A finite element analysis of a ‘S’ cam brake

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A FINITE ELEMENT ANALYSIS
OF A "S" CAM BRAKE

by C WATSON

A DOCTORAL THESIS

submitted in partial fulfilment of the requirements for the award of Ph.D. of the
Loughborough University of Technology,
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ABSTRACT

An analysis of a commercial vehicle drum brake fitted with a typical asbestos free friction material has been investigated by the finite element method. The "GAPFRIC" concept has been extended to model in three dimensions the frictional interface between brake linings and drum. This approach incorporates an accurate representation of the brake itself to include such features as the brake drum stiffener and mounting flange. The brake shoe representations include the actual web and platform thus eliminating the need for shoe stiffness approximations to a curved beam of uniform section as used in previous two dimensional work. The mechanical "GAPFRIC" analysis is combined with a thermal analysis of the brake to form a brake analysis package. The package is fully automatic; the output from each stage of the analysis is post processed and the results used to modify the original data file. Variations in physical properties exist between new and used friction material and these are incorporated in a five phase idealisation of the friction pair. Modifications to allow for the change of coefficient of friction with temperature were made by means of tables within the coding. In addition friction material wear was included in the analyses using an empirically derived wear criteria.

Analyses were completed to investigate the effect of combined axial and circumferential distortions on temperatures at the friction interface, interfacial pressure distributions and subsequent brake performance. Predicted results show that high temperatures are reached at certain regions on the rubbing path and the temperatures may fluctuate during a brake application. Pressure variations are seen to exist both around and across the surface of the linings. The coupling of pressure and temperature variations combined with frictional changes over the lining produce changes that result in the frictional drag per unit area tending to be reasonably constant over the interface between drum and lining.

The predicted values of brake torque and brake factor from the three dimensional analyses have been compared with results derived from the earlier two dimensional brake analysis and validated by comparison with measured...
results from a brake mounted on a dynamometer. Similarly predicted brake drum and lining temperatures were compared with measured values and some reasonable trends established. The work itself presents a better physical description of the behaviour at the friction surface during braking to improve the determination of brake drum performance.
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NOMENCLATURE

$a, A$
constant

$\Lambda$
area (m$^2$)

$A_p$
pad area (m$^2$)

$A_s$
rotor swept area (m$^2$)

$[A]$
constant matrix relating nodal displacement and applied forces; $(g) = [A][F]$

$A$
geometrical factor of the infrared thermometer optics

$b, B$
constant

$B$
width of lining (m)

$[B]$
inverse of $[A]; [A]^T = [B][g]$

$b$
population regression coefficient (least squares solution)

$[b]$
column vector of sample regression coefficients

$c, C$
constant

$C_p$
specific heat (J/kg K)

$C_p$
specific heat of pad material (J/kg K)

$C_r$
specific heat of rotor material (J/kg K)

$[C]$
flexibility matrix; $[\Delta\delta] = [C][\Delta F]$

$D$
constant

$d$
width of wear grooves (m)

$D$
wear thickness (m)

$D_L$
drag force on leading shoe (N)

$D_r$
drag force on trailing shoe (N)

$d'$
depth of equivalent section beam (m);

$$d' = \left(\frac{12I}{B}\right)^{1/3}$$

$E$
Young's Modulus (N/m$^2$)

$E_s$
Young's Modulus of the brake shoe (N/m$^2$)

$E_l$
Young's Modulus of the lining material (N/m$^2$)

$e_i$
spectral emissivity of the surface

$e_i$
ith residual (least squares solution)

$f_n$
fraction of abrasive particles that cut or contribute to wear

$F$
tangential force (N)

$F$ normal force (N)

$[F]$
vector of applied loads

$F$
Fourier number; $= \frac{k\delta t}{C_pQe^2}$

$F_c$
cam force (N)
FL input force to the leading shoe (N)
Ft input force to the trailing shoe (N)
f( ) function of ( )
FM constant referring to the friction material composition

G flexibility matrix; $\delta_p = Gp$
{g} vector of increments in the relative displacements of the interface nodes.
[g] column vector of the right hand side of the normal equations

Hb hardness

I rotational inertia (Kgm²)

Jsr Plank radiation function

K constant
K hardness function
K proportion of areas of contact which give rise to a wear particle
K number of tests at each temperature; $\Sigma \left( \frac{W_i}{d} \right)$
k thermal conductivity (W/mK)
k_r thermal conductivity of the rotor material (W/mK)
k_p thermal conductivity of the pad material (W/mK)
l projected pad life (km)
L load (kg)
LSF leading shoe factor

ME constant referring to the rotor metallurgy

n number of particles that contact the surface
N number of brake applications

p pressure (N/m²)
[P] coefficient matrix (least squares solution)

Q_a Arrhenius actuation energy
Q heat flux (W/m²)

R universal gas constant, $R = 8.3143$ kJ/kmol K
R elemental reaction force (N)
r radius (m)
$r_1$ spectral reflectivity
$S_a$ spectral sensitivity of the infrared detector

t time (s)
T torque (Nm)
t thickness (m)
T temperature (°C)
{$\Delta T$} vector of increments in forces applied to the interface nodes
$\Delta t$ time step (s)
$\delta t$ time step for transient temperature calculations (s)
TSF trailing shoe factor

$u$ random variable (least squares solution)
$u_i$ $i$th random variable (least squares solution)

$v$ sliding velocity (m/s)
V infrared detector output

$w$ wear rate (m/s)
$w_i$ measured wear (m)
$w_{isp}$ ratio of measured wear and wear thickness
$\Delta w$ wear increment (m)
$\delta w$ friction material wear (m)
W work done (J)

X independent variable (least squares solution)

Y dependent variable (least squares solution)
y deflection

$\alpha$ constant
$\alpha$ negative rake angle of file teeth
$\alpha$ coefficient of thermal expansion (K$^{-1}$)

$\beta$ constant
$\beta$ population regression coefficient (least squares solution)

$\gamma$ constant

$\delta$ distance (m)
$\delta_s$ distance between rubbing surfaces (m)
$\delta_d$ distance change due to displacement (m)
$\delta_w$ distance change due to wear (m)
$\delta_t$ distance change due to thermal effects (m)
{$\delta$} vector of displacement
\( \varepsilon_s \) strain
\( \varepsilon_{p1} \) nodal strain
\( \varepsilon_{wear} \) wear strain
\( \varepsilon \) distance between closest nodes in the direction of heat flow where thermal shock is severe (m)

\( \theta \) temperature (K)
\( \psi_1, \psi_2 \) lining arc and position; \( \int_{\psi_1}^{\psi_2} p(\theta) \, d\theta \)
\( \theta \) location parameter

\( \mu \) dynamic friction coefficient
\( \mu_p \) ploughing contribution to friction
\( \mu_\lambda \) spectral refractive index

\( \nu \) Poisson's ratio

\( \pi \) constant = 3.14159

\( \rho \) density (kg/m³)
\( \rho_p \) friction material density (kg/m³)
\( \rho_r \) rotor material density (kg/m³)

\( \sigma \) stress (N/m²)

\( \tau_\lambda \) spectral transmission of the infrared thermometer optics

\( \upsilon \) heat partition ratio

\( \Phi_{\text{sh}} \) shape factor

\( \Phi \) total area of contact per unit load (m²)

\( \omega \) angular velocity (rad/s)
1. INTRODUCTION

A pre-requisite for the safe and reliable operation of commercial vehicles is a consistent and predictable brake performance. The performance of a vehicle's brakes and brake systems is determined to a large part by the friction material. This seemingly simple product is in fact a complex composite material which is required to perform consistently under all conditions that may be experienced by the vehicle during braking.

Since the 1960's friction material requirements for commercial vehicles have increased in severity as the progressive increase in engine power and vehicle mass has greatly increased the possible energy to be dissipated by the brake system. An increase in vehicle speed from 65 km/h to 80 km/h results in the equivalent increase in kinetic energy to be dissipated during a stop to rest, as does increasing the gross vehicle mass of a 32 tonne vehicle to 48.5 tonnes when braked from the same speed. If heavy-duty stops are made from high speeds with large vehicle masses then the formulation of the friction material plays a very significant role in withstanding the temperatures reached and in providing effective braking.

The European transport bodies and friction material industries have been quick to react to changing circumstances resulting from technical developments in vehicles to ensure safe braking. Legislation, primarily in the form of EEC Braking Directives 71/320, 74/132, 75/524 and 85/647, provided the impetus required to encourage new approaches to the problem of slowing heavy vehicles in a repeatable, controlled and progressive manner under a wide variety of loads, speeds and climatic conditions.

The change to asbestos-free brake lining materials coincided with this changing brake philosophy which required the newly developed asbestos-free commercial vehicle brake linings to offer significant improvements over the previous asbestos based products. The following criteria had to be met. Asbestos-free lining materials must exhibit greatly reduced wear rates to give compatibility with the higher energy dissipation for each kilometre required by increased vehicle power and traffic densities; be capable of providing high
peak power dissipation demanded by the increased maximum vehicle speeds and weights; provide stable repeatable friction across the range of operating temperatures and speeds; and show improved kindness to the mating surface of the friction pair to satisfy the historic requirement of relining brakes while retaining the original disc or drum.

In order to meet these new requirements a greater knowledge of the braking process is required. When a vehicle’s brakes are applied energy is dissipated by friction heating generated by physical interaction of asperities at the sliding interface causing an increase in surface temperatures of the rubbing pair. This relatively simple sounding process is in fact highly complex, the brake output being dependent upon the interface contact and pressure distributions, which in turn are significantly affected by the properties of the friction material and the geometric stability of the mating surface. Under normal operating conditions temperatures higher than the surroundings can occur locally and their value will fluctuate depending on the size of the contact area. Some contacts can be replaced by an envelope of small contact areas and in drum brakes lead to the formation of bands across the rubbing path. Under severe operating conditions this contact situation gives rise to a macroscopic kind of temperature variation across each heat band leading to a slow moving value that is of much higher temperature relative to the steady state value. Such heat bands and other thermo-elastic effects resulting from high temperatures within the brake can give rise to anomalous brake performance which can be investigated by the finite element method.

The first step forward in the specific use of the finite element technique for the study of frictional effects under high energy sliding contact conditions was made by Kennedy and Ling (Ref 1) who simulated thermo-elastic instability and transient contact changes at the interface of an aircraft type annular disc brake using sintered metal friction material. Frictional forces have also been modelled to determine the performance of drum brakes (Refs 2,3,4). It has been shown how thermal, mechanical and wear effects change the temperature and pressure distributions in the friction pairs during braking. Elements at the friction surface have been shown to move in and out of contact thereby simulating thermal instability.
A limitation of these earlier studies is that they were restricted to two dimensions and therefore could only predict circumferential effects along the lining. A more accurate representation of the lining/rotor interaction can be achieved if the combined effects of mechanical and thermal distortions in the axial as well as circumferential direction are considered. This can be achieved using a three dimensional approach to give a more realistic representation of pressure and temperature distributions on the lining surface.

Such a three dimensional finite element analysis has been applied here to investigate the major parameters that occur between sliding members during braking. The characteristics of the S-cam drum brake pressure distribution are shown and it is also illustrated how the pressure variations affect the performance of the brake. Other factors such as changes in the lining compressibility and brake shoe stiffness are also considered in relation to the behaviour of a commercial vehicle brake. Although elements cannot be small enough to cater for contact areas on a microscopic scale meshes can be chosen to deal with several heat bands on the friction surface. Temperatures at the friction interface are predicted to be well in excess of those which certain constituents of organically bonded friction materials may exist and consequent degradation of the material occurs, thus changes in the physical properties of the degraded material are of major importance. Phase changes can occur and these may be taken into account by changing the properties at certain elements.

Using the three dimensional finite element brake analysis method developed, different liner to drum contact situations and the effect of varying other parameters are discussed, and it is shown that this approach gives a much better understanding of the physical changes that occur during a brake application made by a commercial vehicle drum brake. The work described enables the requirement for greater accuracy in brake performance predictions, necessary to complement other advances being made in brake technology, to be achieved.
2. LITERATURE SURVEY

2.1 FRICTION AND WEAR

Friction occurs when there is relative motion between two or more contacting bodies. Classical theories of friction are based upon some mechanism of surface interaction between a rubbing pair which is made up of similar materials, most frequently metals, under carefully controlled conditions of sliding. During sliding contact between nominally flat surfaces occurs at microscopic peaks or asperities on the surfaces. The resulting contact stresses at these real areas of contact may be sufficiently high to cause elastic, elastoplastic or plastic deformations. Under conditions of ideal plasticity the relationship between the tangential friction force developed at the surface and the normal force is one of direct proportionality consistent with Amontons' Laws of friction. Similarly the frictional force is proportional to the normal force if flat surfaces are covered with hemispherical asperities which themselves carry smaller asperities and these undergo elastic deformation. Attempts have also been made to allow for plastic and elastic effects at surface asperities in determining the coefficient of friction. However due to deformation of the polymeric binding resin Amontons' Laws do not always hold for friction materials. The mechanism of dry friction between conventional friction pairs is considered to consist primarily of adhesion and abrasion. Abrasion, which is related to ploughing and grooving, is predominantly responsible for any departure from Amontons' Laws because of the significant contribution of the deformation component to the friction force generated. Classical theories of friction treat the mechanisms of abrasion and adhesion separately, whereas in most systems both operate and hence an over simplification occurs. However, an understanding of these mechanisms is required as they significantly influence the friction and wear characteristics of friction materials.
2.1.1 Abrasion occurs when material is machined away from the softer surface of a friction pair by irregularities, or asperities on the harder surface, or by foreign particles, or by wear particles themselves. The wear between two nominally flat surfaces occurs at microscopic peaks on the surface and over the area associated with each asperity. The characterisation of asperities in terms of the surface topography was first discussed by Bowden and Tabor (Ref 5) in 1938, and was further investigated by Krushchov in 1956 (Ref 6, 7, 8) who investigated the wear of a range of pure metals when slid against abrasive cloth. It was found that the wear rate, \( w \), was inversely proportional to the flow pressure of the metal and this led to the proposal of the wear equation

\[
W = \frac{KL}{H_p}
\]  

(2.1)

where \( K \) is a constant for pure metal, \( L \) is the applied load and \( H_p \) is the hardness. However, for heat treated steels the \( W \) vs \( H_p \) lines do not pass through the origin and thus for these materials \( K \) is a function of hardness. Spurr and Newcomb (Ref 9) performed similar experiments to Krushchov but included polymer specimens in their investigations. Reasonable agreement was found with Krushchov's wear equation (2.1) but better agreement was seen with a relationship using Young's modulus, \( E \), rather than hardness, which could be expressed in the form

\[
W = \frac{KL (\tan \alpha)}{E}
\]  

(2.2)

when metals were slid against files; where \( K \) is constant and \( \alpha \) is the negative rake angle of the file teeth.

Spurr and Newcomb found that the wear rate was proportional to the ploughing contribution to friction \( \mu_p \), and the diameter of the abrasive particles (over the size range 20 to 120 \( \mu m \)). The effect of ploughing was investigated by Stroud and Wilman (Ref 10) who demonstrated that small quantities of metal are removed compared with the quantity which undergoes plastic deformation when a silver specimen is
slid over emery paper. Mulhearn and Samuels (Ref 11) concluded that not all the particles on a piece of emery paper were suitably shaped or orientated to cause wear, the others merely ploughing through the surface and displacing material to each side. Thus they proposed the equation

\[ w \propto f_a \Phi_a n d = K f_a \Phi_a L/p \]  

(2.3)

to characterise wear behaviour of the system where \( f_a \) is the fraction of abrasive particles that cut or contribute to wear, \( n \) the total number of particles that contact the surface, \( d \) the width of the grooves, \( \Phi_a \) is a geometrical factor relating to the shape of the particles, \( p \) is the flow pressure and \( K \) is a constant.

Spurr (Ref 12) considered a relationship between the ploughing contribution to friction, \( \mu_p \), and the wear rate.

\[ w = K \mu_p L/p \]  

(2.4)

and showed that this equation applied equally well to polymers as it did to metals. Wear was seen to occur around the ploughing particle, displaced material building up around the particle until the tensile strength of the material was exceeded and a wear particle was formed.

2.1.2 Adhesive wear arises when the adhesive force between two mating surfaces is greater than the strength of one of the materials. It is highly dependent on the materials in the friction pair with similar metal pairs wearing more rapidly than dissimilar pairs. Archard (Ref 13) showed that the adhesive wear rate could be expressed by the equation.

\[ w = KL \Phi/3 \]  

(2.5)
where $\Phi$ is the total area of contact per unit load and $K$ is the proportion of the areas of contact which give rise to a wear particle, which is usually quite small.

The quantity $\Phi$ is independent of the apparent contact area of the surface and is therefore quite difficult to determine. Archard’s approach was to assume that $\Phi = p^{-1}$ and indeed some correlation exists between $w$ and $p^{-1}$ although $K$ varies from material to material. The approach used by Hughes and Spurr (Ref 14) was to run a material which softens on heating against a steel disc and then to measure the wear rate at different temperatures. They used montan wax and found that its wear rate was described by the equation

$$w = KL/p$$

when the variation in $p$ with temperature was taken into account. The process by which adhesive wear actually takes place is not very well understood, there being a certain amount of disagreement on the subject. Rabinowicz (Ref 15, 16) considered that a particle will store strain energy due to the repeated loading and unloading as it passes through the contact zone, until the energy stored is such that it provides the necessary interfacial energy to detach the particle. He then went on to attempt to relate the value of $K$ in equation (2.5) to the ratio of the surface energy of adhesion to the sum of the surface energies of the contacting materials.

2.1.3 The wear of polymers can be categorised into the preceding sections. However, the relationship between wear and hardness of polymers is not as simple as that for metals. For example Krushchov stated that wear is proportional to $H^{1/3}$. However, Selwood (Ref 17), in experiments with non-rigid polymers found that harder materials had an increased wear rate over softer ones.

Two types of abrasive wear were found by Lancaster (Ref 18) to occur in polymers. These are, plastic deformation and cutting by relatively
sharp asperities in the abrasive surface, and elastic deformation and fatigue by more rounded asperities. Both mechanisms occur simultaneously in most practical situations, but if any plastic deformation occurs that mechanism will be likely to be the dominant one. Thus it is very important to characterise the topography of the abrasive surface in order to predict the mode of deformation and its consequences. Lancaster found that increasing the roughness of the abrasive surface by a small amount could produce a large increase in wear.

Polymer friction and wear have been found to be highly temperature dependent, mainly because of the marked reduction in elastic modulus with increased temperature. It was found by Lancaster (Ref 19) that the relatively low thermal conductivity of most polymers makes interface thermal effects important, and therefore carefully controlled low speed test conditions are necessary to minimise any associated temperature rise so the effects of test parameters can be isolated from those of thermal softening.

Fibre reinforcement enables the strength as well as the friction and wear behaviour to be improved and Lancaster (Ref 20) noted that levels of friction and wear could be achieved independent of the polymeric binder under dry sliding conditions. This forms the basis of resin bonded composite friction material technology, where heat resistant fibres are used to reinforce a polymeric binder resin so that much greater amounts of frictional heating can be tolerated.

2.1.4 Interactions between the various constituents of composite friction materials can be extremely complex in terms of their effect upon friction and wear. Fillers can have a large effect on the friction and wear processes (Ref 20), altering the friction and wear properties by modifying the topography of the mating surface. Mildly abrasive components such as silica or asbestos can reduce wear by producing smoother surfaces, while more abrasive fillers which make the mating surface rougher as rubbing proceeds can be responsible for an increase in the wear rate.
In 1970 Rhee (Ref 21) proposed an experimental equation for the wear of polymer bonded friction materials sliding against a metal surface in a low temperature regime (below 232 °C) of the form

$$w = KL^n V^b t^c$$

(2.7)

where $a$, $b$, and $c$ are constants. This was a modification of the formula $w = KLVT$ proposed by Lewis (Ref 22) to explain the wear of polymer-matrix friction materials. Rhee (Ref 23) also investigated the wear characteristics of metal reinforced phenolic resins in both transient and constant temperature experiments and concluded that the wear of metal reinforced phenolic resins increases with temperature, and that the wear can be satisfactorily described by the equation (2.7) for both transient temperature and isothermal tests. Further work (Ref 24) showed that metal reinforced resin material behaved in a similar manner to asbestos filled resins in that wear was proportional to the square of the surface roughness. However, the increase in wear of the metal reinforced resin with surface roughness was not as great as that of the asbestos one. Rhee explained this using the theoretical model of the abrasion of non work hardening metals proposed by Mulhearn and Samuels (Ref 11).

Rhee and Liu later examined the high temperature wear mechanisms of both organic and semi-metallic friction materials (Refs 25, 26, 27). They found contrary to the result from earlier papers, that the value of $c$ in equation (2.7) was approximately unity for organic friction materials, thus making the wear of organic friction materials directly proportional to the time of this test. A different result was observed for semi-metallic materials. This was attributed to the fact that the metal constituents are more reactive than asbestos (which is relatively inert at the temperatures experienced in the experiment), hence the composition of the surface of the semi-metallic friction changes with time, thus changing its wear rate.
Modifying equation (2.7) for high temperature (above 232°C) wear Rhee and Liu proposed the exponential equation

\[ w = KL' V^n t \exp \left( -\frac{Q_a}{R_0} \right) \]  

(2.8)

where \( Q_a \) is the activation energy and \( R \) is the universal gas constant. As the thermal degradation of the phenolic resin used in friction materials follows an Arrhenius rate law, it can be concluded that the mechanism that controls the wear at high temperatures of resin bonded composite frictions materials is thermal decomposition or pyrolysis of the binding material which is usually organic in nature.

