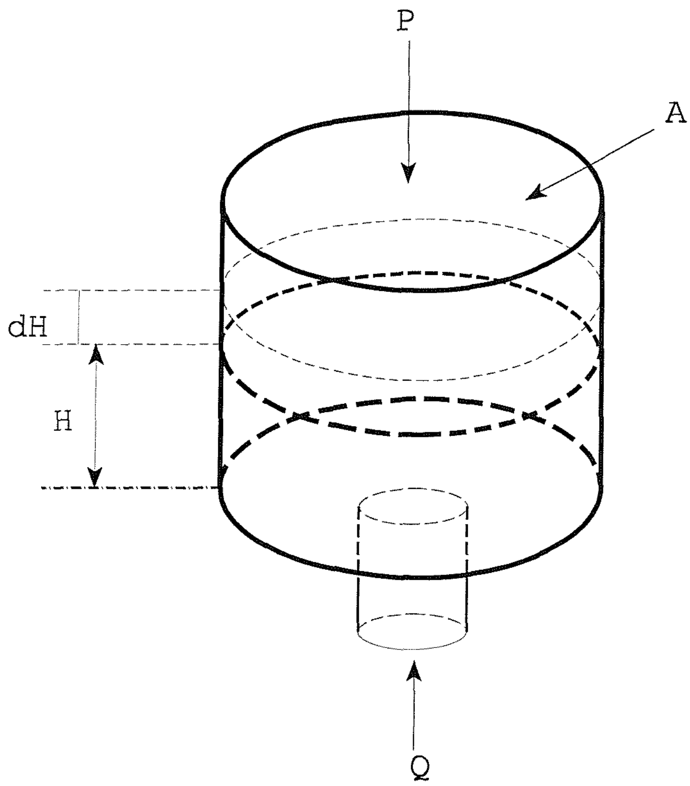
Figure 2.1 : **Simple Analysis of the Hydraulic Accumulator.**

Given :

A = Fluid Surface Area
P = Fluid Pressure
H = Height of Fluid
dH = Change in height of fluid in time **dt**
Q = Volumetric flow rate

Known :

$\mathbf{P = H\rho g}$ ..(1)
ρ = Density of the fluid
g = Gravity
$\mathbf{Q = Av_x}$ ...(2)
$\mathbf{v_x = dH/dt}$ ...(3)
(Flow Velocity)



A relationship between the through (Q) and across (P) variables must be found. Noting that the rate of fluid flow into the tank will change the pressure head :

From (1) : $\mathbf{dP/dH = \rho g}$ .....(4)

Substituting (3) into (2) : $\mathbf{Q = A\ dH/dt}$ .....(5)

Substituting for dH in (4) using (5) :

$$\mathbf{dP / (Q/\{A\ dt\}) = \rho g}$$

or

$$\mathbf{Q = (A/\rho g)\ dP/dt}$$

Compare : $\mathbf{i = C\ dv/dt}$ ... charging of a dc electric capacior.

drawn. The capacitor draws a current until fully charged - its potential is then equal and opposite to that of the supply. Further analysis, such as deriving the equations for the charge / discharge curves for capacitors can then be used to describe the fluid system characteristics. Analogies do not only exist between fluid and electrical systems but exist across all the engineering domains. (REF 2. REF 3).

Analogy is thus a very useful technique and has been used to great effect in analytical methods such as bond graphs (REF 4) where any engineering system can be modelled using simple, basic elements. The relationships between the elements are governed by simple differential equations, the only difference between systems are the values and units of the constants. Similar techniques can be employed to allow electronic simulation packages such as "Saber" by Racal Redac, to be used to simulate mechanical, hydraulic etc. systems.

2.2 **Through and Across Variables, Direct and Dual Analogies.**

The flow of power through any energy domain can be represented by the relationship between two quantifiable variables. For fluids, it is flow rate and pressure. For electrical circuits, it is current and voltage. In the rotary mechanical domain, it is torque and speed. These quantities can be defined for any engineering domain. The link between all of these parameters is the similarity in the way they are measured.

Consider a uniform conductor connected to a d.c. power supply. To measure the direct current, the conductor must be broken. No matter where along its length the conductor is broken, the current measured will always be the same. The current can be thought of as flowing "through" the conductor. For a current flow to occur, the current must be driven. The supply voltage provides this driving force by creating a potential difference. This difference exists across the conductor, giving a direction to the current flow. Voltage can therefore be thought of as existing "across" the conductor.

The power components of other energy domains can now be examined to see if they exhibit these "through" and "across" characteristics. Intuitively, fluid flow and fluid pressure (or pressure difference) provide an obvious and accurate analogy for the fluid domain power components. The identification of through and across variables is less obvious for mechanical systems. Newton's Laws of Motion (REF 49) can help clarify the through and across components. Considering force, Newton III states that action and reaction are equal and opposite.

Thus for a body to remain in equilibrium the forces upon it must cancel each other. This will not only be true for the entire body, but also for any part of it. Hence the forces acting on the body must be acting through it. If one of the forces is removed, Newton II states that the momentum change of the body is proportional to the applied force. Since the mass of the body can be considered to be a constant, the momentum change constitutes a velocity change. The velocity change is in fact a relative velocity difference. The velocity being measured relative to a point in space. This is basically the definition of an across variable. Hence velocity is the across variable in linear mechanical systems. For rotary mechanical systems, the opposed forces become couples. If a couple is removed, rotary motion occurs. Thus, torque can be defined as the through variable and angular velocity the across variable. As a confirmation of these analogies, notice how the vector quantities force and torque differ only by a scalar quantity, distance. The vector quantities linear and angular velocity also differ by the scalar quantity distance. Since a vector multiplied by a scalar is still a vector, the analysis relating force and torque, velocity and angular velocity through Newton's Laws is confirmed by the Vector Laws (REF 50).

Through the consideration of first principles and further analysis as above, the through and across variables can be defined for all engineering domains. This task can be made simpler by employing analogies between the basic elemental devices used to model systems in these engineering domains. In the linear mechanical domain, the three basic system elements are mass, spring, damper (REF 2). By analogy, the system elements in the rotary mechanical domain are; inertia, torsional stiffness and the rotary damper. This analogy is intuitive, but it can be formalised by considering the elemental equations for these devices. Comparing the elemental equations and noting how the through and across variables are related by the elemental coefficients will allow the analogies to be identified. The derivation of these relationships is well documented (REF 2, REF 34 - "Tetrahedron of State"). The analogies for some well known engineering domains are summarized in Table 2.1.

Table 2.1 : Elemental Analogies.

Energy Domain	Elemental Component.		
Linear Mechanical	Mass : m	Spring Constant: k	Linear Damping Constant: b_x
F = Force v_x =Velocity t = Time	$F = m\dfrac{dv_x}{dt}$	$F = k\int v_x dt$	$F = b_x(v_{x21})$
Rotary Mechanical	Inertia : J	Torsional Stiffness: k_t	Rotary Damping Constant : b
T = Torque ω = Angular Velocity	$T = J\dfrac{d\omega}{dt}$	$T = k_t\int \omega dt$	$T = b(\omega_{21})$
d.c. Electrical	Capacitance: C	Inductance: L	Resistance: R
i = Current v = Voltage	$i = C\dfrac{dv}{dt}$	$i = \dfrac{1}{L}\int dvdt$	$i = \dfrac{v}{R}$
Fluid	Fluid Capacitance	Fluid Inertia: I_f	Fluid Resistance: R_f
Q = Fluid Flow Rate P = Pressure Difference A = Pipe CSA ρ = Fluid Density g = Gravity	$Q = (\dfrac{A}{\rho g})\dfrac{dP}{dt}$	$Q = \dfrac{1}{I_f}\int Pdt$	$Q = \dfrac{1}{R_f}P_{21}$

The important observation about Table 2.1 is that for the mass elements, the through variable is related to the time derivative of the across variable. For spring type elements, the through variable is related to the time integral of the across variable. For damping elements, the through and across variables are proportional. Thus the system elements for all engineering domains can be identified with a particular generic group of analogous elements. The generic titles of these groups being taken from the d.c. electrical domain - capacitive, inductive and resistive. The d.c. domain elements have been chosen as the generic titles because they are well known.

Not only are these basic elements related by their through and across variable but the system equations that allow these elements to be combined are also analogous. Thus, Newton's Laws, Kirchoff's Laws (REF 6), the Laws of Thermodynamics (REF 50) all of which stem from the Law of Conservation of Energy can be shown to be analogous and can be summarized as the compatibility and continuity equations (REF 2). The compatibility equation requires the sum of the across variables in a system to equal zero. The continuity equation requires the sum of the through variables to be zero.

Consider the two systems shown in Figure 2.2. The analysis for the electrical system is immediately obvious. Kirchhoff's Second Law tells us : $v = v_c + v_l + v_r$ Knowing the elemental equations for the system components, the system equation can be written:

$$v = \frac{1}{C} \int i \, dt + L \frac{di}{dt} + iR \quad \text{..................} \quad (1)$$

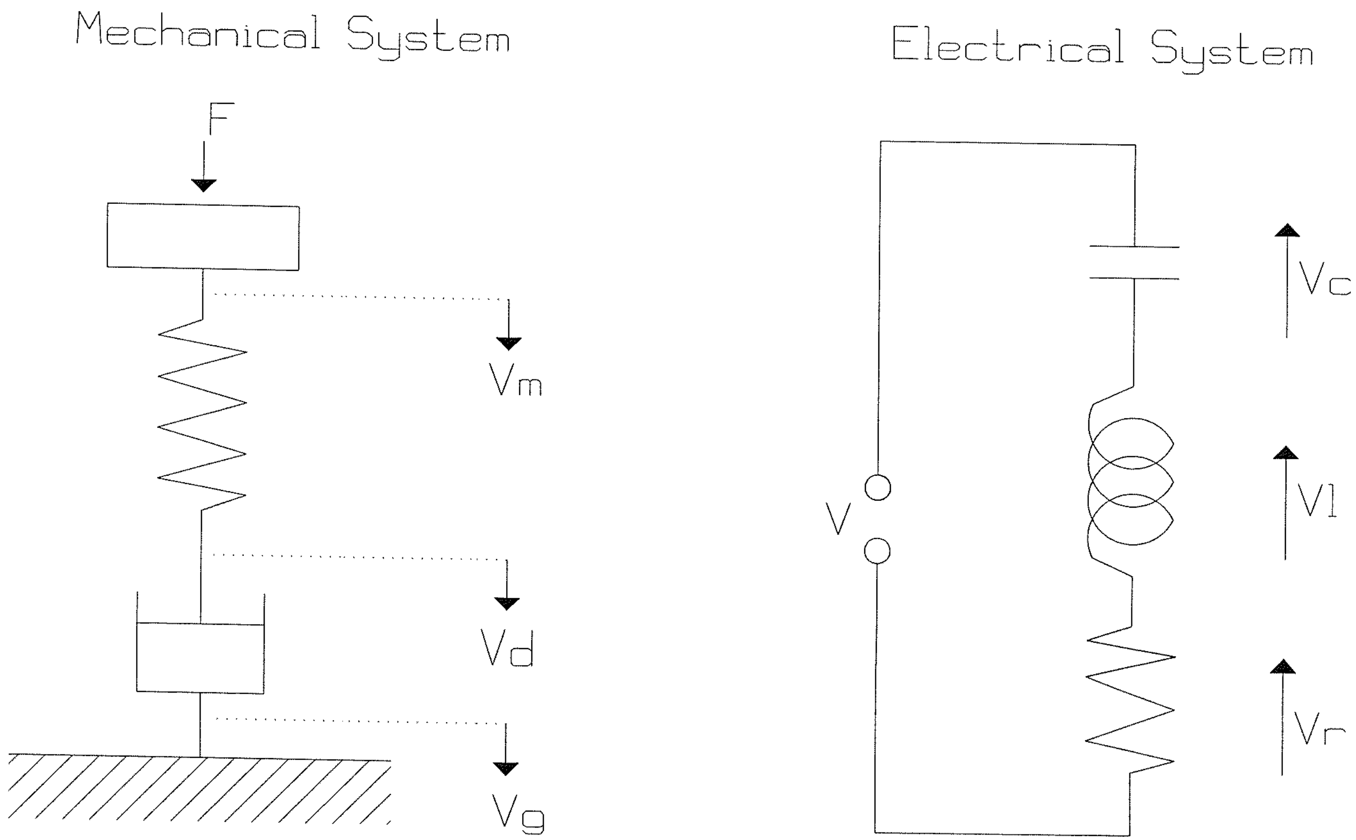
The mechanical system is less easy to analyse, but by equating the system elements and the through and across variables by analogy we have :

$$v_x = \frac{1}{m} \int F \, dt + \frac{1}{k} \frac{dF}{dt} + \frac{F}{b_x} \quad \text{..............} \quad (2)$$

This yields the correct system equation. This type of analogy is known as the direct analogy.

If a more intuitive approach is taken to the behaviour of the system elements, it may be concluded that the "mass" and the "inductive" elements are similar in behaviour. These elements have the effect of attempting to keep the system moving after the system "force"

Figure 2.2 : Two Analogous Systems.



has been removed. Similarly, the capacitive and spring elements can be related by their release of energy after the system "force" is removed. For these analogies to be valid, force and voltage, velocity and current must be deemed analogous. Considering equation (1) and making the following substitutions :

F for v (force for voltage).

v_x for i (velocity for current).

m for L (mass for inductance).

1/k for C (compliance for capacitance)

b for R (damping constant for resistance)

The result :

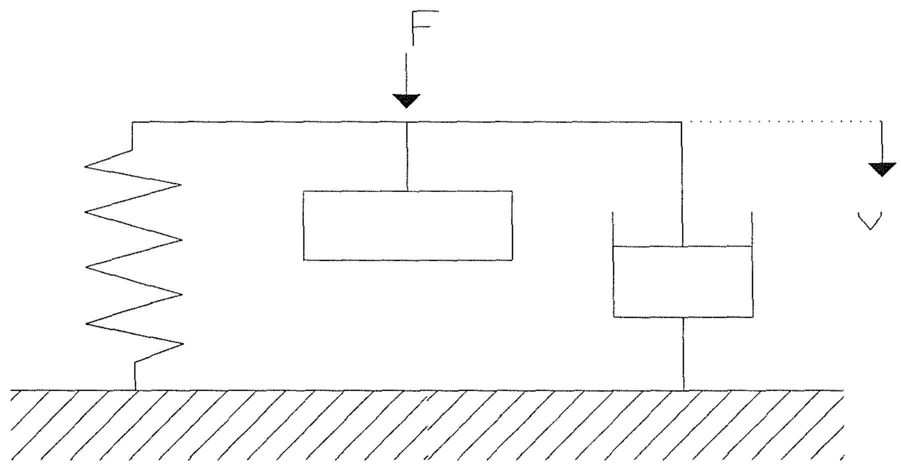
$$F = k \int v_x \, dt + m \frac{dv_x}{dt} + b_x v_x \quad \text{..........} \quad (3)$$

The physical meaning of equation (3) is that the sum of the elemental forces equals the total force. This is the linear mechanical equivalent of Kirchhoff's First Law ; "The sum of the flows into a node is zero". REF 6. Also, the elements have a common velocity, which is now the through variable. For the velocity to be common, the elements must be arranged in parallel !. This yields the system shown in Figure 2.3.

This system contains the system elements originally arranged in series, arranged in parallel. This system will clearly function in a very different way. However, it has been generated from the same basic system equation. The system in Figure 2.3 is called the "dual" or "indirect" analogy of the system shown in Figure 2.2. Systems can be analysed using either direct or dual analogies. Both methods are valid and as long as the elemental equations and the through and across variables remain consistent throughout the system analysis.

In the context of this thesis, both direct and indirect analogies will be considered and used. The type of analogy used will depend on which elemental component of the energy domain under consideration best matches the elemental component from the domain with which it is being contrasted. The major problem is deciding whether an elemental characteristic is "capacitive" or "inductive" in its action. In all cases, comments will be made as to which type of analogy is being tested.

Figure 2.3 : The Mechanical Dual of Figure 2.2.



The "Analogy matrix" shown in Table 2.2 has been developed with the direct analogy rule set. The matrix has been drawn to provide an indication as to how components can be related by their function. The matrix provides a guide to help try and assess how a component or characteristic is functioning within a system. This initial assessment can then be used as the basis for more detailed analysis.

The term "matrix" is used because it is possible to move around it tracing analogies. A row of the matrix will contain components from a particular engineering field ie the linear mechanical field will contain elements such as rams, springs, dampers, masses, levers etc. The columns of the analogy matrix will therefore cut across the engineering fields and it is within these that the components must be grouped to preserve the analogies down the columns. Thus, the column that contains electrical resistors will also contain dampers (both linear and rotary), fluid resistance, thermal resistance etc. It must be noted that terms such as "fluid resistance" are very vague and must be carefully defined for the circumstances in which they are considered. (REF 5). However the vague terminology is used because it allows the research to consider the possibilities of analogies within these areas. Thus, in a rotary mechanical system, the "transformer" element is represented as a gearbox. There are however, several other elements which also perform this function ie belts and pulleys, chains and sprockets and friction discs. Hence when using the matrix to generate analogies, all the elements which perform the "gearbox" function should be considered. Thus, the columns are headed by "generic titles" which attempt to describe the functionality of the components in the column. These titles, which are mainly based on electrical devices, will cover discrete components and components whose overall function fulfils the generic title. These may range from using an ideal component, such as a damper, to a complex assembly of ideal components in a sub system, such as a motor.

2.2 The Flow of Power Through the Matrix

The Law of Conservation of Energy states that "energy cannot be created nor destroyed, it can only be changed from one form into another". This energy conversion can be traced very accurately through the analogy matrix as power flows from domain to domain via the class transformer elements. Power loss through heating, can be traced through the thermal systems, thus all the energy in the system can be accounted for and thereby, the Law is

Table 2.2 : The Analogy Matrix.

	Generic Title							
Energy Domain	Source Elelment	Impedance Transformer	Inductive Element	Capacitive Elelment	Resistive Elelment	Connecting Elelment	Other Elelments	Class Transformer
DC Electrical	DC Generator	Static Inverter	Inductor	Capacitor	Resistor	Wire		Light Bulb
AC Electrical	Alternator	Transformer	Inductor	Capacitor	Damper	Wire		AC Motor
Hydraulic (Liquid)	Pumps	Differential Piston	Hydraulic Accumulator	Fluid Mass	Pipe Friction	Pipe		Piston
Pneumatic (Gas)	Pumps	Differential Piston	Pneumatic Accumulator	Gas Mass	Pipe Friction	Pipe		Turbine
Linear Mechanical	"Piston"	Lever	Spring	Mass	Damper	Rod		Reciprocating Pump
Other Domains								

satisfied. If a particular method of analysis is developed to deal with a particular energy domain, it should, by analogy, be applicable to the other domains. Furthermore, a generalised analysis can be developed to model systems in any of the domains. One such methodology is the Bond Graph technique. REF (4). "Bond Graphs are based on the splitting of engineering systems into separate components that exchange energy through identifiable connections or ports.". "bond graphs are exceedingly suitable to prepare simulation by differing computer programs, including the so called matching analysis of complex mechanical engineering components like heat exchangers, turbines and compressors.". The essence of the Bond Graph approach was to model systems through "generic components" which are identical no matter what energy domain one is considering. This is illustrated in Figure 2.4. The energy domain is irrelevant for the analysis because the technique simply uses the appropriate values of the system constants and it is for the analyst to place a significance on the results. Thus, the bond graph shown in Figure 2.4a can be used to analyse the three simple systems in Figure 2.5. Results specific to one system are generated by entering the particular system constants.

The study of Bond Graph techniques made a significant contribution to the overall philosophy of the approach used when attempting to generate analogies between components. The technique of attempting to break down each component into its fundamental elements was a very efficient method of generating characteristics which could then be looked for in analogous components.

2.3 Generation of Analogous Properties / Characteristics

The need for rapid access to component information has already been outlined in Chapter 1, the use of analogies has been described above, this section attempts to describe how these two areas can be brought together in a profitable way for component selection.

The methodology of the research is described in Chapter 10, the hypothesis of how analogies can be of use in component selection can be described here. Using the analogy matrix, components analogous to that being studied can be identified. Not only can the whole component be studied, but its constituent parts can also be studied by analogy. The analogy matrix will help the investigator generate analogies at both a physical and analytical level.

Figure 2.4 : The Generic Components Used in Bond Graph Analysis.

		Examples		
Generic Name	Bond Graph Symbol	Electrical	Fluid	Mechanical
Simple Bond	$\overset{e}{\underset{f}{\rule{2em}{0.4pt}}}$	Wire	Pipe	Rod / Shaft
Resistive Elelment	$\overset{e}{\underset{f}{\rule{2em}{0.4pt}}}R$	Resistor	Constriction	Stiction
Inertia Element	$\overset{e}{\underset{f}{\rule{2em}{0.4pt}}}I$	Inductance	Mass Action	Mass / Flywheel
Capacity Elelment	$\overset{e}{\underset{f}{\rule{2em}{0.4pt}}}C$	Capacitor	Accumulator	Spring

Figure 2.5 : The Simple, Unaugmented Bond Graph Describing the Simple Systems in Figure 2.4.

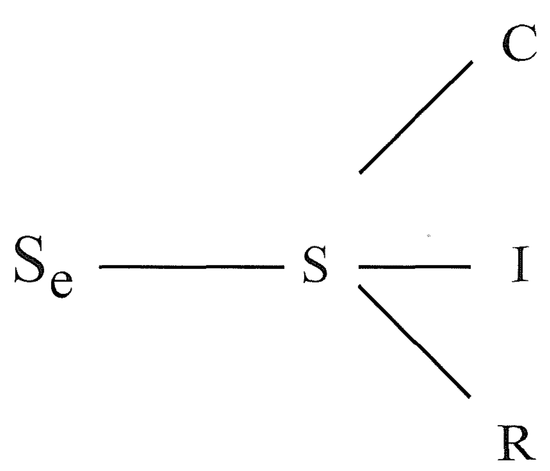
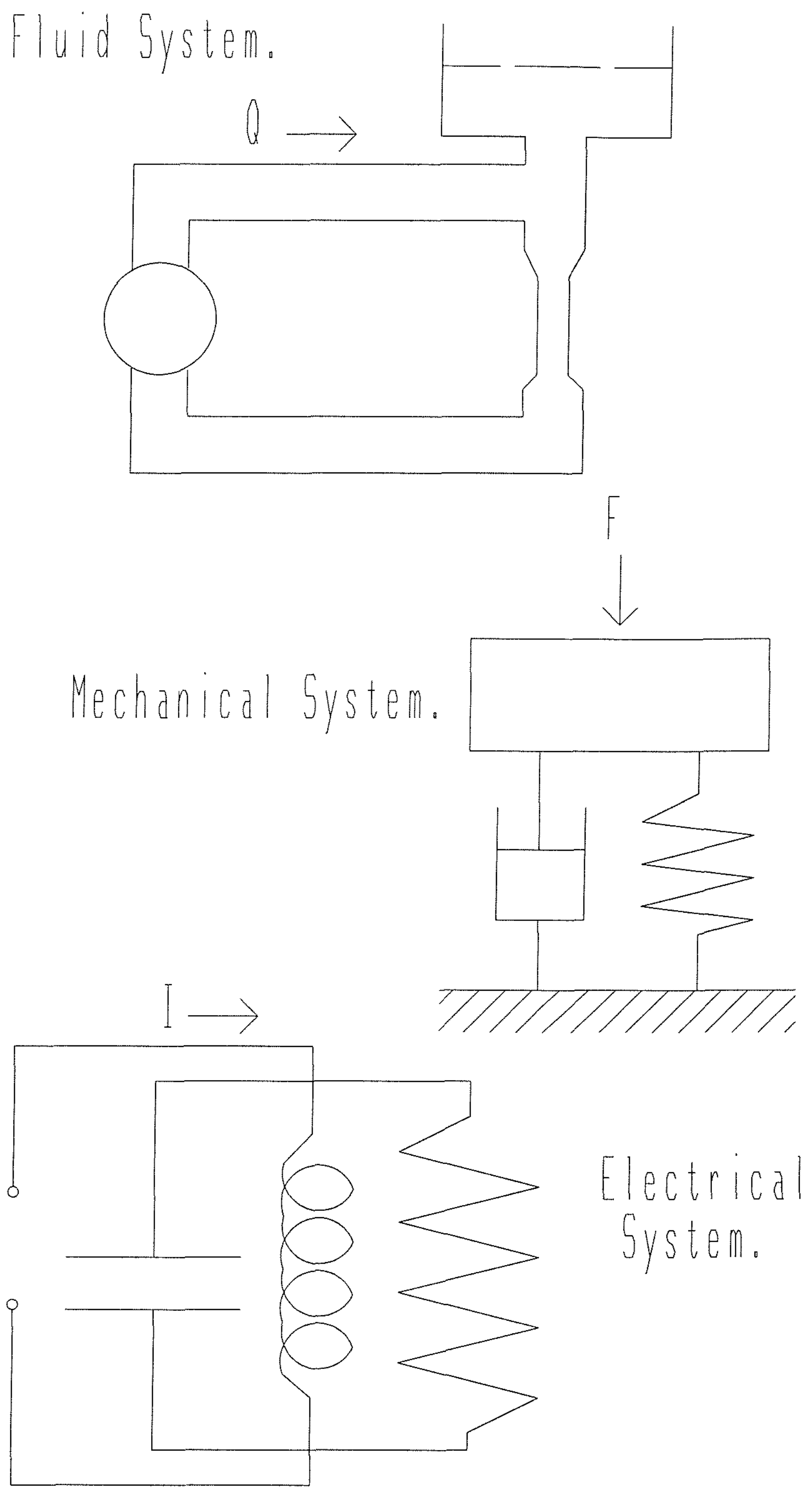


Figure 2.5 : Three Analogous Systems.



The physical analogies tend to be intuitive and are stimulated by the analogy matrix, the analytical analogies will then follow as the physical concepts are quantified. This process has two main results, firstly the clear analogies between components are identified and confirmed. The analogous parameter will usually be a quantifiable characteristic of the component. The significance of the characteristic can also be noted because the analogous components require this characteristic to be specified in order to make a valid selection. Secondly, characteristics used to describe one component may not have readily identifiable equivalents in other components. The energy dissipated when a clutch is engaged does not seem to have an equivalent in a fluid coupling. However, careful study of fluid coupling selection data does in fact allow for the start up energy of a fluid coupling. (This is fully described in Chapter 8). The significance of this discovery is that the starting performance of both these devices is limited by this parameter because excessive heating will cause damage. Hence, not only should this parameter be considered when selecting a clutch, it should also be considered when selecting a component analogous to a clutch - the fluid coupling. Thus the most appropriate set of characteristics can be developed and justified by using analogies. Clearly, not all the analogies will be significant for selection, many will be trivial or of a very abstract nature which are mainly of academic interest (Chapters,4,6,7 & 8) . However, it is the thoroughness of the research which has allowed the author to propose the parameter lists proposed in Chapters 4,6,7 & 8 which have a practical application.

Some time has been spent examining the further implications of the analogy work and how it may be used to help designers. The use of analogy allowed characteristics not considered in certain components, to be identified and their importance towards selection assessed. If designers and engineers performing specification and selection of components were made aware of these characteristics, would they be assisted in selecting better matched components?. Would the process of design benefit by the advantage of more information ?. Also, could the design process be helped by considering analogies ?. For instance, could techniques used to combat a problem in one field, be used to overcome problems in another, in a novel way ?. Could good design be justified by considering how similar problems are dealt with in analogous systems. Many of these additional questions are considered and debated, in the following chapters.

CHAPTER 3

THE STUDY OF MOTORS

3.1 Introduction

The first component to be studied was the rotary motor. It is the source of power in most rotary mechanical systems. Using the principles of the analogy matrix and common sense, the various sources of rotary mechanical power can be identified. These are shown in Table 3.1 in Appendix E. For the purposes of research, the types of motor considered were:

i) a.c. electric motors.
ii) d.c. electric motors.
iii) Hydraulic motors.
iv) Pneumatic motors.

Power sources, such as the internal combustion engines, have been omitted from the study because in the industrial situation the above four types of motor dominate. They are more powerful, efficient and quieter when compared to a similar sized internal combustion engine REF.15. The analogy matrix suggests other components such as rams, generators and pumps should be studied because they are also sources of power in analogous systems. It was decided for research purposes that rams would not be studied because the types of motor listed above gave the research the breadth required. It will be shown later through the principle of commutation that linear and rotary system characteristics can be related through analogy and hence do not need to be studied separately. Furthermore, the construction of simple electric and fluid motors, will allow the machine to work in the opposite sense ie the output shaft is driven. Under these circumstances, a simple synchronous motor will produce an alternating supply in the stator windings when the stator supply is switched off. The transition of a shunt wound d.c. machine from motor to generator is well documented in text books and is put to use practicably in the Ward - Leonard method of speed control (REF 41). Piston, vane and gear type constructions are available for pumps and motors in the fluid field. The differences between motoring and pumping are subtle and are mainly to do with the way seals and clearances are designed (REF 38). Therefore, generators and pumps need not be formally studied as their operating principles etc. would be part of the background work when looking at the motor fundamentals. Analogy will allow comments about fluid pump and electrical generator construction to be made from a detailed study of motors.

3.2 **Generation of the Extended Parameter Lists.**

In Chapter 1, the use of analogy for generating additional characteristics to describe components was outlined. The analogy matrix shown in Chapter 2 showed how analogous components could be identified. Having created groups of analogous components, the characteristics for comparison must be gleaned. This process is described below. Although a laborious task, the compilation of the characteristic lists is the keystone of the research. It is through these lists that the analogies are drawn and the bulk of the academic work performed. Some of the characteristics may seem trivial but for the work to be a rigorous test, all possibilities must be considered. Thus, the analogies between components can be developed through logical deduction and not by random or favourable methods.

The main postulation of the research work is that a generic set of characteristics can be developed for a group of analogous components. Furthermore, characteristics not normally considered as applicable to certain components in the group, but applicable to other group members can be re-appraised. Therefore, a detailed knowledge of the characteristics of one component in an analogous group is not enough because the hypothesis suggests that this will not be a complete description of the component. A complete description can only be achieved by confirming analogies with known characteristics of other components in the analogous group. Thus, it is necessary to research the components in the analogous group, noting their individual characteristics ready for analysis by analogy. This study will yield lists of characteristics which have been called "extended parameter lists". These lists form the basis of the research work because they stimulate the generation of analogies. The depth of these lists contributes to the thoroughness of the research work.

The first objective of studying any component is to become familiar with the basic principles governing its behaviour, its application and how its construction is modified for different applications. The best sources of this information are text books and handbooks. Standards were also consulted, BS 5000 part 99 being the index to all of the relevant British standards. IEC and DIN standards were also obtained. All the relevant standards are given in REF. 10. The significance of these standards is that they indicate the kind of information expected for the description and specification of motors. BS 4999 part 1 is especially helpful. It must be noted that there are no BS or overseas standards for the specification of hydraulic or pneumatic motors.

With the fundamentals understood, a study of the company literature could be made using the Technical Indexes (TI) microfilm library of manufacturers' catalogues. The company literature study was a very time consuming but important part of the basic research. The manufacturers covered by TI amount to around 250 suppliers for all four types of motor. The catalogues from all these companies were studied, the parameters used for selection noted and selection procedures (if given) recorded. The parameters used for selection were noted at two levels. Specific characteristics such as torque, maximum volts, breakaway pressure etc. would be noted. Characteristics such as dimensions would be grouped under the heading "dimensions", this implies characteristics such as shaft size etc., are available, but would be too time consuming to note on an individual basis. The title 'dimensions' is perfectly valid because in most cases the component is represented as a dimensioned drawing. Other characteristics falling into this category would be mount type (many are available), insulation class, type of enclosure etc. Finally, a note was made as to whether any graphical information was present, this would usually be torque / speed curves. Thus for each company brochure studied, a list of parameters used to describe their motors was compiled. An example list is shown in Figure 3.1. Additional examples, Figures 3.2 - 3.8 are shown in Appendix E.

Not all of the companies studied yielded lists. Some brochures consisted of little more than a photograph of the motors produced and quotes for the power and torque ratings. Since these values were quoted by most manufacturers, the catalogue was not "listed".

The end result of this data collection was a set of lists of parameters used for selection and a smaller set of selection procedures. The selection procedures could be supplemented by the definitions and procedures outlined in the relevant Standards. A new, more detailed,

FIGURE 3.1: AN EXAMPLE OF DATA EXTRACTED FROM A COMPANY BROCHURE.

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

DC MOTORS

Dimensions
Mountings
Fan type
Winding type
Balancing
Insulation
Overload current (magnitude and time)
Efficiency at rated values
Type of excitation
Speed range and control
Starting current
Rated volts
Inertia of rotor
Electrical time constant

DATA REQUESTED FOR ORDERING

Nominal power
Service class
Nominal speed and direction of rotation

Control range
Max torque at max speed
Max torque at min speed
Mounting and protection
Ambient temp
Altitude
Supply Details.

GRAPHICAL INFORMATION (and comments) :
Torque v speed curves (straight lines for shunt)
Dimensions and parametric drawing
Power v temp / altitude tables

analogy matrix could now be compiled using these parameter lists. The column headings in this matrix would consist of the general motor types; a.c. electrical, d.c. electrical, hydraulic, pneumatic. The rows of the matrix would consist of analogous parameters. Under each column heading every parameter used to describe the motors concerned would be listed. To help structure the lists, the parameters were broken down into four basic types: input, output, physical and "others". The input characteristics are any parameters concerned with the supply to the motor. The output characteristics cover the motor performance. The physical characteristics encompass the motor dimensions; mounting type, mass, etc. "Others" captures any parameters which do not fall easily into the former three categories ie noise in decibels, type of enclosure etc.

From the catalogue study, it was noted that a.c. electric motors are the most documented. It was decided to use this list as the base list against which the others would be ordered. Firstly, sub lists were prepared, using the four categories given above. No attempt was made to order these lists but, by a natural process, the most common parameters tended to appear at the top of the list. The sub lists for a.c. motors are shown in Appendix E, Figures 3.9 - 3.12 inclusive. These lists were then combined into the "extended lists" which are hoped to cover every possible parameter used by manufacturers to describe the various types of a.c. motor on the market.

A similar process was then performed using the d.c. motor lists, because the structure was already there from a.c. motors, the long list for d.c. motors could be compiled immediately along side the a.c. list. Thus, parameters such as "input power", "rated torque", "dimensions", etc. would appear in the same row in the lists. If a d.c. equivalent did not appear, the row was left blank. Also, if a d.c. parameter occurred which did not have an a.c. equivalent, a blank had to be introduced into the a.c. list. (From a practical point of view, this was not difficult as this process could be carried out on sub lists). Thus when this process was complete two "incomplete" but "total" lists were created. This contradiction can be explained very easily. The lists were incomplete because all the analogies were not listed. The individual lists were however total, because they contained all the parameters used to describe that group of motors. This process was then repeated for hydraulic and pneumatic motors. The incomplete matrix could now be studied and completed by using the principle of analogy. A section of the incomplete, extended, parameter list is shown in

Figure 3.13. All of Figure 3.13 is shown in Appendix E.

When completing the matrix several factors had to be considered. Firstly, there is a fundamental anomaly in the analogies between electric and fluid motors, that is, torque of electric motors is generally governed by the supply current (REF. 6). Torque of fluid motors is governed by the supply pressure (REF. 7). The anomaly is that current and pressure are not analogous. The analogy is current and flow, voltage and pressure (Chapter 2 and REF.2). In terms of fundamentals, the fluid motors are class (or domain) transformers, the electric motors are crossed over transformers or gyrators. (REF.4). For the class transformer, the fundamental output variables are dependent on their analogous input variables. For the gyrator, the fundamental output variables are dependent upon the dual of their analogous input variable. Given this anomaly, it is still possible to complete the analogy matrix provided that this anomaly is respected as it occurs across the rows.

Secondly, care had to be taken with semantics. Different manufacturers employ different ways of describing a parameter, the best example being the use of the adjectives "rated", "continuous", "running", "normal", they all mean the same thing: the value at which the motor will run continuously without overheating or damage. The preferred term is "rated" as described in BS 4999 part 1. Thus, parameters with these prefixes can all be moved to the same row of the matrix. These were left with their different adjectives to highlight the difference. Clearly, not all the manufacturers used different adjectives so only the peculiar ones from any one group were entered, if several existed, they were all entered. The preferred term was only identified when the lists were to be shortened.

Thirdly, if parameters could be derived by simple calculation or by reading a performance curve, the derivations should be noted and the degree of difficulty assessed. An example is the use of percentage values particularly in the case of a.c. motors. Parameters such as pull out torque are often presented as a percentage of full load (or rated) torque. The absolute value can easily be deduced from the absolute rated value. It can however, be a frustrating experience if the absolute rated value must also be calculated from power and speed information (note the speed has to be in radians per second and is always quoted in

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...

revs per minute!). A further implication of derived information is that, at times, the derived data may be the result of some extra thought as well as calculation, in these cases only the initial clue is placed in the analogy row. The actual analogy would require extra work. The best example of this occurs in clutches (Chapter 6), in short, if the heat capacity is quoted and the permissible temperature rise, the limit on the absorbed power can be evaluated, similar inferences can be made in the motor analogy matrix.

Finally, if no obvious analogy could be identified, the potential analogous parameters must be investigated in greater detail in order to find the most suitable equivalent. If successful, the derivation of the analogy can then be noted. If unsuccessful, the element in the analogy matrix must be left blank.

A section of the completed analogy matrix is shown in Figure 3.14. All of Figure 3.14 is shown in Appendix E. An attempt has been made to indicate several points. Where the same characteristic occurs for all motor types, the characteristic is entered across the row, unless there are any semantic changes, in which case the different versions are all entered. Parameters, which must be derived, (from curves or by calculation) and those found through analogy, are denoted by a hatched background. Parameters for which analogies have not been found, have been left blank on a hatched background.

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...

CHAPTER 4

MOTOR ANALOGIES

4.1 Discussion of the Extended Parameter Lists for Motors.

This chapter analyses the analogies described in Chapter 3. The following section moves through the motor matrix highlighting any of the derivations and analogies that may not be immediately obvious to the reader. This section is very extensive, thus the bulk of Section 4.1 has been moved to Appendix A. The major analogies identified are discussed in Section 4.2 below.

4.2 Discussion of the Motor Analogies

In attempting to complete the "extended lists" by filling in the missing parameters across the rows of the extended list matrix, many interesting analogies were identified. These analogies work at a fundamental level and can be use to show how differing concepts in machine design can be traced to the same simple principles.

4.2.1 Phases in a.c. systems - the Fluid Analogy to an Induction Motor.

The power supply for a.c. motors may be single or polyphase (usually three). For selection, knowing the number of supply phases is a secondary consideration (see Chapter 10). The phasing of the a.c. supply is the source of the main motive element in a.c. motors. A phased power supply allows the creation of a "rotating" magnetic field. REF.8. It should be noted that the "speed" of rotation of the magnetic field set up in the stator windings of a.c. motors is proportional to the supply frequency alone. Frequency is a characteristic unique to the a.c. electrical domain. It has no analogies in the d.c. electric or fluid domains.

Using the analogy matrix to look across the analogous engineering domains, the fluid vane motor is the only motor with a similar rotating motive component. Direct current machines do not set up rotating fields but use commutation to "move" magnetic forces - see Section

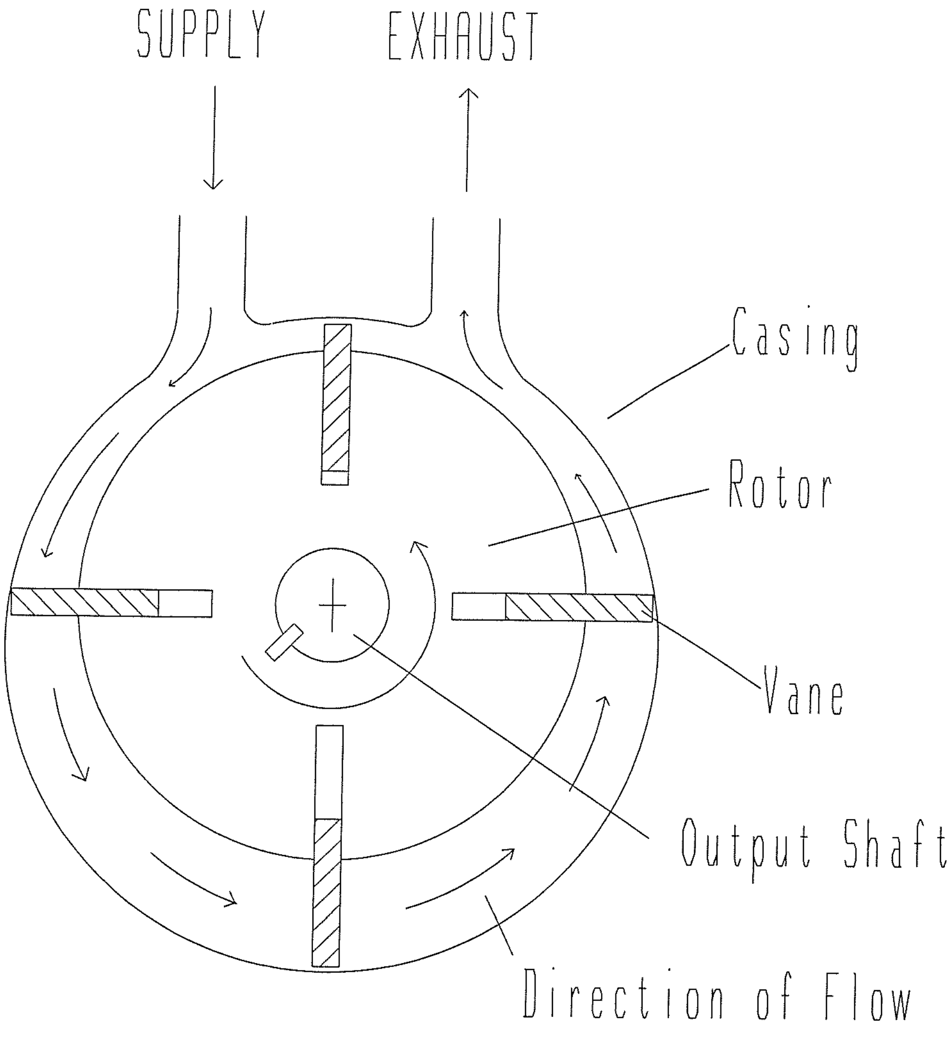
4.2.4. Figure 4.1 is a schematic of a vane motor. Fluid is caused to flow around the casing, dragging the vanes on the rotor with it. The fluid flow around the motor casing can be defined as the analogue of the rotating magnetic field set up in the stator coils of polyphase electric motors. The vanes can then be defined as the analogue of the rotor parts. Noting that both induction and synchronous machines have similar stator fields, the next question is whether the fluid motor performance is analogous to the electrical synchronous or induction motor. Intuition would suggest that the vane motor was analogous to the synchronous motor because the vanes move round at the same speed as the flow - the rate of flow through the motor is a direct measure of the motor speed. However, there are other features of the vane motor that make this intuitive analogy less likely.

Vane motors and induction motors are able to start "direct on line" ie when power is applied, the motor will start to rotate. Synchronous motors require a starter, these machines have to be "pulled up" to the synchronous speed set by the rotating field in the stator coils. The other key difference between synchronous and induction a.c. motors is that the induction motor rotor does not rotate at the same speed as the stator field but runs a little slower. This lag or "slip" (REF. 40) is vital for its operation. If slip can be shown to exist in vane motors the analogy with induction motors is complete.

4.2.2 **Slip in the vane motor**

Slip, in induction motors, is defined as the ratio of the difference of the synchronous speed (the speed of rotation of the magnetic field around the stator) and the rotor speed to the synchronous speed. Slip will therefore always be some value between 0 and 1. When the rotor moves at the same speed as the rotating magnetic field, slip is zero. When the rotor is stationary, the slip is 1, or 100% slip is present. Induction motors usually run at around 3% slip but continue to produce torque throughout the slip range. At zero slip the induction motor produces no torque because the rotor does not cut the rotating magnetic field. Synchronous motors produce torque at zero slip and begin to stall or "pull out" of phase as the value of slip increases. Thus, noting the observations above, the induction motor appears to be the better model for finding an analogy with the vane motor.

FIGURE 4.1 : A SIMPLE VANE MOTOR.



Consider a vane motor with the rotor removed. It is connected to a constant pressure supply and a fluid flow is established. This flow is analogous to the rotating magnetic field in the stator of an induction motor. When the rotor is replaced, a force from the fluid pressure (and fluid momentum to a smaller degree) will act on the rotor causing it to spin. For a constant pressure and a steady flow, the rotor will eventually reach a condition where the torque it receives from the flow is enough to overcome its losses and it rotates at a constant speed. At this point, if the flow rate is measured again, it will have dropped. This is because of the pressure drop across the motor caused by the introduction of the rotor. Thus, in terms of flow rate, there is a slight "flow slip ". The supply pressure and the inertia of the rotor will dictate the degree of slip. If the rotor is now subject to external loading (something needs to be driven), there will be a further torque requirement. This must be met by the supply pressure, this will result in a greater head loss across the motor and a further reduction in the flow rate. Thus the flow slip will increase. As the load is increased, there will eventually come a point where the supply pressure cannot produce the required torque, the motor will stall, the flow rate is zero and the flow slip is 100%. This description is not perfectly analogous to the stall characteristics of an induction motor. There is no "pull out torque" but a steady increase of torque as flow decreases. However the analogy is sufficient at this intuitive, physical level.

The use of the slip analogy concludes that the available torque from the vane motor is inversely proportional to flow slip. This is better expressed as torque being inversely proportional to delivery which is a well known and documented characteristic REF 7. Thus the "flow slip" characteristic, developed through analogy, has produced a recognised motor characteristic. Although the flow slip characteristic has been verified, can the full analogy of the slip characteristic of an induction motor be satisfied?. The full explanation of the induction motor torque speed characteristic is given in REF 6.

A typical characteristic showing the starting, pull-up, pull-out, and rated torques is shown in Appendix F, Figure 4.2. The torque / slip curve has two distinct sections. The section up to the pull-out torque, where torque and slip are inversely proportional and the section after the pull-out torque where torque and slip are proportional. From the study of manufacturers literature, it can be seen that induction motors reach their pull out torque, by at most, 30% slip. So, for the larger part of the torque speed characteristic, torque is inversely proportional to slip. This therefore broadly agrees with the predicted variation of torque and slip by analogy.

4.2.3 The "Synchronous" fluid motor

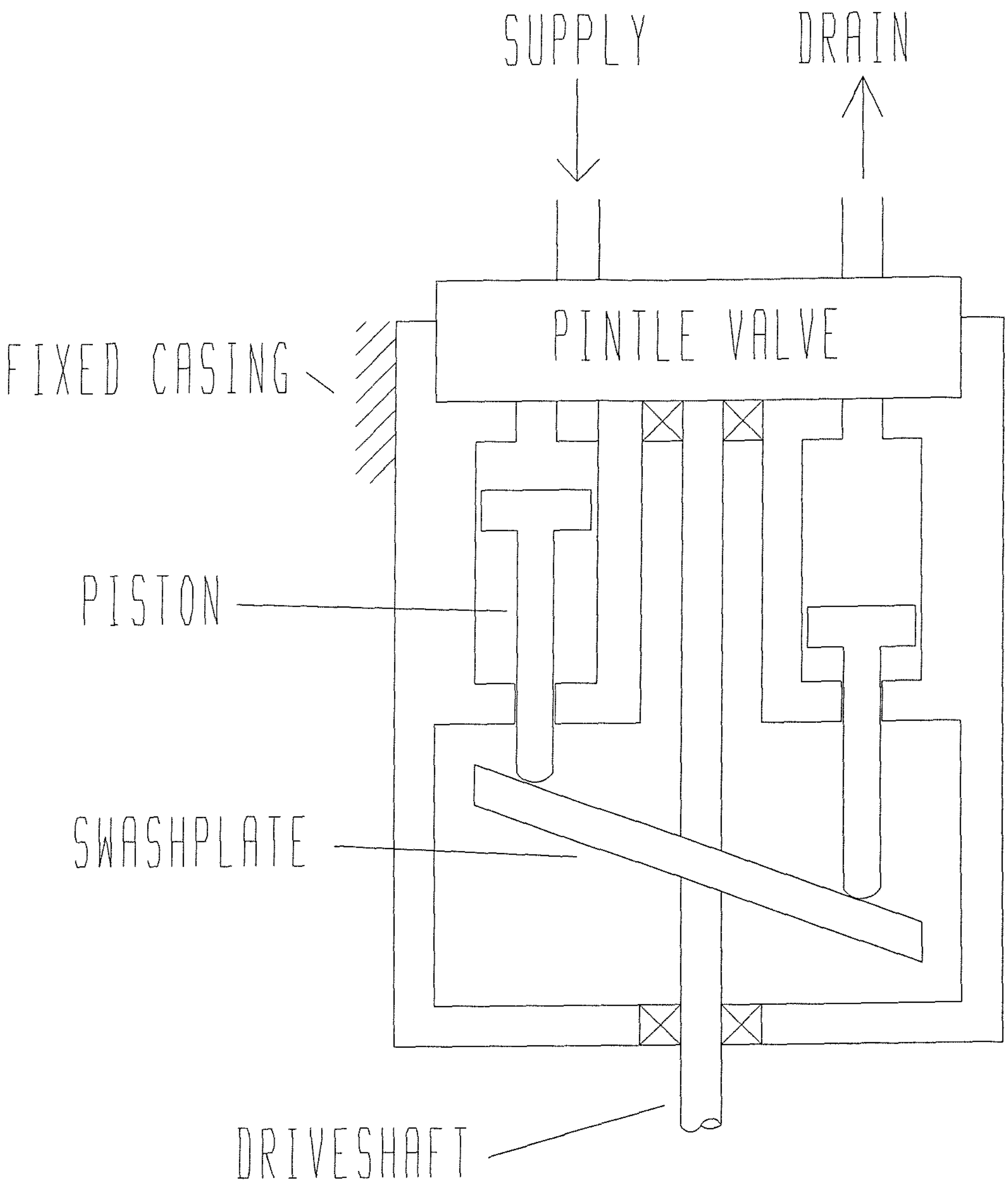
Having made the analogy between the fluid vane motor and the induction motor, it is now possible to redevelop the analogy between the synchronous motor and a fluid motor. The major problem is that all fluid motors are self starters, whereas the synchronous motor cannot start by the synchronous action. Synchronous motors require a starter using another motor principle to accelerate the rotor to synchronous speed (an induction winding may be provided). The postulation is now to analyse the mechanisms of fluid motors to attempt to find some kind of construction that will mimic a synchronous motor.

There are two basic types of fluid motors, the vane type described above and the piston type. The concept of flow slip can also be applied to piston motors so they could be deemed inductive. Consider the action of a simple axial piston motor shown in Figure 4.3. The pistons are supplied in turn by the rotating pintle valve, the pistons force round the swash-plate which is keyed to the drive-shaft. The drive-shaft provides rotary mechanical power at its free end, but also rotates the pintle valve within the motor. Thus, the pintle, pistons, swash-plate and drive-shaft are all synchronised by their mechanical connections.

In the synchronous motor, the rotating magnetic field and the rotor are not physically joined, they are linked by electromagnetic forces. These forces can be overcome, and once out of phase, they will not produce useful torque and the motor will stall. An experiment can therefore be performed with the piston motor. The mechanical links causing the mechanical synchronisation, can be broken, and the effect predicted.

With reference to Table 4.1 (Appendix F), the mechanical connections that can be broken are the pistons / pintle valve, swash-plate / pistons, drive-shaft / pintle valve, drive-shaft / swash-plate arrangements.

FIGURE 4.3 : A SIMPLE AXIAL PISTON MOTOR.



The swash-plate and pistons are not connected because the pistons do not rotate with the swash-plate but reciprocate forcing the swash-plate round, they must therefore be free to slide over the swash-plate surface.

If the pistons were not connected to the pintle valve as in Figure 4.4, the fluid power would not be available to the pistons and the motor would not work.

If the drive-shaft was to be disconnected from the swash-plate as in Figure 4.5, the motor would fail to rotate because although fluid power was available, it could not be ported to the pistons because the drive-shaft was not working the pintle valve. If the drive-shaft was to be rotated, the swash-plate would begin to spin, but, no useful power would be transmitted to the drive-shaft, so, the motor would be useless.

If the drive-shaft was disconnected from the pintle as in Figure 4.6, the motor would again fail to work because the pintle could not supply the pistons in sequence. However, if the pintle was rotated, the pistons would start to reciprocate. The vital point, to now note, is that if the pintle were to be rotated at any significant speed, say 1000 RPM, the swash-plate would never accelerate to that speed in time to continue to receive the piston pushes to perpetuate its rotation. The piston strokes and the swash-plate would be out of phase, (the pistons may even attempt to reverse the swash-plate / drive-shaft rotation) and the motor will doubtlessly stall.

Consider the case where the drive-shaft was somehow spun up to the pintle speed and then the pintle driven. Although the pistons and the swash-plate would be out of phase, the mechanism may well pull up because the swash-plate may be slightly lagging the pistons. A great deal would depend on the amount by which the pintle and the pistons were out of phase. Clearly, if a piston were to come down on the wrong side of the swash-plate, a great deal of damage would result. Thus, ideally, the drive-shaft / swash-plate should be in near synchronisation with the pintle when spun up by the starter.

Starting could be achieved, by knowing the position of one of the pistons and its relative position to its appropriate port (a timing mark). The drive-shaft could be spun up with the

FIGURE 4.4 : PISTON MOTOR WITH THE PINTLE AND PISTONS UNCONNECTED.

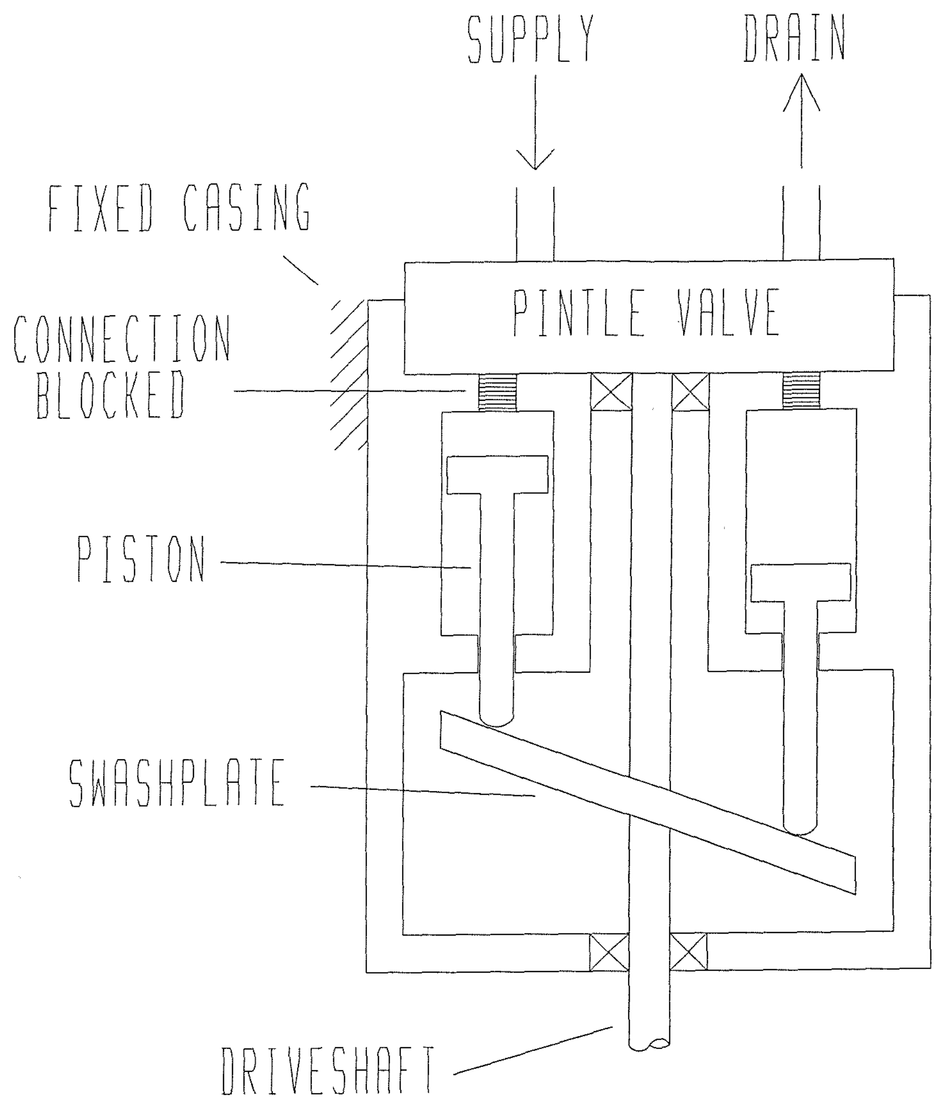


FIGURE 4.5 : PISTON MOTOR WITH THE SWASHPLATE AND DRIVESHAFT DISCONNECTED.

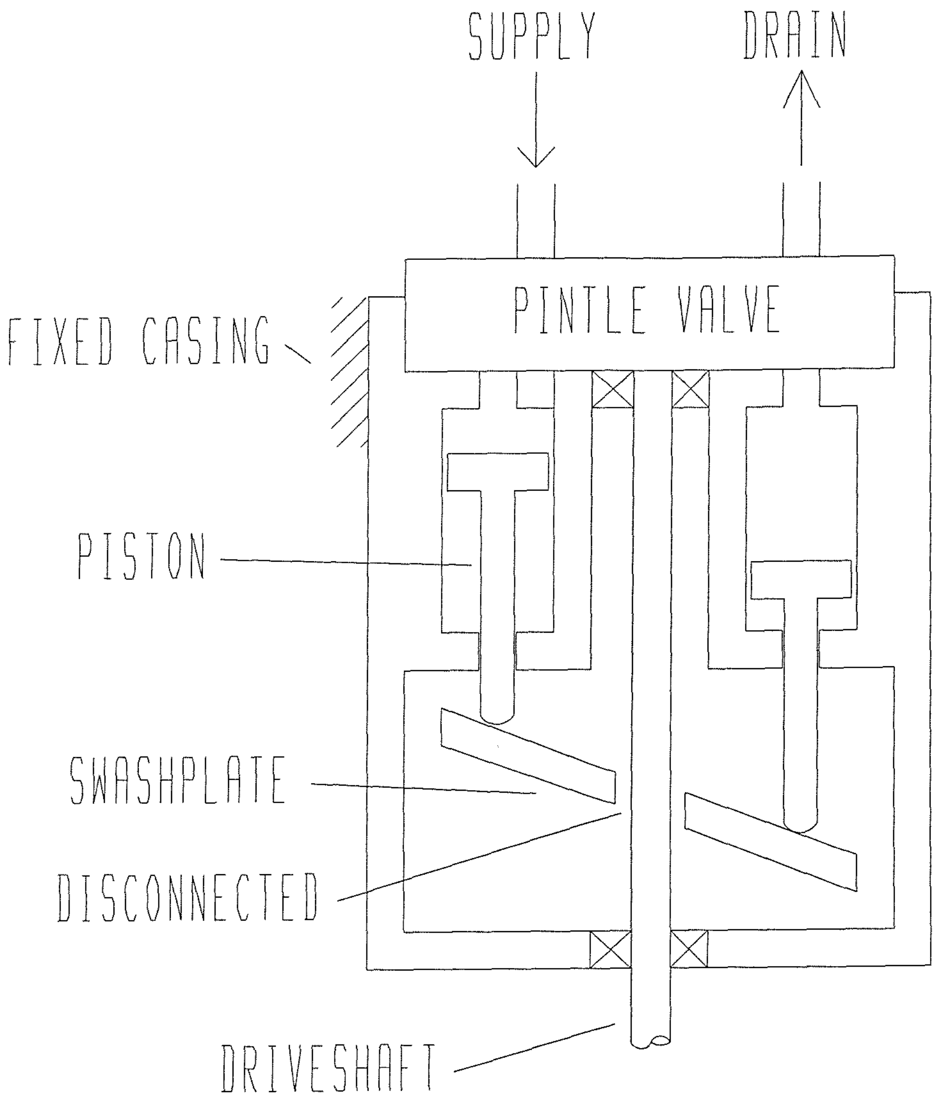
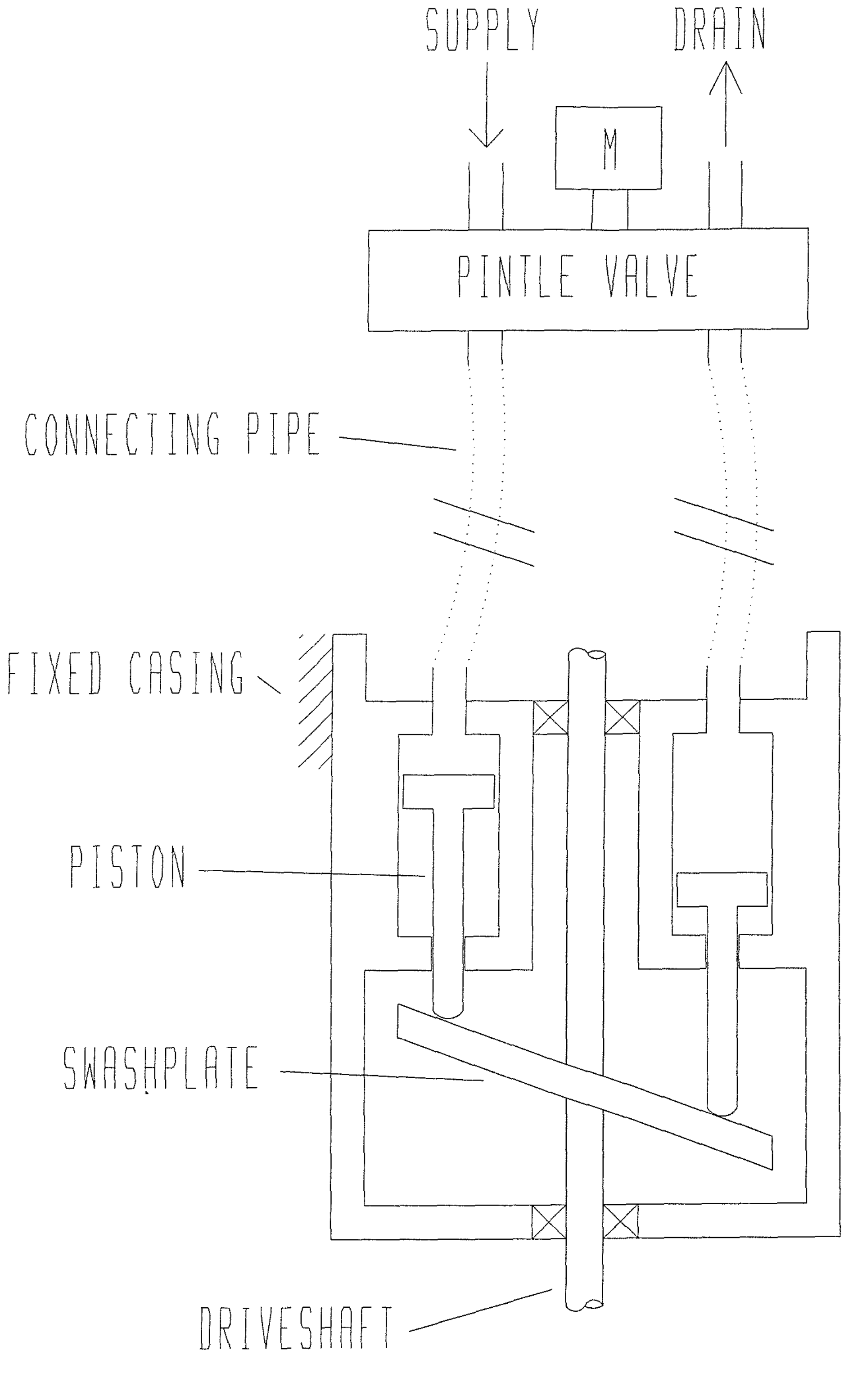


FIGURE 4.6 : THE SYNCHRONOUS PISTON MOTOR.



pintle stationary and when piston and pintle are nearly in-line, the pintle could be driven allowing fluid to pass into the piston, thereby continuing the rotary motion.