Rakowski (Ref 28) investigated the friction surface layer of the friction material and established that wear was mostly by thermal and abrasive action. The surface layers of the material become carbonized and are removed by the roughness of the mating surface. At high temperatures near the surface of the friction material macroparticles of the resin start to decompose and this degraded resin has poor resistance to abrasive wear. The degradation of the resin is aggravated even more by the high temperatures reached locally by any metal particles present. Cracking due to thermal stress also takes place at the boundaries between metal particles and resin binder. Conversely Rakowski showed that increasing the thermal conductivity of the material due to the addition of metal particles can also have a beneficial effect. It can significantly reduce the thickness of the degraded layer by lowering the temperature at the surface, and can reduce thermal stresses by decreasing the temperature gradient. On investigation of the topography of the friction surface of semi-metallic materials it was found that the metal grains protruded above the surface of the surrounding plastic, possibly due to more rapid wear of the degraded matrix, and elastic deformation of the surface layers occurs due to the normal load on the friction surface.

Bark et al (Ref 29) showed that the degradation produced at high temperatures generated during sliding was not as severe as would be
expected from simple pyrolysis of the organic binder under similar conditions, supporting the existence of an ablation type wear mechanism providing sacrificial protection to the subsurface material.

Pavelusu and Musat (Ref 30) tried to determine some relationship between wear of composite brake materials and load and speed. They found that the wear of semi-metallic materials could be modelled by an equation similar to equation (2.7) but in direct contrast to the findings of Rhee found it could not be used to model other types. They then proposed an equation relating projected life, $l$, to the number of brake applications, $N$, and the work done, $W$.

$$l = K_1 N^a W^b + K_2 N^{a'd} W^d = \psi (K_1 N^a + K_2 N^{a'd})$$

The six variables $a$, $b$, $c$, $d$, $K_1$ and $K_2$ were listed for each of these materials which did not contain metal, although no explanation was given as to how these figures were obtained. No theory was put forward to explain the form of the equation, but the fact that the values of wear calculated from the above equation were close to those measured was deemed to validate the equation.

The difference in results obtained by Pavelusu and Musat and those obtained by Rhee and Liu may be explained by the temperature at which the experiments were performed. Some of the tests performed by Pavelusu and Musat were carried out in the high temperature wear region which may account for the need for a more complicated formula than that used by Rhee.

Pavelusu and Musat conclude that the wear of materials which include metal fillers is less than materials which do not because of the superior heat conduction of the metal. This does not significantly reduce the temperature at the actual friction surface, but the temperature gradient at the surface of the friction material is reduced, so less thermal stress is induced.
2.1.5 A Linear Wear Hypothesis (LWH) was proposed by Arsenic, Duboka and Todorovic (Ref 31) to predict the wear of friction materials. They assumed that there is a linear relation between wear of a resin bonded composite friction material and the work done at a given brake temperature; and that there is a linear relation between the work done by a brake in a single application at a given temperature, \( n \), and the total number of brake applications, \( N \), realised at that temperature. That is, the wear of the friction material is directly proportional to the time of the test as found by Rhee and Liu (Refs 25. 26. 27). This can be written as:

\[
\frac{n_1 + n_2 + \ldots + n_x}{N_1 + N_2 + \ldots + N_x} = \frac{K}{1} \left( \frac{n_i}{N_i} \right) = \frac{K}{1} \left( \frac{w_i}{d} \right) = \frac{K}{1} \frac{w_{isp}}{1}
\] (2.10)

where \( w_i \) is the thickness lost as wear of the total wear thickness \( d \), \( K \) is the number of different wear tests made at different temperatures and \( w_{isp} \) is the ratio of measured wear and active wear thickness.

Todorovic et al (Ref 32) modelled the friction and wear characteristics of a resin bonded composite friction material for a light car brake by means of an experimental function of the third degree in the form.

\[
w = C_1 p^{a_1} v^{b_1} \theta^{c_1}
\] (2.11)

\[
\mu = C_2 p^{a_2} v^{b_2} \theta^{c_2}
\] (2.12)

The coefficients in the above functions were obtained experimentally for a specific brake/friction material combination enabling the tribological properties of the friction material to be described as an exponential function.

2.1.6 Rhee (Ref 33) investigated the influence of rotor metallurgy on brake friction materials and concluded that correct matching of the lining to brake drum composition is very important factor in controlling lining wear. He also established that the higher the thermal conductivity of
the drum or disc the lower the wear of the brake lining material by virtue of the lower temperature of the friction face.

Chapman and Hatch (Ref 34) showed that certain trace elements, normally present in grey cast iron, greatly affected both the coefficient of friction between the rubbing pair and their wear rates. Only very small amounts of titanium, vanadium, and less frequently niobium, which form very hard discrete carbide or nitride particles distributed through the matrix of the castings, needed to be present to greatly reduce friction and wear. The presence of 0.5% of any of these elements in the brake rotor was found to reduce the wear rate of the rotor by at least 50% and the wear rate of the friction material by between 30% and 50%. The coefficient of friction developed between the rubbing pair was also generally reduced by between 0.05 and 0.15, being dependent upon the operating temperatures and pressures, generally being greater at lower temperatures and lower pressures.

The surface finish of the brake rotor has also been seen to significantly influence the performance of the friction pair (Ref 35). The coefficients of friction initially developed against turned surfaces are significantly lower than those developed against ground surfaces with the same friction material and that the coarser the turning the lower the friction. Furthermore the friction developed against many turned surfaces is sensitive to the direction in which they were rotated during turning: rotors turned with ceramic-tipped or round-tipped tools give higher friction when rotating in the same direction during service as during turning and rotors turned with sharp carbide-tipped tools give higher friction when rotated in the opposite direction. Obviously the effects of surface finish of the rotor will be less apparent with low titanium rotors since they wear rapidly and would be expected to quickly lose any initial surface condition.

2.1.7 Liu, Rhee and Lawson (Ref 36) investigated how the chemical and microscopic interactions at the friction surface are related to the wear rates of friction materials. They considered the transfer film formed
on the sliding surface of the drum by the transfer of friction material and wear debris (both from the friction material and the drum) to the rubbing surface of the drum. They concluded that a continuous and adherent transfer film is desirable for obtaining low wear rates and that the thickness of the film increases with increased braking severity. A rougher surface as opposed to a smooth surface, was seen by Rhee and Ludema (Ref 37) to promote the formation of transfer films.

Jacko, Tsang and Rhee (Ref 38) proposed that friction film formation on the rotor and friction material is a result of compaction of wear debris generated from the rotor and liner surfaces. During braking, this compacted layer of wear debris is formed and sheared into two parts at the same time, one part remaining on the surface of the rotor and the other on the pad surface. The mechanical integrity of the compacted wear debris becomes weaker as the wear debris generation increases due to increased thermal degradation. As the thermal degradation proceeds, the compacted layer will become very weak and result in a situation similar to a three-body rolling contact in which friction will be reduced.

It has been shown (Refs 28 and 36) that depending on the prevailing starting conditions (initial temperature and speed), boundary layers are formed which are constant under the given operating conditions but which may vary more or less with these conditions. But it is known (Refs 39, 40, 41, 42, 43) that contact between the friction material and rotor occurs over only a small percent of the geometric contact area resulting in very high peak temperatures at small areas on the friction surface under normal and relatively mild operating conditions. The tribological and wear properties are highly dependent on the properties of the intermediate or boundary layer between the friction pair, the properties of which are principally determined by its chemical composition which is in turn dependent on the usage history of the brake. This subject is considered further in subsequent chapters.
2.2 DRUM BRAKE ANALYSIS

The prediction of brake performance, and the expected variation of that performance in service, are vitally important to a vehicle manufacturer if the vehicle is to have safe brake retardation under all normal operating conditions. Early attempts at drum brake analysis used geometric methods whereas modern brake analysis makes extensive use of the finite element method.

2.2.1 Some of the earliest published work was by Watt (Ref 44) who investigated the angular position of the centre of pressure or drag. He idealised the circular lining to set of chords and resolved the reaction on each one into perpendicular components. A vector polygon was then constructed to find the direction of the resultant of the elemental reaction forces. In his construction Watt implicitly uses a cosine pressure distribution circumferentially over the lining surface.

Barford (Ref 45) developed Watt's approach and stated that when the chords are small the polygon becomes a cycloid. He indicated that a point on the circumference of a circle of diameter equal to the drum radius would trace out the appropriate cycloid if it was allowed to roll along the line from the shoe pivot to the brake centre. The lining arc angles are marked off and the line joining these points bisected. Where this bisection intersects the cycloid is the centre of pressure.

Acres (Ref 46) in the first clear general exposition of the method of calculating the forces acting on pivoted drum brakes demonstrated that the locus of the centre of pressure is a circle, the basic circle. He went on to describe a method of calculating the torque developed by pivoted brakes using the triangle of forces and discussed how the torque may vary for different lining to drum contact patterns.

Watt (Ref 44) assumed a cosine pressure distribution acting on the lining surface in his work. This same distribution was deduced by
Dawtrey (Ref 47), who assumed the lining was elastic in compression, and Merritt (Ref 48) who assumed that the wear of the lining at any point on the contacting surface is proportional to the pressure at that point.

2.2.2 An initial investigation of the parallel sliding abutment brake was made by Waller (Ref 49), who instead of constructing the basic circle described a circle about the centre to determine the centre of pressure. This circle and the basic circle touch only on the lining arc bisector and then gradually diverge, thus when the centre of pressure lies near the arc bisector as at a low friction levels the discrepancy between the two approaches is small and Waller’s simple construction can be used. However, the error rises as the coefficient of friction is increased and Waller’s answers become incorrect especially when the spragging zone is reached.

Waller’s work was extended by Robinson (Ref 50) and Oldershaw and Prestidge (Ref 51). Robinson calculated the torque exactly for both parallel and inclined abutment brakes. Oldershaw and Prestidge repeated Robinson’s work noting that contact between lining and drum may be lost as the shoe slides along the abutment for certain values of the coefficient of friction depending on the length and the position of the lining.

2.2.3 A wholly theoretical analysis of an abutment brake was attempted by Steeds (Ref 52) who proposed a pressure distribution of the form \( k_1 \cos \theta + k_2 \sin \theta \) to account for the two degrees of freedom in an abutment brake of sliding and rotation of the shoes. \( k_1 \) and \( k_2 \) are constants to be determined. In order to obtain the constants it was necessary for Steeds to make an assumption about the contact situation at the abutment. He investigated two situations:

(i) no friction at the abutment (pure sliding)
(ii) sufficient friction at the abutment to prevent any sliding (pure rolling)
He was then able to calculate $k_1$ and $k_2$, and hence the torque output. He applied his results to the special case of a pivoted shoe ($k_2=0$) and showed they were in agreement with the standard formulae. The analysis is not applicable to the general case of both rolling and sliding and Steeds does not discuss the validity and implications of his results.

Steeds work was repeated by Stroh and Lawrence (Ref 53) who extended it by allowing for friction at the abutment. They went on to attempt to take drum and shoe deflections into account by representing the shoe as a curved beam subject only to radial loads. From an initial cosine pressure distribution they obtained an expression for the deflection. By assuming the change of pressure to be proportional to deflection they qualitatively discussed how the total pressure distribution might be affected by flexure of the assembly.

Parker and Newcomb (Ref 54) observed that for both disc and drum brakes, the static distribution of interface pressure was altered by the application of tangential friction forces under dynamic conditions. Prior to this work, as the above summary shows, the dynamic pressure distribution had been assumed to be determined by the geometry of the brake. The assumptions of rigid brake shoe and drum lining material which obeys Hooke’s Laws inherent in the graphical techniques discussed above constrain the pressure distribution to be dependent upon the virtual displacement of the brake shoe, generating a sinusoidal form.

The limitations of the assumptions inherent in conventional geometric drum brake analysis concerning the form of the pressure distribution were recognised by Millner and Parsons (Ref 55) who idealized the brake shoe as a thin curved elastic strip and the brake drum as a thin proof ring. The equivalent rigidities were obtained experimentally.

The main limitation of Millner and Parson’s model is that it cannot be easily refined to include thermal effects. The heat input to the brake changes both the friction characteristics of the lining and the geometry of the components, and the combined effects of these changes
have a large effect on the performance of the brake. Another limitation of the model is that the empirical determination of the flexural properties of the shoe and drum makes it less adaptable to investigate the effects of changes in the overall geometry.

2.2.4 Lining wear can also influence drum brake performance and its effect was investigated by Day and Harding (Ref 56). Using results published by Millner and Parsons (Ref 55) as experimental evidence, they concluded that no single type of pressure distribution, "u"-shaped or sinusoidal, assumed or predicted, can be said to apply uniquely to any given brake shoe. A brake drum may experience all types of pressure distribution between the extremes of "heel-and-toe" and "crown" contact affecting its performance correspondingly. They stated that the principal effect of lining wear is to stabilise the brake performance at a particular pressure/temperature condition, and to accentuate the effects of fluctuations in these conditions.

2.2.5 Day and Harding together with Newcomb (Ref 3) produced a combined thermal and mechanical analysis of a commercial vehicle drum brake using the finite element method. The effects of lining wear, empirically related to local values of surface temperature and pressure, were considered in the calculation of interface pressure distributions and consequent brake performance. Lining wear was found to be greatest over regions of high surface pressure and temperature, and such regions will be worn away faster than low pressure regions, thus causing some trend towards a uniform pressure distribution. Furthermore the dynamic pressure distributions predicted by the finite element analysis, which takes into account the pressure of brake shoes, linings and drum, are more cosinusoidal ("u" shaped) than those predicted assuming rigid components. The latter assumption, which leads to a sinusoidal pressure distribution, is adequate for basic design purposes but insufficient for the detailed study of problems arising from service brake performance.
2.2.6 Myers (Ref 57) investigated the effect of "S" cam brake component variation on performance, questioning the assumptions inherent in the theoretical approach to drum brake analysis. This paper was based on the findings of work in response to the introduction of the American braking regulation FMVSS 121. The common approach is to assume:

(i) All the members are rigid.
(ii) Anchor pivot and input mechanism friction is zero.
(iii) Normal lining contact loading is proportional to wear.
(iv) Friction between lining and drum is constant, unaffected by normal contact loading.

In his investigation Myers found that brake shoe rigidity has a major effect on performance, more flexible shoes showing greater mechanical fade and possibly temperature fade. Temperature fade was said to be due to local hot spots on the cam end of the trailing shoe and mechanical fade was due to the greater deflections using up the actuator stroke. The cam and cam roller were also found to have a significant effect on brake performance. The cam profile and concentricity together with cam head hardness were found to be important. Brinelling can be caused by the high Hertzian stress due to roller to cam contact. Assuming the cam roller has adequate hardness to prevent brinelling it can still cause a shift in brake input efficiency. The resistance to rotation due to friction between the roller and pin or shoe, directly reduces the brake efficiency. The higher the contact point on the cam profile the greater the reduction.

Myers went on to determine that if the clearance on the camshaft bush is excessive then the cam can float becoming an equal force mechanism. This latter arrangement influences the behaviour of the brake and its effect was investigated in greater detail by Day and Harding (Ref 58). They found that a change in brake performance during the bedding in stage or the early part of the working life of a set of brake linings can be due to the progression from a floating cam mode to an equal work mode. The full theoretical floating cam mode may not be achieved in
practice if cam bush clearance are small and the brake is well constructed to close tolerances; instead the brake may operate in some intermediate stage between the floating cam and equal work modes depending upon the camshaft bearing compliance.

2.3 THE FINITE ELEMENT METHOD APPLIED TO DRUM BRAKE ANALYSIS

Conventional methods of brake analysis ignore the distortions of shoe and drum and are acknowledged to give inaccurate predictions of brake torque. Whilst they are adequate for very general design purposes the problem of advanced design methods demands a more rigorous treatment. The modern technique of finite elements enables the complex interdependent effects of interface pressure, temperature and friction material wear to be studied in detail.

2.3.1 The finite element technique has been used extensively for stress analysis of brake components, but its use for the study of high energy sliding systems was pioneered by Kennedy and Ling (Ref 1). Fessel (Ref 59) attempted to find the mechanical and thermal stresses of a drum brake by the use of finite element analysis. The drum was assumed to be a solid of revolution and was thus modelled asymmetrically. The loading was expressed in a Fourier series. Each term was analyzed separately and the total response obtained from the summing of the individual responses.

2.3.2 Day, Harding and Newcomb (Ref 2) presented a simple model of a shoe and lining as a means of obtaining both improved accuracy and a development capability for including drum distortions and thermal effects into brake analysis. This model is supported by the theory of a beam on an elastic foundation in order to illustrate the importance of the relative stiffness of shoe and lining. This work was based on that undertaken by Wintle (Ref 60) adapted to PAFEC 75 software.
Day et al modelled the shoe and lining using standard finite elements and the drum was considered to be a rigid circular boundary around the lining circumference. The friction interface was at this boundary and the PAFEC 75 programme was modified to allow for loss of contact at this surface. This was achieved by releasing the constraints automatically at any lining surface nodes which gave tensile reactions. The programme iterated to give a stabilised pressure distribution at this surface. The friction forces were calculated from the radial pressure distribution along the outer circumference of the lining by multiplying each nodal pressure at this boundary by the coefficient of friction and assigning this product as a tangential component calculated at each node in the direction of drum rotation. The torque output from the shoe was calculated from

$$ T = \mu r^2 B \int_{\theta_1}^{\theta_2} p(\theta) d\theta $$

(2.13)

where $\theta_1$ and $\theta_2$ define the lining arc and position. In this analysis thermal expansion was predicted using the temperature distribution developed by Ashworth et al (Ref 61).

Day et al obtained the radial pressure at any point $x$ by considering the lining and shoe as an elastic strip compressed against a rigid drum by a semi-rigid curved beam. This requires the solution of a differential equation for deflection $y$ at any point in the form:

$$ \frac{d^4 y}{d \theta^4} + \frac{2 d^2 y}{d \theta^2} + \left[ \frac{E_B r}{t E_l} + 1 \right] \frac{dy}{d \theta} = 0 $$

(2.14)

where the location parameter $\theta = x / r$ (Ref 62).

$I$ = the second moment of area of the shoe cross section

$B$ = the width of the lining

$E_l$ = Young's modulus of the lining

$E_s$ = Young's modulus of the brake shoe

$t$ = the thickness of the lining

$r$ = drum radius
The solution of this differential equation showed that the pressure distribution is a complex function of the ratio of lining stiffness to shoe stiffness.

2.3.3 Day and Newcomb (Ref 63) built on this above work by simulating thermoelastic instability at the friction interface using special elements and incorporating the wear of organically bonded composite friction materials into the simulation using an empirical wear correlation based on an Arrhenius law. The finite element analysis covered both temperature and stress distribution calculations and the same mesh was used for both. The mesh employed had a very thin row of elements at the position of the friction interface and such thin elements extended in both directions into the mating body and friction material where a steep temperature gradient was expected. The element sizes became larger and the mesh coarser where a less severe temperature gradient was expected. In the model, one line of elements were special, "compression only" elements to simulate the fact that normal reactions at the friction interface can only be compressive. The heat energy generated by the friction process was applied to the nodes on the friction surface of those elements which were referred to as "friction interface source" elements ("fis" elements). The contact situation at the interface was determined from the stresses acting in the single row of "fis" elements. The average of the three nodal stresses along the interface was found to be the criterion which gives the most consistent results for stability in the contact assessment.

In the context of the finite element analysis, the affect of wear, when it occurs at the friction interface, is to reduce the strain \(\varepsilon_p\) at the nodes in the "fis" elements on the friction interface, by an amount equal to the wear strain \(\varepsilon_{wear}\), thus

\[ (\varepsilon_i) = (\varepsilon_p) - (\varepsilon_{wear}) \quad (2.15) \]
In this analysis therefore, the wear was considered to be an additional strain at the friction interface, which is equivalent to the thickness of material worn off at the node in the interface.

For the purpose of this work wear was expressed in the form

$$\Delta w = Dp^a t \exp (-Q/RT) \quad (2.16)$$

where $D$, $a$, $b$ are constants.

As the speed was constant this was simplified to

$$\Delta w = Bp^t \exp (-Q/RT) \quad (2.17)$$

which was then converted into thickness loss and because the primary unknown in the finite element stress calculation is the strain, this is called the "wear strain".

The finite element analysis of brakes was further extended by Day in his investigation into the energy transformation at the friction interface of a brake (Ref 64). The method described above to model the performance of a brake (known as the Combined Stress Transfer Method) enabled a sliding friction pair to be modelled by a finite element mesh containing an interface defined by special no-tension (fis) elements. An alternative approach was now proposed. This was to model each part of the friction pair by a separate finite element mesh, connected together at the friction surfaces by modal "Gap Forces", which have the characteristics associated with the forces transmitted across the friction interface, that is compression only. This method has been extensively used for structural analysis where members which may be initially separated can come into contact under load (Ref 64, 65, 66). In such an event tangential frictional forces may be developed as well as normal compressive forces transmitted, and problems of shrink-fit and bonding have been studied using this method. The slippage which may occur when the relative tangential force exceeds the bond strength
has definite parallels in problems involving static and dynamic frictional contact.

In the Gap Force method the actual dimension of the gap between the two friction surfaces is determined by the specific gap value and not by the dimensions of the finite element mesh. Therefore, wear of the friction material can be easily incorporated into the simulation by increasing the gap size by the amount of wear which has occurred at each node pair. In his work Day adopted the time step approach of Kennedy (Ref 67) for the combined thermal and thermo-elastic analysis of the brake friction interface. Over the duration of each time step the interface contact and pressure distributions are assumed to remain unchanged, and transient temperatures are calculated based upon a heat flux distribution determined from the pressure distribution.