This inability to start, unless the rotor is spun up to the same speed as the pintle, is very similar to the starting problems encountered by a.c. synchronous motors. (In this case, the d.c. rotor field is subjected to a rapidly rotating stator field and cannot accelerate in time to allow the d.c. field to "lock on" to the stator field.) Furthermore, if the pintle is driven at a constant speed and the drive-shaft subjected to a torque greater than it can deliver, the swash-plate and pistons will begin to move out of phase. Once the swash-plate and pistons begin to loose synchronisation, the torque generated, begins to decrease. Then, much like a synchronous motor, once past this maximum slip point, the pistons and pintle will be so far out of phase that a stall is inevitable. A final observation, which leads to the next analogy, is that if the pintle is separated from the pistons and the fluid is considered ideal, the distance between the pintle and pistons can vary from a few millimetres to a much larger distance. The consequence of this observation is that the pintle is acting as a remote supply providing a reciprocating power source. An ideal graph of fluid flow against time, may well look like a d.c. square wave. A real graph may tend to be much less stepped, and more "sinusoidal", caused by the gradual opening and closing of the pintle ports. The flow will grow and decrease and not start and stop instantaneously.

Looking at this remote, reciprocating source, noting that each motor port must receive a fluid supply and a fluid exhaust, it could be likened to an a.c. supply. The machine, can therefore be deemed to be a.c. in nature. The power source (on the input side of the pintle) is genuinely d.c., but the pintle ports it, turning it into an a.c. signal. Thus, although the machine has a "d.c." source and would be termed a "d.c." machine, its action is primarily a.c. The next logical step would be to look at a d.c. electrical machine and to establish if it has a similar d.c. to a.c. conversion in its action.

4.2.4 **Commutation and phasing**

Considering a simple permanent magnet d.c. motor. The d.c. supply is fed to the rotor windings by the commutator. The commutator causes the current in the rotor windings to be reversed to allow the rotor to continue to produce torque in the same direction as it passes through the series of north and south poles created by the field winding (REF. 8). Thus, although the supply into the motor is d.c., the current in the rotor windings will "alternate". The current will look very much like a square wave as opposed to a sinusoid. Figure 4.7 shows the current in a coil with time, each oscillation representing a commutation. There are clear similarities between the actions of the pintle valve and the commutator. However, the nature of the modification necessary to the construction of the d.c. motor to produce a "synchronous" motor is not immediately apparent. Breaking the flow of power between the commutator and coils is not as easy as breaking the link between the pintle and pistons. The reason being that the commutator and coils are directly connected, the pintle and pistons are not connected physically, they are ported and are joined by only a fluid flow. To detach the commutator from the coils in a similar fashion to the piston motor, a great deal of redesign would have to be done, a possible solution is shown in Figure 4.8. The idea is possible, but very impractical and expensive because two commutators would be used along with many times more brushes than usual. Another way to achieve a synchronous d.c. motor without having to change the construction greatly, would be to create an "inverted" synchronous motor. If the field magnets were driven round as in Figure 4.9, the rotor may not be able to accelerate fast enough to allow the rotor magnetic fields and the field fields to complement each other. No torque will be produced and hence, no rotary mechanical power is generated. However, if the rotor were accelerated to the field speed by some means, the motor would function as it would do normally. This kind of inversion is the basis of the construction of the "brushless d.c." type motor. The rotor has permanent magnets and the stator field is "rotated" by a fast switching controller (REF 43).

The above examples show exactly how analogies have been used to help suggest possible design developments which have been confirmed by noting practical examples. It should be pointed out that the development of the brushless d.c. motor is more to do with the development of electronic controllers than the analogy hypothesis developed in this thesis.

4.2.5 **Analogies between Losses in Motors**

Another area where analogies were investigated in depth was concerned with the equivalent

FIGURE 4.7 : VARIATION OF ARMATURE CURRENT WITH TIME.

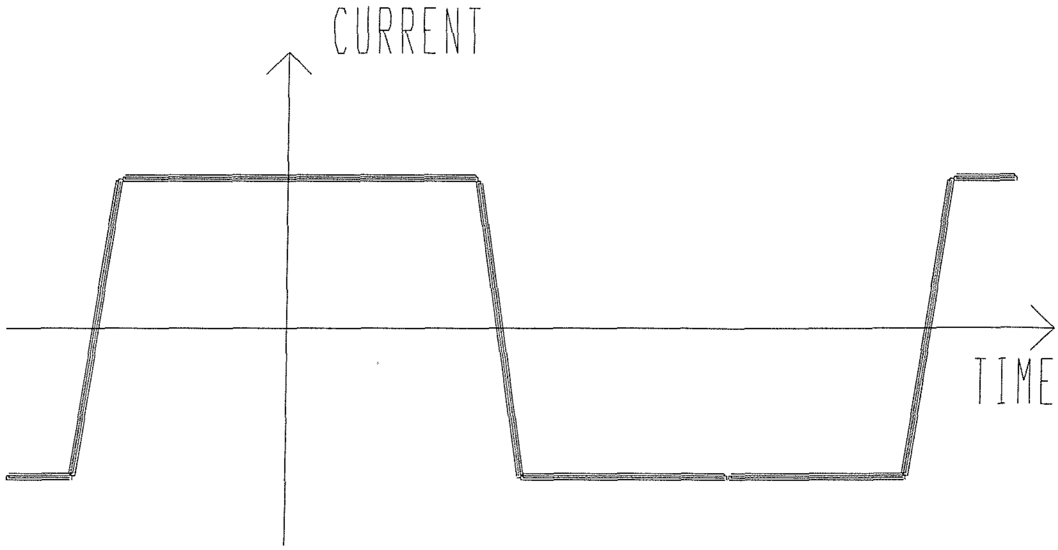


FIGURE 4.8 : "SYNCHRONOUS" DC MOTOR USING REMOTE COMMUTATION.

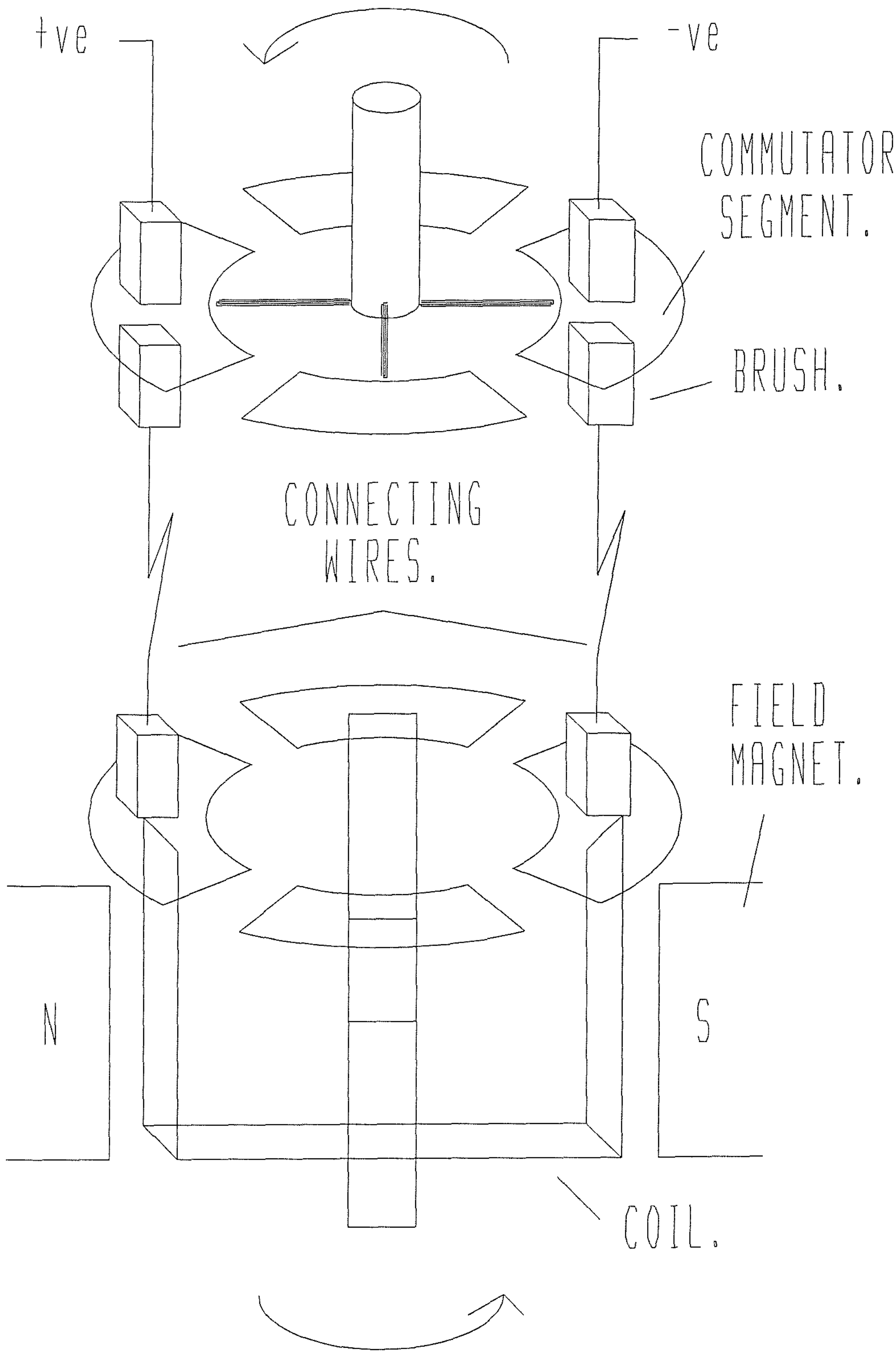
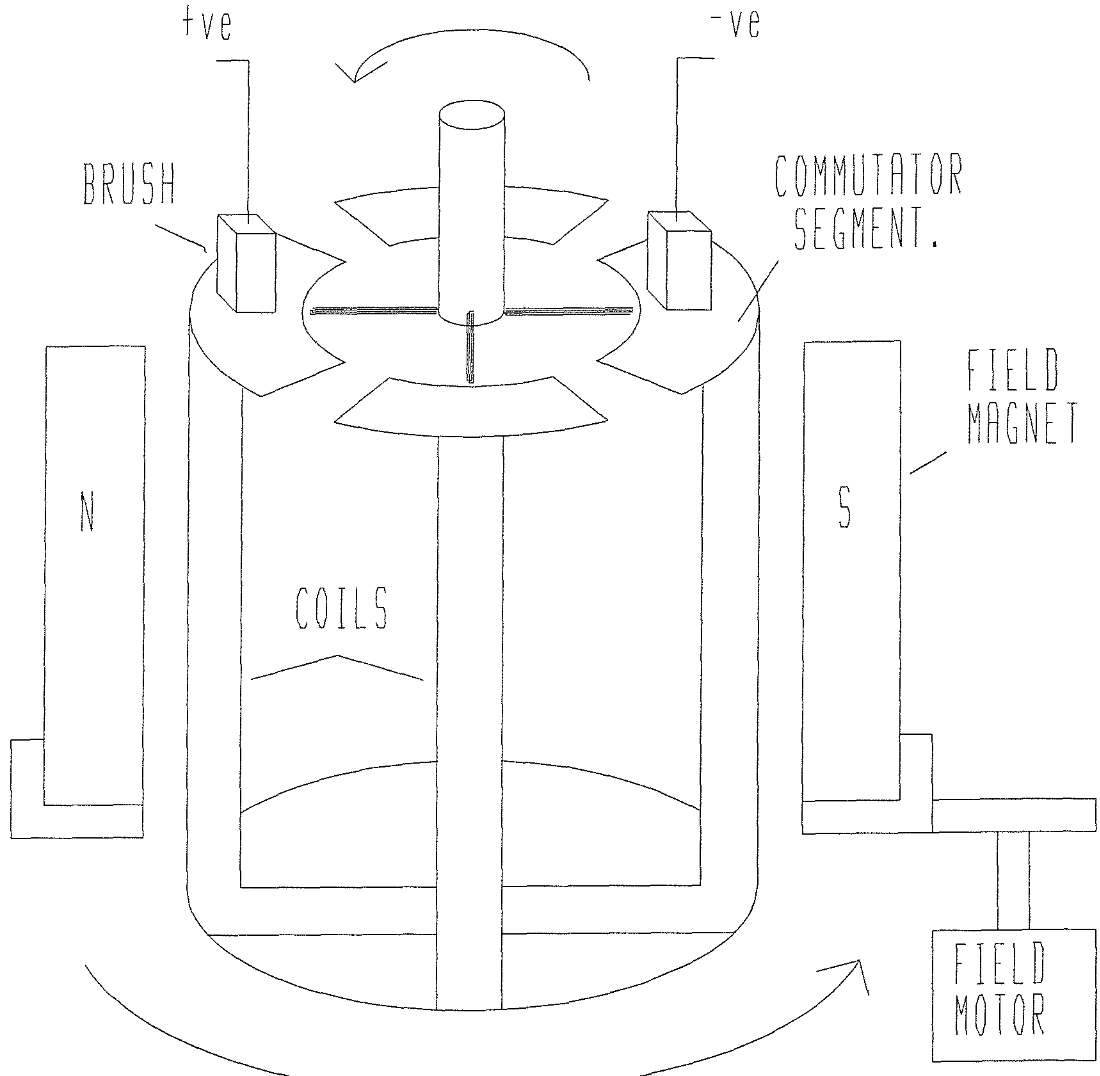


FIGURE 4.9 : "SYNCHRONOUS" DC MOTOR USING DRIVEN FIELD MAGNETS.



losses in fluidic and electrical machines. It must be highlighted here that electrical machines have an intermediate mechanism for producing mechanical torque. Magnetic flux is the phenomena that generates the mechanical forces in electrical machines. With fluid machines, the supply components are directly responsible for the mechanical power output. Thus, flux losses or any mechanisms affecting the ability of the magnetic flux produced in electrical machines to work efficiently will be difficult to compare in fluid machines.

Looking first at electrical machines and using the a.c. / d.c. terminology when appropriate, the classically quoted losses are (REFs. 6, 8, 11);

Armature losses - or copper losses caused by the resistance of the copper armature, or rotor, windings.

The stator wildings of both a.c. and d.c. machines will also have copper losses.

Hysteresis losses - caused by the residual magnetism of the armature / rotor core, requiring additional current to demagnetise the core, before it is remagnetized in the opposite sense.

Eddy current losses - currents which are induced in the magnetic core will cause magnetic fields, which will hinder the production of the core magnetic field.

Hysteresis and eddy current losses are known as core or iron losses because they are peculiar to the ferrous material used to create the armature / rotor field.

Flux loss - the stator and armature fields will not interact perfectly. Some of the generated flux will play no part in producing torque.

Commutation losses - these are losses caused by the inefficient transfer of current from the brushes to the commutator segments / slip rings.

All of the above losses can be grouped under the broad heading "excitation losses". They are the losses associated with creating a useful magnetic field.

Mechanical losses - friction and windage are the main problems.

In fluid machinery, the common losses are (REFs. 9, 17, 18) :

Volumetric or flow losses - these are mainly pressure dependent and can be identified as ;

i) Leakage between moving parts ie piston and bore, port plate, slipper pads etc.
ii) Fluid compressibility.

Mechanical losses - these are a function of the motor speed ;
i) Viscous drag, this causes friction between the moving parts.
ii) Bearing friction.
iii) Churning.
iv) Inlet pressure drop, this is mainly a function of porting geometry.

4.2.6 **Comparison of the losses and identification of analogies**

Perhaps the best way to consider the large number of losses listed above would be to start broadly and then to move to specific examples and analogies. In all cases, the electrical characteristics will be considered first and fluid analogues suggested.

The broad electrical characteristic; "excitation losses" can be defined as the electrical power needed to create enough flux density for the magnetic field or fields produced to generate enough torque to overcome the mechanical resistances in the system. These are primarily the rotor inertia and the bearing stiction. Fluid motors also have moving parts such as rotors and pistons. The use of seals and sliding parts in fluid machines would cause the perceived rotor inertia and bearing stiction to be greater than that for a similar sized electric machine. The excitation power in both these cases does not produce any useful torque, to produce useful torque, more supply power would be needed. Thus, in all cases, the motor can never be perfectly efficient because there will always be a power requirement to energise the motor.

It would now be useful to trace the power through an electrical machine in order to identify where the power is lost and hence find analogies in hydraulic machines. Considering firstly the machines that require power to be routed to a rotating part. These machines will have some kind of slip ring or commutation element. Both of these mechanisms require an electrical connection to be made between two conductors in intimate contact. By its very nature, this will be an area of higher resistance and hence there will be a voltage drop. This voltage drop is not a voltage drop caused by the working elements of the machine and is hence a loss. There is an immediate analogy with the pressure loss caused by the porting geometries of fluid machines. At this intuitive level, the commutation loss and the porting loss have similar physical effects ; a loss of voltage / pressure at the interface between the supply and the drive. The fluid pressure loss can now be studied more carefully in an attempt to provide some corroborative analysis. There are several ways of representing resistance in fluid systems (REF 5). The porting loss is one associated with changes in geometry caused by piping connections and fluid porting. It is most akin to the head losses caused by fitments in fluid pipelines (REF 5). The analysis of these fitments quantifies the head loss as a factor that relates an equivalent of pipe of the same bore. Pipe roughness is a major cause of head loss in pipes (REF 27). Thus the longer the pipe the greater the head loss. It is clear that this factor is analogous to electrical resistance. Since an unwanted voltage drop in an electric circuit can also be described as a high resistance, the porting geometry pressure loss is directly analogous to the voltage drop at the slip ring or commutator in the electrical machine.

Electrical machines produce their useful power by producing magnetic fields. Magnetic fields are caused by current flow in a conductor. Whenever a current flows through a conductor, heat is generated. Since electrical machines do not use this heat to produce torque, the heat is a loss. Thus windings of electrical machines have an inherent loss. Windings are often made of copper and the loss is deemed the "copper loss". The copper loss is the power passing through the winding. Fluid machines do not perform a power conversion to generate useful torque, power is generated directly from the fluid. There is a pressure head loss in addition to the mechanical and porting losses already mentioned. This is the pressure drop within the fluid machine caused by the fluid simply having to flow though the working parts.

Intuitively, churning in fluid motors and windage in electric motors may seem analogous.

Careful consideration of their propagation and effects, will show that they are not analogous. The windage loss is caused by the rotor of an electric motor moving through the air within the rotor casing. This results in a retarding torque on the rotor, tending to slow it. Churning in fluid motors is the result of the fluid flow swirling whilst moving through the motor. It produces heating through friction. The churning is a loss because working pressure must be used to overcome the friction causing the heating. This pressure drop across the fluid motor has already been considered. Therefore churning must be considered part of the losses analogous to copper losses. Therefore, windage in electrical machines has no direct analogy in fluid machines.

As well as the pressure losses through the fluid machine, there are also flow losses caused by leakage past seals etc. Fluid leakage will cause an additional drop in the available working pressure thereby reducing the useful output power. When looking at leakage paths, the model of a constriction or small orifice in the flow path in parallel to the main flow path is the most suitable. When the leakage path is short, the pressure drop across the orifice is a function of flow rate and area. (REF 5). The analysis for this model yields a simple equation, analogous in form to: $V = I R$, where voltage is analogous to pressure drop and current is analogous to flow rate. The analogous value of resistance is a function of orifice area. The smaller the orifice, the larger the resistance. This is in keeping with the practicalities of fluid machines. When seals are working correctly, or a slight controlled leakage is used, the "orifice resistance" is very high. Thus, only a little fluid is lost and the power conversion is maximised. When seals leak, the "orifice resistance" is very low, fluid leaks, working pressure is lost and output power is reduced.

It is very difficult to say exactly which of the resistive losses in electrical machines the orifice loss represents. It is a small loss in comparison to the losses which will occur at the porting of the fluid machine. The leakage paths are a genuine loss and will affect the performance of the machine. If fluid is not used to produce power but instead leaks past the pistons, vane tips etc., there will be a drop in efficiency. This loss is indicated by the casing drain in fluid machines. Perhaps the small resistive losses in the electrical connections, excluding the brush contact loss, could be modelled by the orifice losses. These losses will all be high resistance losses, only very small currents will flow.

Magnetic hysteresis is dependent on the composition of the magnetic core, the supply frequency and the magnetic flux density (REF 6, 26). Hysteresis is considered a liability because it perpetuates a magnetic field after the magnetising current has been removed. Thus, in order to remagnetize the core in the opposite sense, additional power must be used to firstly demagnetise the core. This residual magnetism is essentially an inertial type effect. It could perhaps be described as "magnetic inertia". The best way to attempt to find an analogy in fluid machines is to consider if it is possible for the fluid power to perpetuate once the supply has been stopped. In vane motors, it may be conceivable that the fluid momentum will cause the fluid in the casing to continue to move, thus pushing the vanes round. This would be a very small effect because the inertia of the rotor would be very much more significant and bearing fiction etc. would also be large in comparison. Also, the force on the rotor would not persist like the residual magnetism caused by hysteresis. A more interesting effect will occur in a piston motor with a compressible fluid. Any work done in compressing the fluid will not be recovered fully when the piston extends. This would be analogous to the loss associated with having to demagnetise the iron core of an electro-magnetic machine before re-magnetising it in the opposite sense. This represents only part of the full analogy to hysteresis because there is no perpetuation of the motive force. If the motor was stopped with the compressed fluid trapped in the bores, energy would be available to cause rotary motion. Depending on the porting geometry, the stored energy will tend to brake or perpetuate the rotary motion. These residual forces can be deemed "hysteresis".

No analogy has been found for eddy current loss in fluid machines. Eddy currents are caused by relative motion of a conductor and a magnetic field, there appears no direct analogy with phenomena in fluid machines.

Table 4.2 summarises the analogies developed above.

TABLE 4.2 : ELECTRICAL / FLUID LOSS ANALOGIES.

Armature Exitation (resistive loss).	Leakage Losses
Hysteresis	Inertia / Compressibilty
Eddy Current	————
Commutation / Slip ring	Inlet / Outlet porting
Friction	Bearings / Viscous drag
Windage	————
————	Churning

4.2.7 Number of Phases

Analogies can also be developed by looking at other characteristics extracted from manufacturers' catalogues. The number of phases is often quoted for a.c. electric motors. As mentioned above, the d.c. commutator allows an alternating d.c. voltage to be supplied to the rotor windings. The relationship between adjacent commutator segments is that they are in contact with the brushes at slightly differing times, allowing the appropriate conductors to carry current as they pass under the stator poles. They are essentially phasing the supply. This is more easily understood if one were to invert the motor's action. Imagine the rotor locked and the stator (or casing) rotating. What would the field magnets "see" ?. They would see a series of rotor conductors having a current passed through them one after the other, the currents would flow in an identical fashion, but lag each other as they approached the field magnet. They can be said to be phased. The conclusion is that a d.c. commutator has as many phases as half the number of commutation segments (a segment pair will be in contact with the positive and negative brushes, therefore, one phase equals two segments.).

Using the pintle / commutator analogy previously developed, the analogy for phases in fluid motors can be considered. The pintle valve is a feature of piston motors. Each piston, in turn, is supplied with fluid and then exhausted. The pistons are therefore phased to extend sequentially in such a fashion as to perpetuate rotary motion. The phases can therefore be said to be the number of pistons. If the motor is set up with opposed piston pairs, one phase would be one pair. The phase angle will be the angle between the pistons. Interestingly, both fluid and d.c. motors have more phases than a.c. motors. The a.c. systems use a maximum of only three phases, why is this possible? The basic reason is that a.c. systems have a high frequency, so the time between the phases is small. Thus, a good approximation to a steadily changing magnetic field is possible. In d.c. and fluidic machines, the nature of the frequency of operation is very different. In fluid machines, the frequency is basically flow dependent, the more flow, the faster the pistons reciprocate. In d.c. machines the frequency is dependent on the speed of rotation. In both of these cases, the starting frequency will therefore be low. Hence, to provide a "smoothly rotating motive force", the number of phases must be increased, this reduces the phase angle. Reducing the phase angle means that the distance between the phase peaks is reduced. Thus, the rotor will travel a shorter distance to receive its motive force. The motor will be able to start smoothly because the power will not appear to be supplied in discrete bursts. The forces on the rotor will be a series of overlapping impulses making a smooth progression of force on the rotor. Also, because of the smaller phase angle the mean force on the rotor will be higher giving good starting characteristics.

The number of poles used in the construction of the motor have an important effect on the performance. In the case of synchronous motors, as the number of rotor poles increases, the synchronous speed decreases. In d.c. motors, an increase in the field poles reduces the final speed of rotation. In both cases rotation is caused by the interaction of a permanent magnetic field with a transient one.

When considering the derivation of fluid analogies to poles, it should be noted that fluid motors do not work by the same mechanisms as electric machines. Electric motors rely on the interaction of two magnetic fields, fluid motors use the pressure and flow of what is essentially one fluid power source. It is recognised that d.c. motors use only one supply but split it to supply the armature and field windings. Fluid motors do not work in this fashion. Fluid flows do not interact to produce rotary power, they impinge on solid parts (rotors, pistons etc.) of the motor. The fluid pressure / momentum causes a force to act on the motor parts resulting in motion. Thus, it is difficult to find a fluid equivalent of magnetic poles, particularly the stationary field poles of d.c. machines. However, considering, exactly what the magnetic poles do, a reasonable fluid model can be described. Using a four pole d.c. machine as an example, consider the relationships between the poles. Magnetic flux flows from north to south, south to north around the machine. This is the stationary magnetic field with which the transient field in the rotor interacts to produce a torque. Could the fluid flow in a balanced vane motor be considered to act in the same way, as it flows from high to low pressure ports?. This type of fluid motor can then be said to have four poles. In the case of piston type motors, if the idea of inlet / outlet ports is pursued, the rotor will have as many pairs of poles as pairs of pistons extending / contracting. A similar idea was expressed above when considering phasing.

An interesting observation concerning the inlet / outlet pairs is that piston motors tend to have odd numbers of pistons so as to achieve minimum ripple (REF 17). Thus, the pistons cannot be set up as opposed pairs. The identification of poles then becomes less clear. The number of ports in the pintle valve plate may be a more suitable indication.

4.2.8 **Free Speed**

Very high speed, pneumatic motors, have a theoretical maximum speed called the "free speed". It is a theoretical maximum because the motor produces no useful torque at this speed. The motor is run at full pressure with no load thus allowing the maximum possible flow rate and hence, the maximum or free speed to be achieved. When required to produce torque, the motor will run at a speed below this because of the reduced flow rate caused by the head loss across the motor needed to produce torque. Induction motors also have such a theoretical maximum speed, namely the synchronous speed. It is an unachievable target for the rotor because the rotor bars must cut the rotating flux to produce torque, if the rotor bars and the flux were to be moving at the same speed, no flux would be cut, and no torque produced. This concept, known as "slip" was discussed in detail in Section 4.2.2. The free speed of a d.c. machine is dependent on the type of construction used. Generally the motor speed is limited by the supply voltage and the variation in armature current. (REF 6). Shunt wound machines cause the armature current variation to be small and therefore the speed is determined by the supply voltage which is usually held constant. The free speed of the motor is therefore very close to its operating speed. In series machines, the maximum speed also occurs when the machine is operating under no load. Thus, free speed has significance for fluid, a.c. induction and d.c. series wound motors. Thus, the relative difference between low speed, rated and free speed will give an indication of the motor type and hence its torque characteristic. This is an important characteristic differentiation which helps with selection. See Chapter 9.

4.2.9 **Static Friction Torque**

Smaller electric motors often have a value for static friction torque quoted. This parameter has a very vague definition, it attempts to account for all the mechanical losses in the bearings, the drag caused by commutators and perhaps magnetic effects that may stop the rotor from spinning freely. Fluid motors do not use this characteristic but suffer from this phenomenon far more severely, particularly where moving seals are used (vane tip to casing, pistons in bores). A measure of static friction torque is however provided by a parameter called the "breakaway pressure". It is the pressure needed to turn the motor over with no load. It is a fraction of the rated pressure. Given that output torque and supply pressure are proportional, it should be possible to deduce a value of static friction torque from the full load torque by interpolation. This may seem an unimportant parameter, but it interestingly it has been highlighted in Section 4.2.5 above as part of the analysis of losses in fluid machines.

4.2.10 **Coil Capacitance**

Whenever insulated conductors carrying a current are placed together, a certain amount of capacitance will occur. (This is the definition of the parallel plate capacitor - REF 26.) Thus, whenever a change of state occurs within the device containing the conductors, a capacitive effect will ensue. Windings in electrical machines are obvious sources of unwanted capacitive effects. Motors having an a.c. supply will be affected more by the capacitive effects because the supply components will be affected every cycle. A similar effect will be present within fluid motors as the working fluid is compressed. If the fluid is highly compressible, there will be a loss as extra work must be done on the fluid to compress it before any work can be done by the fluid. Heat will also be generated, this serves no useful purpose. This capacitive type behaviour is considered a loss because, although the potential for work remains available eg the compressed fluid, it cannot be recovered usefully. It is therefore a true loss. This behaviour was predicted above in Section 4.2.6.

4.3 **Creation of the Lists for Selection**

With the extended lists complete and the analogies investigated as far as possible, an attempt can be made to distil the best selection criteria for use with a computer based retrieval. This will mean the minimum amount of information, using discrete values which are readily obtainable and practicable. A useful segregation has already been made by breaking the extended lists into input, output, dimensional and other information. To assist with compiling the lists it would be wise to firstly consider the relative importance of these four categories.

A motor is a source of rotary mechanical power, its job is to produce a certain torque and speed. It may have to do this within other constraints such as space or environment. If it cannot drive the system it is connected to, it cannot be selected despite meeting the latter criteria. The conclusion of this reasoning is that the output characteristics are of premier importance (REF 19).

The type of supply available will also influence motor selection. If only a particular power source were available and the motor selected to use this supply proved to be too large for the available space, what would happen ?. Firstly, if the motor was to drive an already designed and proven system, it would be unlikely that the system would be modified to allow a smaller motor to drive it. The space available for the motor would be maximised, this may require a certain amount of physical redesign. If these alterations failed, perhaps methods of providing alternative power supplies would be considered if other motor types were more suitable. Thus, output characteristics, then input and dimensional information is the suggested order of parameters in the selection list.

Consultation with TI concluded that dimensional information was vital. There are two reasons for this, one is logical, the other "traditional". Clearly, the size of the motor will affect the way it is integrated into the system (mounting type etc.) and also, designers are always much happier when dimensioned drawings are available. The size of a motor gives an idea of cost and power. Input data is something that can be neglected until installation is being considered. Perhaps, simply limiting the input data to the input "type", i.e. the power domain from which the motor receives its motive power would be sufficient.

Thus, the general format of the selection list will be mainly output characteristics, some dimensional information and an indication of the power source. Remembering that the retrieval system is intended to run in parallel with the TI microfilm system, (Chapter 2), this smaller set of characteristics should be enough to pin-point exact motors within the microfilm system from where the details can be studied. Also, as a computer based system is to be used the characteristics must lend themselves easily to a computer based retrieval.

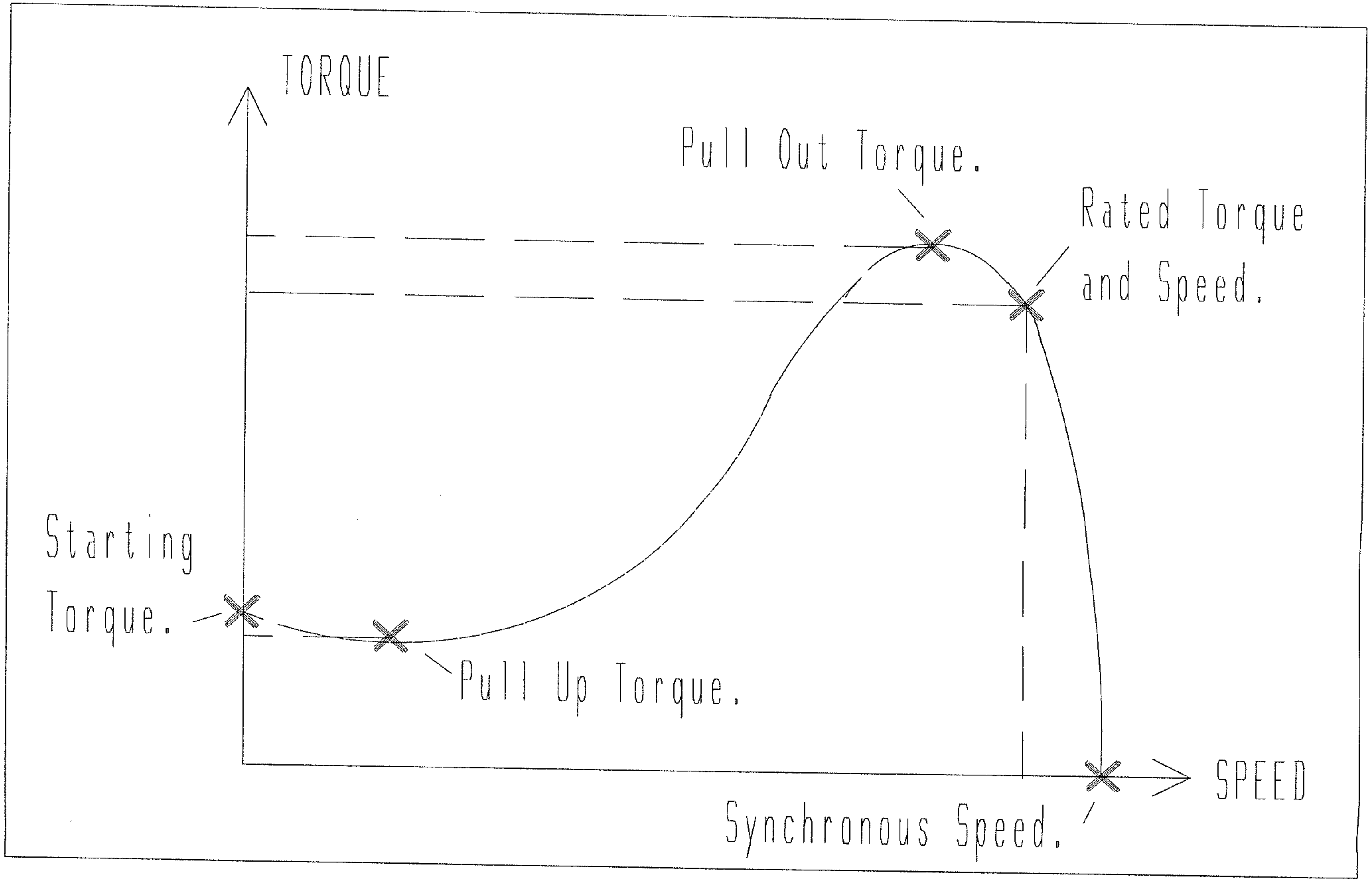
To this end, the characteristics should be simple and discrete. The use of graphical information in the form of performance curves etc., is to be avoided, because of the problems with capturing this information, as well as presenting it to the user in an informative way.

4.3.1 **Handling Graphical Information**

In the case of motors, the most common graphical information is the torque / speed curve. In certain fluid motors, this graph does not appear in this form, but results from the combination of flow / speed curves and pressure / torque curves. The study of a.c. electric motors highlighted that various points on a typical a.c. motor characteristic could be identified. Each point defines a specific state of the motor along its torque speed curve. To be able to construct an a.c. motor performance curve, the designer would need the following points on the curve; starting torque, pull up torque, pull out torque, rated torque, rated speed. These are values that are defined in standards and are discussed in Appendix A. Thus, the torque / speed curve for any a.c. motor can be drawn by simply holding five discrete values. Figure 4.10 illustrates these points using a typical a.c. motor performance characteristic.

Considering d.c. motors and fluid motors, the starting torque, rated torque and the rated speed should all be available for selection purposes. Depending on the shape of the torque / speed curve, the pull up torque and pull out torque may be applicable to these motors. Fluid and d.c. motors are noted for their controllability over a large part of their operating range. The performance characteristic varied is usually speed, torque is often a constant (ie most fluid machines and torque based, shunt wound d.c. motors). Thus an idea of the motor's speed range is also important. The speed range is usually a region around the rated speed where the motor can be controlled without running inefficiently. It is rare for a motor to be perfectly controllable between zero RPM and its free speed. Thus, an indication of the minimum and maximum speeds the motor can run at successfully would be a sufficient representation of the speed range of the motor.

FIGURE 4.1Ø : Typical Torque / Speed Curve for Induction / Synchronous Motors.



4.3.2 **Summary of the Performance Characteristics**

Using the parameters discussed above, the torque / speed curves for all common motor types can be defined. These parameters are commonly quoted by suppliers or can be gleaned from torque/ speed curves. A parameter that links all of these characteristics is the output power of the motor. As explained in Appendix A4.1.1 it is the product of rated torque and speed. It gives a very good 'feel' for the capabilities of the motor without necessarily being specific about the relationships between the performance characteristics. It is a parameter that designers try to minimised to keep costs down. Rated power can therefore be added to the list of performance characteristics identified so far :

1. Rated Power
2. Rated Torque
3. Rated Speed
4. Starting Torque
5. Pull Up Torque
6. Pull Out Torque
7. Maximum Speed
8. Minimum Speed

Designers are familiar with these characteristics and will have few problems exploiting them in a search that will encompass many different types of motor.

4.3.3 **Dimensional information**

Representing the large amount of dimensional data provided for engineering components, in a simple and effective way is something that required a great deal of careful consideration. Most suppliers do not provide unique drawings for each motor but a parametric representation which allows one drawing to depict a family of motors with similar construction. The drawings provide detailed information about shaft sizes, keyways etc. which are important for allowing the motor to be connected into the system. Noting that this information would be available through the TI microfilm, for retrieval purposes it would be ideal to limit this information. The aim is to limit dimensional data, whilst retaining the intuitive value of "how big" the motor is. The most convenient way of doing this would be to extract the maximum dimensions of the motor, and use them to define a 'box' whose boundaries the motor would not exceed. This technique would tend to oversize the motor because they are generally cylindrical in nature. Also, protrusions such as terminal boxes and lifting rings, which are movable or detachable, would increase the overall dimensions in a misleading way.

The dimension which includes the motor shaft would definitely mislead the designer as to the size of the motor casing if it was unclear as to whether the length of the shaft was included in the dimension. However, once aware of the limits on these dimensions, the user should have no problems. Defining the three largest dimensions in the x, y and z planes will preserve the notion of performance to size ratio, which serves a useful comparison between the motor types.

4.3.4 **Mounting arrangements**

Most large manufacturers provide a variety of mounting types for any one of their motors and keep them in stock. Less common mounting types may have to be ordered specially. Since most manufacturers tend to promote their stock items in catalogues and users are usually searching for the cheapest, readily available component that can be integrated into a design, specifying the mounting type is important. The most common mounting arrangements are foot, flange and face shown in Figure 4.11. Less common arrangements such as rod, skirt and other mountings are included in the term "others" shown in Figure 4.11. The common mounting arrangements are covered very thoroughly in IEC 34-7 (REF 10). A section is shown in Figure 4.12. Figures 4.11-4.12 have been removed to Appendix F.

The comment on the extract in Figure 4.12 is that it is a small section of a larger set of examples dealing with every type of mounting arrangement in a variety of common orientations. Presenting the user with the standard and allowing a choice to be made from over one hundred permutations would be wasteful from a computer memory viewpoint and very annoying for a user to search through a series of very similar drawings until a match is found. Attempting to cut down the number of permutations will only lead to inconsistencies and therefore confusion for the user. It is possible to break the mount type problem down into a series of simple questions one of which would be; "what type of mount is required ?", and then supplementary questions determining orientation. With this information, the software could then generate the IEC mounting code. This is possible, a similar process is described in detail in Appendix G - the gearbox shaft configuration code. It was felt that such a relatively complex function for such a simple characteristic was unnecessary. Thus a simple representation of the mounting types available without an indication of orientation is sufficient. This was born out in the field trials - see Chapter 9.

4.3.5 **Further Considerations**

As previously mentioned, some fluid motors are not reversible. Most electric motors are reversible. It is something that is often assumed with no real thought as to how it will be achieved. "Reversibility" is therefore included in the selection list to provoke thought and to highlight that certain motor types are non reversible.

"Motor type" is a parameter that identifies the power domain from which the motor gains its supply. It is hoped that in most cases the designer will not specify this parameter because it will limit the breadth of the retrieval by cutting out the selection of motors from other domains meeting the specification. This may be necessary where the supply is limited, and it would be uneconomical to install power conversion devices. It would therefore be short sighted for a retrieval system not to offer this option, and it would be very frustrating for a user to inspect retrieved records that do not match the available power supply.

The final parameter used in the selection list is "company name", this is the name of the supplier of the motor. The reason for including this parameter in a system that attempts to be objective, is that it is an option that is often required by users. Designers may be constrained to use a certain supplier's motors because of company policy. If the designer was to specify a company purely through personal preference, it would be wasteful of a fast, broad based retrieval system. The system would only point out motors in a catalogue the designer probably already had.

The selection list takes no account of cost, absolute or relative. This is because most catalogues come without price lists and TI do not provide cost information. TI follow this policy because prices tend to vary and change with the size and type of order. Most suppliers prefer to quote a price when an approach is made by a customer. If pricing details were made available through the retrieval system, care must be taken to ensure the information was up to date and accurate for the type of purchase to be made. The use of price, although it is very important in the final decision making process, would constrict the selection process, removing the breadth it was intended to bring to motor selection.

A comment should be made at this stage regarding the selection procedures provided by motor manufacturers. These procedures help the user evaluate some of the basic performance characteristics quoted above. They also help the user convert design data into selection data. Design data is the specification defined by the system to be powered that the motor must meet. The selection data may be the design data or, the design data modified by imposing a scaling factor upon it, depending on the type of service the motor must provide. This scaling factor is called the "service" or "duty" factor. This factor generally oversizes the motor by increasing the design power. The duty factor is an empirical value and can be easily explained. Consider a motor running continuously, driving a light load with a low starting torque, a generator say. The motor driving this system will have less demands put on it, than a motor driving a conveyer belt in a warehouse. Here, starting torques will be high, there may be overloads for short times, and also impulsive loads. Thus, not only must the motor be correctly selected, but some account must be taken of the "stress" the motor will encounter. A list of typical applications and service factors is given in Table 4.3. in Appendix F.

4.4 **The Selection List.**

The final list used for the specification of motors is given in Table 4.4. The characteristics are listed in preferred order of importance for selection. Consultation with TI resulted in a revised order caused mainly by the loss in priority of the pull up and pull out torques. The TI view was that these characteristics could not be easily identified for motor types other than a.c. electric and were not used by designers when selecting non a.c. motors.

TABLE 4.4: MOTOR INDEXING PARAMETERS.

INDEXING PARAMETER	VALUE	UNITS	WEIGHTING
Rated Power		kW	10
Rated Torque		Nm	10
Rated Speed		RPM	10
Starting Torque		Nm	9
Maximum Speed		RPM	8
Minimum Speed		RPM	8
Maximum Length		mm	6
Maximum Height		mm	6
Maximum Breadth		mm	6
Mounting Type		--	5
Motor Type		--	2
Company Name		--	1
Pull Up Torque		Nm	1
Pull Out Torque		Nm	1
Reversibity		--	1
Motor Frame Code			
Microfilm Number			

The University agreed with this reorganisation noting that the characteristics were still available on the list. To help TI speed up the indexing process each characteristic was given a weighting. The weighting is an attempt to help the TI staff make decisions about how important, and therefore how much time should be spent extracting the particular piece of information from the supplier's catalogue. Data capture time is a limiting factor for a small company like TI attempting to create databases potentially as large as hundreds of thousands. An estimation of data capture times is given in Chapter 9. Thus, a low weighted parameter that is difficult to extract is ignored. Some example motors are shown in Figures 4.13 - 4.17. Only Figure 4.13 is shown below, Figures 4.14 - 4.17 have been removed to Appendix F.

Thus, information contained on these indexing sheets, could then be typed into the database, using the software set up by the University. The software is described in Chapter 9.

The suitability of the characteristics used for selection is discussed in Chapter 9.

FIGURE 4.13: EXAMPLE OF INDEXED MOTOR.

Rated Power = 2.5 kW
Rated Torque = 26 Nm
Rated Speed = 915 RPM
Starting Torque = 54 Nm
Maximum Speed = 1000 RPM
Minimum Speed = 732 RPM
Maximum Length = 435 mm
Maximum Height = 261 mm
Maximum Width = 224 mm
Mount Type = F
Motor Type = A
Company Name = British Brown Boveri
Pull Up Torque =
Pull Out Torque =

Motor Frame Code = MEUC 112M6
Microfilm Number = 3673 - 1144

CHAPTER 5.

STUDY OF GEARBOXES.

5.1 Introduction

The gearbox was studied because of it is the most common element connected to a motor. It is very important to stress that when referring to a "gearbox", the term "impedance transformer" should be borne in mind (REF. 4). The definition of an impedance transformer is a device that changes the system power components by a given value whilst conserving power. In the rotary mechanical domain the system components are torque and speed, the scaling factor is the gearbox ratio. However, this definition is not only valid for the entity "gearboxes" but is applicable to all devices which satisfy the definition. Hence, belt and chain drives must also be considered. The effects of this statement are discussed below in Section 5.2.

The gearbox was the subject of three final year student projects run at the University of Southampton (REFs. 12, 13, 14). Thus, a good deal of background thinking had been done on the subject. The main problem area was a method for accurately defining the positions of the gearbox shafts so that the user, data capture system and retrieval system could describe the shafts positions quickly and unambiguously. A solution to this problem is discussed in Appendix G.

Although the three projects had looked at gearbox selection in great detail, it was felt that the whole subject should be reinvestigated using a similar method to that used to tackle motors. This method had been very successful, especially in generating interesting analogies. None of the student projects produced any analogies which were not already well documented.

5.2 Gearboxes and the Analogy Matrix, the Scope of the Work

In the column containing the transformer type elements in the analogy matrix, the "gearbox" is quoted as the rotary mechanical transformer. As mentioned above, it is not the only rotary mechanical device that satisfies this description. Belts and pulleys, chains and sprockets and gear trains will all modify the torque / speed relationships by an identifiable constant. The difference between these components and a gearbox is that they are sub-systems. They must be designed, selected and assembled before being incorporated into the system. A gearbox is an "off the shelf" item whose characteristics are predefined, the designer can only change these characteristics by obtaining another gearbox. The selection of a gearbox is therefore a very different procedure because it does not involve design.

It was recognised, however, that the results of the study of gearboxes could be put to use in defining the input / output characteristics of these sub-system assemblies. Clearly, the effects that the sub-system has on the system into which it is integrated will be identical to the effects of a gearbox. The result of this observation is that the retrieval procedure will have to branch out from a "linear" retrieval to an iterative procedure where system design is taking place. The retrieval will therefore become part of an optimisation routine where potential solutions are generated and tested for suitability. A more detailed description of this process is given in Appendix H. Optimisation of such systems has also been well researched (REFs. 1, 20). The work by Culley and Vogwell at Bath University, England, being well developed. In short, the writing of optimisation software was considered outside the scope of this project. However, the results of the research will help to define the system parameters that should be considered for any system element, discrete or sub-system.

A second area where confusion may occur is that of the combined motor gearbox units, often referred to as 'drives' by industry. The problem lies in how to classify these components in a consistent way so that they can be compared with motors or gearboxes. The various combinations are listed in Table 5.1, their common titles are included and the component type under which they should be indexed.

TABLE 5.1 : CLASSIFICATION OF MOTOR / GEARBOX ASSEMBLIES.

FIXED MOTOR	CONTROLLED MOTOR	FIXED GEARBOX	CONTROLLED G'BOX	INDEXED AS ...	
✓				MOTOR	
	✓			MOTOR	
✓		✓		MOTOR	
	✓	✓		MOTOR	
✓			✓	MOTOR	
	✓		✓	MOTOR	*
✓		✓	✓	MOTOR	**
	✓	✓	✓	MOTOR	*
		✓	✓	GEARBOX	
		✓		GEARBOX	
			✓	GEARBOX	

NOTE : * - not usually available. ** - "VARI-DRIVE" type devices.

5.2.1 **Justification of the Indexing Types Given in Table 5.1**

A "black box" approach was used. A study was made of the input / output characteristics of the units and a remark as to the probable identity of the component in the black box was made. Thus, in fundamental terms all the motor / gearbox units are seen as "domain transformers" (REF. 4). This is because power was being converted across the device. Rotary mechanical power is produced, but the input could be from any power domain. The definition of a domain transformer whose output is rotary mechanical power is the "motor". This classification is reinforced from practical considerations. Consider the motor characteristic shown in Figure 5.1. The motor has constant torque over most of its operating speed range. The effect of a reduction gearbox on this characteristic is shown in Figure 5.2. The effect of a controllable reduction gearbox is shown in Figure 5.3. In all of these cases, it is effectively the motor characteristic that changes, the gearbox serving as the torque / speed modifier. Similar arguments can be made for the other configurations.

Thus, the main area of research focused on the gearbox and its analogies with "transformer" elements in other systems. The main working analogy being the a.c. electric transformer.

5.3 **Generation of the Extended Parameter Lists**

Before considering the characteristics identified by studying manufacturer's literature, a brief analysis of the data structure used by Technical Indexes Ltd (T.I.) to define gearboxes may help enlighten the reader as to the advantages of the results of the research work.

Within the T.I. indexing catalogue, suppliers producing gearboxes which satisfy certain criteria are listed. Although T.I. have attempted, in many cases, to make the criteria hierarchical, the system is very unhelpful. The various headings are listed in Appendix I, Table 5.2. Firstly, the hierarchy is not well structured, each new heading is not a sub set or parallel group of the proceeding heading. Secondly, the headings are functional and not "performance" based. Thirdly, because of this structuring the user has to inspect a great deal of irrelevant information because the headings are so general. A heading such as "Ratios over 10" would be far more useful to the user. The ratio being a performance characteristic.

FIGURE 5.1 : CHARACTERISTIC CURVE FOR A MOTOR HAVING A CONSTANT TORQUE OVER ITS OPERATING SPEED RANGE.

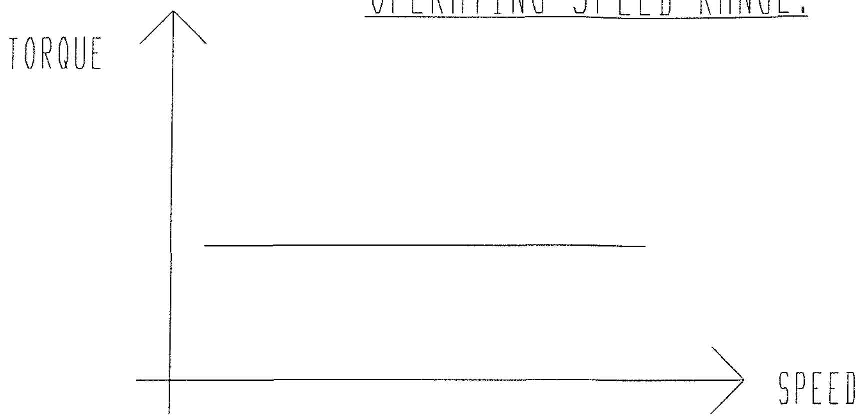


FIGURE 5.2 : THE EFFECT OF A REDUCTION GEARBOX ON FIGURE 5.1

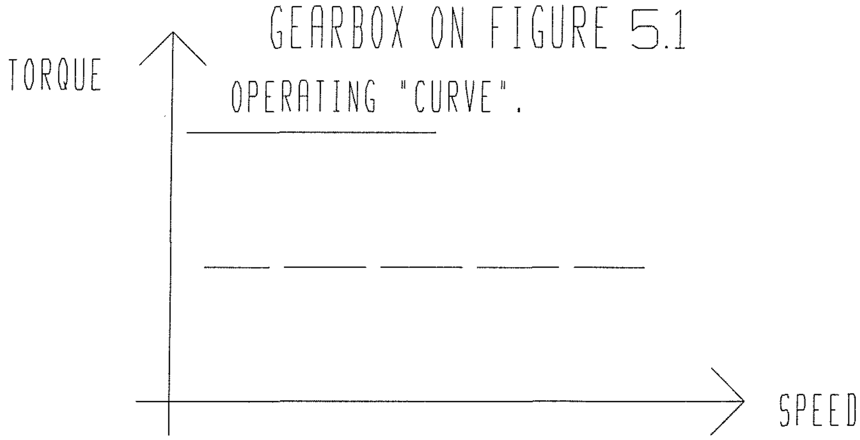
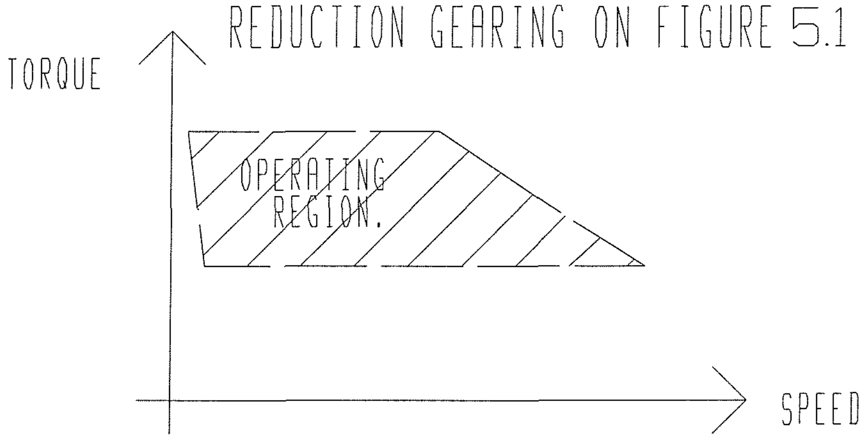


FIGURE 5.3 : THE EFFECT OF A VARIABLE REDUCTION GEARING ON FIGURE 5.1



The section headings for belts, chains and pulleys are shown in Appendix I, Table 5.3. They suffer from the same problem of being descriptive of type and not of performance. A well structured list of performance characteristics would intrinsically infer the general nature of the belt / chain drive used. Table 5.4 contains the list of characteristics identified by a third year project student at Southampton University (REF. 13). The characteristics are clearly more performance orientated with physical information also present. The intention for these parameters is that they are considered as a full description of a gearbox.

5.3.1 **Belt and Chain drives**

The results of the research work should allow the characteristics each side of the gearbox element to be identified. Thus, the input / output characteristics of belts, chains pulleys and sprockets should be the same as that of a mechanical gearbox. The difference between a gearbox and these subsystems is that gearboxes are selected using input / output characteristics. Belts, chains, pulleys and sprockets are optimized (using different characteristics) towards meeting the input / output characteristics. Clearly, when the gearbox is designed a similar process is performed, however, because gearboxes are generally stock components, only the input / output characteristics are considered. The conclusion of this observation is that comparing the selection criteria used for belts etc. would be irrelevant to the study of gearboxes. It could however, be argued that certain parameters may prove stimulating in developing analogies.

5.4 **The Breadth of Analogies Studied**

The primary analogy with gearboxes is the electrical transformer. The transformer although working on very simple principles, is a well developed and hence complicated device. Thus, the analogies used for the research will be kept at a fundamental level and not venture too far into a field of expertise outside the scope of the project. A good example of this philosophy is that BS 171 covers power transformers. Part 1 is the standard containing the general requirements including all the major definitions, requirements for rating plate markings and factors considered important when ordering these power and distribution transformers. This standard alone generates an extensive set of characteristics which are

TABLE 5.4: THE CLASSIFICATION SYSTEM SUGGESTED BY De La Cour (REF. 14)

1) Maximum speed at input.
2) Maximum corresponding torque.

3) Gear ratio.
4) Output shaft positions.

5) Torsional stiffness.
6) Inertia.

7) Rated axial loading.
8) Rated radial loading.

9) "Dimensions".
10) Weight.

11) Cooling requirements.

NOTE :

"Output shaft positions" is a very complex, multi-digit code.

Torsional stiffness is measured at the input with all the outputs locked.

Dimensions are the maximum overall values : LENGTH , BREADTH , HEIGHT.

complete enough to totally specify a transformer. Thus, this data in conjunction with a manufacturers' literature study of general purpose transformers could be considered a very adequate study. Whenever possible the characteristics of special types of transformers can also be considered.

The characteristic analogy matrix can be developed between gearboxes, transformers and belt / pulleys etc.

5.5 The Extended Parameter Lists for Gearboxes.

Noting the above observations and guidelines, the extended parameter lists for gearboxes have been compiled in a similar fashion to the lists for motors. The lists are extensive and have thus been removed to Appendix B.

5.6 The Complete Parameter Lists for Gearboxes.

The complete analogous parameter lists for gearboxes can be generated using the lists of parameters shown in Appendix B.

A section of the list is shown in Table 5.5 Part 1, the full list is given in Appendix I. The list was completed using the technique of tracing analogies as performed in Chapter 3. The more interesting analogies are discussed in Chapter 6.

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 ...

CHAPTER 6

THE STUDY OF GEARBOX PARAMETERS.

6.1 Introduction

This chapter details and discusses the analogies identified in Chapter 5.

6.2 The Gearbox / Transformer Analogy

The gearbox and the a.c. transformer appear in the same column of the analogy matrix (Chapter 2, Table 2.1). In their simplest, ideal forms, both of these devices modify the through and across variables by a certain ratio, whilst preserving power. The main conceptual problem with the analogy between these devices is that the transformer uses an a.c. supply and the gearbox a d.c. supply. Considering this difference in more detail, the transformer is an electromagnetic device, the magnetic flux being an intermediate mechanism by which power is transferred. The transformer is a "double gyrator" (REF. 4) with energy being converted in from one form to another and then back again. In "black box" terms this internal conversion is disregarded and the simple relationships between input and output considered. When attempting to find analogies between transformers and gearboxes, particularly when considering losses, the electromagnetic energy conversion process must be investigated.

6.2.1 The Gearbox / Transformer Analogy - Depth and Breadth

The gearbox / transformer analogy was discussed by three third year project students at Southampton University (REFs. 12, 13, 14). The depth and breadth of the analogies developed did not fully exploit or explain the fundamental mechanisms required for the analogies to be valid - the definition of "flux" for a rotary mechanical system was not considered. The thorough and well documented use of the transformer equivalent circuit developed by electrical engineers, provides an excellent method of understanding the elemental interactions within transformers. From these fundamental representations, analogies can be developed with the elemental representations of gearboxes.

Figure 6.1 (REF. 6) shows the elemental circuit and phasor diagrams for a transformer on no-load. Figure 6.2 (REF. 6) shows the full circuit and phasor for a loaded transformer. Figures 6.3 (REF. 6) and 6.4 (REF. 6) give the effects of the open and short circuit tests and Figure 6.5 (REF. 6) is the "fully referred" equivalent circuit. Figures 6.1 to 6.5 have been removed to Appendix J. All of these circumstances will be discussed and expanded as analogies for gearboxes are developed through them.

6.3 The Rotary Mechanical Analogy of Flux

A current in a coil will produce a magnetic flux (REF. 26). A coil with no electrical power connections will experience a current flow when placed in a changing magnetic flux (REF. 26). These simple principles, allow a sinusoidally varying current in the primary coil of a transformer to induce a sinusoidally varying voltage in the transformer's secondary coil. The flux is common to both windings and possess the same "strength" in both coils at any time during the cycle. Flux is considered as being produced by a current in a conductor and hence, in generic terms is dependent on the 'through' variable.

Considering the fundamentals of meshing gear-teeth, at the point of contact, the gear-teeth must have the same instantaneous linear velocity and the same instantaneous linear force (REF. 21). The differing angular speeds and torques are produced by the different pitch circle diameters of the gearwheels. In the linear case, the through variable is force, so the "flux" in the gearbox must be the forces transmitted through it. As a confirmation of this concept, a loaded transformer requires a larger primary current to counter the demagnetizing flux of the secondary (See Figure 6.2, Appendix J). Similarly, a loaded gearbox will have a larger reactive torque requiring more torque at the input and hence greater forces through the gearbox. The conclusion is that flux can be modelled by the forces transmitted through the gearbox.

6.4 The Rotary Mechanical Analogy of an a.c. Source

In most applications, gearboxes run under d.c. conditions. This essentially means that the torque and speed transmitted are constant at rated conditions. If a gearbox experiences, transients, such as accelerations and reversals i.e. cycling, characteristics which do not greatly affect "d.c." performance become important. When running at constant angular speed, inertia does not affect the system. However, if an angular speed change is required, the effects of inertia must be considered.

A transformer, by its nature, has a constantly changing supply and as previously mentioned, relies on these conditions to operate. Thus, in order to develop the transformer and gearbox analogies, the rotary mechanical supply must be considered to oscillate in a similar fashion to an a.c. supply. For visualization, the frequency of oscillation can be reduced so that the mechanical oscillations can be considered. In a rotary system of this type, the main driver of the oscillations is the torque, the angular speed of the system is the result of the torque.

6.5 Discussion of the Analogies Identified in the Extended Lists

6.5.1 Impedance Matching

When building a.c. electrical systems, for maximum power to be transmitted between components, it is important to ensure that the output / input impedances are equal (REF. 26). In certain cases, this may not be possible. Using the fully referred equivalent circuit of a transformer (Figure 6.5, Appendix J), it can be seen that the impedance of the load can be lumped together with the transformer impedance. Therefore, a carefully chosen transformer connected between two components can be used to adjust the impedance "seen" by the driving component so that the output / input impedances match.