2.3.4 Samie and Sheridan (Ref 68) used the gap approach to analyse the mechanical behaviour of a passenger car brake sliding caliper. They found the gap element approach of Day to be costly for a large finite element model and in addition the iterative scheme employed by NASTRAN finite element code does not converge easily for the case involving sliding friction. Therefore complementary equations for the gaps between adjacent nodes were formulated in terms of the contact compliances. The compliance matrices were determined for each contact surface using NASTRAN (the compliance matrix is the inverse of the stiffness matrix and does not vary as long as the mechanical properties and mesh remain unchanged). The gap equations for each interface node were derived as a function of the compliance matrices, rigid body rotations, applied load and the unknown force distribution. Using this method, parameter studies considering such variables as initial gap, loading, stiffness, etc, were performed with a minimum cost.

2.3.5 A calculation method to predict the operational life and wear rate of drum brake linings was proposed by von Heck (Ref 69). In order to model the frictional interface of an S cam brake a similar approach to that of Samie and Sheridan (Ref 68) was applied. To relate the contact
pressure, $p$, and the distance between shoe and drum, $\delta_s$, a flexibility matrix, $G$, was calculated from

$$\delta_s = G.p$$  \hspace{1cm} (2.18)

The matrix $G$ is composed of a number of finite element calculations. The distance between drum and shoe is decreased by a column matrix $\delta$ which is calculated from

$$\delta = \delta_s - \delta_l - \delta_t - \delta_w$$  \hspace{1cm} (2.19)

In equation (2.19) $\delta_s$ represents the decrease in distance due to displacement, $s$, of the S cam, $\delta_l$ represents the lining wear at the degrees of freedom, $\delta_t$ are the deflections caused by the thermal effects and $\delta_w$ is calculated in equation (2.18). A set of linear equations for the contact pressure distribution and the S cam displacement are then calculated. Since the contact pressure must have a positive solution an iterative procedure is used whereby the solution procedure is repeated until all contact pressures are positive. This approach does not allow for separation between drum and lining ($p = 0$).

2.3.6 The partitioning of the energy dissipated at the brake's friction interface, that is the friction material/rotor split, cannot be accurately determined without a complete thermo-mechanical analysis. Day and Newcomb (Ref 70) found that the formation of surface layers and interfacial wear products have a significant effect on heat transfer from the interface. They found that interface contact resistance leads to different temperatures at the surface of rotor and lining so that heat partition between the two mating bodies cannot realistically be assumed constant under braking conditions. Other investigators (Refs 71, 72, 73) have made an estimate of the energy partitioning, making the assumption of equal average temperatures at the interface. The fraction of the total energy dissipated in the rotor, $\gamma$, is determined from the analysis of two semi-infinite solids coming into contact (Ref 74).
\[ \gamma = \frac{1}{1 + \left( \frac{\rho C K}{\rho_r C_r K_r} \right)^i} \] (2.20)

where \( C \) is specific heat, \( \rho \) density, \( K \) thermal conductivity and the subscripts \( p \) and \( r \) refer to the friction material and rotor respectively. When the rubbing path areas are unequal \( \gamma \) is given by

\[ \gamma = \frac{1}{1 + \left( \frac{\rho C K}{\rho_r C_r K_r} \right)^i \frac{A_p}{A_r}} \] (2.21)

where \( A_p \) is the area of the linings and \( A_r \) area of the rubbing path.

2.4 SUMMARY OF LITERATURE SURVEY

From the literature survey it has been seen that there are many different potential causes of anomalous performance of commercial vehicle drum brakes. During braking energy is dissipated by frictional heating generated by the physical interaction of asperities at the rubbing surface causing an increase in temperature of the sliding pair. The temperatures and forces involved may be sufficiently large to cause non-uniform deformation of components of the brake changing the nature of contact between the friction material and mating surface. Therefore in considering brake performance it is necessary to consider other factors in addition to those of brake geometry and friction material composition. Indeed the action of a brake during a stop is a very complex one involving the interaction of many different parameters, many of which are temperature or pressure dependent.

Wear is one of the most important of these and Day and Harding (Ref 58) investigated the effect of friction material wear on brake performance. They consider that the principal effect of lining wear is to stabilise the brake performance at a particular set of operating conditions, thus accentuating the effects of fluctuations in these conditions. This is because the torque output of a drum brake is significantly affected by the circumferential distribution of pressure along the brake lining arc. Wear of friction materials has been found to be greatest over regions of high surface pressure.
and temperature. Such regions will be worn away faster than low pressure
regions thus causing some tread towards a uniform pressure distribution. When
the brake operating conditions change however, there is a change in the drum
temperature and hence thermal expansion, changing the contact pattern between
brake drum and friction material. It has been found that the lining to drum
fit as a result of usage history can have a significant effect on brake
performance particularly at low actuation pressures.

The wear of the friction material is dependent not only on the brake operating
conditions but also on the composition of the material. Although much work
has been done comparing asbestos based and semi-metallic brake lining
materials much of it is contradictory. Rhee and Liu (Ref 25, 26, 27) found
that wear of organic friction materials was directly proportional to the time
of test but not so for semi-metallic materials. This was attributed to the
fact that the metal constituents are more reactive than asbestos, hence the
composition of the surface of the semi-metallic friction material changes with
time, thus changing its wear rate. In contrast Pavelusu and Musat (Ref 30)
found that the wear of materials which include metal fillers is less than
materials which do not because of the superior heat conduction of the metal.
This does not significantly reduce the temperature at the actual friction
surface (this can only be reduced significantly by increasing the thermal
capacity of the drum hence lowering the wear of the brake lining material by
virtue of the lower temperature of the friction face), but the temperature
gradient at the surface of the friction material is reduced, so less thermal
stress is induced. This contradiction can be explained by the work of
Rakowski (Ref 28) who found that the degradation of the resin binder can be
increased or decreased by any metal particles present. Cracking can be
increased due to thermal stress at the boundary between the metal particles
and the resin binder leading to increased wear. Conversely the increased
thermal conductivity of the material due to the addition of metal particles
can have a beneficial effect by significantly reducing the thickness of the
degraded layer by lowering the temperature at the surface.

Friction rarely occurs directly between the friction material and rotor but
between transfer layers or third body layers on the surface of the lining or
drum or both. These coatings are formed by chemical and physical interactions at a range of temperatures at the rubbing interface. The composition of the transfer layers is determined by the composition of both members of the friction pair and the usage history of the brake. The usage history of a brake has been identified (Ref 75) as a major influence on subsequent brake performance.

Conventional methods of brake performance calculation are limited by assumptions concerning the distribution of interface pressure whilst interdependent thermal and wear effects associated with brake operation have not normally been covered by such analyses. Increasingly the use of finite element methods is being made to take into account factors such as the flexure of brake shoes, linings and drum in the analysis of drum brakes. The friction interface can be modelled by the use of "Gap" elements which allow for relative movement between each member of the friction pair. Sliding friction was incorporated into the gap approach by Day (Ref 4), allowing tangential friction forces to be developed as well as normal compressive forces transmitted. A two dimensional approach was employed by Day (incorporating brake shoe stiffness approximations) enabling the circumferential pressure distribution around the lining are to be determined. To investigate the non-uniform deformations that may occur across the width of the brake rubbing path under conditions of high temperature and pressure a three dimensional analysis is necessary and this is one of the principal aims of this present study. This work is supported by experimental investigations to provide the properties of friction materials to be included in the analytical model and also to compare experimental data from brake tests with results obtained from the theoretical analysis applied to commercial vehicle drum brakes.
3. FRICTION MATERIALS

3.1 FRICTION MATERIAL COMPOSITION

3.1.1 Thermal Considerations

Possibly the single most significant factor to be considered when designing a friction material is temperature (Ref 76). All other factors - pressure, running speed, frequency of application of the brakes - manifest themselves ultimately as temperature changes. The problems encountered at extremes of temperature, or by significant temperature changes over relatively short periods of time, form the basis of friction material development work.

Temperature stability and thermal stability in a raw material or a friction material composite are considered as two separate properties. If a material is placed in an oven at elevated temperatures, and its physical and chemical properties do not change significantly then it is considered "temperature stable". However that same material when used in a friction composite may be thermally unstable when subjected to the complex thermal and mechanical stresses present at the friction interface during braking. Recent developments have seen a dramatic change in friction material compositions as the asbestos content has had to be substituted by organic man-made fibres or steel fibres. Man-made fibres with suitable temperature/heat resistance are very expensive and hence there has been a significant reduction in the volume of such fibres in any formulation. Steel fibre contents can be high but there are associated problems from the greater proportion of heat generated being transferred into the material. Friction material formulations, in particular the organic content, have had therefore to be modified to compensate for these changes.
3.1.2 Friction Material Elements

There are five main elements in the composition of a friction material; fibres, abrasives, lubricants, fillers and organics. Each element has a primary role to play in the performance characteristics of the friction material, but each will have secondary effects or may perform a dual role. Secondary effects are usually detrimental to the performance of the friction material, and hence are as important as the primary effects.

The volume of each element, and the volumes of the fractions which comprise each element, varies from formulation to formulation, giving each its own particular performance characteristics. Different formulations can be divided into families in which the proportions of the elements are appreciably different from family to family, for example High Ferrous, Low Ferrous, Non Metallic, High Organic, etc. Each of these families has different frictional characteristics, and each will therefore be better suited to different vehicle/brake installations.

3.1.2.1 Fibres

The primary objective of adding fibres to a composite formulation is to increase strength. Fibres impart physical strength increasing the liner strength and impact resistance as well as providing resistance to thermal shock. Rapid fluctuations in temperature, particularly at the surface of materials with low thermal conductivity, result in high surface stresses. These surface stresses can result in thermal cracking and crack propagation into the body of the material.

Asbestos fibre provides a unique combination of properties highly desirable in friction materials. The very strong and high modulus fibrils are tightly held in bundles giving apparent fibre flexibility and are commercially available in fibre length grades which give a high level of reinforcement to friction material composites, even at
elevated temperatures. The impact strength of such composites is relatively high and their thermal conductivity is low. The bundles of fibres, in high concentrations, provide a medium for internal damping, inhibited only by the modulus and inter-lamellar shear of the surrounding organic matrix. Both the physical and chemical properties of asbestos are stable throughout the temperature range experienced by friction materials. Simple formulations containing high proportions of asbestos display the mineral's good abrasion resistance, naturally high friction level and kindness to metal mating surfaces.

In addition asbestos is of low cost and easily processed. A diverse variety of mixing techniques can be used during production as the fibres are resistant to breakdown and are readily compatible with the organic binder systems common to friction materials. However, despite all these beneficial properties, asbestos has had to be replaced in friction materials because of health and safety considerations. It was hoped initially that a single fibrous material would be found which would encompass most of the above features and hence be a direct substitute in both formulation and performance.

Glass, used in GRP (glass reinforced plastic) mouldings, was the first choice, having good heat resistance and thermal stability. It gave a high level of reinforcement and could be made compatible with the organic matrix through surface coatings (silanes etc). However, it had an inherently low friction level and tended to score the mating face. In addition it was difficult to process, the fibres breaking down easily to form a non-reinforcing powder, and the mouldings were more brittle than those produced from asbestos. The friction compositions were also prone to noise generation and transmission.

Second choice was steel fibre which had been used previously during the Second World War when asbestos was in short supply. When used in high concentrations steel fibres give good material strength, good thermal stability and good compatibility with the organic binders. The fibres are stiff and robust, however the fibre shape, length to diameter
ratio, particle size and distribution curve are important. Correct grade steel fibres are kind to the mating surface and give very low wear rates. However the inherent friction level is low and tends to rise with temperature and its thermal conductivity is high.

Carbon fibres appeared initially to be an ideal substitute for asbestos despite a relatively high cost. The fibres are strong, light, temperature resistant and compatible with the binder systems. However, most friction materials produced from these fibres, when rubbing against cast iron, exhibited two major faults. Firstly, under light to medium duty running levels, a steady persistent increase in friction level was observed. Over prolonged periods the final level could reach a value twice the formulated or initial value. Secondly, under heavy duty conditions, the fibres decompose to graphitic type structures at the friction interface producing erratic and unpredictable low levels of friction. This is possibly due to orientation effects of the matrix bound fibre.

With the advent of polyaramid fibre came a man-made organic fibre which was flexible, heat resistant and thermally stable. It is easily processed, the material fibrillates and is compatible with the organic matrix. Polyaramids are effective in low concentrations but their high cost restricts their use to where absolutely necessary.

Although vermiculite is non-fibrous, its other properties are very similar to those of asbestos. Vermiculite is a type of exfoliated mica. It has low thermal conductivity, a good inherent level of friction and abrasion resistance and is compatible with organics. It is relatively cheap and when blended with flexible reinforcing fibre its properties, both physical and frictional, closely approximate to those of equivalent asbestos composites.

Other fibres investigated by the friction material industry include ceramics and mineral fibres. Generally these materials, although
thermally stable, tend to have such short fibre lengths that they impart little, if any, reinforcing effect into their composites.

The two fibres most commonly used today are steel and polyaramid. The two fibres are very different. Steel is an inorganic stiff fibre with high thermal conductivity. Polyaramids are organic, flexible fibres with low thermal conductivity. Steel is relatively cheap and polyaramids are very expensive; hence commercially the volume of polyaramid in a formulation is necessarily low. Because of its high thermal stability polyaramids are not usually considered as part of the organic element.

3.1.2.2 Organics

The organics are the most diverse and versatile element in a friction material. They can be dispersed within the main matrix as discrete particles or as a constituent part of the matrix itself. The processing conditions can be critical in achieving the desired homogenous matrix form from two or three (or more) organic components, despite uniform dispersion on a molecular scale of the components at the mixing stage. There are also problems of solubility at the mixing stage; a common solvent is required if phase separation is to be avoided. The organic element is the most unstable, and hence least predictable element in the friction material.

High proportions of asbestos in the friction material have a stabilising effect on performance. The volume of organic material in the main matrix of an asbestos based friction material tended to be considerably lower than that in non-asbestos materials. Asbestos based friction materials are thus characterised as high fibre (flexible fibre) content and low organics. A non-asbestos friction material can contain a high proportion of steel (rigid fibre) combined with a low organic content but the composite has a high thermal conductivity and low flexibility.
As already mentioned the high cost of polyaramid fibre restricts its use to low concentrations. Therefore the additional volume previously occupied by fibres in earlier high fibre formulations has to be filled by other material. Particulate fillers are non-reinforcing, actually weakening the structure, hence a proportion of the "vacant volume" has to be filled with additional binder if the material integrity is to be maintained. Thus there has been an appreciable increase in the organic element giving rise to added instability, combined with the loss of the stabilising influence of asbestos, previously present in high proportions. The combined effect of these two changes has been a destabilising of friction performance which had to be countered by a different approach to formulating.

The organic matrix of non-asbestos friction materials has been elevated from a relatively minor consideration when used in asbestos based products, to a major factor in the formulation. The use of Phenol/Formaldehyde resins as straight "binders" in the days of asbestos has been superseded by the multi-role part played by the organic matrix in an asbestos free friction material. Resin technology has had to advance alongside brake lining technology with modifications to the basic Phenol/Formaldehyde system designed to relieve specific problems resulting with the move to asbestos free formulations.

3.1.2.3 Abrasives

Abrasives are added in relatively small proportions to a formulation to increase the general level of friction, and different abrasives have optimum effects in different temperature zones (figures 3.1). A relatively simple formulation may, at low temperatures, give the required level of friction. As the temperature at the friction interface increases changes to the organic matrix result in a lowering of friction, or fade, as a result of decomposition, or merely softening, of the binder. It is therefore necessary to bolster the friction level by the addition of abrasives. The selection of a particular abrasive, its particle size and the proportion in which it is to be
added, are very important. Particular abrasives are most important within particular temperature ranges, for example the more abrasive materials such as Alumina, are most effective at the highest temperatures. Therefore in any friction material formulation there will be several abrasive materials, each being effective at a particular temperature. A suitable blend of abrasives will result in a continuous but approximately constant friction level throughout the required temperature range.

The conditions under which the rubbing surface reaches a given temperature will effect the performance of the abrasive content of the friction composite. A steady rise to a high temperature presents one set of criteria to be satisfied. A rapid rise to the same temperature can present different criteria. The control of friction level through abrasive content has to be a compromise between the two, but the formulation may be biased if it is felt that one set of criteria are predominant. A wide range of abrasives is used although refractory oxides and silicates are generally preferred. The effect on counterface wear is complex being coupled to the particle shape and size; this is discussed at some length in Chapter 2.

3.1.2.4 Lubricants

Lubricants are added in relatively small amounts to reduce wear of both members of the friction pair. Different lubricants have an optimum effect at different temperatures (Figure 3.2). At lower temperatures materials such as graphite are most effective. At higher temperatures materials such as copper may be necessary. An inclusion of steel fibres, primarily to give strength, has a dual effect of improving wear, since it corresponds to a large area asperity having the good wearing properties of mild steel on graphitic/perlitic cast iron. The effect of adding a particular lubricant will be dependent on the proportion added. As with abrasives there will be an optimum inclusion for a particular set of circumstances and a blend of lubricants maybe required to cover the whole temperature range requirements. The
formulation of the blend can be biased towards a particular temperature range to suit a specific brake/vehicle installation.

Graphical representations of friction level and wear against temperature are depicted in figures 3.1 and 3.2. They show the general trends for the additions of abrasives and lubricants to a friction material. Under gentle braking conditions mating face temperatures may settle within a single zone, hence a wear rate could be read off directly for this steady state. However, when heavy duty braking is experienced, the temperature will climb and fall through several temperature zones during each braking cycle. Each set of circumstances should ideally have its own blend of lubricants to give optimum wear characteristics. This is not possible in a single friction material so a compromise must be reached.

3.1.2.5 Fillers

Fillers are generally low cost additives and are added in relatively high proportions. They can be divided into two groups; fillers added for cost reduction, and fillers added to enhance the friction material performance. Performance enhancers, such as barytes, can be subdivided into mild abrasives and mild lubricants. Since fillers can be added to relatively high inclusion levels, they can have a significant effect on the performance of a friction material despite their being characterised as "mild". Most fillers are non-reinforcing and hence stiffen and weaken the organic matrix.

3.2 FRICTIONAL HEATING EFFECTS DURING BRAKING

3.2.1 During braking frictional energy is dissipated as heat, and this heat plays a significant role in brake friction behaviour. When a system as complex as a friction material is subjected to heating, chemical and physical changes occur and the nature and extent of these changes effect the performance of the friction material. The chemical changes
may be of various types, including reaction between constituents or degradation of a particular constituent, changing the friction material characteristics and physical properties.

### 3.2.2 Chemical Changes in the Organic Constituents of Friction Materials

Phenolic resins, which make up the bulk of the organic constituents in friction materials, may promote two major changes during braking. Firstly the volatile products of degradation can have an effect on brake performance (brake fade), particularly at higher temperatures when the amounts evolved are often large, and secondly the chemical nature of the residual polymer will to some extent determine subsequent frictional behaviour, (Ref 29,42,77).

The behaviour of the friction material organics or polymers can be interpreted in terms of char formation, or "ablation". There are several factors which affect the ability of polymers to form stable char of which chemical structure is the dominant one. The majority of the work concerned with phenolic resin ablation has been due to the USA space programme. The hulls of many space craft have been treated with phenolic resin preparations in order to protect the metal from the generation of large amounts of heat during re-entry into the earth's atmosphere by the phenolic resin undergoing ablation and in so doing absorbing the heat generated. A similar process occurs at the brake friction interface where the phenolic resin absorbs heat and in so doing suffers degradation resulting in the formation of volatile by-products and a stable residue, the char. The ablation process is to some extent governed by the amount and composition of the resulting pyrolysis products; the chemical reactions occurring in the char layer playing a significant part in determining the overall ablationary behaviour. The degradation of the phenolic resin can also be affected by the interaction of the phenolic resin with the inorganic constituents of the friction material, although the effects differ widely depending on the inorganics used.
3.2.3 Chemical Changes in the Inorganic Constituents of Friction Materials

The chemical changes occurring in inorganic friction material constituents are present to a much lesser extent than those involving organic constituents. A huge array of inorganic compounds have been proposed for use in friction materials, and any chemical change undergone by any of these components may effect the performance of the friction material. For example molybdenum disulphide, used as a friction material lubricant, begins to oxidise at relatively low temperatures (100°C) and proceeds rapidly at higher temperatures (approximately 400°C). Surface oxidisation of the molybdenum disulphide affects the formation of transfer films at the brake interface, effecting the friction performance.

3.2.4 Physical Changes to the Friction Material

The physical changes to the friction material that occur during frictional heating are of less significance to the performance of the brake than chemical changes over the temperature range normally experienced during braking. It has been shown by Tanaka et al (Ref 78) that during braking friction is strongly controlled by the lubricating action of the resin decomposition products, there being no correlation between the overall friction level and the mechanical properties of a resin based composite. However, the physical properties of the material can strongly influence the contact conditions at the friction interface and hence brake output. The variation of friction material physical properties with temperature is investigated in section 3.4.

3.3 Friction Material Idealisation

The aforementioned chemical changes to a friction material can be incorporated into an idealised model consisting of several layers to represent the phase changes in the friction material due to frictional heating. Although the chemical changes through the depth of material are continuous three separate
layers can be identified. At the rubbing surface there is a layer which is predominantly char, under which there is a layer in which some degradation is apparent, and thirdly there is a layer of unreacted or virgin material. The depths of these layers are determined by the usage history and thermal conductivity of the friction composite. When the brake is applied friction occurs not directly between friction material and rotor, but usually through the interaction of third body or transfer layers on either the friction material surface or drum rubbing path, or both. Therefore a five phase idealisation can be made of the friction pair (Figure 3.3).

**Phase 1:** Virgin friction material which exists in a relatively unchanged state from ambient temperature to about 180°C.

**Phase 2:** The reaction zone is the phase in which degradation occurs, in the temperature range 180°C to 400°C.

**Phase 3:** The surface layer of char which is the residual from the reaction zone, and exists within the temperature range 400°C to 1,000°C.

**Phase 4:** Interface layer; it includes material transfer, or coating, to the mating surfaces, and wear debris. This phase consists mainly of inorganic material and, once sliding is initiated and wear has occurred, can be present through the temperature range ambient to 1,000°C.

**Phase 5:** Metal mating body, which is assumed to be perfectly smooth and elastic, and unaffected by wear in the finite element analysis. Grade 14 and 17 grey cast iron is commonly used for brake drums although other non-ferrous materials may be used in specialised applications.
3.4 DETERMINATION OF FRICTION MATERIAL PHYSICAL PROPERTIES

3.4.1 Friction Material Physical Properties - Assumptions

The physical properties of friction materials are not only time dependent but due to the methods used in friction material production, highly directional. For example, three different values of tensile strength can be obtained for a friction material sample for each orthogonal direction. However, when a brake is applied the friction material is subjected primarily to forces normal to the rubbing surface, thus the response of the friction material to these forces is considered to be most significant. It is assumed therefore that the friction material is linear elastic in compression and that the property measured in the direction of interest also applies to the other orthogonal directions.

The variation in physical properties of the friction material with temperature is incorporated in the brake analysis by using tables of relevant values within the coding.

3.4.2 Young’s Modulus E

The Young’s Modulus is calculated for the material in compression using a Teves Compression Apparatus to BS AU180 part 1 1983 (ISO 6310). A pad of friction material is produced, relieved of residual stresses resulting from production, and then loaded to determine its compressibility. Readings are taken at stresses of 4,000 kPa, 6,000 kPa and 8,000 kPa at ambient temperature, and then the process is repeated at 200°C and 400°C. The Young’s Modulus of the friction material is then calculated from $E = \sigma/\varepsilon$ at each temperature.

Shown in Figures 3.4 and 3.5 respectively are compressibility results for material X, asbestos free, and material Y, asbestos based. It can be seen that the compressibility of both materials is significantly increased with temperature (material X exhibits a fivefold reduction in...
stiffness between ambient and 400°C; the Young’s Modulus varying between $25.8 \times 10^8$ and $4.3 \times 10^8$ N/m$^2$).

### 3.4.3 Poisson’s Ratio $\nu$

Poisson’s ratio is a difficult parameter to measure for polymeric materials because of their anisotropic behaviour; therefore a value must be assumed for the purposes of finite element simplification. Conventional measurement techniques using "right cylinder" shaped specimens may yield values of $\nu$ as high as 0.5 (which is theoretically inadmissible for elastic isotropic materials). Early work (Ref 4) has shown that good correlation can be achieved using a value similar to that of the shoe, that is $\nu = 0.25$. This may be justified since the friction material is used in a configuration where the thickness is small in comparison with other dimensions, and bonding or rivetting of the lining to the shoe reduces the lateral strains introduced by compressure applied forces. In addition high surface friction produces a lateral stiffening effect even where friction drag creates tangential tensile forces.

### 3.4.4 Density $\rho$

The specific gravity is calculated to BS AU 142: 1968 part 3.2.2, method 2. The weight of the specimen is determined in air and then in water (temperature of the water being between 18°C and 24°C) and its specific gravity calculated from

$$SG = \frac{W_a}{W_a - W_b}$$  \hspace{1cm} (3.1)

where: $W_a$ = Weight in air in grammes
$W_b$ = weight in water in grammes.

It is found that there is little variation in density over a wide range of temperature.
3.4.5 Thermal Expansion

Thermal expansion is a temporary increase of brake lining thickness due to temperature increase. Initial thermal expansion refers to the brake block thickness dimensional change during the first brake usage, otherwise known as "green swell". The largest portion of green swell occurs when the friction material is subjected to temperatures in excess of those of the manufacturing process. As the brake block is heated volatiles generate internal pressure within the friction material. At the same time the lining softens from this heat enabling residual thermal strains to relax. The binder resins may also undergo additional cure. These factors combine to cause brake blocks to swell much more, and quite unevenly, during the first significant heating during service usage. Following green swell, repeated heating will provide a much smaller steady state thermal swell as the residual volatile material has been baked out and thermal strains relaxed. Thus steady state swell involves simple thermal expansion, but remains somewhat non-linear.

To determine the coefficient of linear thermal expansion of a friction material a Stanton Redcroft TMA 691 is used. A sample of friction material is heated several times to remove green swell effects. The thermal expansion is then calculated for the temperature range ambient to 400°C using the formula

$$\alpha = \frac{E}{T \times L} \tag{3.2}$$

where:
- $E =$ sample expansion at maximum temperature
- $T =$ maximum temperature - starting temperature
- $L =$ initial thickness.

This temperature range encompasses phases 1 and 2 in the friction material, that is the virgin and reaction zone material, but is not representative of phases 3 and 4, the char and interface layers.
However the char and interface layers are only thin (surface effects) relative to the bulk of the material (phases 1 and 2), so their effect on the bulk thermal expansion of the material is small. The coefficient of thermal expansion for phases 1 and 2 can therefore be assumed for the friction material.

3.4.6 Thermal Conductivity \( k \)

The thermal conductivity of a material is defined as the quantity of heat in the "steady state" condition passing in unit time through an area forming part of a slab of uniform material of infinite extent and with flat and parallel faces. It is proportional to the area and to the temperature difference between the faces, and inversely proportional to the thickness of the slab. The thermal conductivity of a friction material is a characteristic property of that material and its value may depend on factors such as the material density, porosity, moisture content, fibre diameter, pore size and type of gas in the material as well as temperature.

The thermal conductivity is measured using a modified Lee's Disc Method (Figure 3.6). With reference to the figure the thermal conductivity is derived from the following equations.

\[
K_s = \frac{Q 	imes L_1}{A 	imes dT_2}
\]  
\[Q = \frac{K_s 	imes A 	imes dT_1}{L_2}
\]

where \( Q = \frac{K_s 	imes A 	imes dT_1}{L_2} \)

and

- \( L_1 \) is the friction material sample thickness
- \( L_2 \) is the distance between the thermocouples measuring \( dT \)
- \( A \) is the area of the steel reference block
- \( K_s \) is the thermal conductivity of steel
- \( K_n \) is the thermal conductivity to be determined.
The values of thermal conductivity presented are for reaction zone material (Figure 3.3). Different conductivities would be obtained for virgin material and the char layer at the rubbing surface, however the thermal properties of the reacted material are considered the most relevant when determining temperatures at the friction surface.

3.4.7 Specific Heat $C_p$

The specific heat of a friction material can be calculated from its formulation. Friction materials consist of a number of different constituents. The specific heat of the friction material is obtained from summing the individual contribution (specific heat of the constituent multiplied by its percentage by weight of the friction material) of each constituent. However this method does not allow for changes in specific heat of the friction material as it undergoes the transformations associated with pyrolysis. This difficulty was overcome by Lagedrost, Eldridge and Stone (Ref 79) who measured the specific heat of various friction materials using a differential scanning calorimeter and values of specific heat have been determined based on their work.

3.4.8 Coefficient of Friction $\mu$

The coefficient of friction of a friction material can be determined for a range of temperatures using a Chase friction materials test machine. A 25.4 mm square sample of friction material, radius ground to drum diameter to give a finished sampled thickness of between 5.85 mm and 6.10 mm rubs against an externally heated drum. The drum has a constant rotational speed of 434 rev/min giving a rubbing speed of 6.34 m/sec², and a contact pressure of 13.8 Bar (200 lb/in²) is maintained. An FM3 Test Schedule (Appendix 1) is used to obtain a preliminary assessment of the friction/temperature characteristics of friction material. Table 3.1 contains the friction temperature characteristic of a typical asbestos free friction material (Material X).
The small lining sample used to determine the friction coefficient, although being only a guide to the average friction level of the installed lining does provide the frictional characteristics required for the elemental approach adopted in finite element modelling.

**Table 3.1 μ - Temperature Characteristics of Friction Material**

<table>
<thead>
<tr>
<th>TEMPERATURE (°C)</th>
<th>μ</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>0.32</td>
</tr>
<tr>
<td>200</td>
<td>0.32</td>
</tr>
<tr>
<td>300</td>
<td>0.32</td>
</tr>
<tr>
<td>400</td>
<td>0.30</td>
</tr>
<tr>
<td>500</td>
<td>0.28</td>
</tr>
<tr>
<td>600</td>
<td>0.24</td>
</tr>
<tr>
<td>1,000</td>
<td>0.1</td>
</tr>
</tbody>
</table>

The value of friction coefficients cannot be determined at temperatures approaching 1,000°C using a Chase machine. The friction of composite friction materials composed of resin, fibres and additives, is known to vary considerably at higher temperatures, being found (Ref 78) to be low, of the order of 0.1 and these values have also been established from measurements made using a disc brake mounted on an inertia dynamometer.

**3.4.9 Summary of Friction Material Properties**

Table 3.2 compares the physical properties of an asbestos free drum brake lining material (Material X) with those of an asbestos based material (Material Y).
Table 3.2: Physical Properties of Commercial Vehicle Drum Brake Liners

<table>
<thead>
<tr>
<th>MATERIAL</th>
<th>PROPERTY</th>
<th>TEMPERATURE/°C</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>25</td>
</tr>
<tr>
<td></td>
<td>E (N/mm²)</td>
<td>2580</td>
</tr>
<tr>
<td></td>
<td>v</td>
<td>0.25</td>
</tr>
<tr>
<td>X</td>
<td>ρ (kg/m³)</td>
<td>1990</td>
</tr>
<tr>
<td></td>
<td>α (K⁻¹)</td>
<td>2.5x10⁻⁵</td>
</tr>
<tr>
<td></td>
<td>k (W/mK)</td>
<td>0.72</td>
</tr>
<tr>
<td></td>
<td>Cp (J/kgK)</td>
<td>820</td>
</tr>
<tr>
<td>Y</td>
<td>E (N/mm²)</td>
<td>2460</td>
</tr>
<tr>
<td></td>
<td>v</td>
<td>0.25</td>
</tr>
<tr>
<td></td>
<td>ρ (kg/m³)</td>
<td>1848</td>
</tr>
<tr>
<td></td>
<td>α (K⁻¹)</td>
<td>1.5x10⁻⁵</td>
</tr>
<tr>
<td></td>
<td>k (W/mK)</td>
<td>0.38</td>
</tr>
<tr>
<td></td>
<td>Cp (J/kgK)</td>
<td>903</td>
</tr>
</tbody>
</table>

3.5 PHYSICAL PROPERTIES OF CAST IRON

Compiled in Table 3.3 are common values for the variation with temperature of the physical properties of Grade 17 cast iron. Of these properties only thermal conductivity and specific heat show any significant change with temperature. However due to the influence of specific heat on the element size and time step used in the finite element analysis (see Equation 5.11) its value is constrained to be constant. In practice the effect of this variation in specific heat on brake drum temperatures will, to some extent, be reduced by the corresponding changes in the thermal conductivity of the cast iron. Therefore constant physical properties, as shown in Table 3.4, are assumed for cast iron in this work.
### Table 3.3 Physical Properties of Grade 17 Cast Iron

<table>
<thead>
<tr>
<th>PROPERTY</th>
<th>TEMPERATURE/°C</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>25</td>
</tr>
<tr>
<td>$E$ (N/mm$^2$)</td>
<td>$125 \times 10^3$</td>
</tr>
<tr>
<td>$\nu$</td>
<td>0.25</td>
</tr>
<tr>
<td>$\rho$ (kg/m$^3$)</td>
<td>7100</td>
</tr>
<tr>
<td>$\alpha$ (K$^{-1}$)</td>
<td>$1.1 \times 10^{-5}$</td>
</tr>
<tr>
<td>$k$ (W/mK)</td>
<td>59</td>
</tr>
<tr>
<td>$C_p$ (J/kgK)</td>
<td>544</td>
</tr>
</tbody>
</table>

### Table 3.4 Standard Properties of Cast Iron

<table>
<thead>
<tr>
<th>PROPERTY</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$E$ (N/mm$^2$)</td>
<td>$125 \times 10^3$</td>
</tr>
<tr>
<td>$\nu$</td>
<td>0.25</td>
</tr>
<tr>
<td>$\rho$ (kg/m$^3$)</td>
<td>7100</td>
</tr>
<tr>
<td>$\alpha$ (K$^{-1}$)</td>
<td>$1.2 \times 10^{-6}$</td>
</tr>
<tr>
<td>$k$ (W/mK)</td>
<td>54</td>
</tr>
<tr>
<td>$C_p$ (J/kgK)</td>
<td>586</td>
</tr>
</tbody>
</table>
FIGURE 3.1
SCHEMATIC EFFECT OF ABRASION ON
FRICTION AND FADE PERFORMANCE

FRICTION LEVEL

0.8
0.6
0.4
0.2
0

TEMPERATURE

100 200 300 400 500 600 700

INITIAL  ABRASIVE 1  ABRASIVE 2
FIGURE 3.2
SCHEMATIC EFFECT OF LUBRICANT ADDITIONS ON MATERIAL WEAR

SPECIFIC WEAR RATE mm\(^2\)/MJ

TEMPERATURE

INITIAL  LUBRICANT 1  LUBRICANT 2
FIVE PHASE MODEL OF THE BRAKE FRICTION PAIR

<table>
<thead>
<tr>
<th>PHASE</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Virgin Friction Material</td>
</tr>
<tr>
<td>2</td>
<td>Friction Material Reaction Zone</td>
</tr>
<tr>
<td>3</td>
<td>Char Layer</td>
</tr>
<tr>
<td>4</td>
<td>Interface Layer</td>
</tr>
<tr>
<td>5</td>
<td>Metal Mating Body</td>
</tr>
</tbody>
</table>
FIGURE 3.4
COMPRESSIBILITY MATERIAL X
ASBESTOS FREE

[Graph showing strain versus pressure for different temperatures: Ambient, 200 C, and 400 C.]
FIGURE 3.5
COMPRESSIBILITY MATERIAL Y
ASBESTOS BASED

Strain

MPa

Ambient  200 C  400 C
FIGURE 3.6
APPARATUS FOR MEASUREMENT
OF THERMAL CONDUCTIVITY

Cooling Platen

Steel Ref Block

Brass Disc

Sample

Brass Block

Insulation

Aluminium Plate

HEAT
4. FRICTION MATERIAL WEAR

4.1 INTRODUCTION

The tribological properties of friction materials have been seen (Chapter 2) to depend on a number of different factors such as pressure, rubbing speed, interface temperature as well as the composition of the friction pair. In addition the surface finish of the metal member, the friction material type, density and modulus of elasticity, together with the design and geometry of the friction mechanism all contribute to the wear of the friction material during braking. These factors can be included in a generalised relationship of the form:

\[ w = f (p, \nu, \theta, D, FM, ME) \]  

(4.1)

where 
- D is the effect exerted by the brake design
- FM is the effect exerted by the friction material
- ME is the effect exerted by the metal member

Representation of the tribological properties of friction materials in terms of all these factors is not practically possible. Many attempts have been made to study the tribological properties of friction materials in terms of the predominating factors. Rhee and Liu (Refs 25, 26, 27) found the wear rate of resin bonded composite friction materials to be controlled by the thermal decomposition of the phenolic resin. The thermal decomposition of the phenolic resin has been shown to be an energy activated process following an Arrhenius rate law where

\[ \text{Reaction Rate} = B \exp \left( \frac{-Qa}{RT} \right) \]  

(4.2)

Thus Rhee and Liu described wear of the friction material by the relationship

\[ \Delta w = \beta v^p t \exp \left( \frac{-Qa}{RT} \right) \]  

(4.3)
Although Rhee and Liu considered equation (4.3) to be effective only at temperatures above 232°C, the Arrhenius rate law on which it is based, equation (4.2), is valid at all temperatures. Equation (4.3) when applied to sliding conditions of constant velocity can be expressed in the form:

$$\frac{dW}{t} = \lambda p \exp \left( -\frac{B}{T} \right)$$  \hspace{1cm} (4.4)

where A and B are values that can be determined from wear measurements made from small sample testing. Values A and B have been determined (Ref 4) as $4.73 \times 10^{-12}$ and 9200 respectively for commercial vehicle friction materials.

Investigations of the tribological properties of friction materials are based on the assumption that the factors from equation (4.1) can be classified into two basic groups.

(i) Factors influencing the conditions at the friction interface ($p$, $v$, $\theta$).

(ii) Effects exerted by the brake design ($D$), friction material composition ($FM$) and the metal member ($ME$). These factors are highly inter-related.

Certain mutual dependence exists between these groups of factors; but it is not possible to consider all the complex interactions. However, if one specific friction mechanism is selected, i.e., one specific brake type, it is possible to assume that the effects of the factors from group (ii) are constant, thus,

$$w = f \left( p, v, \theta, C \right)$$ \hspace{1cm} (4.5)

Using this approach the friction material wear can be described in terms of the factors determining the actual operating conditions and loads of the brake under consideration.
Todorovic et al (Ref 32) showed that equation (4.5) could be used to model the wear of automotive friction materials when expressed as an experimental function of the third degree

\[ w = C_i p^a v^b \theta^c \]  

(4.6)

On the basis of the dispersion analysis conducted, it was concluded that the parameters \( p, v, \) and \( \theta \) were significant factors effecting the friction and wear of friction materials, and that the exponential function of the third degree allows adequate modelling of these parameters.

4.2 EXPERIMENTAL DETERMINATION OF FRICTION MATERIAL WEAR

4.2.1 Introduction

There are several ways of determining the wear of friction materials. Some methods are based on comprehensive road tests (Refs 80 and 81), others include inertia dynamometer tests or small sample tests. Road tests tend to be long lasting and hence costly procedures to determine friction material life under a range of operating conditions. The operating time and expense involved with road testing can be reduced by the use of inertia dynamometers to simulate a range of different operating conditions within the laboratory. The time and cost can be further reduced by small sample testing although in this case the results obtained are strictly related to the test conditions leaving only slight possibilities for the interpretation of results under different conditions.

4.2.2 Experimental Procedure

The approach used by Todorovic to express wear as an exponential function of the third degree, Equation (4.6), is adopted here. For the purpose of the experimental determination of the friction material
wear, tests were conducted on an inertia dynamometer. The brake chosen for the investigation was an hydraulic two leading shoe brake of diameter 330 mm and width 127 mm. Two friction materials were considered; material X, a typical asbestos free polyaramid fibre based material and material Y, a typical asbestos based product, to determine numerically the different wear characteristics of these two types of friction material.

A schedule of experiments (contained in Appendix 2) consisting of 4000 braking cycles was developed to provide the experimental data necessary to determine $C$, $\alpha$, $\beta$ and $\gamma$ in Equation (4.7) for each brake and friction material combination. Each braking cycle was determined by preset initial conditions of brake rotational velocity, actuation pressure and brake drum temperature, the values of which were carefully chosen to cover the normal operating conditions of the brake. The large number of cycles was necessary to provide measurable wear for each set of test conditions.

4.2.3 Numerical Procedure

Wear of a friction material may be expressed as

$$w = C p^a v^b \theta^c$$  \hspace{1cm} (4.7)

This function can be linearised and then solved by the method of least squares using linear regression.

$$\log w = \log C + \alpha \log p + \beta \log v + \gamma \log \theta$$  \hspace{1cm} (4.8)

Equation (4.8) is now a linear relationship of the form

$$Y = \beta_0 + \beta_1 X_1 + \beta_2 X_2 + \ldots + \beta_p X_p + u$$  \hspace{1cm} (4.9)

between a dependent variable $Y$ and $p$ independent variables $X_1$, $X_2$, $\ldots$, $X_p$. The parameters $\beta_0$, $\beta_1$, $\ldots$, $\beta_p$ are the unknown population regression
coefficients, and \( u \) is an unknown random variable which is a measure of the departure of \( Y \) from exact linear dependence on the \( p \) independent variables. It is assumed that the random variable, \( u \), has a mean of zero. The principle of least squares can be applied to Equation (4.9) to derive estimates of the population regression coefficients.

Suppose that the estimates of the population regression coefficients \( \beta_0, \beta_1, \ldots, \beta_p \) are \( b_0, b_1, \ldots, b_p \), then

\[
Y_i = b_0 + b_1 X_{i1} + b_2 X_{i2} + \ldots + b_p X_{ip} + e_i
\]

where \( e_i \), the \( i \)th residual, is an estimate of the \( i \)th random component \( u_i \). An estimate of the \( i \)th value of the dependant variable on the basis of the multiple regression is

\[
Y_i' = b_0 + b_1 X_{i1} + b_2 X_{i2} + \ldots + b_p X_{ip}
\]

Therefore \( Y_i = Y_i' + e_i \),

and the \( i \)th residual provides a measure of the departure of the \( i \)th observation on the dependent variable from its value as predicted from the regression. The sum of squares of these residuals gives some measure of the goodness of fit of the general linear model to the data. According to the principle of least squares the sample estimates \( b_0, b_1, \ldots, b_p \), of the population regression coefficients are chosen so as to minimise the sum of squares of the residuals, thus

\[
\Sigma e_i^2 = \Sigma (Y_i - b_0 - b_1 X_{i1} - b_2 X_{i2} - \ldots - b_p X_{ip})^2 \tag{4.10}
\]

It can be shown that Equation (4.10) is minimal when \( b_0, b_1, \ldots, b_p \) satisfy the following system of \( n \) linear algebraic equations.
Equations (4.11), known as the normal equations, may be represented simply in matrix notation as

\[
[P][b] = [g] \tag{4.12}
\]

where \([P]\) is the coefficient matrix of order \((p + 1) \times (p + 1)\), \([b]\) is the column vector of the sample regression coefficients and \([g]\) the column vector of the right hand side.