By their very nature, gearboxes match the output characteristics of a driver to the input requirements of a load. It can be argued that if the driver were connected directly to the load, poor performance would result so the gearbox must be "impedance matching". The concept of impedance matching in a.c. systems is however more subtle. The internal 'losses' of the transformer play an important part in determining the impedance match. The internal losses in gearboxes, are generally ignored in the rotary mechanical matching process. Thus, it can be concluded that impedance matching does occur in rotary mechanical systems but only at a very superficial level.

6.5.2 **Gearbox Running Steadily With no Load**

Consider a gearbox being driven continuously, in one direction, with no load connected. The torque being supplied is simply that required to overcome the losses within the gearbox. These small, steady losses will be made up of bearing friction loss, the drag caused by the gearwheels moving through the lubricant and the sliding friction at the gear-teeth interface (REF. 52). These losses sum to create a torque opposing the motion of the gearwheels. The bearing friction losses increase slightly with speed (REF. 24) but can be said to be constant in practice. The drag torque is proportional to the square of the gearwheel speed through the lubricant but again is very small and can be considered constant (REFs. 5, 24). The sliding friction is very small and will remain constant over much of the operating range (REF. 21).

Transformers also have losses associated with their construction. These are identified using the well known "open circuit" and "short circuit" tests (REF. 6). The open circuit test as shown in Figure 6.3 (Appendix J) yields the power necessary to create enough flux to produce the secondary voltage. It is therefore a measure of the current needed to magnetize the core. This magnetization current is a measure of the "iron loss" of the core. The loss is small and constant (REF. 6) over the operating range of the transformer. It is caused by eddy currents and hysteresis in the magnetic core. These small, steady losses cause the flux produced in the primary winding to be grater than it need be. This is similar to the torque on the input side of a gearbox having to overcome the losses (or retarding torques) set up by motions within the gearbox. The eddy current loss is a demagnetisation effect, this can be thought of as a demagnetising flux, a flux retarding or opposing the primary coil flux. The hysteresis effect is more complex and is discussed in Section 6.7.3 below.

The short circuit test as shown in Figure 6.4 (Appendix J) identifies the copper or resistive losses in a transformer. These losses are current dependant. Therefore they change as the transformer loading changes. As described above gearboxes have small steady torque losses which will not vary as greatly as the flux and current in a transformer.

The flux produced by the transformer windings is not all carried by the transformer core. Some will leak. Thus to achieve the expected useful magnetic flux to create the correct secondary voltage, the supply voltage must be slightly greater than that required. Referring this observation to the gearbox, the actual input speed must be slightly greater than the theoretical input speed to produce the output speed in a gearbox. A possible mechanism for a speed loss in a gearbox could be sliding between the gear-teeth, therefore, only a component of the speed is transmitted. Thus, the analogy can be confirmed and it does not contradict the notion of flux outlined in Section 6.3.

Consider now the transformer analogy of a gearbox. The action of a gearbox uses torque and speed at the input to produce torque and speed at the output. A transformer on no load, has output voltage but no output current. Thus it is not directly analogous with an unloaded gearbox. The transformer condition that is more analogous to a gearbox running on no load is that when the secondary winding is short circuited. In this condition, the transformer produces secondary voltage and current.

The above analogy, although satisfying many characteristics does not consider the oscillating case for the gearbox. When cycled, the supply torque must drive the gears and shafts one way and then in the other. For this to happen the inertial energy of the gearbox must be dissipated as the system slows before a reversal. If this is to happen quickly without any external damping, energy must be dissipated within the gearbox. The bulk of this energy absorption must occur as strain energy as the gear-teeth distort and the drive shafts twist as the load is reversed. This strain energy will hopefully be below that required to permanently distort the working parts. This rapid straining will cause heating which will be the main mechanism for energy dissipation. This method for energy absorption upon reversal is clearly not suitable for any system other than small low speed, low torque applications. Most systems will require external damping.

Consider a transformer with the secondary winding left open circuit. If the current in the primary winding of a transformer is rapidly reduced, the flux set up in the transformer core will collapse but in doing so will induce a voltage in the primary winding which will in turn cause a current to flow. The direction of this current will be in the same direction as the primary current thereby attempting to stop the flux from collapsing. Thus, when a sinusoidal voltage is applied to the primary coils of a transformer, the current and hence flux will tend to appear to have "inertia", it will always change slightly slower than the applied voltage. The nature of electrical power allows the driving voltage and resulting current to be decoupled in this fashion. In the mechanical domain, changes in velocity cannot be made to "lag" the torque change unless the driver and load are decoupled by some sort of energy dissipation device. In the oscillating case, this device will allow the driver side to dissipate energy whilst absorbing the driven energy. Eventually, the driven side will have lost all of its kinetic energy allowing the driver to accelerate it in the opposite direction. During this reversal, a great deal of energy will have been dissipated. The time constant for such a system will be much longer than an electrical system. If the gearbox is driving a load, the ability of a mechanical system to change direction will also depend heavily on the nature of the load inertia.

When the secondary winding of a transformer is short circuited, secondary current flows and a demagnetizing flux is produced. This opposes the primary flux. It reduces the flux in the system. Thus, as the flux collapses with the alternating supply current, the "inductive lag" as described above is reduced because the flux is reduced. The analogy with a gearbox would be to reduce the effects of inertia. This would mean running the cycling process slowly with heavy damping, the system would then follow the torque characteristic almost perfectly. The need to dissipate the inertial energy would be diminished by the low speed and inherent high damping. The torque applied would be dedicated to overcoming the damping and not the inertial torque of the system.

If the transformer is a fully loaded transformer. The demands of the secondary current and hence demagnetizing flux will cause the primary circuit to operate at fully rate values. However, as was shown earlier, the net current required to cause magnetisation is very small, so the inductive lag is still reduced. This situation does not occur with loaded gearboxes because the inertial effect is not countered internally.

6.5.3 **The Hysteresis and Backlash Analogies**

The effects of hysteresis on the magnetising current of a coil are best explained using Figure 6.6(a) (REF. 6). Hysteresis causes a distortion of the magnetising current because of residual magnetism.

Before attempting to identify an accurate analogy, consider an easily visualized, rotary mechanical situation; a high inertia load is subjected to a sinusoidal driving torque through a reduction gearbox. Assuming the system is at rest, when the torque is applied, the system will only accelerate once sufficient torque is provided by the driver. As the driver torque reaches peak torque, the system approaches peak acceleration and constant velocity. After the peak torque, the driver torque decreases. However, the high output inertia will not decrease in speed at the same rate as the driver torque decreases. It will tend to remain constant. It will essentially be driving the driver. Not until the driver torque has reversed will the load slow significantly, stop and then reverse rotation as the driver torque increases in the reverse direction. This whole sequence has been plotted out in Figure 6.6(b). The driver torque is the sinusoid, the output speed, the kinked waveform.

Having shown what is happening physically, a more accurate analogy can be developed. Flux and magnetising current are analogous to torque. What Figure 6.6(b) shows is that the sinusoidal driver torque does not reverse "early" enough to cause the output speed to reverse "on time" thereby following a sinusoidal pattern. If the torque sinusoid were to distort by reversing earlier, it would pull the output speed down earlier thereby making it follow a sinusoid. The result would be that the torque sinusoid would distort to a shape very similar to that of the magnetising current shown in Figure 6.6(a). The output speed would then be controlled to a sinusoid. Thus, an analogy between system inertia and residual magnetism has been identified.

FIGURE 6.6a : FLUX AND MAGNETISING CURRENT IN A TRANSFORMER COIL.

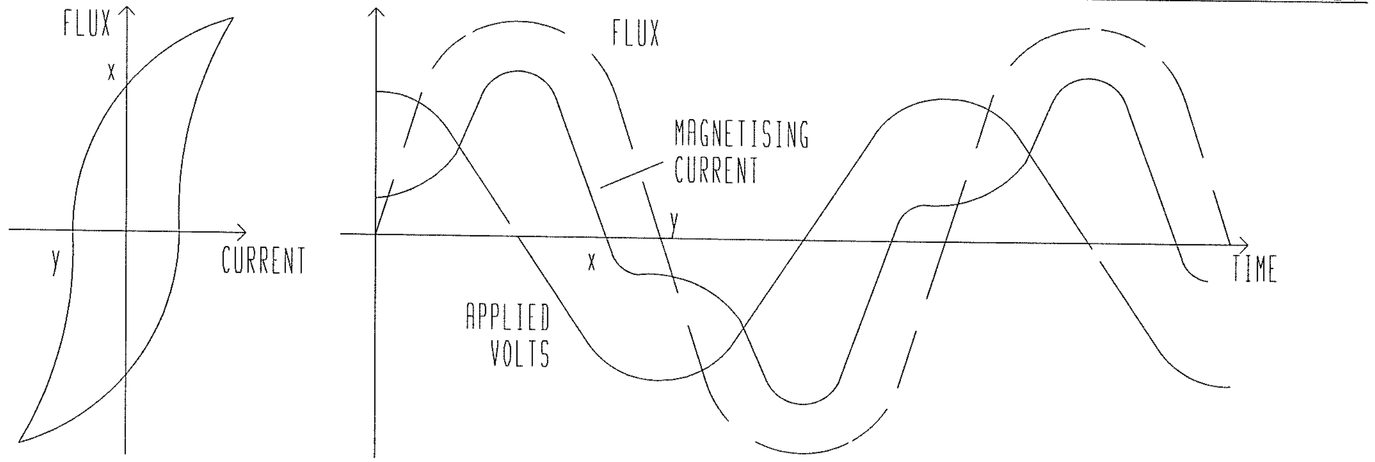
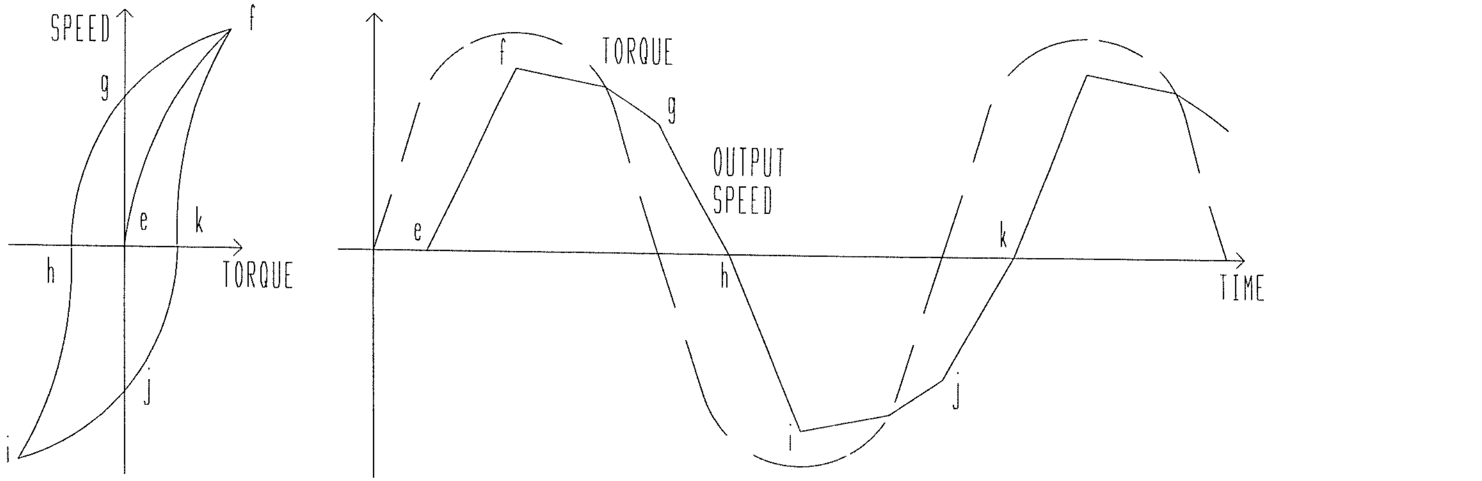


FIGURE 6.6b : TORQUE AND GUESTIMATED OUTPUT SPEED FOR A LOADED GEARBOX.



The effect of backlash on these curves would be to introduce a discontinuity when the torque changes. With a sinusoidal driving torque, the torque changes are so gradual that the system inertia will take up the backlash and smooth out the discontinuities.

6.5.4 The Rotary Mechanical Analogy to Voltage Regulation

Voltage regulation is the percentage change in the secondary voltage between no-load and full load conditions. Since the secondary voltage is a function of the primary voltage, the regulation can be worked out in terms of the primary change (REF. 6). It is a measure of the losses in voltage incurred in the transformer as load changes. The losses are the voltage drops caused by the primary and secondary impedances.

Using the voltage / velocity analogy, the change in input speed between no-load and full-load conditions must be studied. It was stated in Section (6.7.2) that the velocity of sliding between the gear-teeth must be a component of the input velocity and hence the input velocity was in fact greater than that required to produce the output velocity. In very high reduction gearboxes, such as worm and wheel gearboxes, the amount of sliding will increase with increased torque which is why these devices are less efficient (REF. 52). In the general case, it is difficult to see how additional speed losses can be identified because the larger load causes a larger torque to accelerate it to the final speed. Hence this analogy cannot be pursued any further within this thesis.

6.5.5 The Epicyclic Analogy

The analogy between transformers and epicyclic gearboxes produces some very interesting ideas. Consider first, some very simple observations. (Assuming the sun, planet, carrier type construction for the gearbox, each element being able to be driven).

With the carrier held and the sun driven, the planets rotate giving a fixed ratio. The torque available is a function of this ratio. With the planets fixed, and the carrier free to rotate, the gearbox exhibits a different ratio, again, the torque is a function of this ratio.

This change in ratio can also be achieved by a transformer by using two output tapping. If only one is used at a time, the other being left open circuit, a different turns ratio is created. This is illustrated in Figure 6.7.

When the planet and carrier are both free to move the ratios remain the same but the torques transmitted change with load. This is because the total torque on the system must equal zero.

The transformer analogy to this would be two tappings on load as shown in Figure 6.8. Also shown is a possible electrical model for the transformer loads. The currents in the branches of the parallel circuit are load dependent. The adjustable impedance is present to keep the ratio n1/n constant. Breaking either arm would produce the equivalent of Figure 6.7a or 6.7b.

The full analysis of epicyclic gearboxes and their equivalent transformer circuits are out of the scope of this project. However, it is hoped that the above observations show that a possible analogy exists and that further investigation is required.

6.5.6 **The Condition for Maximum Efficiency for Transformers**

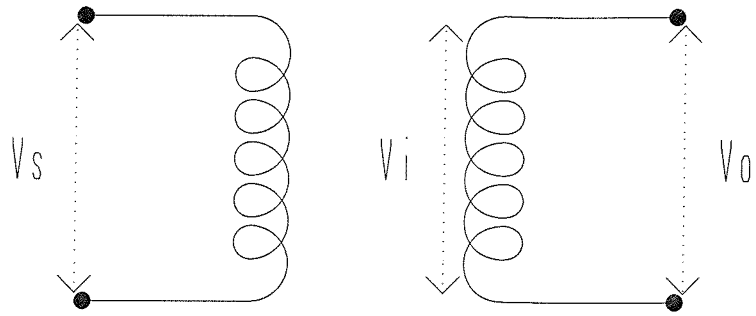
Transformers are at their most efficient when the constant core losses are equalled by the current dependent copper losses in the windings (REF. 6). The analogy to iron losses for gearboxes is discussed in Section 6.7.2. The transformer copper loss increases with current and in fact represents the voltage drop cause by the reactance of the windings. This increases with primary current and therefore with load. The rotary mechanical analogy to this phenomenon was explored in Section 6.7.4 and the results were inconclusive.

The total copper losses can be determined by the short circuit test for transformers. The rotary mechanical equivalent would be a gearbox transmitting rated torque, running at the lowest possible revolutions. The low speed would minimize the speed related friction losses outlined in Section 4.7.2 thereby maintaining the analogy with the core losses. However, because rated torque is being transmitted through the gearbox, it infers that the full flux is

FIGURE 6.7 : EFFECTS OF TAPPINGS.

a) General Case:

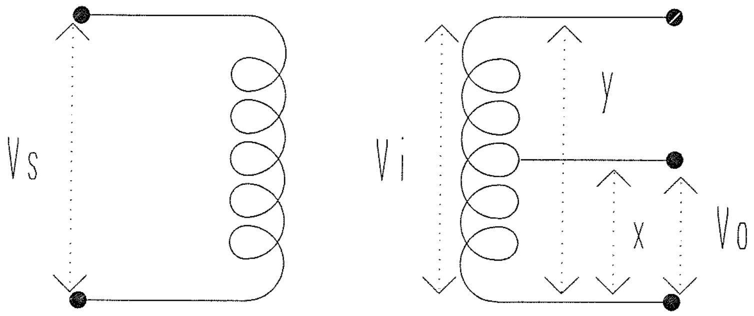
Turns ratio = n.



$$V_o = V_s \ (\text{ if } n = 1 \ )$$

b) Tapping Added to Output:

Turns ratio = n



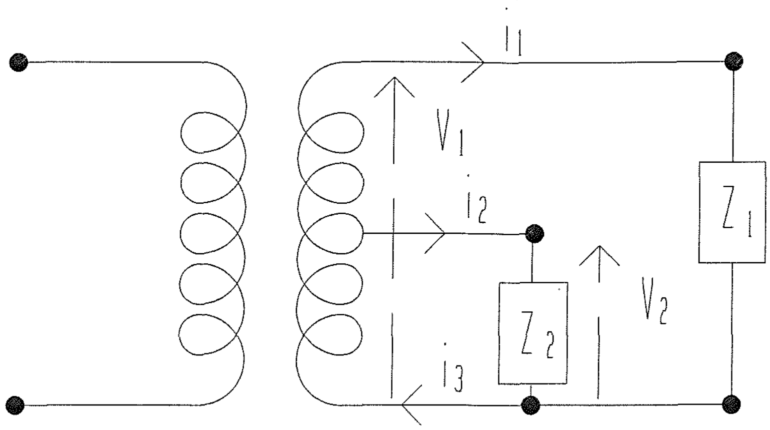
$$V_o = V_s * x/y \ (\text{ if } n = 1 \ )$$

NOTE :

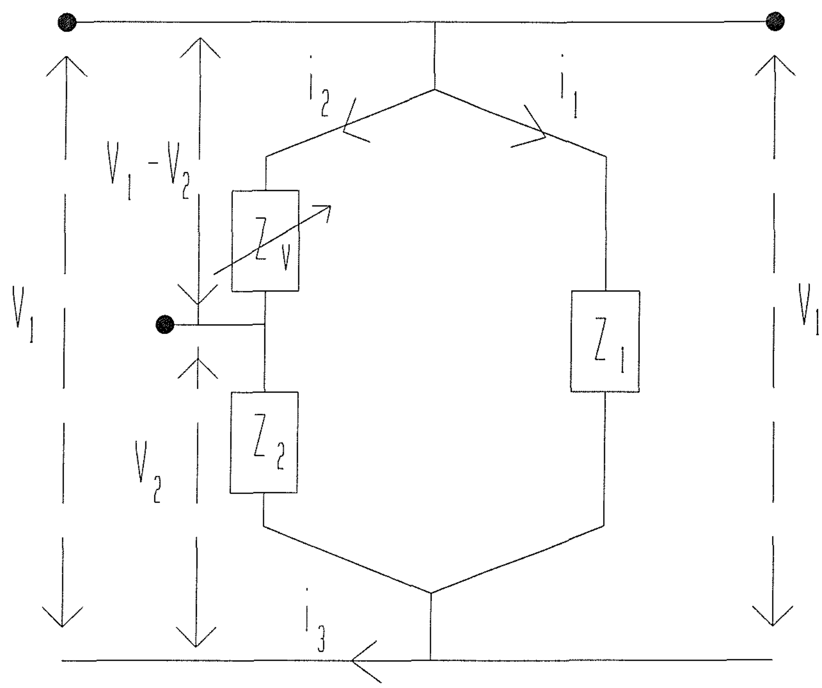
$$V_s * n = V_i$$

FIGURE 6.8 : TWO TAPPINGS ON LOAD AND A POSSIBLE SIMPLE, NON-MAGNETIC EQUIVALENT.

Transformer with two tappings on load.



Equivalent circuit using an AC source.



present in the transformer. This contradicts the theory of the short circuit test because minimal flux is maintained. Thus, the final conclusion of these observations is that there is no apparent analogy in gearboxes to the copper losses in transformers. Thus, the condition for maximum efficiency of a gearbox cannot be related to the criteria required to ensure maximum efficiency in transformers. Common sense suggests that a gearbox will be most efficient when all the internal losses are minimised.

6.5.7 **The Capacitance / Stiffness Analogy**

For any wound coil to exhibit the characteristics of "turns", it is vital for the individual turns to be insulated from each other. This is achieved by coating the wire from which the coil is wound with a thin layer of insulating material. Thus, within the coil there will be a series of connected conductors separated by insulators. This is essentially a description of a parallel plate capacitor. The resistance of the coil will create a potential difference across the coil and hence there will also be a slight potential difference between adjacent coils. Therefore, in fundamental terms, some capacitive effects must be present. Along the length of the coil, the capacitors are in series. This results in the net capacitance being small. The small surface area of the wire will also cause the capacitance per coil to be small. The capacitive loss will be proportional to the applied voltage and the frequency of the supply and hence be said to be virtually constant and small. With reference to Figure 6.5 (Appendix J), the equivalent capacitance could be placed in parallel with the core losses represented by the parallel inductive and resistive elements.

The rotary mechanical equivalent of capacitance under these circumstances can be identified as strain energy stored in the gearbox shafts and gear-teeth. This strain energy will manifest itself under extreme conditions such as very high loads and high torques. As a high inertia is accelerated, the gearbox shafts and gear-teeth will deform slightly before any motion is transmitted to the load. When the load begins to move, the accelerating torque will continue to act thereby holding the deformation within the shafts and gear-teeth. Once the load has ceased to be accelerated the twist in the shaft will be relieved, it will in fact have contributed to the acceleration of the load by untwisting when the twisting torque has been removed.

In most gearboxes, the shafts, gear-teeth and casing will have been sufficiently over-designed to render such stored energy insignificant. Situations may occur where a rapid and exact response is required and any twisting in the transmission will clearly hinder the desired result. Therefore, if the designer is not aware of this possibility, it cannot be eliminated at the design and selection stage of system assembly.

6.6 Creation of the Lists for Selection

With the extended parameter lists complete, a shorter list, designed to make computer aided retrieval commercially viable can be gleaned. As with the motor selection list, the parameters must be recognisable by designers, unambiguous and discrete. The extended lists for gearboxes were structured in a similar way to the motor parameter lists. The broad heading being input, output, input / output, dimensional and other. The additional heading "input / output" deals with the gearbox ratio, the shaft relationships etc. This is an important group of parameters because they describe how the gearbox links the components each side of it. Thus, the input / output information must be readily available to the designer.

To match the speed of adjacent components, any two of the following three parameters must be known: input speed, output speed, gearbox ratio. For selection purposes, all three parameters must be available for the user to enter what is known. If the gearbox has a variable ratio, the output shaft speeds will have a range. Defining the limits of this speed range is something that must be given consideration. The most common types of variable ratio gearbox provide a continuously variable ratio between an upper and lower limit. Thus, identifying the gearbox as continuously variable and quoting the limits of the output speed range would adequately deal with this particular problem. A less common type of gearbox provides a series of fixed ratios between upper and lower limits. The above parameters would certainly define the limits of the performance of this type of box, but would give no indication of the intermediate ratios. Quoting the intermediate ratios would be possible but because these boxes are less common, dedicating a retrieval field (or fields) for them would be wasteful. The epicyclic gearbox can also provide two ratios depending on how it is applied. Using the criteria outlined above, the ratios and speeds of the epicyclic box can also be captured.

Given that the gearbox can provide the correct speed relationship, it is vital to check that the torque it can transmit is adequate. In general, the gear-teeth are "designed" to transmit a certain torque and hence, excessive torque will damage them. Therefore, simply noting the maximum torque the gearbox can transmit will be sufficient to avoid damage to the gear-teeth. It does not matter whether this is at the input or output or if the gearbox is step up or step down, the largest torque will always appear at the slowest shaft and that is where the manufacturer will have measured it.

The set of characteristics above describe the performance characteristics of the gearbox in general terms. It is now necessary to define them exactly for selection purposes. The parameters are:

Box Type
Input Speed
Output Speed Range
Ratio Range
Maximum Torque

6.6.1 Gearbox Type

The gearbox type attempts to indicate what kind of ratio the gearbox has. Four main categories exist:

(a) **Fixed**: The most common type of gearbox. The gearbox has one discrete ratio value. The final ratio may be the result of a series of reductions but, in "black box" terms, the gearbox has one fixed ratio.

(b) **Continuously Variable**: The output speed can be continuously adjusted between two fixed limits. These devices are often called "variators".

(c) **Selectively Variable**: A series of discrete ratios between two limits. These are less common.

(d) **Epicyclic**: These gearboxes have three basic ratios. In common applications, the sun shaft is usually driven and therefore the sun / carrier and sun / planet ratios are of most significance.

Having identified the type of ratio the gearbox has, the relationships between input and output become very much clearer.

6.6.2 **Input Speed**

This was fully defined in Appendix B4.6.6. It is a parameter the designer will know because it is the output from the preceding component. It is common to express the speed in terms of the non-S.I. unit RPM.

The output speed range will be made up of the maximum and minimum output speeds.

6.6.3 **Fastest Output Speed**

The fastest output speed the gearbox can produce. This will be the rated speed for fixed ratio boxes. Units: R.P.M.

6.6.4 **Slowest Output Speed**

The slowest output speed the gearbox can produce. This will be the rated speed for fixed ratio boxes. Units: R.P.M.

The ratio range will be made up of the maximum and minimum ratios.

6.6.5 **Upper Ratio**

The ratio of the input speed to the slowest output speed required. This will in most cases be a number greater than one (the gearbox is a reduction box).

6.6.6 **Lower Ratio**

The ratio of the input speed to the fastest output speed required. The upper and lower ratios will be identical in a fixed ratio box.

The above five parameters cope very adequately with the gearbox ratio. A designer would be able to use a mixture of these parameters to specify the ratio required. They neglect to convey the relative direction of rotation of the input and output. This could be solved by making the output speeds and ratios positive or negative. If positive, output and input rotate in the same direction, if negative, they rotate in opposite directions. Problems immediately arise when trying to define the rotations. One solution would be to define rotation as clockwise or anticlockwise along the axis of the shaft, towards the gearbox. This option was not taken. The reason was data capture time. Also, quite surprisingly, direction of rotation of shafts are not always quoted by manufacturers.

6.6.7 **Maximum Torque**

The largest torque the gearbox can transmit. The various values of torque are discussed in Appendix B, Sections B4.6.3, B4.6.4, B4.6.11, B4.6.12. In most cases, the value for starting torque will be defined.

6.6.8 **Rated Power**

To complete the performance characteristics, the rated power of the gearbox must be quoted. This parameter is useful to the designer because it firstly ensures compatibility with adjacent components, it gives an indication of the size of the component, and given a rated speed, the rated torque can be calculated. The efficiency of gearboxes is high in most cases, so whether the power is quoted at the input or output is unimportant.

6.6.9 **Shaft Configuration Code**

This is fully explained in Appendix G. The shaft configuration code allows the user to define the relative positions of the gearbox shafts by answering a few simple prompts. From the user inputs, a code is generated that is easily handled by the retrieval system.

Having defined the performance and "shape" of the gearbox, the size of the gearbox can be defined. Much like motors, it is possible to draw a box around the awkward shape of gearbox casings and to thereby define three orthogonal dimensions.

6.6.10 **Length / Breadth / Height**

These are the three orthogonal dimensions mentioned above. They follow the same format as for motors i.e. measured in millimetres and length > breadth > height.

No dimensional information regarding shaft sizes, flange sizes etc. has been included. It is recognised that this information is important if the gearbox is to be connected into the system. However, once a gearbox has been selected by its primary characteristics (those above), details such as shaft sizes can be found from the dimensioned drawings given by manufacturers in their catalogues. Also, because the gearbox shafts must be joined to other shafts, couplings must be used.

6.6.11 **Mounting Type**

The gearbox is usually supported in one of the following ways :

i) **Foot**: The gearbox rests on small feet protruding from the casing. These are suitably drilled to accept a securing bolt.

ii) **Flange**: A flange is part of the casing, this can be bolted onto a mounting or another component.

iii) **Shaft**: The gearbox has a female keyway rather than an input shaft. This allows the gearbox to slide onto a drive shaft. It is restrained from spinning by a tie bar and an output shaft is available for use.

iv) **Other**: This covers any of the less common mounting types.

6.6.12 **Gear Type**

Four main gear-tooth profiles have been identified as commonly used in bought-out gearboxes. These are listed in the Technical Indexes Ltd. catalogue as: Spur, helical, bevel and worm. A fifth category, "other" is included to cope with less common tooth profiles and combinations of profiles. The type of gear-teeth tell the designer a great deal about the gearbox. Spur gears are relatively cheap. Bevel gears have shafts at right angles, worm and wheel gears usually give a high reduction. Profile combinations are more expensive but may have advantages of quietness etc.

6.6.13 **Company Name**

In many design environments preferred suppliers are used. Hence, there is often a need for the designer to target particular requirements at a known supplier. This can be done by stating the suppliers name as one of the search criteria. It is hoped that this option is not used often because it narrows the breadth of search which is one of the advantages of a computer aided retrieval.

6.6.14 **Inertia**

The most important analogous parameter developed was gearbox inertia (Sections B4.6.6, 4.7 et al). Inertia is rarely quoted by manufacturers of above fhp gearboxes, but is something that should be considered in transient conditions. The gearbox inertia will contribute to the overall system inertia. Knowing the system inertia and the driver torque, it would be possible to estimate how big the gearbox inertia must be before it significantly effects the ability of the driver torque to accelerate the load. The gearbox selection can then be made using this value as a maximum. In general terms, the inertia of the gearbox will be a small part of the system inertia and the maximum torque the driver can produce is usually up to one and a half times the steady state torque. Therefore, the gearbox inertia need not be considered for applications where the load has a high inertia compared to that of the gearbox. Also, using current manufacturer's literature, inertia is not quoted. Thus, in order to capture this information manufacturers would have to be contacted. Given the size, cost and time constraints of the whole date capture exercise, contacting manufacturers for data not readily available from catalogues would be prohibitive. Inertia has been included in the selection list as a prompt to the user that it ought to be considered. Technical Indexes did not enter any data for this parameter when creating the trial databases, so it would ruin a retrieval if this data was entered.

Another parameter identified through analogy as useful was stiffness of the gearbox. This was not suggested as a retrieval parameter for two main reasons. Firstly, it is not provided by manufacturers for above fractional horse power gearboxes. This would therefore require extra research by Technical Indexes staff to determine it. Therefore like the inertia parameter it would prove very expensive to capture. Secondly, quantifying the term stiffness would be difficult. For instance, would it be the stiffness of the shafts and gears ?, the gearbox casing ?, or that of the surrounding structure ?. All of these items will flex when torque is transmitted, assessing which one is most significant is extremely difficult. Therefore, this parameter cannot be used without complex analysis and would not be suitable for the type of retrieval system to be used (See Chapter 9). It has therefore been omitted from the short retrieval list.

The selection parameter list is shown in Table 6.1. The assessment of the parameter list is given in Chapter 9.

TABLE 6.1 : Gearbox Indexing Parameters.

INDEXING PARAMETER	VALUE	UNITS	WEIGHTING
Rated Power		kW	10
Ratio Type		--	10
Input Speed		RPM	9
Slowest Output Speed		RPM	9
Fastest Output Speed		RPM	9
Lower Ratio		--	9
Upper Ratio		--	9
Highest Rated Torque		Nm	8
Shaft Configuration Code		--	8
Maximum Length		mm	6
Maximum Breadth		mm	6
Maximum Height		mm	6
Mount Type		--	5
Gear Type		--	3
Company Name		--	1
Motor Frame Code			
Microfilm Number			

CHAPTER 7

COUPLINGS.