Applying Equation (4.12) to the wear equation given in Equation (4.8) yields,

\[
[P] = \begin{bmatrix}
      n & \Sigma \log_{10}(p) & \Sigma \log_{10}(v) & \Sigma \log_{10}(\theta)
      \\
      \Sigma \log_{10}(p) & \Sigma \log_{10}(p)^2 & \Sigma \log_{10}(p^1) & \Sigma \log_{10}(p^1 \log_{10}(\theta_1))
      \\
      \Sigma \log_{10}(v) & \Sigma \log_{10}(v)^2 & \Sigma \log_{10}(v) & \Sigma \log_{10}(v) \log_{10}(\theta_1)
      \\
      \Sigma \log_{10}(\theta) & \Sigma \log_{10}(p^1 \log_{10}(\theta_1)) & \Sigma \log_{10}(v^1 \log_{10}(\theta_1)) & \Sigma \log_{10}(\theta)^2
    \end{bmatrix}
\]

\[
[b] = \begin{bmatrix}
    n \\
    \alpha \\
    \beta \\
    \gamma
  \end{bmatrix}
\]

\[
[g] = \begin{bmatrix}
    \Sigma \log_{10}(w_1) \\
    \Sigma \log_{10}(p^1) \log_{10}(w_1)
    \\
    \Sigma \log_{10}(v^1) \log_{10}(w_1)
    \\
    \Sigma \log_{10}(\theta^1) \log_{10}(w_1)
  \end{bmatrix}
\]

The solution of Equation (4.12) is given by

\[
[b] = [P^{-1}][g] \tag{4.13}
\]
where \( [P^{-1}] = \text{adj} [P] | P | \)

Equation (4.13) is solved using the Gauss-Jordan method of elimination.

### 4.2.4 Results

Applying the above analysis to the experimental data yields the following mathematical models. For material X

\[
w = 1.4684 \times 10^{-5} p^{-0.028} v^{1.881} \theta^{0.417}
\]  

(4.14)

and for material Y

\[
w = 1.3183 \times 10^{-7} p^{-0.017} v^{5.246} \theta^{0.732}
\]  

(4.15)

A comparison of these mathematical models with measured data is made in Figures 4.1 to 4.6 as follows:

- **Figure 4.1** - Material X Wear vs Pressure
- **Figure 4.2** - Material X Wear vs Rubbing Speed
- **Figure 4.3** - Material X Wear vs Drum Temperature
- **Figure 4.4** - Material Y Wear vs Pressure
- **Figure 4.5** - Material Y Wear vs Rubbing Speed
- **Figure 4.6** - Material Y Wear vs Drum Temperature

### 4.3 DISCUSSION OF RESULTS

#### 4.3.1

Plotted in each of Figures 4.1 to 4.6 are two sets of experimental data and the calculated results. Apparent from the figures is the variation in wear between the two experimental tests typical of results from abbreviated wear testing. Because of the non-homogeneous nature of friction materials (as discussed in Chapter 3) the conditions at the friction interface are constantly varying changing the friction and
wear characteristics of the friction material. Therefore over the relatively short period of a dynamometer wear test (each point in Figures 4.1 to 4.6 represents 500 brake applications) the average conditions at the interface can differ from test to test resulting in different wear rates. Despite these variations the theoretical results show good agreement with the experimental values for both Material X and Material Y, the predicted values usually falling between the two experimental results throughout the test series.

4.3.2 The wear equations (4.14) and (4.15) for the two materials are both of the same form and examination of this form provides an insight to the materials' wear behaviour. The average interface pressure, $p$, is raised to the power $-0.028$ for the asbestos free material (Material X) and $-0.017$ for the asbestos based one (Material Y) indicating that for conditions of the same initial drum temperature and rubbing speed, friction material wear is little effected by actuation pressure. The negative indices indicate that wear decreases with pressure suggesting that wear may be a function of time, the higher pressures resulting in a reduced stop time and hence slightly reduced wear. From the equations it can be seen that both materials exhibit increased wear with increased temperature, with indices of 1.417 and 1.732 respectively for Material X and Material Y, indicating that the wear of the asbestos based material is more sensitive to temperature. The wear of Material X is, as could be expected, proportional to the square of the velocity term, demonstrating that wear of this material is proportional to the energy dissipated. This is not so with the asbestos based material where wear is proportional to rubbing speed raised to the power 5.246, suggesting a more complicated relationship than that with energy dissipated, possibly arising from the higher drum temperatures attained during brake applications from higher vehicle velocities.

4.3.3 From Figures 4.1 to 4.6 it is apparent that the asbestos free lining material offers significant benefits over the asbestos based product in terms of material wear, both in total material wear over the test series and also in wear stability over the range of test conditions.
This result is typical of the difference between established asbestos based lining materials and the newer asbestos free formulations developed for use in the higher temperature regimes resulting from recent increases in vehicle speeds and masses. However the wear of the friction material is significant not only in terms of cost to the vehicle operator but also in effecting the performance of the brake. The performance of a brake is determined primarily by the coefficient of friction of the lining material and the pressure distribution between the lining surface and the brake drum; where the contact pressure is determined by brake distortions, friction material compressibility and friction material wear. During the operation of a drum brake wear occurs such that the friction material is conformed to the drum shape which may be distorted due to thermal and mechanical effects. Therefore when the brake is applied the contact pressure distribution and hence the brake output is determined by the recent usage history of the brake. Because of the reduced wear rates of asbestos free materials, variations in brake performance may be increased and prolonged due to the longer period necessary for the linings to conform to the brake drum. The action of a non-uniform contact pressure distribution may be to introduce further instabilities into the braking process such as hot spots and bands. These phenomena, although present with asbestos based linings are likely to be exacerbated by asbestos free lining materials due to their resistance to high temperature wear, possibly causing drum damage.

4.3.4 The equations (4.14) and (4.15) represent mathematical models of friction material wear of two materials X and Y, when installed on a specific brake. Although providing an insight into the wear characteristics of the friction material, these equations are not suitable for inclusion in a finite element brake simulation where a time step method of analysis is adopted. Wear of the friction material during each time step can be expressed as

\[
\frac{\delta W}{t} = C_1 P^{\alpha_2} v^{\delta_2} \theta^{\beta_2} \tag{4.16}
\]
where \( C_2, \alpha_2, \beta_2 \) and \( \gamma_2 \) can be obtained by the method of least squares if the friction material wear is assumed to be directly proportional to time during each brake application. Thus for Material X

\[
\frac{\delta w}{t} = 4.1881 \times 10^{-12} \ p^{-0.002} \ v^{1.602} \ t^{0.716} \quad (4.17)
\]

and for Material Y

\[
\frac{\delta w}{t} = 2.6476 \times 10^{-11} \ p^{-0.016} \ v^{2.024} \ t^{0.678} \quad (4.18)
\]

4.4 SUMMARY

4.4.1 The wear characteristics of two friction materials, one considered a typical asbestos based formulation and the other a typical asbestos free product, have been described analytically by exponential functions of the third degree. Two functions have been determined for each material to give the friction material wear both in terms of time and in terms of brake applications. The coefficients of these functions have been determined experimentally from a schedule of tests undertaken on a hydraulically operated two leading shoe brake where good agreement is seen between measured and predicted results. The mathematical models developed provide not only a means of predicting friction material wear under a range of operating conditions, but also an insight into how the different operating parameters effect this wear.

4.4.2 The results obtained from the wear tests are consistent with those obtained in service with the asbestos free friction material exhibiting approximately half the wear of the asbestos based material under the same operating conditions. In addition the wear of asbestos free Material X is less effected by increases in drum temperature and rubbing speed.

4.4.3 The different wear characteristics of the two materials will give rise to different brake performance characteristics in service. These vari-
ations in brake performance can be investigated using a finite element approach to brake analysis. Although the friction material wear occurring in an individual brake application is small the cumulative effect of wear on the contact pressure distribution can be studied. Similarly the effects of uneven wear over the lining surface can be determined.
FIGURE 4.1
MATERIAL X: WEAR vs AVERAGE INTERFACE PRESSURE

WEAR (500 applications) mm

- Experimental
- Experimental
- Calculated

Rubbing Speed = 6.672 m/s
Drum Temperature = 100 deg C
FIGURE 4.2
MATERIAL X: WEAR vs RUBBING SPEED

WEAR (500 applications) mm

Pressure = 130 KN/m^2
Drum Temperature = 100 deg C
FIGURE 4.3
MATERIAL X: WEAR vs INITIAL DRUM TEMPERATURE

WEAR (500 applications) mm

Pressure = 130 KN/m^2
Rubbing Speed = 6.672 m/s
FIGURE 4.4
MATERIAL Y: WEAR vs AVERAGE INTERFACE PRESSURE

WEAR (500 applications) mm

PRESSURE KN/m^2

Rubbing Speed = 6.672 m/s
Drum Temperature = 100 deg C
FIGURE 4.5
MATERIAL Y: WEAR vs RUBBING SPEED

WEAR (500 applications) mm

RUBBING SPEED m/s

Experimental
Experimental
Calculated

Pressure = 130 KN/m^2
Drum Temperature = 100 deg C
FIGURE 4.6
MATERIAL Y: WEAR vs INITIAL DRUM TEMPERATURE

WEAR (500 applications) mm

Pressure = 130 KN/m²
Rubbing Speed = 6.672 m/s
5.0 **FINITE ELEMENT ANALYSIS**

5.1 **THE SIMULATION OF FRICTION INTERFACES BETWEEN ELASTIC BODIES BY FINITE ELEMENT ANALYSIS**

5.1.1 **Introduction**

The Gap Force method is a technique which allows the effects of a sliding friction interface to be studied using the finite element method. Friction interfaces are defined by the position of the interface node pairs which must be positioned close enough together to avoid any significant loss of stiffness across the interface. Under the application of an external load the normal stress or reaction at each interface node is determined to obtain a value of the frictional drag force to be applied to the node in the next solution, or iteration. A stable solution is achieved when, at each interface node, the difference between the frictional drag forces applied in two successive iterations may be regarded as "small" (eg, less than 5%). At this point the friction drag load and the resulting normal reaction at each interface node in contact are consistent and related by Amonton's Law.

A friction interface cannot support a normal tensile direct stress at any point. If this arises the constraint governing the displacement of such a node must be released in the next iteration and the associated frictional drag set to zero. The node is consequently allowed to displace freely in the direction normal to the interface causing local loss of contact. However, at a later iteration the loading and contact situation may give a solution in which such an out of contact node has displaced through the interface boundary. In this case contact is re-established in the next iteration by re-applying the constraint governing the displacement of the node normal to the surface.
5.1.2 Finite Element Analysis of the Friction Interface

Figure 5.1 (a) represents a section of the friction interface between friction material and brake rotor. Nodes A, B and C describe the surface of the friction material and the corresponding nodes A', B' and C' represent the mating surface of the rotor. The separation of the bodies which exists before they are loaded is given by gap values ΔA, ΔB and ΔC respectively. Upon application of the brake displacement occurs such that contact in the region of nodes A and C occurs but not in the region of B as shown by Figure 5.1 (b).

Since nodes B and B' are not in contact no load is transmitted between the bodies in this region in either the normal or tangential directions to the interface. At the interface between the bodies in the region of A and A' load is transmitted between the bodies in the form of discreet nodal forces of $F_A^n$ and $-F_A^n$ respectively in the direction normal to the interface. In addition, the nodes exert frictional shear loads of $F_A^T$ and $-F_A^T$ respectively upon each other. Similarly nodes C and C' exert equal and opposite forces of $F_C^n$ and $F_C^T$ upon each other in the normal and tangential directions to the common interface respectively.

The relative displacement of nodes B and B' is less than ΔB in the normal direction while the relative displacement in the tangential direction does not matter. Node pair B and B' are said to have underclosed since their relative displacement is less than required to close the initial gap which separates them. The normal relative displacement of node pair A and A' is equal to ΔA since the gap separating the mating surfaces has fully closed at this position.

5.1.3 Sliding Friction

If two contacting bodies are in relative tangential motion, ie, sliding across each other, then the magnitude of the force $F_A^T$ is given by the equation

$$|F_A^T| = \mu |F_A^n|$$  \hspace{1cm} (5.1)
where \( \mu \) is the coefficient of sliding friction in the region of node pair A and A'. The direction of \( F_A^r \) on each node is dictated by the sense of relative motion. The convention used is that a positive coefficient of friction gives a positive drag load on node A and a negative drag load on node A'. This would simulate the brake rotor sliding to the right relative to the friction material.

### 5.1.4 Static Friction

If there is no relative motion between the two bodies then the friction force developed between a pair of interface nodes may take any value from zero up to the limiting value \( |\mu F_A^n| \). The magnitude and direction of the friction force is therefore just sufficient to prevent relative tangential displacement of the pair of nodes. If the limiting value of static frictional force is exceeded then the nodes will displace or slip relative to each other in the tangential direction. The direction of this slip must be opposite to the direction of the frictional force. Referring to Figure 5.1, if A and A' are developing a limiting friction force between them \( (F_A^r = \mu F_A^n) \) in the direction shown, then the brake rotor will displace to the right relative to the friction material; the friction force will oppose any such relative motion.

### 5.1.5 Determination of Transmitted Loads

The applied nodal forces and resulting displacements are connected by a matrix equation of the form

\[
[\delta] = [C] \{F\} \tag{5.2}
\]

where \( [\delta] \) is the vector of displacements resulting from the vector of applied loads \( \{F\} \) and \( [C] \) is the flexibility matrix. If the applied forces are incremented by amounts given by the vector \( \{AF\} \) then the resulting displacement increment vector \( \{A\delta\} \) is given by the relationship

\[
\{A\delta\} = [C] \{AF\} \tag{5.3}
\]
In particular, if only the frictional interface nodal forces are incremented by equal and opposite amounts on corresponding nodes then

\[
\{g\} = [A] \{\Delta T\} \tag{5.4}
\]

where \(\{g\}\) is the vector of increments in the relative displacements of interface nodes, \(\{\Delta T\}\) is the vector of increments in forces applied to the interface nodes and \([A]\) is a constant matrix. For each direction of transmitted force on each pair of interface nodes the gap opening (or tangential slip) increment may be calculated for any increment in applied forces, provided \([A]\) is known.

If the matrix form expressed by equation (5.4) is expanded to form

\[
\begin{align*}
\{g\}_1 &= \lambda_{11} \Delta T_1 + \lambda_{12} \Delta T_2 + \ldots + \lambda_{1n} \Delta T_n \\
\{g\}_2 &= \lambda_{21} \Delta T_1 + \lambda_{22} \Delta T_2 + \ldots + \lambda_{2n} \Delta T_n \\
&\quad \vdots \\
\{g\}_n &= \lambda_{n1} \Delta T_1 + \lambda_{n2} \Delta T_2 + \ldots + \lambda_{nn} \Delta T_n
\end{align*}
\tag{5.5}
\]

where the suffixes 1, 2 \ldots n label pairs of corresponding interface nodal degrees of freedom, then the elements of \([A]\) may be determined by using the displacements resulting from the solution of the system created load cases as shown below. Let system load case 1 apply \(\Delta T\), value of unity, with all other \(\Delta T\) values of zero. From equation (5.5) this gives

\[
\begin{align*}
\lambda_{11} &= g_1 \text{ (for load case 1)} \\
\lambda_{21} &= g_2 \text{ (for load case 2)} \\
&\quad \vdots \\
\lambda_{n1} &= g_n \text{ (for load case n)}
\end{align*}
\]

For load case 2, \(\Delta T_2\) is unity and all others are zero.
\(A_{12} = g_1 \) (for load case 2)
\(A_{22} = g_2 \) (for load case 2)

etc

Generally the elements of \([A]\) are determined from the gap opening displacements of the system load cases by applying the equation

\[A_{ij} = g_i \text{ for system load case } j \quad (5.6)\]

Sliding friction pairs of freedoms are not included in equations (5.4), (5.5) and (5.6) because the analysis does not require them. Inverting matrix equation (5.4) yields

\[\{\Delta T\} = [B] \{g\} \quad (5.7)\]

Where \([B]\) is the inverse of \([A]\). Thus, given any set of desired gap opening and relative tangential slip displacement increments, \([g]\), the required increment in equal and opposite loads to be applied to the pairs of frictional interface nodes is given by equation (5.7). These implicitly applied forces provide the necessary coupling between pairs of interface nodes and the iterative procedure adjusts the applied forces, by means of equation (5.7) until the convergence criteria are satisfied.

5.1.6 PAFEC GAPS Module

The Gapfric procedure described above enables the user to analyse elastic structures which contain both static and sliding friction interfaces. The GAPS facility, contained in Level 6 of PAFEC (Ref 82) provides an analysis of structures possessing static frictional interfaces only. Furthermore the iterative procedure contained in the main solution subroutine for the GAPS facility does not appear to operate according to the PAFEC Theory and Results Manual (Ref 83). Consequently it is expected that solutions will be incorrect if the standard Level 6 programme is used to analyse structures containing static frictional gaps.
The main solution subroutine for the GAPS facility in Level 6 of PAFEC is called B15800. This subroutine contains an iterative loop which aims to ensure that, under a given set of boundary conditions and applied loads, all specified gaps in the structure are either open (underclosed) or closed, in which case they transmit a compressive force between the mating surfaces. If the user specifies that any parts of the mating surface are static friction gaps (TYPE = 2 in the GAPS module) then the coding is supposed to ensure that those parts of the surface which are in contact either do not slip in the tangential direction because the frictional shear forces they exert on each other do not exceed limiting friction, or, they slip in the tangential direction and exert on each other a frictional shear force exactly equal to limiting friction in the direction opposed to the slip.

The existing coding does not provide solutions which satisfy these conditions. When static friction is specified degrees of freedom are correctly untied when the transmitted friction force exceeds limiting friction. However, the coding does not apply to such pairs of untied nodes tangential friction forces of magnitude equal to limiting friction. Incorrect solutions will therefore result.

In order to utilise the structure of the PAFEC GAPS module, modification of the existing routines was necessary to install the procedures to analyse sliding frictional interfaces and to properly cater for static friction gaps.

5.2 THE FINITE ELEMENT MODEL OF THE FRICTION INTERFACE

5.2.1 Comparison of Two and Three Dimensional Finite Element Methods

The use of two dimensional finite element gap elements to model the friction interface of a brake has been well documented. (Ref 4, 63, 70). This method of approach assumes that the pressure distribution between the linings and drum varies only in the circumferential direc-
tion ignoring variations across the width of the rubbing path. Both these variations can be investigated using a three dimensional model. This approach can incorporate an accurate representation of the brake to include such features as drum stiffness and its mounting face. By including in the brake shoe representations the web and platform the need for shoe stiffness approximations to a curved beam of uniform cross-section necessary in a two dimensional analysis is eliminated.

Shown in Figure 5.2 is the two dimensional model of the leading shoe of a typical "S" cam brake. Over most of its arc length, the shoe forms a curved T-section beam. The second moment of area $I$ can be calculated from a rectangular section beam of the same width; and of depth $d'$ to give an equivalent rigidity thus:

$$d' = \left(\frac{12I}{B}\right)^{\frac{1}{n}}$$

$$I = \frac{b}{4} (BA_1^3 - bh^2 + aA_2^3)$$

and

$$A_1 = \frac{(ad^2 + bc^2)}{2 (ad + bc)}$$

$$A_2 = d - A_1$$
The model consists of 26 plain strain, eight-noded quadrilateral elements and 117 nodes. The outer layer of elements simulates the friction lining and the inner layer the cast iron shoe with equivalent sectional rigidity as described above. This model can be extended to form a simple three-dimensional model as shown in Figure 5.3. In this model the two-dimensional elements are replaced with eight-noded three-dimensional elements of corresponding thickness. In this manner the stiffness assumptions inherent in the two-dimensional approach are repeated in the three-dimensional analysis enabling a direct comparison of the two methods to be made.

The shoe factors predicted by the two models for a range of friction coefficient are shown in Figure 5.4. The shoe factors, that is shoe drag force divided by shoe tip load, are seen to be similar for both analyses, with those predicted from the two-dimensional model being 5% higher than those from the three-dimensional one. The reason for this discrepancy is clear from the pressure distributions predicted around the lining arc (Figure 5.5). The pressure variations around the shoe is seen to be larger for the two-dimensional analysis indicative that the two-dimensional model is more rigid than the equivalent three-dimensional model resulting in poorer liner to drum conformability.

Therefore there exists a discrepancy in the finite element analysis between two and three-dimensional approaches to the same problem. Results from two and fully three-dimensional analyses are discussed in Chapter 8 and compared with those obtained experimentally.

5.2.2 The Three Dimensional Finite Element Model

The finite element model is based upon an S cam-operated pivoted abutment type of leading and trailing shoe brake assembly of diameter 420 mm and width 175 mm as illustrated in Figure 5.6. To facilitate computing the brake was modelled in two halves as this simplification enables leading and trailing shoe effects to be studied individually. The brake shoes, linings and drum were modified by three-dimensional
finite element meshes (Figures 5.7 a, b and c). This enables investigations to be made of combined axial and circumferential expansions which gave rise to effects such as "bell mouthing", "barrelling" and temperature "banding" across the width of the drum. Each brake shoe was constrained to allow rotation only at the pivots and the surface of contact between lining and drum is defined by nodal pairs. Actuation forces are applied at the cam end of the shoes which are of cast construction each with a single web of inverted ‘T’ section. At the cam end of each shoe a double web provides a mounting for the return spring. The platform is supported by fillets from the web and the platform itself was conveniently represented by a single layer of elements. A second layer of elements then represented the linings in the form of two tapered half blocks, each of 50° arc length. The interface was at a radius of 0.299 m which was the same for both the linings and the drum.

The brake drum was represented by a single layer of elements, with an additional band of elements forming the stiffener at the drum mouth. Provision was also made to incorporate the mounting flange into the model, although the stud holes were ignored. The drum mounting flange was restrained to simulate the clamping action of the wheel and wheel hub.