7.1 Introduction

The term "couplings" covers a wide range of components. Any device that joins two shafts for the transmission of power would be classed as a coupling. Thus, components such as clutches, fluid couplings and rotary dampers would meet this broad specification. For research purposes, a coupling was define as : "A device used to join elements in a rotary mechanical power transmission system so that power could be transmitted from one element to another with minimal power losses. Any physical misalignments between the elements must be absorbed by the construction of the device".

This definition therefore eliminates components such as fluid couplings which have a power loss and do not absorb misalignments. The above definition is a good example of the problem of semantics dealt with so often in this thesis. Components such as fluid couplings, powder couplings etc. are grouped with couplings (i.e. universal joints, bellows, rubber disk etc.). However, taking a "black box" approach, the performance of these components is very different. Attempting to apply the methodology, developed so far would yield a list of parameters which would be confusing for the selection of a simple coupling. For the case where components are supplied with couplings "built in", the coupling function must be ignored until the system performance has been satisfied. This concept was described in Chapter 4, Section 4.2. The performance of the whole is more important and therefore the data for the fluid coupling etc. is the basis of selection. The coupling data will be an added advantage when the system assembly is detailed.

7.2 Scope

Coupling design is well developed and therefore there are many types of coupling. However, as defined in Section 7.1, the aim is to transmit torque and speed from one shaft to another. Thus, much like gearboxes, the energy domain on each side of the coupling is unchanged (rotary mechanical). Using the analogy matrix Table 2.1, analogous components are: pipes, wires, shafts, rods. These are all very simple components and the analogies that can be developed are very basic. A more complex component with the potential for many subtle analogies is the 1:1 transformer. Although from a different column in the analogy matrix, the 1:1 transformer will "look" exactly like a coupling in an electrical system - current and voltage are preserved across the transformer. Thus, potential analogies can be developed and perhaps re-applied to the simpler components.

7.3 Generation of the Extended Parameter Lists

The study of manufacturer's and supplier's literature, selection procedures and other information (standards and text books) was carried out in a similar fashion to Chapters 3 and 5. The T.I. indexing catalogue classifies couplings by "type". Type being a design or salient feature e.g. "bellows". The list of types provided by T.I. is shown in Appendix K, Table 7.1. Also, the selection procedure advised by a body called SEED (Shared Experience in Engineering Design) was studied (REF. 29). The list of selection parameters suggested by SEED is given in Appendix K, Table 7.2.

Comparing Tables 7.1 and 7.2, it is clear that the SEED data is performance orientated, giving the designer clues as to the parameters needed to select a type and then size of coupling. The T.I. system will only help the designer to obtain the correct size of coupling provided the designer knows what type is necessary. This method is inadequate because it requires the designer to know the pros and cons of the numerous designs and the subtleties in the differences between related types e.g. flexible ring, flexible element, flexible disk are all variations on a rubber element coupling. The differences between these devices will appear through performance data and they should therefore be selected using this information.

The extended list was created using the sources of information listed above and is shown in Table 7.3. Table 7.3 is large and has hence been removed to Appendix K. As in Chapters 3 and 5, the list can be broken down into smaller sections. These sections are not the same as those for the previous components because couplings are very much simpler in their operation. Complex input / output relationships such as ratio and shaft position do not exist for couplings. Therefore the power components; torque and speed, the various definitions of power, the misalignments and any other data are the groupings within the extended list. There are no analogous long lists for this component because of the lack of analogies. Interesting analogies will be highlighted in Section 7.4 below and discussed in greater detail in Sections 7.5 and 7.6.

7.4 Discussion of the Extended Parameter Lists

This section, because of its length has been removed to Appendix C. It defines each of the parameters listed. Using this information section 7.5 details the analogies between couplings and other devices.

7.5 Analogies Between the Coupling and the 1 to 1 Transformer

7.5.1 Background

As mentioned in Section 7.2, in black box terms, a coupling could be viewed as a gearbox with a unity ratio, capable of absorbing misalignments. Thus, it follows that a unity transformer is also analogous to the coupling.

The analogy of flux, developed in section 6.3 still holds as torque is transmitted through the coupling. The analogy of voltage and speed will also apply.

The analogies with rigid and torsionally rigid couplings are very simple as they relate fundamental properties. Torsionally flexible coupling and couplings with "cyclic" outputs (these may be rigid or torsionally rigid) provide the more interesting analogies.

7.5.2 Velocity Ratio

This is the variation of output speed for a given input speed. The fluctuations in output speed increase with degree of misalignment. It does not depend on load (REF. 21).

The voltage regulation of a transformer is a measure of the change of output voltage to input voltage as the transformer secondary current increases from no-load to its full load value. As the current in the secondary winding increases, the useful secondary voltage decreases. This is because the increased current results in greater voltage drops across the secondary reactances. In a unity transformer, running under no load, the induced emf in the secondary winding is equivalent to the emf at the primary terminals. Similarly, a coupling rotating with no misalignments has no cyclic variation at the output. Thus, although it would be unwise to make the analogy between transformer secondary current and degree of misalignment, the parameter "speed regulation" could be coined instead of "velocity ratio". It could be quoted as the percentage variation in speed at full misalignment. Much like voltage regulation, the designer will look for this value to be small.

7.5.3 **Cyclic Stressing**

Consider a coupling using rubber or springs to transmit torque and for simplicity assume the coupling is absorbing a pure angular misalignment. The coupling is modelled as two plates rigidly attached to the shafts and linked by rubber or springs as in Figure 7.7. The misalignment causes parts of the coupling to be compressed more than others. This is illustrated in Figure 7.8. Thus, as the coupling rotates, the material between the plates experiences a cyclic increase and decrease in pressure. When compressed, the elastic nature of the material causes strain energy to be stored in the coupling. It is impossible for all of this energy to be restored to the system and the losses take the form of heat energy. To identify the cause of this energy loss and hence which of the power components (torque and speed) it degrades, fundamentals must be considered.

A force is used to compress the coupling material and this must be supplied by the torque the coupling is transmitting. This is augmented by physical observation of what happens when a rubber coupling is rotated and the degree of angular misalignment changes. As the degree of misalignment increases, the torque required to rotate the coupling increases. Losses do not allow all the stored energy to be restored to the system and hence force is required to continuously compress the coupling material as the coupling rotates. Using the torque / current analogy, the current based losses in transformers cause heating. Thus, a heating loss caused by torque is to be expected.

FIGURE 7.7 : A SIMPLE RUBBER ELEMENT COUPLING.

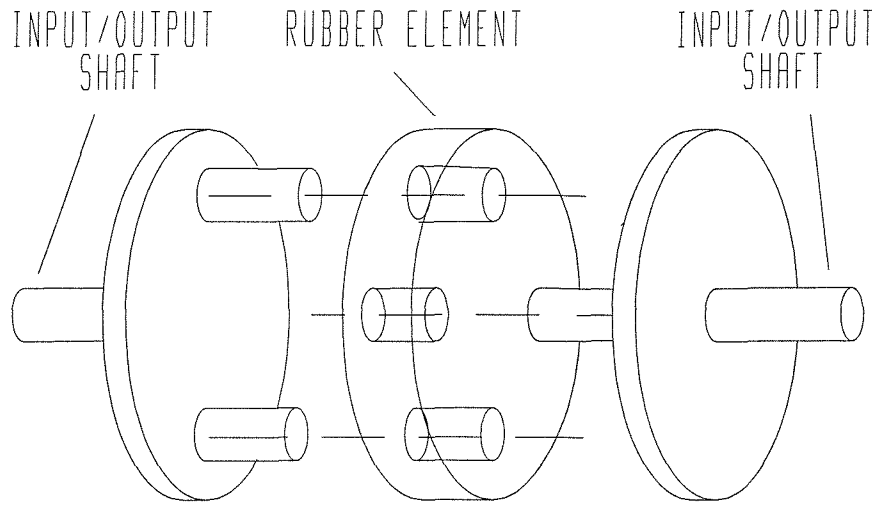
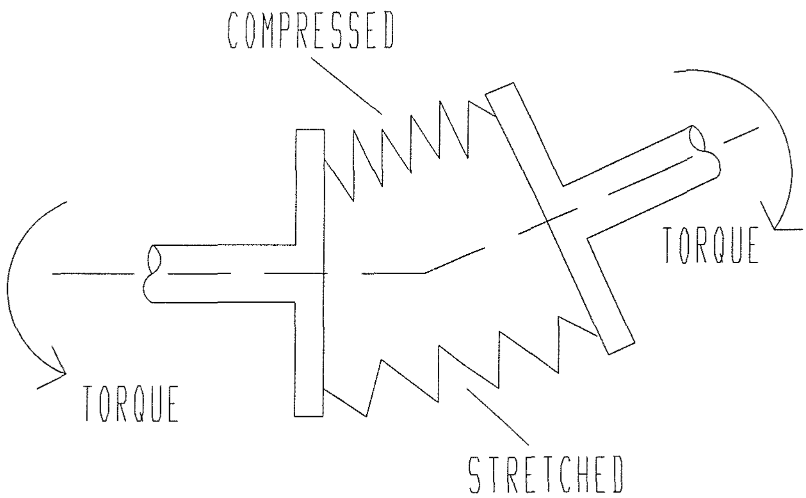


FIGURE 7.8 : FORCES IN A MISALIGNED RUBBER COUPLING.



For a given misalignment the torque needed to rotate the coupling i.e. overcome the internal forces caused by the rubber / springs, will be a constant. It will increase with degree of misalignment but will not change with the amount of torque being transmitted. In general terms, it is very similar to the magnetising current for a transformer which tends to remain small and constant for the load range. The magnetising current is made up of two components, a resistive loss and a hysteresis loss. The heating effect in couplings is clearly equivalent to the resistive loss. Hysteresis will occur in rubber element couplings because the rubber is not perfectly elastic but 'visco-elastic' (REF. 25). If the visco-elastic material was replaced by an element with better elastic properties such as a spring, the hysteresis loss would be less. It should be noted that the use of visco-elastic elements is a deliberate design feature because of the damping it provides. So, although it can be reduced, damping is a deliberate characteristic of these devices. See Sections 7.5.4 & 5 below.

7.5.4 **Torsional Deflection**

The previous section looked at the straining of the coupling in the axial direction. When transmitting torque, most of the straining will take place in the radial plane. The characteristics of the elastic element used will affect the torsional deflection of the coupling. Rubber elements have a rising spring rate, they become stiffer as the torque across them increases. This means that shock loads such as starting or sudden changes in load will be damped as the rubber deforms before the full torque is transmitted. Couplings using springs as the intermediate element will also provide damping by storing energy within the springs during changing load conditions.

There are two modes of torsional deflection that must be considered. The coupling may be driven steadily, or, it could experience changing loads e.g. cycling, starting or shock loads. When running steadily, energy losses on the output side of the coupling will generally ensure that torque must continue to be supplied by the input. Therefore, a flexible coupling will experience a permanent accelerating torque and will run with a constant twist. If running at rated speed and torque, the twist will be the rated torsional deflection. In strict terms, there is now a phase difference between the input and output sides. The magnitude of the phase change is governed by the "resistance" to the input torque on the output side of the coupling. Using the analogy of flux and forces through the coupling, an analogy between the magnetising and demagnetising ampere turns and hence input and output current in transformers can be developed.

An unloaded transformer draws a very small current compared to a fully loaded transformer. When secondary current flows, demagnetising ampere turns are produced. To counteract this, more primary current flows. Applying this to a coupling, the secondary ampere turns can be considered equal to the reactive torque from the load. The primary ampere turns is the supply torque and the net flux is the force through the coupling as measured by the amount of twist in the coupling element. On starting and accelerating up to rated speed, the coupling will flex as high torques are transmitted. Once rated speed is reached, the torque will reduce to rated torque and the coupling will "untwist" to its rated torsional deflection. This is exactly the same as the performance of a transformer having to meet the starting demands of an inductive load.

7.5.5 **Damping**

The twisting of torsionally flexible couplings helps to damp high shock loads. This is because not all the energy associated with the shock is transmitted through the coupling, some of it is absorbed by the coupling and is stored as potential energy or dissipated as heat.

In applications where there is a cyclic variation in torque around a mean torque value - an internal combustion engine say, the coupling will experience a ripple torque around the mean value. The flexing of the coupling element will absorb these variations and hence they will be attenuated on the output side. This is desirable in most applications.

Damping action is achieved by the visco-elastic materials used in many coupling constructions. Energy is dissipated while deforming and is also dissipated while returning to the unstressed state. The stiffness of visco-elastic materials is not constant but increases with the strain the element is experiencing. Thus, as torque increases, the amount of energy dissipated increases, but not in proportion to the increase in torque. Comparing this to the heating losses in the windings of transformers, the copper losses increase proportionally with current drawn (REFs. 6 and 26). Thus, although the analogy is not perfect, in general, the heating losses in both devices increase with the through variable. A complimentary factor is that the heating losses are also proportional to the frequency of oscillation in both cases.

7.5.6 **Capacitance**

In Chapter 4, Section 4.5.2, it was suggested that the capacitive loss in the transformer coils could be modelled as a capacitance in parallel with the core losses. The coupling (the rubber element flexible coupling in particular) has two capacitive characteristics. These have been outlined as stored energy in Sections 7.5.3 and 7.5.4. The stored energy caused by the coupling being deflected as described in Section 7.5.3 can be modelled as the parallel capacitance suggested above. This would compliment the hysteresis effects described in Section 7.5.3.

The energy stored by the coupling twisting must also be equated to the transformer equivalent circuit. Twisting is caused by torque, the through variable. Twisting is also "serial" in nature because it increases with the through variable. Looking at the transformer equivalent circuit, the only serial elements are the transformer coil impedances. These can be broken down into the coil copper loss and the coil inductive reactance. The inductive reactors are used to represent the leakage flux in the primary and secondary windings. They indicate that not all the current flowing in the windings produces useful flux. Similarly, there is a torque "loss" caused by the twisting of the coupling. Thus, the energy storage element is not capacitive in this case but inductive.

The analogy can be explored by considering the situation where a ripple torque is superimposed on a constant torque - the output of a multi-cylinder I.C. engine is the classic example. The coupling will be deflected whilst transmitting the mean torque and will experience a further flexing caused by the sinusoidal component. The strain energy put into the coupling during the "peaks" of the ripple torque will be stored and released during the "troughs" in the ripple torque. The effect on the ripple torque is that it is smoothed. Although damping is present, for this case, smoothing occurs by a redistribution of the energy rather than dissipation. It is most easily envisaged as an inertia kept at constant speed by the mean torque. The peak torque will accelerate the inertia slightly, giving it energy, the trough torque will have the effect of allowing the inertia to slow. However, by its nature, the inertia will react very slowly to these relatively small changes in torque from the mean, hence, the ripple is attenuated. Thus, the analogy between the inductive elements in transformers and inertia is therefore confirmed.

7.6 Analogies Between Couplings and Flows in Pipes

In fluid systems, the couplings between devices are pipes. They allow the working fluid to flow from the high pressure outlet of one device to the inlet of another operating at a lower pressure. The fluid flow rate and the pressure difference across the pipe are the system variables. Unless otherwise stated, the flow in the pipe will be assumed to be a fully developed, steady, incompressible, Newtonian flow. (REF. 27).

7.6.1 Analogues for the System Variables

A hydraulic system is analogous to a d.c. electric system. The through variable is fluid flow rate, the across variable fluid pressure. Thus, the analogies are flow and current, pressure and voltage (REF. 2). In general terms, the ideal pipe will allow the fluid flow and pressure at the outlet of the supply device to be available at the inlet to the driven device without any losses. For a given pressure head, the more pressure losses down the pipe, the lower the fluid flow rate. This is directly analogous to the basic circuit equation $V = IR$.

7.6.2 Flow and Head Loss

It was shown earlier (Section 7.5.2) that couplings transmitting torque by sliding contact tend to have a reduced useful output speed. However, by definition the torque on each side of the coupling must be identical. In pipes, the flow through the pipe is a constant provided there are no flow losses. There is a pressure drop down the pipe caused by friction at the interface between the fluid and the pipe wall. The energy is dissipated as heat.

Thus, the through variable is preserved whilst the across variable is reduced. Also, for couplings and pipes, the "loss" in the magnitude of the across variable appears as a heat loss.

7.6.3 **Pipe Compliancy, Torsional Stiffness and Damping**

Consider a pipe containing fluid under high pressure but with no flow. The pipe will experience a radial pressure. The pressure will cause the pipe to elastically deform. Although there will be no flow out of the pipe, there will be a slight flow into it because of the greater volume created by expansion. Thus, the pipe has "stored some flow" and this flow will be released when there is a pressure change. This is analogous to a torsionally deflected coupling releasing its stored energy as torque when the torque on the shafts is decreased. Again, not all of the strain energy in the pipe will be restored to the fluid, some will be lost as heat thereby providing a little damping.

The above process could be modelled using a perfectly rigid pipe with a hydraulic accumulator attached by a narrow bore tapping. Under high pressure the accumulator would store "flow" which would be returned to the system when the pressure dropped. The narrow bore tapping would ensure a great deal of eddies in the flow causing a loss in energy. Thus, if the system pressure were to fluctuate, as may be the case with a high pressure piston motor, the higher pressure peaks would be absorbed and attenuated by the accumulator and tapping. The pressure drops would be supplemented by stored flow in the accumulator, the narrow tapping ensuring the response was gradual. This mechanism follows the smoothing effects of a rubber element coupling in line with an internal combustion engine.

7.6.4 **Bends in Pipes**

When a fluid flows through a bend in a pipe, several phenomena occur; there is a head loss across the bend, the flow rate remains a constant and there is a force acting on the bend.

The head loss is caused by the fluid becoming turbulent as it is forced to travel at different speeds around the bend, the various actions for turbulence are given in Reference 27. The greater the bend the greater the turbulence and hence more head loss. The head loss over a bend is therefore consistent with the velocity of sliding / degree of misalignment analogy outlined in Section 7.6.2. The sliding and turbulence both produce heat.

All of the above examples have referred to the flow rate remaining constant. This concept is a misleading because a change in head loss, given a constant pressure source, should produce a different flow rate. The confusion lies in the fact that we are discussing system flow rate, meaning that, if no losses occur, the flow into the bend must also come out of the bend. This also holds for a misaligned coupling transmitting torque. The greater the misalignment the more torque needed to overcome the coupling internal losses whilst still providing the output torque. Thus, the torque through the system increases but the useful torque remains the same.

There is a force acting on the pipe bend attempting to "straighten" it. This is caused by the fluid momentum change as the fluid changes direction. Again, the sharper the bend, the greater the force acting on it. The pipe must be adequately supported or robust enough to cope with these forces. Couplings with large misalignments also experience forces other than the torque they are attempting to transmit. Universal joints, for instance, need to be restrained by bearings otherwise, the radial forces generated would tend to flex the shafts.

This is an interesting observation because manufacturers give little information concerning these radial forces and how they should be dealt with i.e. positions of restraining bearings etc. The reason this information may not be provided is that within the safe working limits of the coupling, these forces are usually trivial.

7.7 The Short Parameter Lists for Retrieval

The selection list is shown in Table 7.4.

7.8 Explanation of the Selection Parameters for Couplings

The aim of the short parameter is to make a computer aided retrieval a commercial possibility. This means that the number of parameters chosen must be the minimum possible to allow a decerning selection. The parameters must be familiar to designers and easily gleaned from manufacturers catalogues.

TABLE 7.4: COUPLING INDEXING PARAMETERS.

INDEXING PARAMETER	VALUE	UNITS	WEIGHTING
Rated Torque		Nm	10
Maximum Speed		RPM	10
Maximum Power		kW	9
Misalignments :			
Maximum Angular		Degs	9
Maximum Radial		mm	9
Maximum Axial		mm	9
Minimum Bore		mm	8
Flange Diameter		mm	8
Maximum Diameter		mm	5
Coupling Type		--	4
Company Name		--	2
Motor Frame Code			
Microfilm Number			

7.8.1 **Rated Torque**

The limiting factor for couplings is the torque that they can transmit. If they are not transmitting torque, they are under very little mechanical stress and therefore the life of the coupling will be lengthened. Thus, in order to select a coupling that will not fail in operation, the selection should be made using the rated torque. This selection will of course be most energy efficient and economic. From general system characteristics, most couplings will be constructed to be able to withstand intermittent overloads and shocks of about 170% rated torque. This will allow them to be used with direct-on-line starting motors. Thus, there is no need to select using additional torque parameters.

7.8.2 **Rated Speed**

This parameter and the rated torque are the components of the rated system power. All the other elements in the system will have been designed to satisfy these values. The rated speed will affect the service life of the coupling. If run at speeds above the rated value continuously, failures due to fatigue and overheating are more likely.

7.8.3 **Rated Power**

This is not given precedence over torque and speed as in the previous components because although it is the product of the rated torque and speed and gives a fair indication to the size of the component, it is not as limiting as the rated torque. Using the power to select a coupling without specifying one of the power components would be meaningless in the context of system performance because specific torque and speed requirements must be met. This parameter is given a high rank because it helps the designer to ensure compatibility - the rated power will be similar to surrounding components. The "size" of the coupling can be judged and finally, the closer the rated power to that of the other system elements, the more likely the coupling is to be efficient and economic.

7.8.4 **Angular Misalignment**

This has already been described generally in Section C5.4.20-C5.20.23. Manufacturers will quote the maximum safe value for continuous use - the rated value. In certain cases, the angle of misalignment and the rated torque / speed are dependent. For these instances, the maximum value for the misalignment must be quoted and the other parameter regarded as independent. This would cause the coupling to appear to perform better than it actually can, however, it would be unwise to select a component to operate continuously at its limits. This is a common-sense observation that most designers would be aware of.

For selection purposes, it is necessary to formally define angular misalignment. Angular misalignment is best defined as an acute angle ranging from 0^D upwards. Zero misalignment would mean perfectly aligned shafts. 45^D misalignment would mean severely misaligned shafts. Figure 7.9 shows how the angle of misalignment is measured. This is the definition used by most manufacturers.

7.8.5 **Radial Misalignment**

The maximum radial (or parallel) misalignment can also be quoted independently of torque and speed. It is defined as the maximum perpendicular distance between the axes of the shafts measured at the ends of the shafts (see Appendix C, Figure 7.5).

7.8.6 **Axial Misalignment**

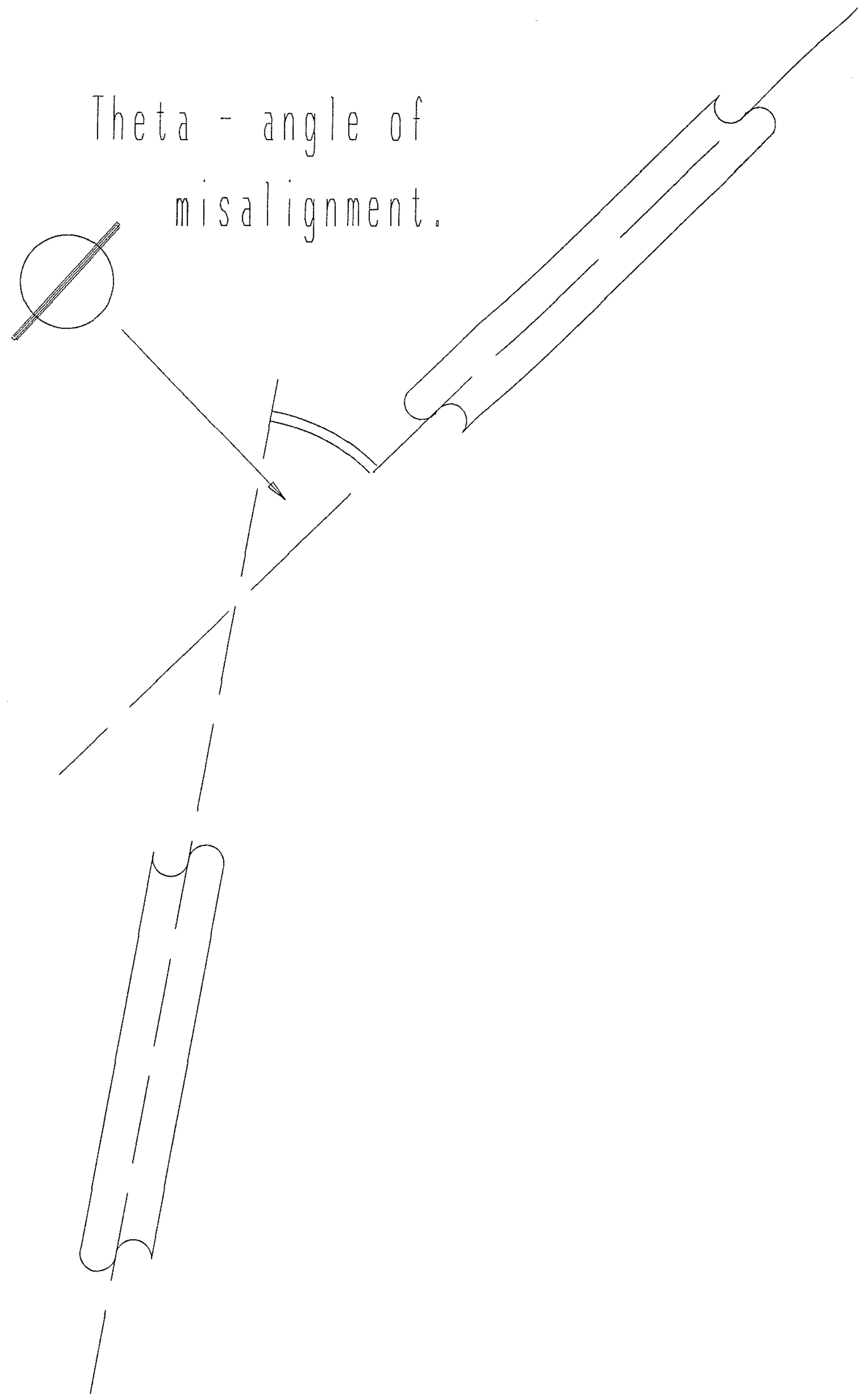
Also called "end float". This distance by which the ends of the shafts can move along the shaft axes. This is generally caused by thermal expansion of the shafts or by a slight amount of play in the shaft bearings. Again the maximum continuous value should be quoted. This measurement is illustrated in Appendix C Figure 7.4.

The above three misalignments are quoted as maximum values. However, this does not therefore imply that the coupling can absorb all of these maximums simultaneously. The designer must firstly by good design, attempt to minimise the system misalignments. If the coupling is expected to run at its misalignment limits the coupling must be selected with a sufficiently large safety factor to bring the stressing it will experience well within its design limits.

7.8.7 **Torsional Stiffness**

The ability of the coupling to resist twisting is a characteristic the designer may wish to know in cases where precise rotation of shafts is important - indexing, feeding systems, synchronised actions etc. The significance of twisting couplings was highlighted in Section 7.5.4.

FIGURE 7.9 : DEFINITION OF ANGULAR MISALIGNMENT OF TWO SHAFTS.



A problem occurs when attempting to assign a value of torsional stiffness to rigid couplings, which, for practical purposes, have a stiffness of infinity - they do not deform when a torque is imposed upon them. From a retrieval viewpoint entering data as infinite into records would be impossible and entering a very large number could be misleading. It would be better to leave the field blank because designers selecting rigid couplings could ignore this field. Also, the field; "coupling type" is available and "rigid" could be specified within it. This is not a perfect solution, and if the designer were to enter a value for torsional stiffness, no rigid couplings would be retrieved because a blank field will yield a "miss" if used for selection with a value anything other than "blank".

7.8.8 **Inertia**

Inertial information is readily available in most coupling catalogues. The significance of inertia was discussed in Sections 6.5.2 and 6.6.14 of Chapter 6. Thus, to ensure compatibility, inertia should be included in the selection list for couplings.

7.8.9 **Rough Bore and Flange Diameter**

Couplings are used to join components. Thus, they must be able to connect with adjacent components. Couplings are usually left with a rough bore which can be machined to accept keyways or bushes. The maximum final diameter of the finished bore should be quoted. If a flange is fitted, the flange diameter should be available. This will be the flange O.D. and not the diameter on which the centres of any pre-drilled holes lie. The reason for this is because the flange could be modified by the designer to ensure compatibility with adjacent components and hence, the basic "matching" dimension is more important.

7.8.10 **Maximum Diameter**

As with motors and gearboxes, dimensional information should be kept to an absolute minimum. This will ensure that the coupling is selected by performance rather than size. As a rotating device on a shaft, couplings tend to be cylindrical in nature. The length tends to be less relevant because they fit over shafts and the distance between flanged components should always be adjustable because of the lack of tolerance for this kind of fixing. Thus, the diameter of the coupling is its major dimension. It is important to ensure that the coupling can rotate freely and it is not in contact with surrounding structures.

The above parameters have been used as fields for records in a demonstration database. The conclusions as to their suitability for data capture and retrieval are discussed in Chapter 9.

CHAPTER 8

CLUTCHES, ROTARY DAMPERS AND FLUID COUPLINGS.

8.1. Introduction

The study of clutches, rotary dampers and fluid couplings builds on the experience developed the previous chapters. It will demonstrate that the principles, techniques and conclusions outlined in the preceding chapters are valid and will augment the research methodology which will be discussed in Chapter 10.

The components covered by this chapter, on inspection, seem very different, but using the technique of analogy, these components can be shown to behave similarly. This similarity can be taken further and a set of common parameters derived to describe the components. Figure 8.1 shows the breath of the components studied under the heading clutches, rotary dampers and fluid couplings. (Hereafter referred to as "clutches et al"). The reasoning behind how these components can be considered in parallel is discussed in Section 8.4 below.