5.3 THERMAL CONSIDERATIONS

5.3.1 Frictional Work

The braking torque generated in a friction brake is calculated from:

\[ T = \int_{x_{1}}^{x_{2}} \mu r(x)p(x)A(x) \, dx \]  

(5.8)

where a pressure distribution \( p(x) \) exists over a friction surface of area \( A(x) \). Assuming that 100% of the work done by the brake is
converted to heat energy generated at the nodes at the friction interface, the heat flux to each node can be calculated from the instantaneous power dissipation

\[ \Delta Q = \Delta T \omega \]  

(5.9)

by using the relevant value of \( \Delta T \) applied as a ramp change over a time-step \( \Delta t \) from \( \omega_1 \) to \( \omega_2 \). The heat flux is applied and the transient temperature distributions calculated using the standard PAFEC package.

### 5.3.2 The Finite Element Model

The finite element mesh used for the thermal calculations required very much smaller elements than those used for the mechanical analysis because of the very steep temperature gradients that exist below the rubbing surface of the friction material. In order to minimise the complexity of the model, heat conduction through the shoes to the friction material was considered negligible during the time interval considered thus allowing the brake shoes to be ignored. Shown in Figure 5.8a is the finite element mesh used for the thermal analysis of the friction material. Clearly seen in the inset figure is the very small element thickness below the rubbing surface of the friction material to accommodate the steep temperature gradients in this region.

The friction material was modelled by six circumferential layers of elements and the cast iron brake drum by three annular rings of elements (figure 5.8b). This resulted in a combined thermal analysis mesh consisting of 930 three-dimensional Laplacian elements comprising a total of 3056 nodes (figure 5.8c).

During braking frictional heat generation can be assumed to occur in the surface layers of the lining material (Ref 70). In a drum brake lining temperatures vary according to local conditions and friction, while for each complete revolution, brake drum temperatures will approximate to some time-average constant value around the circumference. This time-averaged value can be determined experimentally by the use of a rubbing thermocouple. By comparing the measured drum tempera-
ture, one, two and three seconds into the brake application with that predicted from the model, an accurate value for the heat partitioning between the drum and lining can be determined. This eliminates assumptions concerning partitioning or interface contact resistance, therefore accurate predictions of lining surface temperature can be made.

Heat flows other than conduction were neglected in the analysis as they have only a small influence on the temperatures reached in a single brake application.

5.4 THE SIMULATION TECHNIQUE - "MULTI"

5.4.1 The thermal and mechanical analysis (Sections 5.4 and 5.3 respectively) are combined in one brake analysis system known as "Multi" (Refs 84, 85, 86). This is a fully automatic system, a flow chart of which is shown in figure 5.9, which enables a full brake application to be simulated without the need to manually process data between each stage of the analysis. The "Multi" package of routines has been developed for use with the PAFEC finite element system.

5.4.2 In order to simulate a brake application a data file is created. This file specifies the initial geometry of the brake, including the contact condition at the interface between the brake drum and friction material, and the time step to be used in the analysis. Also contained in the file are tables allowing for the variation of friction material properties with temperature. The initial conditions of the brake application; angular velocity, inertia and actuation forces, are specified by the operator.

The pressure distribution at the rubbing surface is calculated using the "Gapfric" analysis outlined in Section 5.2. The analysis of the friction interface predicts the mechanical distortion within the brake as well as determining the work done from the calculated drag forces.
The output from the mechanical analysis is post processed to provide the input to the thermal analysis (Section 5.4). The nodal drag forces are converted to a nodal heat flux and the basic temperature distribution is calculated using the standard PAFEC routines. Once the temperature distribution within the brake is known the thermal distributions can be determined. Friction material wear is calculated (Chapter 4) from the formula

\[
\frac{\delta w}{t} = 4.1881 \times 10^{-12} p^{-0.002} v^{1.602} 0^{0.716} \tag{5.10}
\]

where the nodal pressure is determined from the mechanical analysis and the nodal temperature from the thermal analysis. The mechanical and thermal distortions and wear are combined and their net effect used to re-define the model geometry at the end of the time step. At the end of each time step the local friction material properties are determined from the nodal temperatures. For the new time step each friction material node is assigned properties according to its temperature by means of tables determined experimentally as detailed in Chapter 3.

5.4.3 For each time step the interface contact conditions are assumed constant. Therefore the friction material properties and the heat flux applied to each node are fixed over the time step, whereas in practice they are constantly changing. As the time step is reduced so this effect will be reduced. However, the time step, friction material thermal conductivity and the minimum PAFEC element dimensions are related by the relation

\[
F = \frac{k \delta t}{c_p \rho \varepsilon^2} \tag{5.11}
\]

where

- \( \delta t \) = time step
- \( k \) = thermal conductivity
- \( c_p \) = specific heat
- \( \rho \) = density
- \( \varepsilon \) = distance between closest nodes measured in the direction of heat flow where thermal shock is severe.
In transient temperature calculations it is essential for the non-dimensional Fourier number, \( F \), to have a value of approximately unity for all nodes where there is a severe thermal shock. If the Fourier number has a value greater than one, i.e., the time step is too large, then oscillations may arise in the calculated temperatures; if the time step is too small, i.e., the Fourier number is less than one, the application of a thermal shock may result in the temperature changing in the wrong direction.

5.4.4 The "Multi" brake analysis system is installed on a UNIXSYS NX32 computer. The simulation of a five second brake application for either a leading or trailing shoe takes approximately 24 hours, or nearly five hours per time step. Although the finite element mesh used for the mechanical analysis is not especially large by present standards the "Gapfric" analysis incorporates an iterative procedure, which together with the more complicated finite element mesh required for the thermal analysis accounts for the amount of time required. Within each time step the thermal analysis accounts for approximately two-thirds of this time.
FIGURE 5.1
REPRESENTATION OF THE FRICTION INTERFACE

During Braking

Friction Material

Friction Material

Non Braking Condition

Brake Rotor

Brake Rotor

\[ \begin{align*}
F_A^N & : A \\
F_A^T & \rightarrow B \\
\end{align*} \]

\[ \begin{align*}
F_c^N & : C \\
F_c^T & \rightarrow B \\
\end{align*} \]
FIGURE 5.3
SIMPLE THREE DIMENSIONAL
FINITE ELEMENT MODEL
FIGURE 5.4
LEADING SHOE ANALYSIS 2D AND 3D

Shoe Factor

Friction Coefficient

- 2D Analysis
- 3D Analysis
FIGURE 5.5
PRESSURE DISTRIBUTION 2D AND 3D

Pressure Normal Force /N

Position Around Lining Arc /Degrees

- 2D Analysis  
- 3D Analysis
FIGURE 5.6
S CAM BRAKE
FIGURE 5.7a

3D FINITE ELEMENT MESH OF THE BRAKE SHOE AND LINING
FIGURE 5.7b
3D FINITE ELEMENT MESH OF THE BRAKE DRUM
FIGURE 5.7c
3D FINITE ELEMENT MESH
OF THE BRAKE ASSEMBLY
FIGURE 5.8a
F.E. MODEL - THERMAL ANALYSIS
FRICCTION MATERIAL
FIGURE 5.8b
F.E. MODEL - THERMAL ANALYSIS
BRAKE DRUM
FIGURE 5.8c
F.E. MODEL - THERMAL ANALYSIS
BRAKE ASSEMBLY
FIGURE 5.9
THE BRAKE ANALYSIS PACKAGE "MULTI"

- REDEFINE MODEL
- MECHANICAL DISTORTION
- WEAR
- TEMPERATURE DEPENDENT VARIABLES
- MECHANICAL ANALYSIS INCORPORATING "GAPFRIC" TO DETERMINE FRICTION FORCES
- THERMAL ANALYSIS TO DETERMINE HEAT FLUX AND SUBSEQUENT TEMPERATURE DISTRIBUTION
- THERMAL DISTORTION
6. **PREDICTED TEMPERATURE AND PRESSURE DISTRIBUTIONS IN A S CAM BRAKE ASSEMBLY**

6.1 **INTRODUCTION**

Many factors influence the performance of a drum brake. Some exist before the brake is applied. For example, all brakes include tolerances; these may be irregularities in the mating surfaces from machining effects in the drum surface or inclusions in the friction material. During assembly eccentricity and local friction variations between drum and lining may occur. These can be controlled to some extent at the manufacturing stage but they still contribute to those changes that occur during a brake application.

When the brake is applied a pressure variation occurs over the rubbing path which is dependent on the deformation of the brake components from the actuation force. Both bulk and localised distortions change the nature of the contact between drum and linings. At the contacting areas over-heating is possible resulting in high temperatures, thermal expansion and wear. These effects are inter-dependent and give rise to local fluctuations of pressure and temperature at the friction interface. The temperature and pressure effects that occur at the friction interface between the sliding members during braking can be investigated using the finite element method.

6.2 **NORMAL DUTY BRAKE OPERATION**

6.2.1 **Brake Conditions**

Finite element predictions of the temperature and pressure distributions have been made for the brake outlined in the previous chapter. Normal duty operating conditions assume a wheel load of 5,000 kg, an initial speed of 60 km/h, and a vehicle deceleration of 0.2g. At the commencement of each brake application a drum bulk temperature of 100°C is assumed, while the lining temperature varies from 100°C at the rubbing surface to 20°C at the brake shoe platform. The temperature of
the brake shoe is assumed to be a constant 20°C throughout the brake application. At the commencement of braking perfect liner to drum contact is assumed. This is the condition in which all nodes across the friction material surface simultaneously make contact with the brake drum as the brake is applied and is taken as the condition which most closely represents well bedded linings.

6.2.2 Predicted Pressure Distributions

Figures 6.1 to 6.4 show the dynamic pressure distributions on the leading and trailing shoes at one and five seconds into the brake application. These results show that on the trailing shoe higher pressures occur at the cam end of the shoe. For the first second of the brake application a maximum pressure of 2.98 MN/m² is predicted at the cam end of the shoe. After five seconds of braking, during which time the shoe tip load was constant, the pressure increased to 3.53 MN/m². Throughout the stop the pressure over the rest of the trailing shoe is relatively uniform, increasing slightly as the stop progressed, but remaining below 1.2 MN/m². Some axial variation in pressure occurs at the cam end of the shoe and indicates that shoe distortion is occurring at the areas of high contact pressure (the pressure variation is greater later in the stop when higher pressures were attained). Higher pressures are predicted in the centre of the rubbing path in the area where the shoe platform is well supported by the shoe, decreasing towards the edge of the lining where the platform is less well supported.

Less circumferential variation of pressure at a fixed axial distance across the drum rubbing surface, is calculated on the leading shoe than on the trailing shoe. However more pronounced variations in pressure occur in the axial direction in the early stages of braking. The maximum contact pressure during the first second of the brake application is predicted to be 1.65 MN/m² towards the drum flange at the pivot end of the lining, decreasing to 0.59 MN/m² at the drum mouth. At the cam end of the shoe the maximum contact pressure is 1.42 MN/m². Further
into the stop the axial variation is reduced although circumferential variations in contact pressure remain with maximum pressures of 1.44 MN/m² and 1.0 MN/m² at the cam and pivot ends of the shoe respectively. Overall the pressure distribution on the leading shoe is little changed throughout the stop maintaining a slightly cosinusoidal or "U" shaped profile. The axial variation in contact pressure evident early in the application is due to distortion of the brake drum (barrelling). This is reduced as a result of a reduction in leading shoe factor as the coefficient of friction of the friction material decreases owing to the high temperatures reached (ie fade occurs).

6.2.3 Predicted Temperature Distributions

Plotted in Figures 6.5 to 6.8 are the interface temperature distributions over both shoes at two intervals during the brake application. These results show a rapid build up of temperature at the surface of the trailing shoe immediately as the brake is applied and also that temperatures are higher at the cam end of the shoe than those attained over the rest of the liner. The temperature over the bulk of the two lining blocks is fairly uniform and at around 250°C is approximately one quarter that of the maximum value reached at the cam end. After more prolonged braking there is a slight rise in temperature over the entire lining with the high initial temperature being maintained at the cam end.

The temperature distribution on the leading shoe exhibits variation both across the rubbing path and circumferentially around the lining. After one second the temperature profile is "U" shaped or cosinusoidal with higher temperatures at both ends of the linings and towards the outer edge (towards the drum mounting flange) of the lining. These results are consistent with predictions of pressure distribution discussed previously in that the local temperatures are affected by local pressures. High pressures give rise to high temperatures and the changes in temperature are similar to the variation in pressure distribution observed. Because the drum is stiff at the mounting
flange, less distortion occurs in this region resulting in a greater pressure on the lining accounting for the observed effects. After a further four seconds of braking the variation in temperature across the rubbing path increases with temperatures at the inner and outer edges of the lining varying from 200°C to 600°C respectively.

The thermal analysis also readily enables brake drum temperatures to be determined and a series of results are presented in Figures 6.9 to 6.12. Figures 6.9 and 6.10 show the temperature profile early into the brake application due to the contributions from each of the leading and trailing shoes respectively as the brake is modelled in two halves. As expected the temperature profiles are similar because in the fully bedded condition both shoes of a S cam brake do the same amount of work and generate the same amounts of heat. The temperature across the brake drum is found to be reasonably uniform with temperatures at the centre of the rubbing path of 136°C at one second into the stop, rising to 152°C after five seconds (Figures 6.11 and 6.12). Again temperatures are highest towards the centre of the drum rubbing path due to the heat sink effect of the drum mounting flange and the reduction in contact pressure towards the inner (free) edge of the rubbing path.

6.2.4 Work Done Over the Friction Surfaces

The work done at each node at the surface of the friction material is a product of the normal force and friction coefficient. The normal force is determined from the pressure distribution and the friction coefficient can be determined from the temperature distribution if the frictional behaviour of the friction material with temperature is known (Table 3.1). Utilising this data enables the drag force per unit area over the two brake shoes to be determined at any time throughout the brake application and typical results are shown in Figures 6.13 to 6.16. These figures indicate that during the early stages of the application, when a uniform coefficient of friction is assumed for the friction material surface, the profile of the drag force per unit area is the same as that of the pressure distribution. It is of interest to
note that at one second into the brake application the maximum drag force per unit area on the trailing shoe is 954 KN/m² and 527 KN/m² on the leading shoe. Further into the application the profiles are completely changed, the drag force per unit area becoming nearly uniform over both shoes in the region of 300 KN/m².

Investigation of the work done per unit area over the linings shows that when the brakes are applied the initial profile of work done per unit area over the liners, is, as would be expected, the same as that for the pressure distribution. However, the energy input to each area of lining is dependent on the work done by that piece of lining, and hence on the pressure distribution and friction level of the material. If the friction level across the lining is uniform at the commencement of a brake application, then the heat input to the lining will vary according to the pressure distribution resulting in a non-uniform heat input to the lining and hence, because of the low thermal conductivity of friction materials, a non-uniform temperature profile across the friction material. If the temperature varies across the friction material, then the local friction level will vary according to the μ temperature characteristics of the friction material. As most friction materials fade or display a fall in μ with temperature then there will be a tendency for the friction level of the areas of the lining receiving large thermal inputs to fall. Under the conditions of a normal duty brake application the work done per unit area tends towards uniformity as the stop progresses. Thus a uniform drag force is developed over the surface of the linings despite a non-uniform pressure distribution existing during the brake application.

6.2.5 Comparison of Three and Two Dimensional Approaches

The interface pressure distributions predicted from a two dimensional analysis for the leading and trailing shoes are shown in Figure 6.17, and the lining friction surface temperatures are shown in Figure 6.18, both at a time of one second into the brake application. At the beginning of the stop a "U" shaped pressure distribution occurs over
the leading shoe and the corresponding surface temperatures range from 285°C over the central region of the lining rising to 1,000°C, or more at the ends. As braking progresses the lining pressures become more uniform, with less variation around the friction surface. Results on the trailing shoe, however, exhibited a non-uniform pressure distribution throughout the stop. This commenced with high pressures at both ends of the lining and low pressures at the crown region. This pressure pattern changed appreciably towards the completion of braking when the pressure becomes highly concentrated over the cam end of the lining and the contact pressure is reduced over the leading half of the lining; that is at the anchor end.

Using the same operating conditions as for the two dimensional analysis the three dimensional results (Figures 6.1 to 6.4) illustrate that a more uniform pressure is attained over the lining surface of the leading shoe, being only slightly "U" shaped in the circumferential direction. Some variation in pressure is apparent across the width of the lining with higher pressures predicted early in the stop at the drum mounting flange, especially near the pivot end of the lining. As the braking continues the axial variations in pressure decrease significantly. The behaviour in the pressure distribution on the trailing shoe predicted from the three dimensional analysis is different to that calculated using the two dimensional approach. The more complicated analysis gives a fairly uniform pressure over most of the lining surface rising to higher values at the cam end of the shoe. This higher pressure persists throughout the application.

An advantage of the three dimensional finite element approach to drum brake analysis is that it eliminates the shoe stiffness approximations necessary in the two dimensional analysis. This enables pressure and temperature effects both across and around the rubbing path to be determined, allowing investigation of non-uniform liner to drum contact patterns as a result of mechanical and thermal distortions of the brake. Having studied such effects during a normal brake application it
is of interest to make a comparison with those obtained under heavy braking.

6.3 INVESTIGATION OF THE EFFECT OF BRAKE DISTORTIONS ON BRAKE PERFORMANCE

6.3.1 Heavy Duty Brake Operation

For the simulated heavy duty brake application the same initial conditions are assumed as for the normal duty application, that is a wheel load of 5,000 kg and a vehicle speed of 60 km/h. During the heavy duty stop a vehicle deceleration of 0.5 g is assumed compared with a 0.2 g deceleration for the normal duty application. The same brake temperatures are assumed at the commencement of the application and perfect initial liner to drum contact is modelled. The normal duty application is typical of a vehicle exiting a motorway and slowing down for a round-about, the heavy duty application is representative of the vehicle operator having to stop quickly, in a controlled manner, for stationary traffic on a motorway.

6.3.2 Predicted Pressure Distributions

Figures 6.19 to 6.22 show the dynamic pressure distributions over the two shoes at one and five seconds into the heavy duty brake application. The shape of the pressure distributions for both load cases is similar, although the variation in pressure across the linings is lower for the stop at the lower deceleration. Maximum pressures of 6.95 MN/m² are predicted at the cam end of the trailing shoe lining one second in the brake application, rising to 7.87 MN/m² after five seconds. Early in the stop the pressure over the rest of the shoe is relatively uniform but later on these pressures increase and axial variations occur with higher pressures being obtained at the drum mounting flange, decreasing towards the drum mouth. Distortion of the cam end of the shoe is evident throughout the stop with higher unit loadings occurring in the more supported central region of the shoe. At the highly loaded
cam end of the trailing shoe compression of the friction material distributes the loads over a larger area of lining material.

The pressure distribution on the leading shoe is, as for the normal duty application, predicted to be more uniform than on the trailing shoe, being slightly "U" shaped or cosinusoidal in profile. Maximum pressures for the first second of braking are predicted to be 3.61 MN/m² at the pivot end of the shoe towards the drum mounting flange. Significant variation across the rubbing path is apparent early in the stop at the pivot end of the shoe, the pressure reducing 62% from the drum mounting flange to the drum mouth. Five seconds into the application this variation is significantly reduced.

Comparing the pressure distributions for the two duty cases demonstrates the similarity in the pressure profiles on the two shoes. Any differences between the two stops arise from increased distortion of the brake components under the increased actuation loads. Consequently at high actuation loads the brake shoe stiffness, brake drum stiffness and friction material compressibility may all influence the liner to drum contact pattern and the effect of these factors on brake performance is now considered.

6.3.3 Brake Distortion

Distortion of brake components has been identified as a potential contribution to brake torque variation experienced with drum brakes (Ref 57 and 58). The effect of friction material compressibility and brake drum and brake shoe stiffnesses on brake output has been investigated using the "Multi" brake analysis system. In this work the lower duty load conditions (0.2 g vehicle deceleration) and other assumptions described previously were used. For the purpose of this investigation the shoe factors for both leading and trailing shoes were calculated for the brake in its design condition (where shoe factor is defined as the sum of all the friction drag forces developed at the lining rubbing surface divided by the shoe actuation force). These values provided
the data against which alterations to the brake could be quantified. Against this background the Young's Modulus of the linings, brake drum and shoes, were each increased and decreased individually by a factor of two and the calculated results are contained in Table 6.1.

6.3.4 Friction Material Compressibility

Increasing the Young's Modulus of the friction material from $8.2 \times 10^8 \text{ N/m}^2$ to $16.4 \times 10^8 \text{ N/m}^2$ increases the leading shoe factor by 1.3% and the trailing shoe factor by 0.4%. Reducing the modulus to $4.1 \times 10^8 \text{ N/m}^2$ reduces the shoe factors by 0.8% and 0.6% respectively. As expected, reducing the stiffness of the lining material reduces variations in the interface pressure distribution. Under the assumed conditions of perfect initial liner to drum contact, the pressure distribution is cosinusoidal or "U" shaped, giving rise to higher pressures at the heel and toe of the lining.

### Table 6.1 - Predicted Shoe Factors

<table>
<thead>
<tr>
<th></th>
<th>TRAILING SHOE</th>
<th>LEADING SHOE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design Condition</td>
<td>0.467</td>
<td>1.250</td>
</tr>
<tr>
<td>Friction Material</td>
<td>0.461</td>
<td>1.240</td>
</tr>
<tr>
<td>$E = 4.1 \times 10^8 \text{ N/m}^2$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Friction Material</td>
<td>0.469</td>
<td>1.266</td>
</tr>
<tr>
<td>$E = 16.4 \times 10^8 \text{ N/m}^2$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Brake Drum</td>
<td>0.471</td>
<td>1.264</td>
</tr>
<tr>
<td>$E = 62.5 \times 10^9 \text{ N/m}^2$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Brake Drum</td>
<td>0.466</td>
<td>1.244</td>
</tr>
<tr>
<td>$E = 250 \times 10^9 \text{ N/m}^2$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Brake Shoe</td>
<td>0.468</td>
<td>1.280</td>
</tr>
<tr>
<td>$E = 62.5 \times 10^9 \text{ N/m}^2$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Brake Shoe</td>
<td>0.469</td>
<td>1.233</td>
</tr>
<tr>
<td>$E = 250 \times 10^9 \text{ N/m}^2$</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Reducing these high pressures to give more uniform values across the shoe will reduce the shoe factors and conversely increasing these high pressures will increase the shoe factors, but the overall effects are predicted to be small. Changing the compressibility of the friction material is likely to have a more significant influence on brake performance under conditions of non-uniform liner to drum contact by promoting improved conformability between drum and linings hence reducing thermal damage to the mating surfaces.

6.3.5 Brake Drum Stiffness

Reducing the Young's Modulus of the brake drum is predicted to increase the leading shoe factor by 1.1% and the trailing shoe factor by 0.9% while increasing the brake drum stiffness is predicted to reduce both shoe factors. The effect of reducing the brake drum stiffness is to increase drum distortion for a given set of actuation loads thus increasing the shoe factors; whereas increasing the drum stiffness reduces drum distortions therefore promoting a more even pressure distribution and hence reducing the shoe factors.

6.3.6 Brake Shoe Stiffness

Under the assumed conditions of perfect liner to drum contact the brake shoes are supported around their arc length by the friction material reacting the actuation loads on the brake drum. Under these conditions changing the stiffness of the trailing shoe has very little effect on its output. Distortion of the leading shoe is predicted to increase the shoe factor by 2.4% when the shoe stiffness is reduced by a factor of two. The shoe factor is reduced by 1% when the Young's Modulus is doubled. The brake shoe and brake drum stiffnesses, although predicted to have only a small effect on brake output under normal operating conditions, may have a significant influence on cyclical torque variations as a result of thermal and mechanical distortions of the brake.
6.3.7 Brake Torque Variations

The brake modelled in this investigation incorporates large cast shoes and a well stiffened brake drum. Therefore when altering the stiffness of an individual component this component is supported by the integrity of the rest of the brake. Consequently it is possible that a combination of changes would have had a more significant effect on brake output than that obtained from altering the stiffness of each component individually. However, it is found that mechanical changes to the brake components have only a small effect on brake output under normal operating conditions. Therefore changes in brake output can be concluded to be caused by local changes in the coefficient of friction between the friction pair. The $\mu$-temperature relationship of the friction material (Table 3.1) significantly affects the brake output due to localised heating of the friction surface. Investigation of work done per unit area over the linings has shown that when the brakes are initially applied, the profile of work done per unit area is the same as the pressure distribution. Areas of high pressure will be subjected to a large heat input as the frictional energy is converted to heat. This, due to the low thermal conductivity of the friction material, will result in a non-uniform temperature profile across the friction material. If the temperature varies across the lining surface, then the local friction level will vary according to the temperature characteristic of the material and this is the principal factor that causes changes in the torque output of the brake.
Fig. 6.1 Pressure Distribution Leading Shoe at $T = 1s$

Fig. 6.2 Pressure Distribution Trailing Shoe at $T = 1s$
Fig. 6.3 Pressure Distribution Leading Shoe at T = 5s

Fig. 6.4 Pressure Distribution Trailing Shoe at T = 5s
Fig. 6.5  Temperature Distribution Leading Shoe $T = 1s$

Fig. 6.6  Temperature Distribution Leading Shoe $T = 5s$
Fig. 6.7  Temperature Distribution Trailing shoe \( T = 1 \)s

Fig. 6.8  Temperature Distribution Trailing shoe \( T = 5 \)s
fig. 6.9 Temperature Profile Across Drum Rubbing Path
Leading Shoe Analysis  $T = 1s$

Fig. 6.10 Temperature Profile Across Drum Rubbing Path
Trailing Shoe Analysis  $T = 1s$
Fig. 6.11 Temperature Profile Across Drum Rubbing Path
Leading Shoe Analysis  $T = 5s$

Fig. 6.12 Temperature Profile Across Drum Rubbing Path
Trailing Shoe Analysis  $T = 5s$
Fig. 6.13 Drag Force per Unit Area Leading Shoe at $T = 1s$

Fig. 6.14 Drag Force per Unit Area Trailing Shoe at $T = 1s$
Fig. 6.15 Drag Force per Unit Area Leading Shoe at $T = 5s$

![Diagram of Drag Force per Unit Area Leading Shoe at $T = 5s$]

Fig. 6.16 Drag Force per Unit Area Trailing shoe at $T = 5s$

![Diagram of Drag Force per Unit Area Trailing shoe at $T = 5s$]
Fig. 6.17  2D Interface Pressure Distribution

MN/m$^2$

![Graph of 2D Interface Pressure Distribution](Image)

Cam

Leading shoe

Pivot

Pivot

Trailing shoe

Cam
Fig. 6.18 2D Interface Temperature Distribution
Fig. 6.19 Pressure Distribution Leading Shoe T = 1s

Drum Mouth

Fig. 6.20 Pressure Distribution Trailing shoe T = 1s

Drum Mounting Flange
Fig. 6.21 Pressure Distribution Leading shoe $T = 5s$

Drum Mouth

Cam

Pivot

$2.5 \cdot m^2$

Drum Mounting Flange

Fig. 6.22 Pressure Distribution Trailing shoe $T = 5s$

Drum Mouth

Cam

Pivot

$6.0 \cdot m^2$

$3.0 \cdot m^2$

Drum Mounting Flange
7. **AN INVESTIGATION OF NON UNIFORM LINING TO DRUM CONTACT**

7.1 **INTRODUCTION**

Work in the previous chapter has shown how both bulk and localised distortions of the brake can change the nature of contact between the brake drum and friction material. During braking energy is dissipated by friction heating generated by physical interaction of asperities at the sliding surface causing an increase in surface temperatures of the rubbing pair. At the contacting areas local temperatures can occur and their value will fluctuate depending on the size of the contact. Some contacts can be replaced by an envelope of small contact areas resulting in hot bands (strip braking) as shown in Figure 7.1. Under severe operating conditions this contact situation gives rise to a macroscopic kind of temperature variation across each heat band leading to a slow moving value that is of much higher temperature relative to the steady state value. Such heat bands and their rates of displacement across the rubbing surface are mostly affected by the pressure distribution and the thermal deformation of the contacting bodies during braking. These non-linear effects can be investigated using the "Multi" brake analysis system. It should be noted that although elements cannot be small enough to cater for contact areas on a microscopic scale meshes can be developed to investigate the temperatures associated with several heat bands across the rubbing surface.

7.2 **EXPERIMENTAL INVESTIGATION**

7.2.1 **Experimental Procedure**

A S cam brake was mounted on a dynamometer to enable extended constant speed braking to be achieved. Brake applications were carried out for twenty seconds at a constant actuation pressure of 0.5 bar and an angular velocity equivalent to a vehicle speed of 100 km/h. Each brake application was commenced at an initial drum temperature of 80°C. It
is in this type of high speed extended brake application, typical of those made on motorway-slip roads, that transient thermal effects are most often encountered, usually in the form of brake judder.

In order to measure the drum rubbing surface throughout the brake application an array of ten rubbing thermocouples was mounted to the cam housing to measure the temperature at 18 mm increments across the drum rubbing path. The response of the rubbing thermocouples, being a time average around the drum circumference, is too slow to identify individual hot spots on the drum rubbing surface, but is sufficiently quick to indicate any non-uniformity in temperature across the brake path.

7.2.2 Experimental Results

Figure 7.2 shows the axial temperature profile on the brake rubbing surface for two consecutive brake applications. The position of the ten rubbing thermocouples is shown at the bottom of each figure and the temperature at each thermocouple is plotted vertically every second as the sliding progresses. From the figure it can be seen that the drum temperature is not uniform across the rubbing path, both high and low temperature bands forming during the brake application. These bands, usually two or three in number, are not fixed in position between, or even during, an application.

The temperature variation circumferentially in each band was investigated experimentally by embedding two thermocouples into the drum, both on the same axial rubbing path, but positioned 90° away from each other. Results indicate that the temperature of the "hot" bands is not uniform circumferentially, causing high and low temperature diameters. The differential thermal expansion on these two diameters produces transient drum ovality causing brake judder. Moreover the temperature of the cool bands is nearly uniform circumferentially, these bands probably acting as restraints, limiting the amount of ovality and hence
judder which can be achieved due to the thermal deformation of the hot bands.

7.3 FINITE ELEMENT ANALYSIS OF STRIP BRAKING

7.3.1 Band Contact

During the experimental investigation of strip braking, hot bands were usually seen to be produced on the drum surface following a period of high speed light duty braking. Typically hot bands would be initiated six to eight seconds into the brake application and then develop as the stop progressed. Although the finite element approach used in this work is capable of reproducing the conditions of a light duty extended period application, this is not practicable because of the amount of computer time necessary to simulate brake applications in excess of twenty seconds. To overcome this limitation the friction interface in the simulation was altered to promote strip braking. The contact pattern was modified to allow the actuation forces to be reacted predominantly over three bands on the lining surface; three bands were chosen as being typical of that seen experimentally. In order to accelerate the development of the strip braking effects, the operating conditions of the normal duty application discussed in Chapter 6 were assumed. These conditions assume a wheel load of 5000 kg, an initial speed of 60 km/h, and a vehicle deceleration of 0.2g. At the commencement of each brake application a drum bulk temperature of 80°C was assumed (the same as the initial drum temperature during the dynamometer tests), while the lining temperature varied from 80°C at the rubbing surface to 20°C at the brake shoe platform.

Calculations were then made of the pressure distributions on the leading and trailing shoes one second into the brake application and the results are shown in Figures 7.3 and 7.4 respectively. These figures show that the pressure distributions on the shoes are similar in shape to those predicted at the same stage of the normal duty full liner to drum contact application. On the trailing shoe a maximum
pressure of 2.3 MN/m² is predicted at the cam end of the shoe, decreasing towards the shoe pivot. Across the shoe width the three bands of higher pressure are evident. At the cam end of the shoe there is very little difference in pressure across its width despite the initial contact conditions. The pressure differential between the high and low pressure areas is most apparent at areas of generally low pressure where the effect of friction material compressibility is less marked. On the leading shoe the pressure distribution is slightly cosinusoidal or 'U' shaped, with maximum pressures of about 1.5 MN/m² being developed at both the cam and pivot ends of the lining. Due to the lower actuation forces that are applied to the leading shoe, there is less compression of the friction material and hence three distinct higher pressure bands can be seen. Also evident is a significant pressure variation across the pivot end of the lining, with higher pressure being predicted towards the drum mounting flange. Three seconds into the brake application the pressure distributions on both shoes is significantly changed as shown in Figures 7.5 and 7.6. On the trailing shoe, Figure 7.6, very high pressures, 15.2 MN/m², are predicted at the cam end of the lining. These pressures are greatest towards the drum mounting flange, decreasing towards the drum mouth. Lower pressures are predicted over the crown area of the shoe, with contact being lost over areas of the cam block. At the pivot end of the shoe the pressure rises from 1.0 MN/m² at the drum mounting flange to in excess of 12.0 MN/m² at the drum mouth. The pressure profiles across the width of the shoe indicate that torsion of the brake shoe is occurring. Torsion of the leading shoe is also apparent from examination of Figure 7.5. Here maximum pressures are predicted towards the drum mounting flange at the cam end of the lining, and towards the drum mouth at the pivot end. Contact between the friction material and the brake drum has also been lost over all but the leading edge of the cam block. As a result of the combined effects of wear, thermal expansion and friction material compression, the three bands of high pressure resulting from the initial contact conditions cannot be distinguished on either the leading or trailing shoe three seconds into the brake application.
The predicted temperature profiles on the two shoes are shown in Figures 7.7 to 7.10. The temperature distributions on both shoes at one second into the brake application, Figures 7.7 and 7.8, are similar in profile to the pressure distributions early in the stop. Three hot bands can be clearly seen on both shoes. As the stop progresses, the temperature distributions, like the pressure distributions, are modified. Three seconds into the brake application very high temperatures, up to 1,500°C are predicted at the cam end of the trailing shoe, Figure 7.10. The temperature over the rest of the surface of the cam block is much lower, between 200°C and 500°C, due to the reduced contact pressure in this area. An axial temperature variation also exists on the pivot block with higher temperatures being determined towards the drum mouth. Similarly the temperature profile on the leading shoe three seconds in the stop, Figure 7.9, is also influenced by the changing pressure distribution. High temperatures are predicted at the cam end of the lining towards the drum flange, these temperatures decreasing across the lining towards the drum mouth. At the pivot end of the lining the temperatures increase from the drum flange across the rubbing path towards the drum mouth. The changes in temperature on the surface of the linings are not as marked as the changes in contact pressure due to the low thermal conductivity of the friction material. Because the heat input into the surface of the linings cannot be readily dissipated by conduction, the temperature at the lining surface will tend to reflect the recent work history of that area of lining and hence lining surface temperatures will change at a slower rate than the contact pressures.

The drum temperature profiles for the leading and trailing shoe analyses are contained in Figures 7.11 to 7.14. The leading shoe analysis indicates that early in the stop three distinct hot bands are evident and this temperature profile is only slightly changed during the stop. The temperature in the hot band nearest the drum mounting flange is reduced slightly, and the temperature between this band and the central band is increased. The drum temperature profile from the trailing shoe analysis (Figures 7.13 and 7.14) is different from that
of the leading shoe and varies as the stop progresses. Early in the stop there are three distinct hot bands but as the stop progresses the hot bands become less distinct due to an increase in the temperature between the bands. Results also show that the temperature of the hot band nearest the drum flange is reduced, while that in the hot band towards the drum mouth is increased.

The work done per unit area distributions for the leading and trailing shoes one second into the brake application are shown in Figures 7.15 and 7.16 respectively. As expected these profiles resemble those of the respective pressure distributions. As the stop progresses the work done over the surface of the linings is influenced by the temperature and pressure distributions. Three seconds into the brake application the work done per unit area profile on the leading shoe (Figure 7.17) reflects these influences. The effect of high temperatures at areas of high pressure is to reduce the friction coefficient and hence the work done in these areas, thus the work done distribution after three seconds into the application does not demonstrate the large variations evident in the pressure distribution. It can also be noted that as the stop progresses the majority of the frictional work generated by the leading shoe is generated on the trailing (ie pivot) block.

The temperature profile on the drum surface predicted from the trailing shoe analysis, Figure 7.18, was seen to change as the stop continued; the hot bands becoming less distinct due to an increase in temperature of the rubbing path between the bands. During the brake application the work done at the surface of the lining in between the initial contact bands increases, therefore imparting a greater heat flux to the drum rubbing path in these areas. Also apparent is the reduction in work done at the edge of the lining towards the drum flange, and the increase towards the drum mouth, these changes causing a reduction in temperature of the drum rubbing path at the mounting flange and an increase at the drum mouth. Hence the temperature of the drum rubbing path, at any distance across that path, would appear to be determined by the work done over the surface of the friction material in that
circumferential path, whereas the surface temperature of the friction material is determined by the recent heat flux to each individual area of the lining.

7.3.2 Contact Across Half The Rubbing Path Width

During the investigation into the effects of limiting the initial contact between the brake drum and friction material to three bands, significant shoe distortion (torsion) was predicted to occur during the stop. The effects of non-uniform liner to drum contact can be further investigated by assuming that on the application of the brake, initial contact occurs only over one half of the rubbing path from its centre line to the drum mouth under the same braking conditions used previously.

At the onset of braking significant distortion of both shoes occurs immediately as can be seen from Figures 7.19 and 7.20 which show the pressure distributions for the leading and trailing shoes one second into the brake application. High pressures in excess of 6.0 MN/m² are predicted on both shoes at the edge of contact in the centre of the drum rubbing path. On the leading shoe high pressures are predicted at this edge for the full arc length, decreasing across the shoe width, so that contact is lost between the lining and drum over the majority of the lining arc length at the drum mouth. Clearly the shoe has twisted about its pivot under the action of the actuation forces. A greater degree of twist is predicted of the trailing shoe due to the larger actuation forces. The twist of the brake shoe, combined with the compression of the friction material at the heavily loaded edge of contact, causes contact to occur between the liner and drum at the cam end of the lining in the area rebated to provide the initial contact condition. High pressures are predicted along the edge of contact on the cam block, decreasing towards the drum mouth with contact being lost as the mouth is approached. On the leading (pivot) block the pressure variation across the width of contact is less pronounced with contact being maintained across the full contact width. As the brake
stop is continued the contact pattern over both shoes is modified as shown in Figures 7.21 and 7.22. These results show that after three seconds of braking contact has been lost over the majority of the leading block of the leading shoe, with the exception of the cam end at the edge of contact where very high pressures of 9.0 MN/m² are predicted. On the trailing block the contact pressure increases from the edge of contact at the centre of the drum rubbing path to the drum mouth where maximum pressure occurs towards the shoe pivot. Therefore the twist of the shoe about its pivot evident early in the stop has been replaced by torsion of the shoe along the length of the contact area. A similar contact pattern is apparent on the trailing shoe, albeit at higher pressures due to the greater actuation forces. Under the assumed contact conditions significant distortion of both shoes occurs because the area of liner to drum contact is limited to a largely unsupported part of shoe platform.

Very high temperatures are observed at the edge of contact on both the leading and trailing shoes throughout the brake application (Figures 7.23 to 7.26). Further into the stop these high temperatures decay only gradually due to the low thermal conductivity of the friction material. This is apparent on the leading shoe where, at the commencement of braking, high temperatures are developed along the edge of contact, decreasing towards the drum mouth. Later in the stop the pressure distribution is significantly changed, with contact being lost over the majority of the cam block and higher loadings occurring towards the pivot end of the shoe. Consequently higher temperatures are predicted on the pivot block and also at the areas of contact on the cam block, although high temperatures are maintained on the areas of lining out of contact with the drum due to its low thermal conductivity. The temperature variation on the trailing shoe more closely follows the pressure distribution as better contact is maintained at the friction surface.

The calculated drum surface temperatures from the leading shoe analysis are shown in Figures 7.27 and 7.28. One second into the brake
application very high temperatures are determined at the edge of the rubbing contact in the centre of the drum friction surface. With further braking the heat input into this area is reduced allowing the temperature to fall due to the high thermal conductivity of the cast iron drum. The predicted drum surface temperatures from the trailing shoe analysis are shown in Figures 7.29 and 7.30. Early in the application the temperature profile is similar to that predicted from the leading shoe analysis, with a central band of high temperature at the edge of the contact area. However, unlike the leading shoe analysis, the hot band increases in temperature as sliding continues, and an additional hot band is formed at the edge of the lining towards the drum mounting flange. After more prolonged braking the effects of high temperature wear, thermal expansion and the frictional characteristics of the friction material will cause the second hot band to develop and the initial band to decrease. As outlined earlier the drum surface temperatures result from the work done at the surface of the two shoes throughout the application and the latter is shown in Figures 7.31 to 7.34. The work done per unit area distribution on the leading shoe initially matches the pressure distribution with large forces being developed at the centre of the shoe, decreasing towards the drum mouth. Later the work done over the central region of the shoe is reduced, and that over the trailing block is increased. This change in the drag force distribution over the surface of the friction material is reflected in the changes seen in the temperature profile of the drum. Similarly the changes in the drag force distribution on the trailing shoe are reflected in changes in the temperature profile on the drum surface from the trailing shoe analysis. Early in the brake application two distinct contact areas can be seen on the trailing shoe. In the first of these, the contact area from the centre of the drum rubbing path to the drum mouth, the work done per unit area decreases from a maximum value of 2.078 MN/m² at the centre of the cam end of the shoe, to zero at the drum mouth where contact is lost between the drum and lining. In the second area at the cam end of the lining towards the drum flange a drag force per unit area of 1.2 MN/m² is developed. This is maintained three seconds into the application.
while that in the first area is reduced, thereby promoting development of the hot band on the drum surface coincident with the second area, and a reduction in temperature of the band corresponding to the first area.

In Chapter 6 changes in the stiffness of individual brake components were investigated and found, under normal operating conditions, to have only a small effect on brake performance. It was proposed however, that these stiffnesses could have a more significant effect under conditions of non-uniform liner to drum contact. To investigate this proposal the friction material modulus has been increased and decreased by a factor of two for liner to drum contact only across the mouth half of the drum. The results are shown in Table 7.1.

Table 7.1 The Effect of Friction Material Modulus on Brake Output

<table>
<thead>
<tr>
<th>Friction Material Modulus N/m²</th>
<th>Leading Shoe Factor</th>
<th>Trailing Shoe Factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.1 E8</td>
<td>0.99</td>
<td>0.465</td>
</tr>
<tr>
<td>8.2 E8</td>
<td>1.23</td>
<td>0.468</td>
</tr>
<tr>
<td>16.4 E8</td>
<td>1.22</td>
<td>0.468</td>
</tr>
</tbody>
</table>

The friction material used in this analysis has a Young’s modulus of 8.2 E8 at 200°C. Reducing the stiffness of the material reduces the leading shoe factor by 20%. By reducing the stiffness of the material its conformability to the drum over the reduced contact area is significantly increased, reducing pressure variations across the surface and hence reducing the shoe output. Increasing the stiffness above that of the standard friction material on the leading shoe has a
negligible effect on brake performance. This is because the friction material in its design condition is sufficiently stiff to resist the actuation loads and hence shoe distortion. Lining stiffness plays a less prominent role on the trailing shoe because of its more stable nature. This is illustrated in Table 7.1 where even under non-uniform conditions little change in shoe factor is observed when the modulus is half or double that of the standard material.