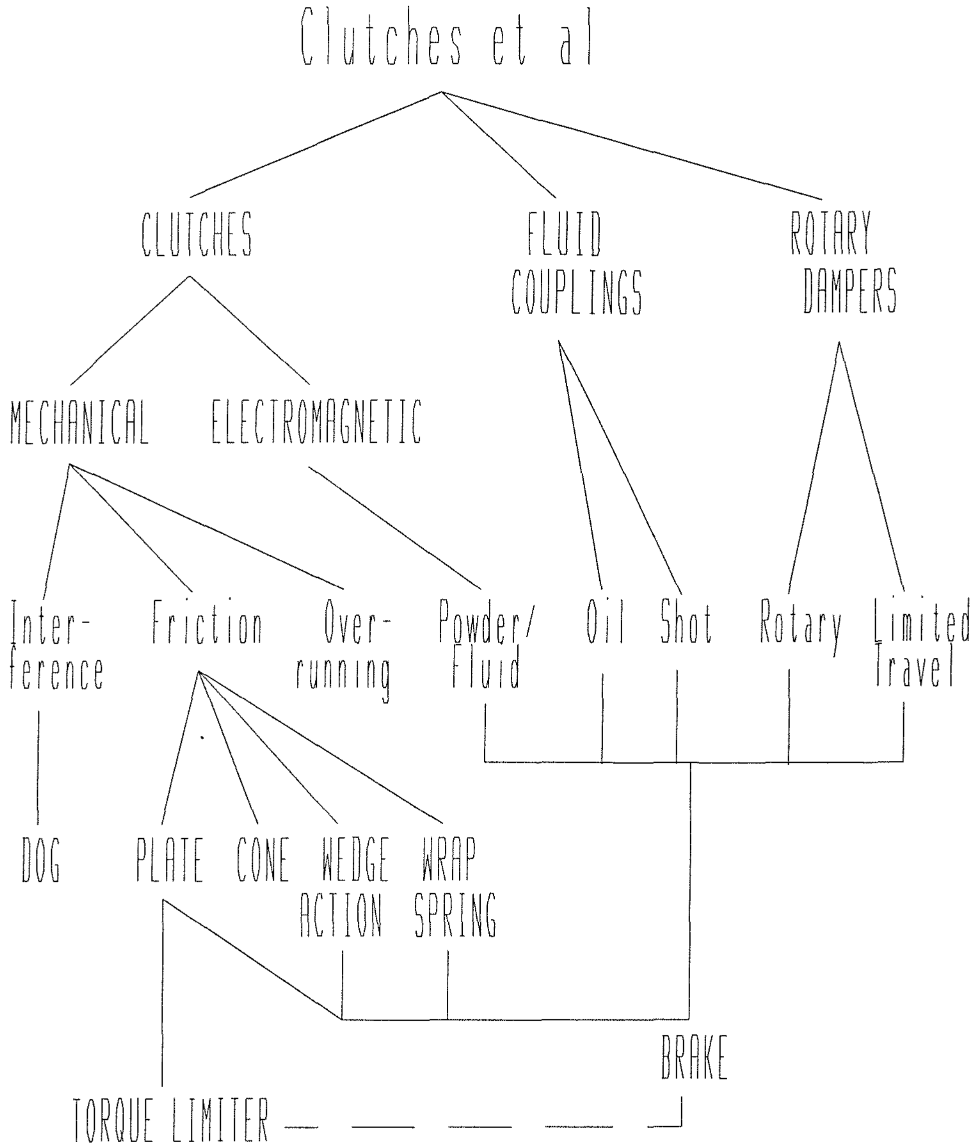
8.2 Scope

From the analogy matrix (Table 2.1), clutches have analogies with electrical switches. Dampers are analogous to d.c. resistors and fluid couplings have similarities with transformers. These elements can be studied in a similar way to motors. The analogies between the devices will be sufficient to give the research the depth required.

8.3 Generation of the Extended Parameter Lists

The extended parameter lists were created using the same techniques used in Chapters 3, 5 and 7. The component group clutches produced a great deal of information because of the many types and applications of these devices. Fluid couplings and rotary dampers produced less comprehensive lists, rotary dampers, in particular are limited to one supplier on the T.I. system.

FIGURE 8.1: The Breadth of Components Studied Under Clutches et al.



The extended lists were broken down into torque, speed, power, response and physical information. The broad heading "response" relates to the method of actuation and factors affecting the speed with which the device engages and disengages. This sub-list has been unnecessary with the other components studied. Gearboxes and couplings are permanently engaged and motors, although they must be switched on and off, produce sufficient starting torque to be considered "on-line" almost instantly.

The complete, extended parameter lists for clutches are shown in Table 8.1. Only part appears below. The complete table is shown in Appendix L.

8.4 Development of the General Analogies that Relate Clutches, Fluid Couplings and Rotary Dampers

Clutches cover a very wide range of devices whose functions are diverse. Figure 8.2, Appendix L, lists the many common type of clutch available. Reference 32, written by the author, through expertise gained in the research, explains the uses of all these devices.

Conventional clutches can be broken down into three groups. The engage / disengages type, such as a dog clutch, friction clutches, such as a plate clutch and over running clutches, the "sprag" clutch being the most well known.

The engage / disengage type require the input and output shafts to have no relative velocity. Conventionally, these devices are at rest when engaged. They are simple components requiring only a few performance parameters i.e. torque, speed and power would be adequate for a performance based selection. The term "friction clutch" covers a great many designs. The important characteristic of a friction clutch is that the input and output shafts can have a relative velocity when the clutch is engaged. If slip occurs, additional parameters must be considered and the selection is more complex. Over running clutches will permit the driven shaft to transmit torque in one direction but not in the other. Again, these are simple devices and although they have numerous applications, the parameters used to specify them are few.

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 ...

The friction clutch is a component used in a variety of applications and hence many designs have been developed. One of the most important considerations when selecting these devices is the ability to dissipate the energy generated when slip occurs. Hence, the "grey area" of clutching / braking. In many cases, the design of a clutch and a brake are identical. Distinction is made when selecting the device. In the case of a brake, more emphasis is placed on energy dissipation. Certain types of torque limiting clutches also have heat dissipation as part of their selection procedure. With all of these components, if the heat generated when slipping cannot be readily dissipated, the component will suffer from heating damage. Hence, a common, linking characteristic, is the ability to dissipate the heat energy caused by friction. Thus different types of clutch can be linked by the parameters concerned with heat dissipation.

Fluid couplings are also used as a type of slipping clutch where slip decreases as the driven shaft is accelerated. Thus, in the time taken for the driven shaft to accelerate, the working fluid is heated by "churning". This heat rise is often significant and the working fluid must be cooled before damage is caused. Thus, a significant part of the selection procedure is dedicated to calculating the heat rise on starting. When running at rated conditions, the slip is low and heating caused by churning reduced. The parameters for power transmission are now similar to a torque limiting clutch with a slight slip.

The rotary damper, by definition, dissipates energy. The parameter "damping rate" defines the torque per unit angular velocity absorbed by the damper. The power absorbed by viscous shear can therefore be calculated. Thus, the parameter power dissipation is also part of the selection process for rotary dampers. As a dissipative device, the rotary damper will have analogies with brakes and friction clutches. Also, rotary dampers and fluid couplings have the general operating similarity that the torque transmitted increases with input speed up to the power dissipation limit of the device.

If it were possible for the viscosity of the fluid used in rotary dampers and fluid couplings to be increased until the fluid became a solid, the component would then transmit torque with no slip and have no heating problems. This solidification process is performed in "shot filled" couplings. These are similar to fluid couplings except that the fluid is replaced by "shot". Once at operating speed, input and output are locked together by tightly packed shot.

The shot locks the system as it is thrown outwards by centrifugal force. Thus as the coupling accelerates, the shot appears to become more viscous. A similar viscosity change is achieved using electromagnetic powders and fluids. The attraction between the magnetic particles (and hence "viscosity") is controlled by the applied magnetic field. Thus, slip can be controlled though the apparent viscosity and hence the torque transmitted. Thus, there is a link between slipping friction clutches, torque limiters, fluid couplings and rotary dampers. Figure 8.3 summarises these relationships.

8.5 Discussion of the Analogies for Clutches et al

The characteristic lists for clutches et al have been moved to Appendix D. The details of the analogies are also within Appendix D.

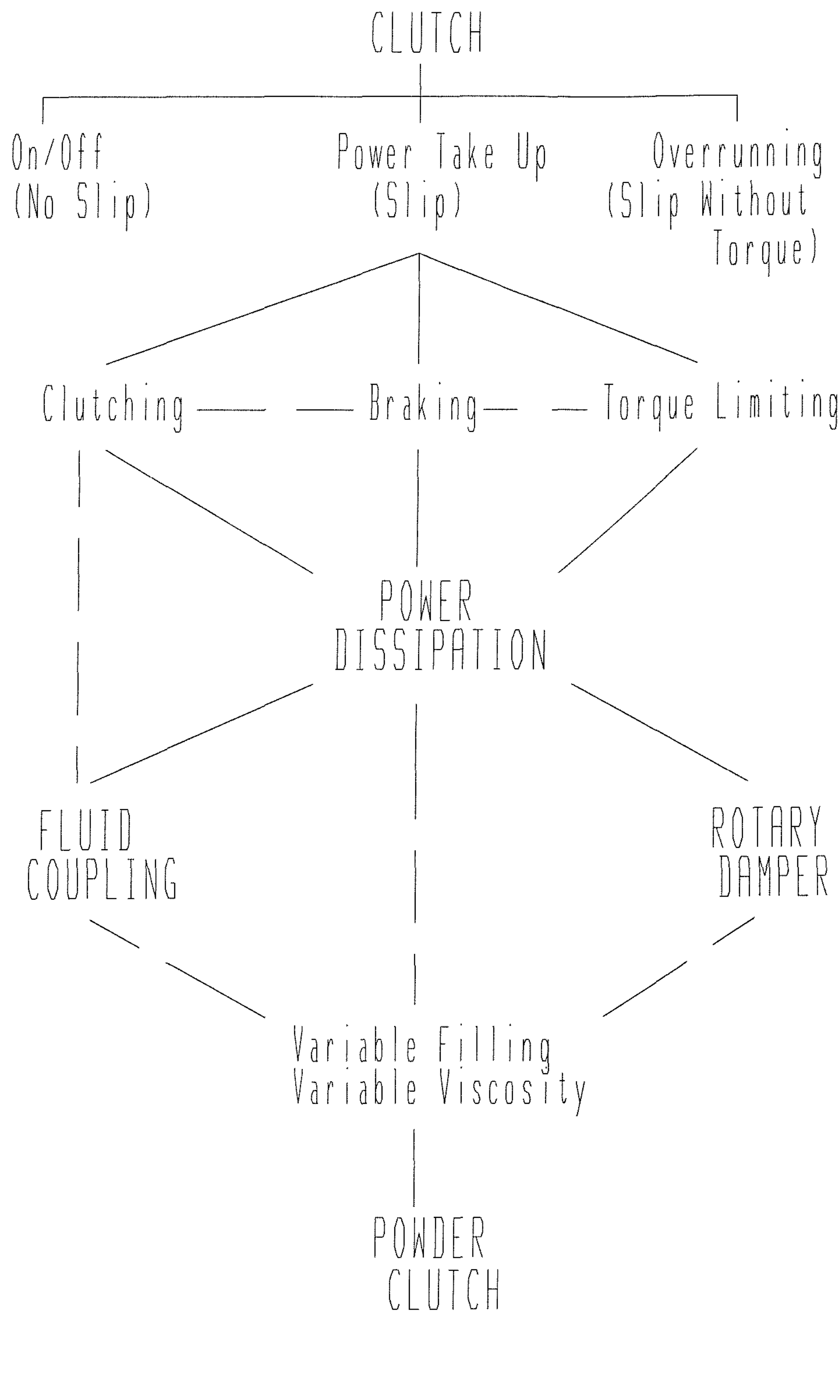
8.6 The Retrieval Parameters for Clutches et al

From the study of the extended character lists and consideration of the analogies between various types of clutch, a set of primary characteristics can be put forward for use with a computer aided retrieval system. The section below describes the need for each parameter and Section 8.6.2 will fully define each parameter for the main types of clutch.

8.6.1 Characteristics Used in the Retrieval Lists

Clutches are similar to couplings because they are connected between components and power flows through them. In the general case, it is hoped that the clutch will preserve speed and torque so that it does not waste system power. To propagate rotary motion in a system where losses occur, torque must be supplied and transmitted through the system. Thus, the rated torque that the clutch must transmit is important for selection and will be a quantity the designer will be aware of.

FIGURE 8.3 : GENERAL RELATIONSHIPS FOR CLUTCHES ET AL.



Having defined the rated torque the clutch must transmit, it would be logical to specify the speed related to that torque. However, a parameter that implies the rated speed and also provides an excellent feel for the size of the device is continuously rated power. Again, it is a parameter that should be readily at hand to a designer because it will be the system power. Designers will attempt to select components whose power closely matches that of the system power so that components are not oversized and run at near rated values. For instance, if a friction clutch is grossly oversized, the increase in inertia of the system will cause the prime mover to work harder to start the system. If the prime mover is forced to start within a more demanding duty cycle, its life will be shortened.

Having broadly sized the device using torque and power, the system maxima must be evaluated to ensure that the clutch can cope with the brief periods of overload. Thus, the maximum torque and the maximum speed the clutch must transmit should be specified. These extremes are not endured simultaneously. They are usually intermittent values. The torque limit is often set to avoid slipping and the speed limit is set to avoid damage by the centrifugal effects.

As highlighted throughout section 8.5 (Appendix D), the ability of a slipping clutch to dissipate the heat generated is vital to its performance. To avoid complex calculations by the designer, three simple characteristics can be used to specify the requirements of the slipping clutch. These values will be related to the continuously rated power and hence, no additional cooling will be needed for a clutch selected within these limits. The designer should readily know two of the following three parameters; continuous slip power, continuous slip torque and maximum continuous slip.

The continuous slip power is the power the clutch can dissipate whilst slipping continuously. It is not the transmitted power whilst slipping. Also the continuous slip power is not equivalent to the "engagement power" which is a transient characteristic and is described below.

The continuous slip torque is the torque available at the output shaft when the clutch is slipping. It will generally be less than the maximum available torque at the input shaft and is dependent on the design of the clutch. Thus, if a clutch is selected to transmit torque with slip, it is vital that the designer ensures that the slip torque is sufficient because it may be less than the continuous torque.

The maximum continuous slip is a measure of the speed difference between input and output shafts. The greater the slip, the more heat is generated. Thus, if a coupling is selected to run within the slip limit, the heat generated should not exceed that stated by the manufacturer for continuous use without additional cooling.

The above characteristics cover the selection of continuous heat parameters, however, most clutches will have an intermittent heat characteristic. As an intermittent value, it is larger than the continuously rated characteristic and will allow an efficiently selected clutch to cope with intermittent overloads, periods of high slip, starting conditions etc. The characteristics can be called response power because in most cases it is the large amount of energy dissipated when the clutch engages. A parameter closely related to the response power is the response time. This is the time taken for the clutch to engage i.e. produce continuously rated torque and power at the output. In most applications, the response time is fractions of a second and is ignored. However, in devices which have a run up time e.g. fluid couplings ,centrifugal clutches, the response time may be up to minutes.

The inertia of the clutch must be considered because it will affect the system response. To evaluate the starting torque of the system the inertia of the system elements should be added to the load inertia. In many cases the system inertia may be much greater than the system load e.g. high speed pick and place units for small components. Also, large fluid couplings and multiple plate friction clutch have high inertias compared to a compatible electric motor. Much like couplings, clutch manufactures tend to provide inertial data for their components so the information is readily available.

Having defined the general performance of the clutch using the above characteristics, details such as the method of activation and reset can be specified. These details tend to be among those that the designer has specified before the performance characteristics. They form part of the basic description of the clutch. Remembering the research philosophy will explain why these characteristics are not at the beginning of this section. The exact way these characteristics can be applied to clutches et al is detailed in Figures 8.4 and 8.5. These lists have been moved to Appendix L.

The method of activation and reset combined with the performance characteristics given above should allow someone familiar with clutches to identify the clutch type. Although it could be argued that the above characteristics are sufficient to define a clutch completely, the option to "name" the clutch should be available if a specific type must be selected. How this parameter is handled for specification and retrieval is discussed below.

Dimensions and mounting type must also be considered. As cylindrical devices, the dimensions of the clutch can be reduced to the major parameters; axial length and diameter. Specific dimensional details can be gleaned when final selection is made. The mounting type for the clutch is an important consideration when integrating it into the system. The adjacent elements may have different mounting types and thus the clutch may have different mounting arrangements at each end. Thus, the user must be able to specify one or two mounting types as well as being given a "help" screen listing various common options.

Finally, the selection of components from particular manufactures must be possible because of the policies and preferences of establishments and individuals. This parameter is not given a high priority because it undermines the breath of retrieval.

For retrieval, the manufacture's identification code and the location of the component on the T.I. microfilm system must be available.

The short parameter list is given in Table 8.2 and the applicability of the characteristics to each component is summarised. The exact definitions of the characteristic for each type of component are given below.

8.7 The Formal Definitions of the Characteristics the Retrieval Lists

As can be seen in Figure 8.1 there are many types of "clutch" covered in the research. Where a characteristic can be generally applied to a component, no reference will be made to a particular component. Characteristics that need clarification or are not applicable to a component type will be defined for that particular component.

TABLE 8.2 : SELECTION CHARACTERISTICS FOR CLUTCHES FLUID COUPLINGS AND ROTARY DAMPERS.

SELECTION PARAMETER	CLUTCH	FLUID COUPLING	DAMPER
Continuous Torque	*	*	*
Continuous Power	*	*	*
Maximum Torque	*	*	*
Maximum Speed	*	*	*
Maximum Continuous Slip	*	*	*
Continuous Slip Torque	*	*	*
Continuous Slip Power	*	*	*
Type of Clutch	*	*	*
Response Time	*	*	/
Response Power	*	*	/
Method of Actuation	*	/	/
Method of Reset	*	/	/
Diameter	*	*	*
Length	*	*	*
Mounting	*	*	*
Inertia	*	*	*
Company Name	*	*	*
Frame Code			
T.T. Microfilm Number			

KEY : * - SHOULD BE AVAILABLE THROUGH MANUFACTURER\S LITERATURE

/ - NOT APPLICABLE

8.7.1 **Rated Torque**

The torque the clutch can transmit continuously without damage. For most friction type clutches this will be the static torque (see Appendix D, Section 8.5.4). For brakes it will be the rated brake torque which should be quoted by the manufacturer. In torque limiting devices, it will be the maximum torque limit. In over-running devices it will be the continuous locked torque. Fluid couplings and rotary dampers both run with slip and hence the torque limit will depend on the limit for continuous power dissipation. This value should not have to be calculated for rated conditions and should be quoted.

8.7.2 **Rated Power**

The power the clutch can transmit continuously without damage. For most friction clutches and torque limiters this will be the power transmitted with no slip. In brakes it is the continuous brake power which will be available under slipping conditions. Over-running clutches can only transmit power in the locked configuration and this should be quoted by manufacturers. If not, the locked rated torque and rated locked speed (both of which should be available) can be used to calculate the rated power. For fluid couplings, it will be the power available at the output shaft when the input is running at rated torque and speed. For rotary dampers, it will be the power dissipated by the damper with the input running at rated torque and speed with the casing locked.

8.7.3 **Maximum Torque**

For friction clutches and torque limiters, this is the maximum torque the clutch can transmit before slipping, it may be the same as the continuously rated value. For brakes, it will be the maximum brake torque which may be an intermittent value. For magnetic powder / fluid clutches it will also be an intermittent value dependant on the overloads the electrical supply is capable of handling. For over running devices, it will be the maximum torque the clutch can transmit without damage to the over-running mechanisms. This will be an intermittent value designed to cope with the sudden shocks encountered when the clutch engages. Fluid couplings and rotary dampers have a maximum intermittent torque limited by the heating considerations.

8.7.4 **Maximum Speed**

This is an intermittent value. It is quoted in most cases to avoid damage through centrifugal forces or to limit the engagement speed so that excessive heating is avoided. This is particularly true for fluid couplings and rotary dampers which can exceed their rated speed limit for brief periods to allow acceleration or extra dissipation of energy.

8.7.5 **Continuous Slip Power**

It is very important to note that is the power "removed" from the system power when the device slips. This parameter is not applicable to over-running devices because when over-running there is no significant power loss. When locked, there is full power transfer. Fluid couplings run with slip under all conditions, the heat generated at rated values is what must be quoted in this case. Rotary dampers have a similar characteristic and a continuous power limit is usually provided by manufactures.

Friction brakes will certainly have a continuous slip power specified by the manufacturer. The designer will also be aware of this value because the amount of energy needed to be removed will be known. Friction clutches are usually run with minimum slip to avoid wear to the friction surfaces. However, certain types ("wet clutches") are capable of continuous slip for low values of slip. In these cases, the continuous slip power will be quoted by manufacturers. Friction clutches used as torque limiters where complete disengagement does not occur and electromagnetic powder / fluid clutches running with slip will also have this value available.

8.7.8 **Continuous Slip Torque**

This characteristic is not applicable to over-running devices although there is a small drag torque (caused by the freewheeling mechanisms) when the device is over-running. However, it is not a useful torque and need not be considered. For fluid couplings, it is the torque available at the output shaft for rated input conditions and therefore is no different from the rated torque of the device. A similar definition exists for the rated input torque for rotary dampers.

The continuous slip torque is significant for torque limiters because it indicates how the device is to react under overload conditions. If the slip torque and rated torque are identical the device ensures constant torque. If the slip torque is zero, the limiter completely disengages on overload. Any intermediate values of slip torque denotes that the device slips with reduced torque and will also suggest that this value is controllable by some means. The method of torque control is a specific detail that can be considered once an appropriate device has been selected.

8.7.7 **Maximum Continuous Slip**

Over-running clutches and rotary dampers will have this value set at 100%. Fluid couplings have a rated continuous slip of 3-5%. By varying the amount of oil filling this value can be increased without additional heat generation. For fluid couplings capable of this action the manufacturer will quote the slip value or the output speed range. Quoting the output speed range will allow slip to be calculated.

Torque limiters which completely disengage will have a maximum continuous slip of 100%. Those which slip at maximum or reduced torque will have the slip range specified by the manufacturer. Friction clutches designed for slip and torque limiting applications will have the maximum safe slip value quoted.

8.7.8 **Response Power**

This is the intermittent maximum power that a clutch can dissipate whilst engaging. It is the "engaging power" for friction plate and centrifugal clutches. For fluid couplings it is the starting or run up power.

Rotary dampers are continuously engaged. The power dissipated depends on the input speed. Hence, the maximum power that can be dissipated depends in the maximum input speed. The maximum input speed will be an intermittent rating. It can be evaluated from torque speed graphs or may be quoted directly. Given the maximum input speed, the maximum intermittent power can be calculated. For brakes, it is the maximum intermittent braking power.

8.7.9 **Response Time**

This is the time taken for the clutch to activate. For over-running devices, the time taken to lock can be considered almost instantaneous and therefore zero. Most friction clutches are designed with minimal plate clearance and if selected correctly, will be able to accelerate the load, without snatching in milli-seconds. The response time is often quoted by manufactures of electromagnetically actuated clutches because it is a function of the electromagnetic circuit.

Fluid couplings and centrifugal clutches have a run up time dependant on input shaft speed. For rated input speed, the full output torque will take from seconds to minutes to develop at the output. The run up time will be quoted for given input speeds.

Electromagnetic powder / fluid clutches have a response time dependant on the magnetic field build up time, for rated values, this will be milli-seconds.

8.7.10 **Inertia of Rotating Parts**

Many clutches consist of input and output parts and therefore manufactures define these separately as they will effect the adjacent components differently. This is best illustrated by a clutch used to separate a motor from the load. If the clutch is disengaged and the motor started, the motor must accelerate the input side of the clutch. When engaged, the motor must accelerate the inertia of the load and the output side of the clutch. Once running, the total clutch inertia should be lumped with the system inertia and will therefore change the system response. For retrieval it is suggested that the total clutch inertia is used because the inertia of the clutch parts will generally be small compared to the system inertia.

8.7.11 **Method of Actuation**

The various options available to the user are detailed in Figure 8.4 (Appendix L). For retrieval purposes the options will be available through a help screen.

Centrifugal, friction and over-running clutches will have both method of actuation and method of reset as "automatic". Fluid coupling and rotary dampers will have "continuous" for method of action and "N/A" (not applicable) for method of reset. Torque limiters will have method of actuation as "automatic" and method of reset as one of the options shown in Figure 8.4.(Appendix L)

8.7.12 Method of Reset

The various options available are detailed in Figure 8.5 (Appendix L). For retrieval, these options must be displayed on a help screen.

8.7.13 Type of Clutch

The research has identified the major types of clutch and these can be displayed as options on a help screen. These clutch types are listed in Figure 8.6.(Appendix L).

8.7.14 Dimensions

The dimensions for clutches et al are summarized in Figure 8.7 (Appendix L). As rotating devices on shafts, clutches tend to be cylindrical in nature. The axial length excluding protruding shafting but including other fixings such as flanges, pulleys etc. can be easily read off dimensioned drawings. The maximum outside diameter is also readily available. It is unlikely that axial length will prove a great problem in system assembly. The diameter of many clutches is large in relation to components of a similar power rating. Hence clearance with the surroundings must be checked.

8.7.15 Company Name

This is simply the option to choose a particular manufacturer when selecting a clutch. A help screen giving a list of clutch manufacturers in the database should be available.

8.7.18 Component Frame Code / T.I. Microfilm Number

These characteristics will not be known by the designer but must be provided by the retrieval system.

CHAPTER 9

APPLICATION.

9.1.1 System Requirements

As described in the previous chapters the end results of the research work were extensive lists of parameters which should be used to specify components. These complete lists could be used by manufactures to restructure their catalogues. The catalogues would then become more useful to designers because they would provide all the information they need in a consistent format. If manufacturers catalogues were more consistent, component selection would be simpler because data could be quickly extracted. There would be a less repetitious calculations and comparisons between components from differing manufactures could be made. The characteristics identified by the research may cause designers to consider parameters they had previously ignored or had not considered significant. This would not only be beneficial for components from the same engineering field but would allow direct comparisons between components from many engineering fields. Thus the merits of electrical, pneumatic and hydraulic systems could be compared directly. If manufactures were to restructure the data in their catalogues into a format so that the former propositions were fulfilled, a great deal will have been done to simplify the process of design (REF. 1).

The research also developed sorter lists for component description, these lists were deliberately designed to be used as the format for the records in a computer aided retrieval system. It would be these parameters the designer defined in order to retrieve a component.

It was realised early on in the project that the databases would have to be set up by the Technical Indexes Ltd. (T.I.). The resources on hand at T.I. would allow this labour intensive task to be done quickly and in parallel with the research work. Chapter 1 noted that the hardware to scan pages of information into a computer is available. If the scanning process is correctly set, only the relevant information will be captured from the page. This data could then be post processed into the form required by the database. This post processing would allow preferred units to be used or information to be derived from the captured data. Thus, the daunting task of manual data capture (estimated by T.I. to be 5 man years for the fifty thousand records for the motor database) will hopefully be reduced to a much shorter time utilising less qualified staff. The "expert" staff would only have to set up the data capture technique and define the requirements of post processing. This task could be simplified by developing software geared to allow these tasks to be quickly defined and handled.

An estimation of the data capture times is shown in Appendix M.

A further possible option for creating the databases is that manufactures could create databases compatible with the T.I. database but containing only there own products. This would allow T.I. to simply merge all the company databases and offer a comprehensive service similar to their current system. This would also reduce the time needed to construct the large databases required for a thorough system. Individual companies would benefit by having their components in the database and could also set up their own in-house systems. They would essentially be replacing their catalogues with a computerised version, a development which many already have in place.

Thought was given to assessing the best method of delivering the system to end users. A multi-component database would be a quarter of a million records with eighteen fields per record, the fields would contain a mixture of alpha numeric and numeric data. The component data files would be around five mega bytes and the associated indexing files would be roughly the same size. The most obvious delivery method would be to use a mainframe and for users to access the database via a modem. The advantages of this system would be that T.I. would only have to maintain one database. The power of the mainframe could be put to use; access to a large amount of up to date information, rapid search times and the user is only invoiced for time spent using the system. The disadvantages would be that the user would need a modem and be prepared to wait for results if there were many other users. Attempting to download sections of the T.I. database into the users computer would prove to be time consuming and it would be difficult to ensure simple and accurate transfer because of the various incompatibilities of the hardware the market. It should be noted that with the success of the "Internet" and the variety of computer "dial up" services available, data transfer down phone lines can be considered routine.

An alternative would be to make the system PC based and offer the user sectionalised versions of the full databases. The obvious delivery method would be floppy disks. The growth of optical disk technology (CD ROM and WORM disks) offering the advantages of very large data storage and robustness makes a "disc through the post" system very easy. With a floppy based system, users would be able to tailor the choice of database sections to exactly their requirements. Updating by post would be cheap and floppy disks can be reused. With regards to optical disks, read / write and read only types are available. The choice of which medium`to use is an economic one. Read / write favour small batches of product. Read-only is best suited to high volume production. One notable user of a read only catalogue on CD is RS (Radio Spares) who now offer their catalogue on CD ROM. This allows their customers with a PC and a CD ROM drive to rapidly interrogate the very large RS catalogue for suitable products. Although at the moment non-reusable and more expensive than equivalent number of floppy disks, CD's are relatively cheap for the amount of data stored. The price of reusable optical discs price will also fall as their application increases.

The demonstration system was therefore, to be PC based. The major reason that a PC based system was chosen is that desktop computing is commonplace and most companies will have a PC available to try out the system.

9.1.2 **Software Requirements**

The research produced sets of parameters for describing components which would lend themselves easily to be handled by a computer aided retrieval system. This would allow many of the commercially available data base management systems (DBMS) to be used. An attempt was made to define the appropriate software through liaison with the Technical Indexes systems analyst. This involved completion of a standard layout for the description of business software. This was called "The Statement of Requirements" it is reproduced in Reference 30, Appendix 10.

T.I. were unable to identify a suitable software system through the "Statement of Requirements" and therefore the University was forced to investigate this area because setting up a demonstration system was of great importance to the success of the project.

Thus, sources such as PC magazines and the PC Year book were consulted so that DBMS's could be identified and investigated.

The requirements for the software as seen by the University were ;

1. PC based.
2. Language interface (Basic or Fortran preferably).
3. Alphanumeric and numeric searches.
4. Relational and logical operators handled (=,>,<,and, etc.)
5. Large databases could be searched rapidly.
6. Relational databases could be constructed.

As discovered in Chapter 5, it would be necessary at times to present the user with pictorial information in order to clarify the required input. This type of prompt is defined as a "graphic prompt". The prompt may take the form of a simple picture making the user aware of the options available for selection i.e. the various mounting types for motors. The prompt could also perform more complex tasks, it derives information for the user. In this case, the user is asked supplementary questions and the answers processed into the form needed for retrieval. The derivation of the shaft configuration code for gearboxes (see Appendix G) is the best example from the components studied so far. These types of prompts, particularly where graphics are used are not readily available through most DBMS's. The reason for this is that they have their own command languages. These languages are designed to handle the data base records and fields within the records, being able to produce graphics is out of their specification. However, there is sometimes a facility to communicate with the operating system or a higher level language. If communication with a higher level language could be established, the power and flexibility of the language could be used to generate the graphics, the user responses manipulated and the results sent back to the DBMS to perform the retrieval. The exact details of this mechanism such as whether the DBMS call the language or the language calls the DBMS will be discussed later.

The use of Basic or FORTRAN was the result of the author's previous experience with these languages. The University expected to do very little programming making learning a new programming language unnecessary. Unfortunately, the University undertook all the programming because T.I. did not have the internal expertise or the financial resource to commission a software house. The language interface chosen was Basic because it was recognised that results would appear faster using a language the author was familiar with.

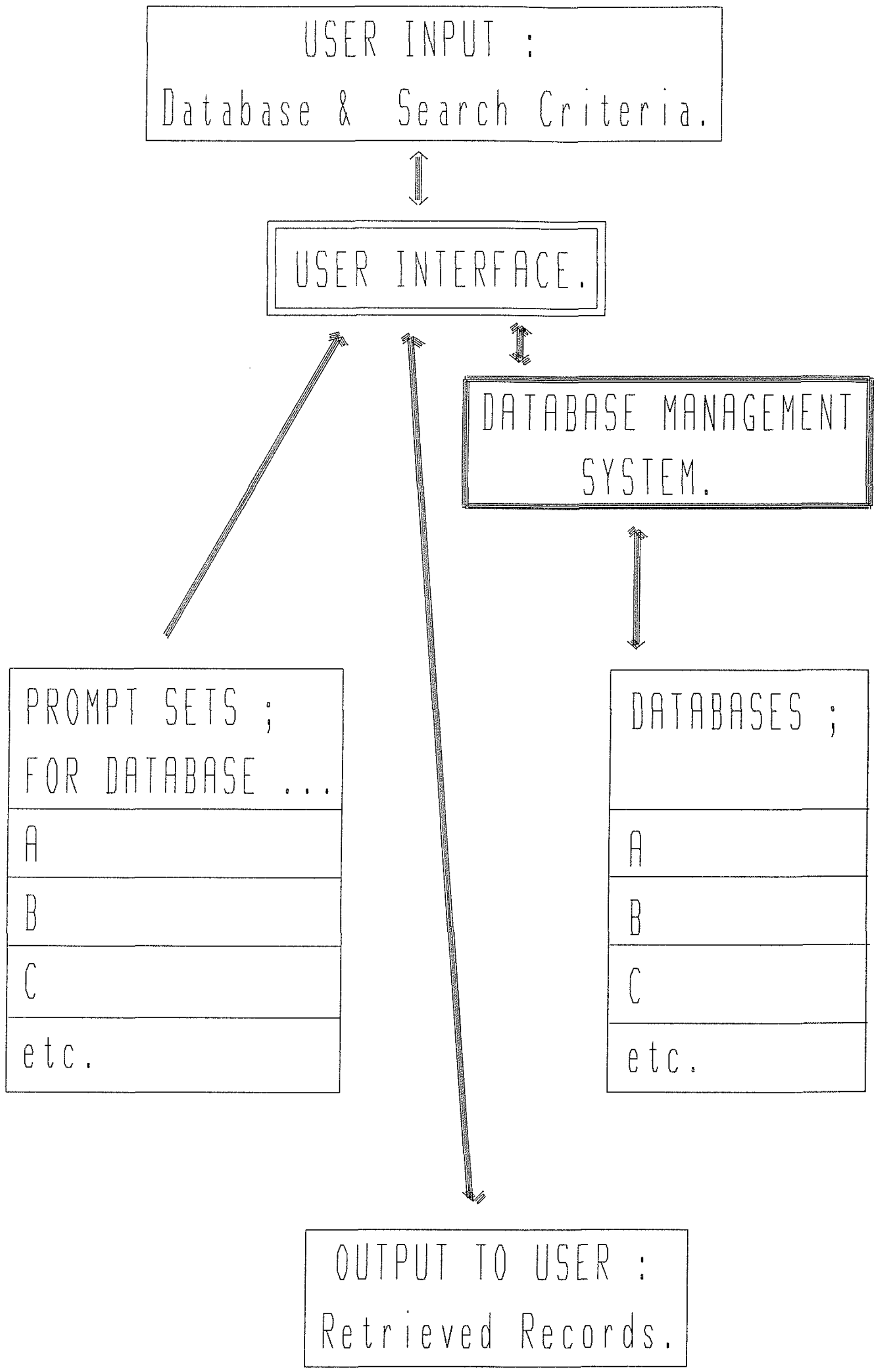
The DBMS would be expected to retrieve both numeric and alphanumeric data. Additionally, if a relational database were to be used, company records could be included. This would simply be the full address and telephone number of the suppliers of components in the database. The company record and the component records would be linked by the common item "Company Name", thus having retrieved a component record, supplier details can be retrieved using the company name.

The numeric data held in the database will be discrete, real numbers, the user will require discrete value answers but will associate a tolerance with them. The user will be looking for a component whose characteristics fall within the predefined tolerances. Thus the DBMS must be able to handle the relational operators and combinations of them. The most common form being "> x and < y". Hence the logical operators (AND, OR, NOT etc.) must also be handled. The most important logical operator for this application is AND. The AND operator could be implied by performing two searches, but this would be crude.

9.2 Design

The DBMS would have to be able to access and retrieve data for many different components. The separate component records would have a very similar format but data within the records would be different because it would only be relevant to that type of component. Ideally the retrieval process would be the same for all components, only the database and the characteristics searched for would change with the component being selected. Also, each database would have a set of graphic prompts associated with it, these would somehow have to be "pulled in" by the DBMS so that they can be displayed when needed. The advantage of such a system is that it can grow quickly. The only development which must be carried out would be creating the databases and the graphic prompts. The retrieval program can then access any of the databases and a retrieval made. Figure 9.1 is a schematic representation of how such a system may work.

FIGURE 9.1 : Schematic of Retrieval System.



9.3 Implementation

Section 9.2 described how the system was envisaged working. This section describes how it was actually done and contains details as to the mechanisms of the DBMS which must be described so that the software can be understood.

The DBMS chosen for the demonstration system was "Superfile 16". It was one of the few commercial packages that met all of the requirements of Section 9.2. The database manager is memory resident program which means it is loaded and then remains in the machine "ontop" of the operating system. As a resident program, it remains dormant until called, so operating system commands can still be executed. In this case, the operating system is MS DOS.

The DBMS operates by loading additional application modules (written in C) which allow the user to construct and interrogate databases. Data is then communicated to the database manager via predefined areas in system memory. These application modules act as "user friendly front ends" enabling the user to prepare data for use by the data base manager. Superfile can also be controlled by an application program written in other programming languages i.e. Basic. The problem of compatibility of a program written in Basic and the memory resident program written in C is overcome by an interface program provided with Superfile that takes the output from the Basic program and puts it into a form acceptable to the data base manager. All that is required of the Basic program is that it presents the information to the interface in a predefined format. Thus, a simple programming language like Basic can be used to perform graphics etc. whilst the power of the DBMS can be used for data retrieval. The problem with this approach is that an entire application program must be written. If one considers the alternative solution, where the DBMS calls the language. Only the tasks requiring Basic (i.e. graphic prompts) are required and the rest of the user inputs can be handled by the already written application programs. This may sound trivial but it is very significant. The special application programs are sophisticated in the way they handle and control data. They are designed to exploit the DBMS to the full. Consider now a program written by the author, attempting to mimic these programs and provide the additional facility of graphic prompts. Clearly, the sophistication of the data verification and speed of operation will not be as good as the professionally written software. Also, from an academic viewpoint there is nothing gained in and developing such a program. Thus, it should be noted that if the DBMS could call the language, only the special programs would have to be developed, whereas with the system chosen, an entire application program had to be created. Given this point, the retrieval program written has been successful, very user friendly, and at the time of writing was much more attractive to use that its commercial equivalent. The structure of the retrieval program is shown in Figure 9.2.

The databases are constructed using an application program provided with the Superfile software. This program does not need the graphics facility because data is being entered by personnel who fully understand the data formats. The extensive data verification facilities allow the data entry to be carefully controlled so that few errors are made.

The original application program suffered from being slow. It was slow to respond to user inputs and slow in finding records. The response to the user is due to the efficiency of the front end program. The time taken to load and run graphic prompts was particularly noticeable. This load speed was increased when the program was rewritten so that all the graphic prompts were included within the main program as sub-routines. This was an inelegant solution because the main program became very large. It now included all the information for all the databases so far created. Each set of graphic prompts takes about 7k and the retrieval program itself is around 10k. Thus the retrieval program and a ten component database would be 80kb. Noting that most modern PC's have a minimum working memory in the tens of Megabyte range, the program size is unlikely to be a problem. Graphic handling speed was greatly increased by compiling the program. The slow loading of the graphics prompts was replaced by an almost instantaneous change to the graphic. The whole system appeared more professional because of the "slick" screen handling. Thus the responsiveness of the system became more acceptable.

Having increased the speed of response to user inputs, the more significant issue, the rate of retrieval of records, needed to be addressed. With compilation, the rate of retrieval did not increase a great deal. The retrieval rate depends much more on the speed of the PC. The largest database at the moment is around 1000 records. Searching this database with a worst case set of criteria, the software was tested for retrieval rate. The assessment was done on a variety of machines, the slowest having a 6 MHz clock speed, the fastest at

FIGURE 9.2 PT1 : Simple Flow Chart for the Retrieval Software.

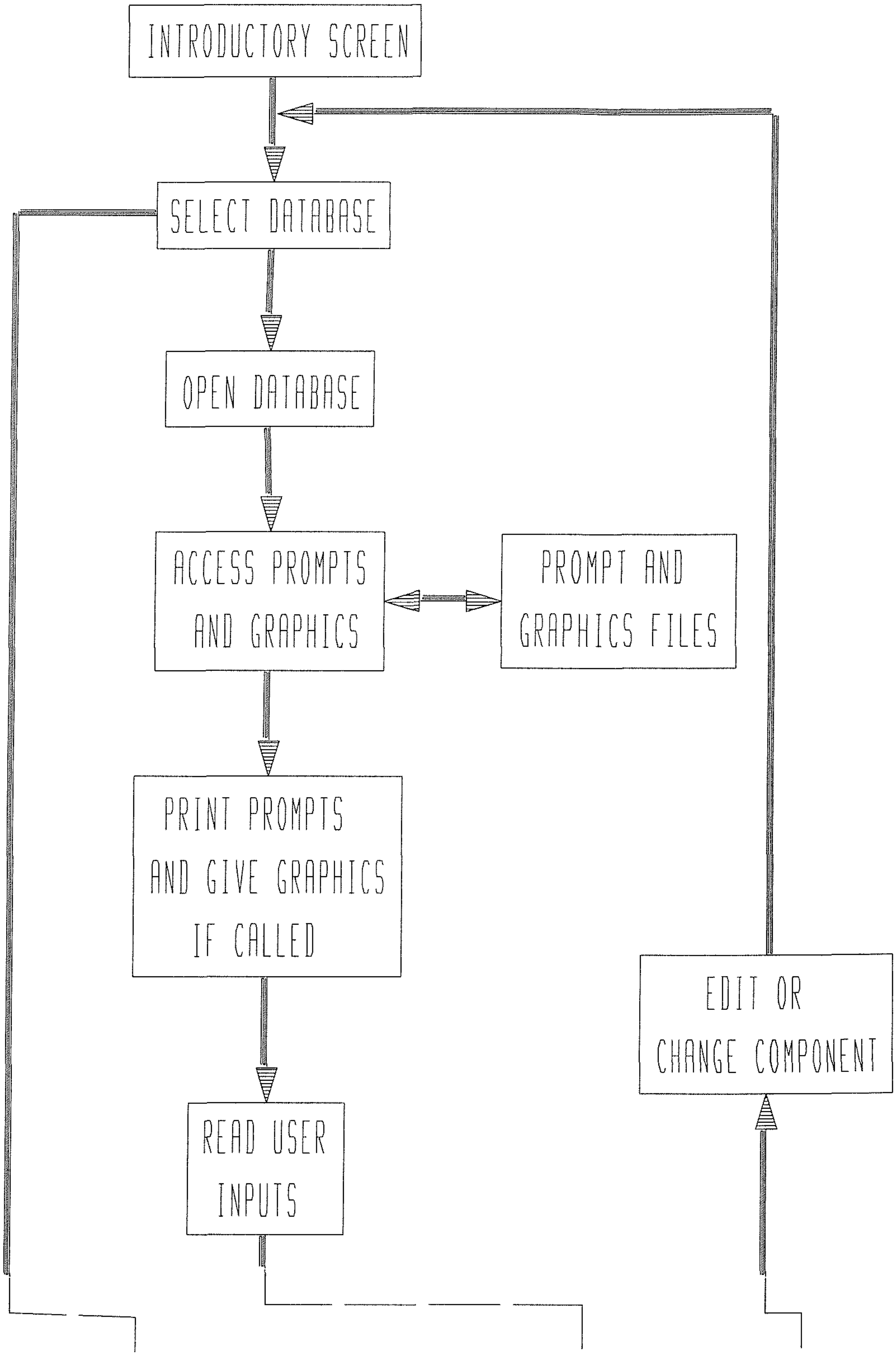
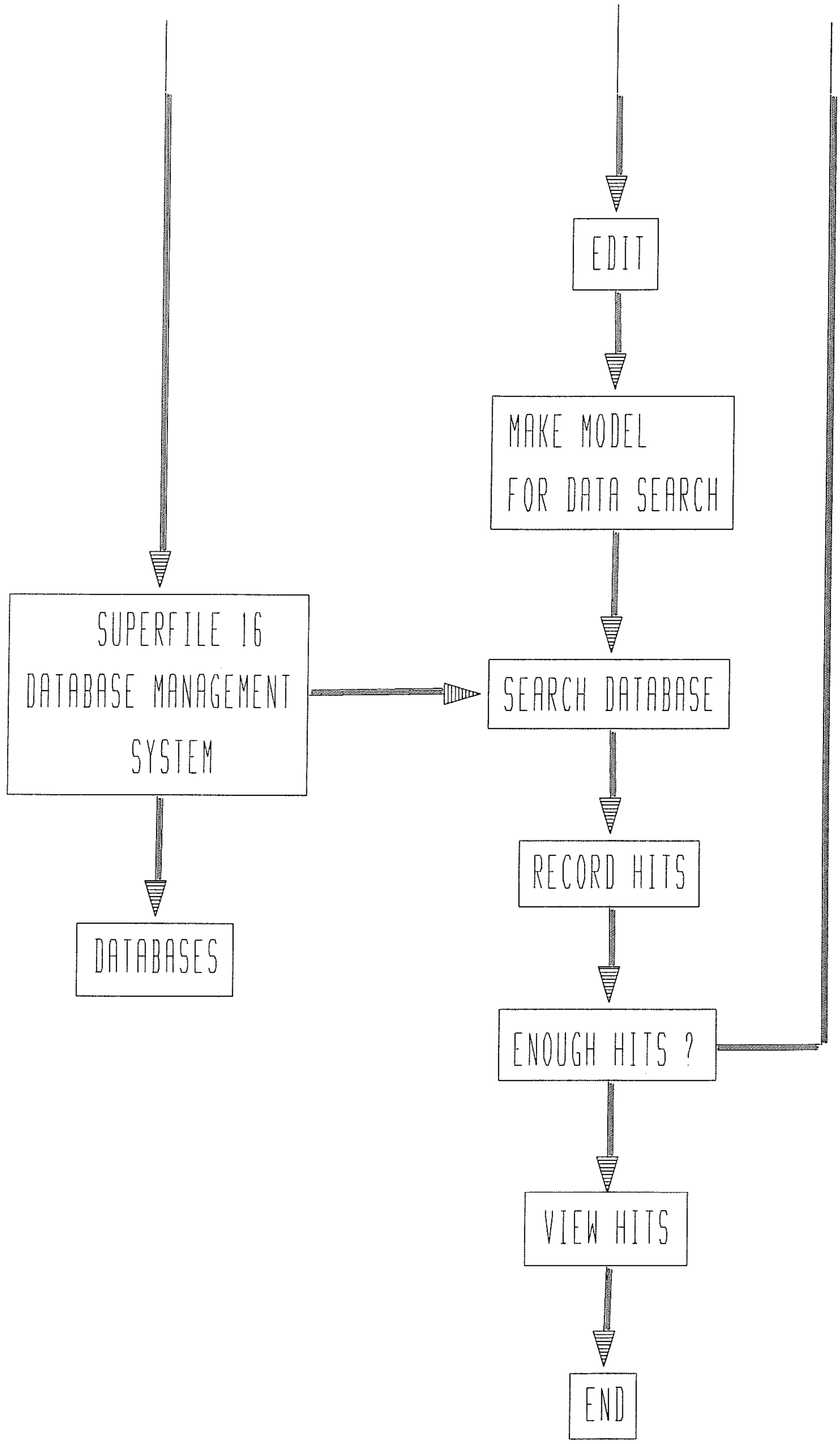


FIGURE 9.2 PT2 : Simple Flow Chart for the Retrieval Software.



16MHz clock speed. All of the machines were 8086 or 80286 machines. Retrieval time for 1000 records ranged between 220 and 40 seconds. Thus, the worst case search for a full database (50,000 records) would take about half an hour on the fastest machine. Noting that modern machines are now averaging 400 MHz clock speeds and have faster architectures than the "803xx" series ie Pentium, Pentium II and III, these search times will become less significant. Also databases will be offered in sectional form and the user will search within certain constraints. The DBM will thus be handling a smaller amount of data and searching on a restricted number of record files. The worst case in these circumstances would be a few seconds.

Breaking the databases up into smaller sections will help to speed up data retrieval and will allow designers to stay within their "power range". The databases would be a series of overlapping sections. Although overlapping, it is important to select the correct ranges. In the case of motors, the power range 1 to 10 kW is considered the medium size motor range despite the fact that motors have ratings from fractions to thousands of kilo watts. Thus, suitable power bands for motors would be below 1 kW, 0.75 to 15 kW, 10 to 100 kW and above 100 kW. Creating theses databases individually would create administrative problems for T.I. Ideally, one large database should be created which could then be used to generate smaller databases. This is what has been done T.I. enter all the data into one large database. Copies of this database can be "trimmed" to the appropriate "record range" by deleting records outside the specification. The trimmed database is then "tidied". Tidying means that the software rewrites the database indexes, optimizing them for the new database. This new database can be re-named making it unique.

9.4 The ROCCI Field Trials

9.4.1 Introduction

With the retrieval program developed to a usable state and with three reasonably large data bases prepared by T.I., the system, christened ROCCI (Retrieval of Compatible Component Information) was tested in-house by T.I. It then underwent field trials. This consisted of demonstrations on the premises of potential users and recording their comments on a formal (questionnaire) and informal (hands on and comments) basis.

To assess the parameters identified for selection and to test the software on naïve users, the ROCCI system had to be offered to potential users in industry and their reactions noted. Establishments using the T.I. system to select power transmission components could be identified by T.I. However, to make an on-site demonstration worthwhile, designers selecting components already on the ROCCI system had to be found.

9.4.2 Identifying Potential Test Sites

A simple questionnaire was prepared by the University and Technical Indexes. The questionnaire is extensive and has been moved to Appendix N, Part 1. It was designed to establish the suitability of a company as a potential test site. The questionnaire was sent to companies selected by T.I. Those replying who met most of the requirements, would be contacted to arrange a site test.

One hundred T.I. users were contacted, 29 responded, and 9 were selected for field trials.

9.4.3 The Field Trials

The trials were conducted by T.I. staff who were trained in the use of ROCCI system and followed a pre-defined format. The format was constructed by T.I. and the University to enable the demonstrations to be conducted consistently and scientifically so that specific questions could be answered and comments focused. A full copy of the demonstration notes is given in Appendix N, Part 2. The demonstration could be divided into two sections; the first section was carried out before the system was seen to avoid any prejudice against, or influence from, the system. The intention of this section was to cause the users to describe how they currently selected components with or without the T.I. microfilm. They were asked to list the parameters that they used at the moment to select components. Also, they were asked to select one of several statements which attempted to summarise the attitude and expectations of the user when selecting components. The questions were, however, phrased so as to reflect the aims of the ROCCI project. This caused the user to comment on some of the fundamental aims of the ROCCI system whilst not being aware of it.

Having completed the first part of the demonstration, the system was shown to the users.

During and after the demonstration, the users were asked about aspects of the system such as; usefulness of the selection parameters, clarity of help screens and how much a system might benefit them. Also, they were asked to price a system such as ROCCI if it were available to meet most of their expectations. This was an interesting question because ROCCI, although eventually intended to stand alone, currently only arguments the T.I. microfilm system. Thus ROCCI would impose an additional cost and its advantages may not justify funding to those who do not fully understand its power. Finally, any criticisms (apart form the size of the databases) were recorded.

9.4.4 **Discussion of the Field Trial Results.**

The characteristics listed by the users for component selection, closely matched those identified for use in the ROCCI system. The users considered all of the parameters available through ROCCI to be important.

Users were also allowed to add parameters that they would like to have available in addition to those provided. All of these additional characterises, except balancability of couplings are contained in the relevant extended parameters list in Chapters 4,6,7 and 8. Most of the additional parameters are not performance based but physical and environmental i.e. shaft sizes, cooling type. They also tended to be specific to particular types of component i.e. "bearing type" was added to the coupling list. Not all couplings have bearings and therefore this parameter cannot be used in a system set up to select on "performance" rather than "features". Remembering the need to keep the amount of information as low as possible to avoid excessive memory usage, including these additional parameters would not be advisable. The problem of being able to specifically identify a particular type of component is solved by the "type" field contained in the ROCCI selection lists. This will allow a component to be named and will therefore satisfy any need to identify the component type. The "type" option will generally describe the "features" the user associates with the component. The problem with specifying the "type" of component is that the breadth of the retrieval is ruined.

All the users found the help screens useful. The text screens indicating various choices were un-criticised. The graphics screens were noted to be of poor quality. The quality is purely a problem of hardware and better written software. The important fact is that the objectives of the graphics screen were seen to be correct.

The lack of shaft details, such as diameters and keyways for motors, gearboxes and couplings was highlighted by the users. Also, for the gearboxes, input / output details were expected. The absence of these parameters has already been explained in Chapter 4, 6, 7 and 8.

The ROCCI system was seen as an improvement over the T.I. system because it allowed the rapid and accurate access to a varied selection of components using a simple, standard format. This was one of the aims of the project. The perceived price for the system (software and databases) varied from a few hundred to the high thousands of pounds. This wide range was therefore, fairly unhelpful to T.I. as a guide to pricing the system. It should be noted that the Superfile 16 database management system that the ROCCI software uses, costs around £1000. This cost would reduce with the multiple orders but would not become a trivial amount. The ROCCI software is free but the cost of building the large databases would also add to the package cost. An estimation of the time required to capture the information is shown in Appendix M. To gain a reasonable profit and to maintain such a system, T.I. would have to charge a least £5000 for a system including at least three databases. If ROCCI were to stand alone, it would probably undercut the current T.I. rental costs. If used to supplement the microfilm system, it would represent a considerable additional investment. The time saved by ROCCI can be rarely accepted by the financial controllers within organisations.

Finally, any criticisms of software architecture were noted. These are summarised as better editing facilities. This means more programming.

The conclusions of the field trial report were that ROCCI was technically competent. Systems of the ROCCI type would be used by industry. Large, comprehensive databases must be available and their price must justify what is on offer. T.I. felt that cost savings could be made by lowering the number of selection parameters from around 14 to 3 or 4. This would undermine the fundamental work done by the University. Also, when selecting from large databases, the lack in depth of search would result in an excessive number of hits. This is shown to be true using the small trial databases. Selecting using just four parameters resulted in tens of hits. Adding two more constraints brought hits to around ten or under. From here users would be happy to "view" records. It is vital that the search parameters identified by the University are not reduced and more effort should be placed in producing the databases cheaply.

In addition to the field trials, the ROCCI system was demonstrated to many visitors, both academic and industrial, who came the University. The response was always favourable.

The ROCCI software developed by the University, although quite satisfactory for a demonstration system, falls far short of the thoroughness usually associated with commercial packages. Since the University felt that the system so far could be considered developed enough to help justify and exploit the research. The improvements to the software have not been made because although they are trivial, software development and debugging is time consuming. A brief documentation of the ROCCI system is given reference 31.

9.5 List of Improvements to the ROCCI System

This section has been moved to Appendix O.

CHAPTER 10

DEVELOPMENT OF THE RESEARCH METHODOLOGY.

10.1 Introduction.

This chapter attempts to summarise the techniques and principles used by the research in order to achieve the research aims. These were a complete description of a given component generated by using manufacturers literature and analogies with other components. The distillation of these extensive lists led to short lists of discrete, clearly defined and 'user friendly' characteristics which could then be used as the basis of a computer aided retrieval system. Chapters 4,6,7 and 8 produced short lists for specific components. This chapter will help the reader to perform similar work with other rotary mechanical components, or, to use components from another engineering field.

10.2 Creation of the Extensive Characteristic Lists

The extensive character lists are the result of a thorough background study of the component. Characteristics from a theoretical, practical and application viewpoint must be identified, understood and defined. The extensive character lists should specify the component completely. During the normal design process, the majority of these characteristics would be considered. The extensive list can then be subjected to analysis by analogy and additional characteristics identified and defined. A technique of no theoretical significance but extremely useful for the process of identifying analogies was the grouping of the extensive list characteristics into sections with broad descriptive titles. Input, output and dimensional were the three most common. Other additional headings specific to the subject component may also be needed. The extensive list could then be handled as a series of smaller lists which would make changes and additions easier. This is particularly important when analogies are considered.

10.3 **Generation of Analogies**

Analogous components to that being studied can be generated using the analogy matrix, the section relevant to power transmission systems is shown in Table 2.1. At least two analogous components should be used. Extensive lists for these components must also be created. These analogous lists can also be broken down into groups of characteristics under the broad headings used for the subject component. Then, using the principle of analogy, characteristics can be cross referenced between the components. This will generally cause the extensive list for the subject component to be increased as additional, previously unconsidered characteristics are added. This reinforces the completeness of the extensive list and contributes to the thoroughness of the specification. By the nature of the research, not only is the subject component thoroughly described but the analogous components are also prepared for similar investigation because analogies are explored in both directions. Hence, the work tends to propagate itself by preparing the basic research in other fields.

Whenever possible, direct analogies should be made i.e. output torque is analogous to output current using the gearbox / transformer analogy. However, often, the analogy is less clear and highlights inferred properties within the characteristic. A good example is the speed against power table or graph provided by many belt manufacturers. This table defines the safe operating region for the product. It is basically the torque limit for the belt. Thus, the speed and power table is analogous to maximum output current using the 'gearbox' / transformer analogy. To avoid confusion, where this kind of derived analogy occurs, the "parent" characteristic is quoted, it is used to provide the clue to the underlying analogy. Also, with a view to a computerized retrieval, being able to break down tabulated or graphical information into discrete values in this way is extremely helpful. Cases where analogies cannot be identified or are not appropriate can also be included by leaving spaces in the lists where the analogous characteristic should appear.

When happy that the analogies have been investigated to the full the analogous lists can be written out side by side enabling the complete list to be seen for each analogous component. These lists are now a very detailed analogy matrix for specific components. An attempt must now be made to reduce the extensive list to shorter lists for use with a computer aided retrieval system.

10.4 The Retrieval List

The aim of the retrieval list is to allow a designer to efficiently define a component so that a rapid selection of five to ten suitable components can be made from tens of thousands. It could be argued that the user should be presented with the extended list and a search performed using the data the user has available. This suggestion is inappropriate because firstly, data capture would be an enormous task and secondly, the system would be clumsily overdetailed for the user. The expertise is to reduce the extensive lists in such a way as to sharply define the component using easily recognisable characteristics. In the case of rotary mechanical components, it was decided that the performance characteristics were of primary importance and other characteristics, such as dimensional information could be looked at generally rather than in detail. The reasoning was that components that would not perform could not be selected even if they met other criteria such as noise levels, cooling requirements etc. As long as the component performed, problems with general integration such as shaft sizes etc. could be accommodated. This hypothesis allowed the extensive lists for rotary mechanical components to be rapidly reduced.

The lists could also be further reduced by eliminating characteristics that could be inferred or assumed to be met by satisfying other characteristics. Efficiency of motors can be exempted from the selection lists because the motor should be selected to operate at rated values. Then, by definition, efficiency will be at its maximum. By this process, it was found that the components studied reduced to around eighteen characteristics. Roughly half were performance based, the others relating more to physical details such as maximum dimensions, mounting types etc. The appropriateness of these characteristics for selection was validated by the response of the retrieval software and also by the results of the field trials (Chapter 9).

10.5 Compatibility and Consistency

The selection lists created by the research are usable for independent component selection but also help to build systems by their compatibility. The constraint of using mainly performance characteristics contributed to this compatibility by ensuring that many criteria were consistent between lists i.e. power. However, a less obvious consistency appeared that reinforced the fundamentals and thoroughness of the research. The best and clearest example was that of inertia. The study of analogies between gearboxes and transformers suggested that gearbox inertia should be included within the selection list. The main reason being that gearbox inertia would affect system response significantly and should be part of the system inertia referred to the prime mover. Since inertia was not often quoted for above fractional horsepower gearboxes, it was not readily available for data capture. It was hence (under the advice of Technical Indexes Ltd.) given a low priority on the selection list and a low weighting for data capture and was thus ignored. However, as the lists for other components were developed, inertia was found to be significant and available for selection of these components. Thus, the academic reasoning for the need of inertia was born out by the practical consistency of this characteristic.

The compatibility of components will contribute to the rapid assembly of matched components. Matching performance characteristics will ensure an efficient system. Thus the whole process of system optimisation can be speeded up giving immediate economic benefits. If current manufacturer's literature is studied, it becomes clear that components that are usually adjacent in rotary mechanical power transmission systems are not presented in a compatible format, even by manufacturers who supply system building elements.

Finally, the research methodology has proven successful over the limited number of components studied. Only if the techniques are tried with more rotary mechanical components and components from other engineering fields will the overall philosophy be justified.

CHAPTER 11

CONCLUSIONS

11.1 The Study of Analogies

From the study of motor analogies it was found that a.c. electric, d.c. electric and fluid, positive displacement motors can be reduced to a system that requires an "alternating" power supply to produce rotary power. In the case of a.c. and d.c. electric motors it is the analogies between the actions of an a.c. phase and the d.c. commutator that create the common analogous mechanism. In the case of fluid piston motors, the action of the pintle valve is analogous to the d.c. commutator and hence these motors can also be related back to their a.c. analogues. Taking these ideas further it was possible to suggest fluid and d.c. analogues to the common a.c. electric motor types. Whist some proposals remain firmly theoretical e.g. the fluid synchronous motor, other suggestions such as the rotating field d.c. motor have practical results e.g. the brushless d.c. motor.

The analogies between gearboxes and transformers yielded two major characteristics that are often neglected when selecting gearboxes. Gearbox inertia is a characteristic that should be considered whenever a gearbox is introduced into a system. Generally it can be neglected as it is a fraction of the load inertia, however, if the load is light, or the system is to be speed controlled, the gearbox inertia must be considered. The stiffness of the gearbox also becomes significant in high torque, low speed applications where positional control is important. The tendency of the gearteeth and shafts to bend or twist under load and to then release this energy when the torque is removed could cause a control system to appear under damped. The significance of stiffness was further emphasized as analogies were made during the study of couplings. The lack of stiffness in certain coupling designs is used as a system benefit as it helps to damp rapid transients. Thus, as a gearbox can be considered a "coupling" between source and load, the designer should have the stiffness of a gearbox available as part of the selection process.

The significance of analogies as a tool for analysis was highlighted when considering the analogies between components as diverse as clutches fluid couplings and rotary dampers. Using analogies the characteristic of power dissipation was found to be common between all of the devices. This then allowed a common set of characteristics to be developed that brought all of the components under one single characteristic set for retrieval.

11.2 **The Retrieval System**

The field trials of the ROCCI demonstration software clearly showed that designers would welcome such a system. It also proved that the characteristics selected to use for retrieval are those that designers need to make an accurate selection. The accuracy of the retrieval is significant in two ways, firstly, the designer will select the correct component because the retrieval has been made using the correct characteristics. Secondly, the number of characteristics used is large enough to cause the retrieval to be specific, limiting the number of retrieved components to those most suitable for the application. These factors will become very significant if large databases are created.

As a demonstration system, the ROCCI package was very successful. However, if it were to be commercially exploited, a more modern, professionally written user interface would be necessary.

11.3 **The Retrieval System and Current Day Application.**

The philosophy, techniques and concepts generated by the research have resulted in three academic publications and dissemination. REFS (19,29,32).

The research has brought together many of the trends being followed by individual manufacturers who have computerized their catalogues. However, manufacturers have not chosen to redefine components by characteristics but to simply reproduce their catalogue electronically and to then provide a search engine that uses the original selection method. The best example of this is the Radio Spares CD-ROM. It is simply a "soft" version of the catalogue with a simple search engine based on the standard catalogue indexing system. It does not give users the additional dimension of narrowing searches by specifying a characteristic. Consequently, the electronic catalogue has the same faults as the manual system - the final selection is made manually with many of the potential components being unsuitable.

The research methodology and the application through the ROCCI software promise a lot more than an automated catalogue. The methods by which this could be achieved have been identified, tried and proven. The stumbling block has been the commercial constraints of time and money for creating the large databases necessary to make the system attractive to users and commercially viable for sponsors.

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