7.3.5 Summary

Three different contact conditions have been investigated and it is shown how conditions of reduced liner to drum contact increase the localised heat input to the lining and drum. The findings indicate that the heat input to the drum is determined by the drag forces developed over the surface of the friction material in each circumferential path, whereas the surface temperature of the friction material varies according to the drag force developed at each local area. These friction forces are dissipated as heat so that the temperature of each local area is determined by the recent work done in that area. The frictional properties of the lining will be modified by this heat input, possibly causing mechanical distortions of the shoes to occur as a result of the actuation forces being reacted unequally over the lining surface. This situation can persist until sufficient liner wear has occurred to produce an even contact pressure distribution. However, bands of high temperature have been seen experimentally to move across the drum rubbing path, indicating that the contact pressure distribution between the linings and drum is changing. This can lead to thermal damage of the drum. With the metal drum the high temperatures can result in permanent damage which may be a phase change within surface layers or from cracking or crazing. Spreading the load at the interface may initially minimise these effects but this condition is hard to achieve over long periods of braking if conformability of the lining material changes with temperatures and time.
FIGURE 7.1
HOT BANDS GENERATED DURING BRAKING
FIGURE 7.2a - MEASURED TEMPERATURES ACROSS THE DRUM RUBBING PATH: TEST 1
FIGURE 7.2b - MEASURED TEMPERATURES ACROSS THE DRUM RUBBING PATH: TEST 2
Fig. 7.3 Pressure on the Leading Shoe at $T = 1s$
Non-Uniform (bands) Contact.

Drum Mouth

Cam

MN/m$^2$

1.5

1.0

0.5

Drum Flange

Pivot

Fig. 7.4 Pressure on the Trailing Shoe at $T = 1s$
Non-Uniform (bands) Contact

Drum Mouth

Cam

MN/m$^2$

2.0

1.5

1.0

0.5

Drum Flange

Pivot
Fig. 7.5 Pressure on the Leading Shoe at $T = 3s$
Non-Uniform (bands) Contact.

Fig. 7.6 Pressure on the Trailing Shoe at $T = 3s$
Non-Uniform (bands) Contact.
Fig 7.7  Temperature on the Leading Shoe at \( T = 1s \)
Non-Uniform (bands) Contact

Drum Mounting Flange

Fig 7.8  Temperature on Trailing Shoe at \( T = 1s \)
Non-Uniform (bands) Contact

Drum Mounting Flange
Fig. 7.9 Temperature on the Leading Shoe at T = 3s
Non-Uniform (bands) Contact

Drum Mouth

Fig. 7.10 Temperature on the Trailing Shoe at T = 3s
Non-Uniform (bands) Contact

Drum Mounting Flange
Fig. 7.11 Drum Temperature $T = 1s$ Leading Shoe Analysis
Non-Uniform (band) Contact

Fig. 7.12 Drum Temperature $T = 3s$ Leading Shoe Analysis
Non-Uniform (band) Contact
Fig. 7.13  Drum Temperature $T = 1s$ Trailing Shoe Analysis
Non-Uniform (band) Contact

Fig. 7.14  Drum Temperature $T = 3s$ Trailing Shoe Analysis
Non-Uniform (band) Contact
Fig. 7.15 Work done per Unit Area on the Leading Shoe at T = 1s
Non-Uniform (bands) Contact

Drum Mouth

Cam

Drum Flange

Fig. 7.16 Work done per Unit Area on the Trailing Shoe at T = 1s
Non-Uniform (bands) Contact

Drum Mouth

Cam

Drum Flange

140
Fig. 7.17  Work done per Unit Area on the Leading Shoe at T = 3s
Non-Uniform (band) Contact

Fig. 7.18  Work done per Unit Area on the Trailing Shoe at T = 3s
Non-Uniform (bands) Contact
Fig. 7.19  Pressure on the Leading Shoe at T = 1s
Non-Uniform († shoe) Contact

Fig. 7.20  Pressure on the Trailing shoe at T= 1s
Non-Uniform (‡ shoe) Contact
Fig. 7.21 Pressure on the Leading Shoe at T = 3s
Non-Uniform (½ shoe) Contact

Drum Flange

Fig. 7.22 Pressure on the Trailing Shoe at T = 3s
Non-Uniform (½ shoe) Contact

Drum Flange
Fig. 7.23  Temperature Leading Shoe  T = 1s  
Non-Uniform Contact ( rubbing path)

Drum Mouth

Fig. 7.24  Temperature Trailing Shoe  T = 1s  
Non-Uniform Contact ( rubbing path)

Drum Mouth
Fig. 7.25  Temperature Leading Shoe $T = 3s$
Non-Uniform Contact (rubbing path)

Drum Mouth

Drum Mounting Flange

Fig. 7.26  Temperature Trailing Shoe $T = 3s$
Non-Uniform Contact (rubbing path)

Drum Mouth

Drum Mounting Flange
Fig. 7.27  Drum Temperature $T=1s$ Leading Shoe Analysis
Non-Uniform ($\parallel$ rubbing path) Contact

Fig. 7.28  Drum temperature $T=3s$ Leading Shoe Analysis
Non-Uniform ($\parallel$ rubbing path) Contact
Fig. 7.29  Drum Temperature T = 1s Trailing Shoe Analysis  
Non-Uniform (¼ rubbing path) Contact

Fig. 7.30  Drum Temperature T = 3s Trailing Shoe Analysis  
Non-Uniform (¼ rubbing path) Contact
Fig. 7.31 Work done Per Unit Area on the leading shoe at $T = 1s$
Non-Uniform (½ shoe) Contact

Fig. 7.32 Work done per Unit Area on the Trailing Shoe at $T = 1s$
Non-Uniform (½ shoe) Contact
Fig. 7.33 Work done Per Unit Area on the Leading shoe at $T = 3s$

Non-Uniform (1 shoe) Contact

Drum Mouth

Cam

Pivot

Fig. 7.34 Work done per Unit Area on the Trailing Shoe at $T = 3s$

Non-Uniform (1 shoe) Contact

Drum Mouth

Cam

Pivot

Drum Flange
8. EXPERIMENTAL VERIFICATION

8.1 COMPARISON OF EXPERIMENTAL AND PREDICTED RESULTS

8.1.1 Introduction

The comparison of experimental and predicted effects at the friction interface is very difficult because of the inability to directly observe this interface. Because direct measurements of the dynamic pressure distributions cannot be readily undertaken it is usual to study changes in contact patterns on the friction material during dynamometer testing. This, together with measurements made of the brake torque to determine the brake factor for comparison with calculated values, provides a means of relating theory and practice.

Predicted values from the three-dimensional analysis can be compared with results derived from the earlier two-dimension analysis. Both finite element methods of analysis have been used to calculate brake torque, friction drag and brake factor of the S cam brake. The results predicted from the two-dimensional analysis were in agreement with those previously obtained (Refs 2,3,4). These results can then be compared with those obtained from inertia dynamometer tests. The dynamometer tests were made under conditions to simulate those under which the brake was analyzed. Before any measurements were taken the linings were carefully bedded to achieve good visual contact with the drum as this was felt to be most representative of the assumption of perfect initial liner to drum contact used in the models.

Calculated temperatures may also be compared with experimental values although measurement of temperatures requires the use of more than one technique. The temperature at the rubbing path of the brake drum, being an average around the brake drum, can be compared directly with that obtained experimentally using a rubbing thermocouple of low thermal capacity and quick response time. Temperatures predicted at the surface of the linings cannot be so easily verified and requires the use of infrared thermometers.
8.1.2 Brake Torque Output

Consider a brake application made on a S cam brake mounted on the dynamometer from an initial drum speed of 300 rpm (corresponding to a vehicle speed of 60 km/h). The brake actuation torque is given by

\[
\text{Chamber area} \times \text{line pressure} \times \text{lever length} = \text{Actuation torque}
\]

where:

- Brake Chamber area = 0.0155 m\(^2\)
- Lever length = 0.14 m
- Line pressure = 3 \times 10^5 N/m\(^2\)

therefore: Actuation torque = 651 Nm \hspace{1cm} (8.1)

Now assuming 100% efficiency

\[
\text{Actuation torque} = \text{camshaft torque}
\]

where:

- camshaft torque = cam effective radius \times \text{camforce} \(F_c\)
- cam effective radius = 0.014.

\[
651 = 0.014 \times F_c
\]

\[
F_c = \frac{651}{0.014} = 465000 N \hspace{1cm} (8.2)
\]

From the dynamometer testing, brake torque under these operating conditions is 5750 Nm.

The drag force at the friction interface is given by

\[
\frac{\text{brake torque}}{\text{rubbing radius}}
\]

where the rubbing radius is 0.21
therefore drag force $= \frac{5750}{0.21} = 27381$ N \hspace{1cm} (8.3)

Now brake factor $= \frac{\text{drag force}}{\text{actuation force}}$

which from 8.2 and 8.3

\[
\frac{27381}{46500} = 0.589 \hspace{1cm} (8.4)
\]

The measured and predicted brake factors for $\mu = 0.32$ are contained in Table 8.1. The predicted brake factors are calculated from the individual shoe factors as follows.

Brake factor, $B$, $= \frac{\text{drag force}}{\text{input force}}$

where drag force is the frictional force developed by the shoe and the input force is the force applied to the shoe tip therefore eliminating the effects of actuation efficiency.

\[
B = \frac{D_L + D_T}{F_L + F_T}
\]

\[
= \frac{D_L + D_T}{F_L + F_T}
\]

\[
= \frac{D_L}{F_L + F_T} + \frac{D_T}{F_L + F_T}
\]

\[
= \frac{\text{LSF}}{1 + \frac{F_T}{F_L}}
\]

Where subscripts $L$ and $T$ refer to leading and trailing shoes respectively. Assuming the brake is in the fully bedded condition then each shoe of the $S$ cam brake does equal work therefore

\[
B = \frac{2\text{LSF}}{1 + \frac{\text{LSF}}{\text{TSF}}}
\]
where LSF = leading shoe factor
   TSF = trailing shoe factor

The measured and predicted brake factors from $\mu = 0.32$ are contained in Table 8.1.

<table>
<thead>
<tr>
<th></th>
<th>Two-dimensional Analysis</th>
<th>Three-dimensional Analysis</th>
<th>Measured</th>
</tr>
</thead>
<tbody>
<tr>
<td>Leading Shoe Factor</td>
<td>1.62</td>
<td>1.25</td>
<td>-</td>
</tr>
<tr>
<td>Trailing Shoe Factor</td>
<td>0.51</td>
<td>0.47</td>
<td>-</td>
</tr>
<tr>
<td>Brake Factor</td>
<td>0.775</td>
<td>0.679</td>
<td>0.589</td>
</tr>
</tbody>
</table>

In Table 8.1 the measured brake factors are less than those predicted from either the two or three dimensional analyses, because the predicted brake factors neglect the effects of actuation efficiency. It is assumed that all the actuation forces are applied to the shoe tips, whereas some part of the actuation forces are required to overcome friction in the cam shaft bushes and cam rollers. To equate measured and predicted brake factors a brake efficiency of 87% is predicted from the three-dimensional analysis, and an efficiency of 76% from the two-dimensional analysis. The brake efficiency of 87% predicted from the three-dimensional analysis is in very good agreement with published brake efficiencies of around 90%. Higher brake performance is predicted using the two-dimensional approach, due in part to the necessary shoe stiffness approximations to a curved beam. The three-dimensional approach incorporating axial as well as circumferential distortions, enables brake performance to be predicted to a high degree of accuracy providing an improvement over that attainable with the simple two-dimensional approach.
8.1.3 Temperatures

The temperature of any path in the brake drum rubbing surface can be measured by means of rubbing thermocouple and this value compared with the predicted temperature throughout the brake application. A thermocouple of low thermal capacity was mounted on the drum surface to measure the temperature in the path corresponding to the second row of elements from the drum mouth. The predicted and measured temperatures for a stop from 300 rpm at a line pressure of 3 bar and an initial temperature of 80°C are contained in Table 8.2.

Table 8.2 Comparison of Predicted and Measured Brake Drum Temperature

<table>
<thead>
<tr>
<th>Time/s</th>
<th>Predicted Temperature/°C</th>
<th>Measured Temperature/°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>117</td>
<td>98</td>
</tr>
<tr>
<td>2</td>
<td>109</td>
<td>120</td>
</tr>
<tr>
<td>3</td>
<td>130</td>
<td>128</td>
</tr>
<tr>
<td>4</td>
<td>130</td>
<td>128</td>
</tr>
<tr>
<td>5</td>
<td>130</td>
<td>125</td>
</tr>
</tbody>
</table>

The calculated and measured brake drum temperatures, after a slight discrepancy early in the stop, show good agreement as the stop progresses, with both predicted and measured values stabilising after three seconds at around 130°C. The discrepancy early in the stop is due to the time step used in the analysis. At the friction surface local conditions are constantly varying, whereas in the finite element model conditions can only vary at the end of each step. This allows higher temperatures to be generated at areas of high contact pressure than would be seen in practice. If an area of lining initially at a low temperature, is subjected to a high pressure then high drag forces and consequent heat fluxes will be generated. These heat fluxes will increase the lining surface temperatures and reduce the local friction.
coefficient, moderating the work done in this area. However, such modifications can only occur at the end of the time step allowing high temperatures to be generated at both friction surfaces. At the end of the time step the friction coefficient is reduced in line with the high lining temperatures, causing a reduction in work done, and due to the high thermal conductivity of the brake drum, a reduction in drum surface temperatures.

The predicted temperature at the rubbing surface of the friction material cannot be so readily compared with measured values. If areas of very high temperatures at the lining surface are neglected as being time step effects, then the remaining lining surface temperatures are still predicted to be considerably hotter than the drum temperature in the same rubbing path. This supports the existence of a high interface contact resistance between the friction pair as proposed by Day (Ref 87) and this effect can be investigated using infrared techniques.

8.2 INVESTIGATION OF FRICTION MATERIAL SURFACE TEMPERATURES

8.2.1 Infrared Thermometry

All bodies above absolute zero (-273°C) possess thermal energy causing them to emit radiation. It can be shown that in the temperature range 0 to 2000°C most of the emitted radiation is in the visible and near visible medium infrared parts of the spectrum. At both shorter and larger wavelengths the emitted radiation is for practical purposes negligible although it is never zero. The measurement of this emitted radiation enables the temperature of a given surface to be determined.

The radiation emitted by a surface is defined if the temperature of the surface and its emissivity are known. The emissivity of an object is defined as the ratio of the emitted radiation to that which would be emitted by a black body at the same temperature and is dependent on the material and surface conditions of the object. Emissivity varies
between 0 and 1.0 as shown in Figure 8.1, where 1.0 is the emissivity of the black body.

When the (black body) radiation inside a solid body reaches the boundary some (depending on the refractive indices of the body and the surroundings) is reflected and some passes through the boundary and is emitted. The proportion reflected is the spectral reflectivity, \( r_1 \), which is given approximately by

\[
\frac{1}{r_1} = \left( \frac{\mu_1 - 1}{\mu_1 + 1} \right)^2
\]

(8.5)

where \( \mu_1 \) = spectral refractive index

The fraction emitted is \((1 - r_1)\) which is defined as the spectral emissivity of the body \( E_1 \).

Thus \( E_1 = 1 - r_1 \) 

(8.6)

In practice it is found that the spectral emissivity of a surface can vary quite markedly with wavelength (especially for highly oxidised metals for example) but varies little with temperature. It can also vary with time if the surface condition changes as, for example, increased oxidation of a metal surface. As most objects emit radiation over a band of wavelengths a mean emissivity is defined. The mean emissivity, \( E \), is the spectral emissivity weighted by the Planck function and thermometer characteristics. Planck's Law defines the radiation existing in a black body enclosure.

\[
J_\lambda \, d\lambda = c_1 \lambda^3 \left( e^{c_2 / \lambda T} - 1 \right)^{-1} \, d\lambda
\]

(8.7)

where \( J_\lambda \) = energy emitted in the wavelength band \( \lambda \) to \( \lambda + d\lambda \) by a black body at \( T \) K

\( c_1, c_2 \) = constants
The weighting of the Planck function varies with temperature, consequently the mean emissivity varies with temperature. The radiation from the surroundings is similarly treated by the surface and the same fraction, $r_s$, is reflected back to the surroundings, a fraction $E_s$ being absorbed by the surface.

Thus, if the temperature and emissivity of a surface are known then the amount of radiation emitted by that surface at a single wavelength, or over a band of wavelengths, can be calculated. If the optical characteristics of the thermometer are known it is possible to calculate the amount of radiation reaching the detector, and if the sensitivity of the detector is known its output can in general be calculated from the equation

$$ V = A \int_0^\infty E_s J_{\lambda T} \tau_s S_s d\lambda $$

where
- $V$ = detector output
- $A$ = geometrical factor of the thermometer optics
- $E_s$ = spectral emissivity of the surface (8.2)
- $J_{\lambda T}$ = Planck radiation function
- $\tau_s$ = spectral transmission of the thermometer optics
- $S_s$ = spectral sensitivity of the detector

Conversely, if $V$ is measured and $A$, $E_s$, $\tau_s$ and $S_s$ are known, $T$ can be calculated. This is the basic principle of radiation thermometry. In practice a radiation thermometer is calibrated by pointing it at a source of known temperature and emissivity, usually unity, and observing the output. By varying the source temperature a calibration table can be built up from which the temperature of any source of known constant emissivity can be read off once the thermometer output when viewing this source is known.

The amount of energy radiated by a surface varies markedly with temperature, as does the spectral distribution of this energy. In
general at low temperatures the bulk of the emitted radiation is in the medium infrared region, say 2 to 20 μm. To measure low temperatures a total radiation thermometer is required. As the temperature increases the radiated energy increases as \( T^4 \) and the bulk of energy moves to shorter and shorter wavelengths. It can be shown that the fractional increase of energy per unit increase of target temperature is proportioned to \( 1/\lambda \). Thus to reduce errors due to emissivity uncertainty, errors in measurement, electrical noise etc, the smallest possible wavelength should be used for measuring temperature.

There are three basic parts to an infrared thermometer:

(i) The detector is some device which converts a given amount of radiation into a measurable signal - usually electrical. It may also define partially or wholly the spectral response of the thermometer.

(ii) The optics which serve two functions. First they define the angular field of view of the thermometer. Secondly they may modify the spectral response of the thermometer.

(iii) The reference source which may be physically incorporated in the thermometer or be situated in a calibration laboratory.

In addition there may be incorporated other components such as a pre-amplifier to raise the signal level from the detector to the point where electrical pick-up is negligible.

8.2.2 Experimental Procedure

A Land fibre optic thermometer combined with a Landmark signal processor was used to investigate the lining surface temperature. The full specification of this equipment is contained in Appendix 3. The fibre optic thermometer comprises an amplifier unit, a flexible fibre
optics and a fibre optics head. The use of a flexible fibre optics between the head and body of the thermometer allows the head to be mounted close to the brake drum exterior surface during dynamometer testing while the more vulnerable detector and electric part of the thermometer can be mounted securely away from the brake, free from heat, dust and vibration. The thermometer was specially produced to give a response time of 1 ms to 98% accuracy at the expense of some extra noise, over the temperature range 300 to 750°C (although this is modified to a range of 350 to 750°C by the signal processor). When mounted 75 mm from the target, a target diameter of 5 mm is required.

The infrared thermometer and ancillary equipment used to measure the friction material surface temperatures is shown in Figure 8.2. The measurement of the lining surface temperature during braking required a novel approach. A thermal camera could be used to measure the lining surface temperatures but only when the shoes have been withdrawn from the brake drum. On the other hand a thermocouple embedded in the friction material could measure temperatures during the brake application, but not the temperatures on the surface of the lining. It was therefore decided that the lining surface temperature could be measured every revolution of the brake drum by cutting an inspection hole in the rubbing path of the drum and focusing an infrared thermometer onto the lining. Using this approach the temperature around the lining arc could be determined by moving the thermometer head (positions 1, 2 and 3 in Figure 8.2). To facilitate this measurement a slot 20 mm long and 10 mm wide was cut in the drum rubbing surface path (calculated from the thermometer target size, and response, and the drum angular velocity).
The emissivity of the friction material used in this test was determined by heating it in an oven and recording the thermometer output at known temperatures. At 400°C the emissivity of the material was 0.6 - 0.65. This is slightly lower than the emissivity of cast iron at similar temperatures (Figure 8.1) so to prevent the signal from the friction material being masked by that from the cast iron the outer surface of the drum was polished along the path of the thermometer. In addition a displacement transducer was mounted to the brake drum to identify the signal from the friction material should the exterior surface of the drum become oxidised during testing. As the emissivity of the drum rubbing surface may be highly variable around its circumference due to smearing and oxidation of the surface, the temperature of the drum rubbing path was recorded with a rubbing thermocouple mounted close to the path of the infrared thermometer. The signals from the thermocouple, infrared thermometer and displacement transducer were recorded on a TEAC CS-XR5000 recorder.

The brake used for this investigation was a Lucas 13" x 5" hydraulically operated twin leading shoe brake assembly as shown in Figure 8.3. Although this brake is different to that modelled in the finite element analysis careful choice of the operating conditions enables the same energy dissipation per unit area of lining to be achieved for the two brakes. Thus comparison of the measured and predicted results will provide an indication to the validity of the calculated results.

8.2.3 Measured Results

Figures 8.4 to 8.6 show the measured temperatures during braking. Each figure displays friction material surface temperature and drum rubbing surface temperature against time for the duration of a single brake application. Positions one, two and three refer to the position of the infrared thermometer around the leading shoe arc (Figure 8.2). So as to promote lining surface temperatures in excess of 350°C (the lower temperature limit for the infrared system) brake applications were made from 400 to 25 rpm with a line pressure of 100 bar with an initial
drum temperature of 300°C. This heavy duty application was chosen to give the same energy dissipation per unit area of lining as for the heavy duty brake application discussed in Chapter 6.

When the brake is applied the lining surface temperatures at all three measurement positions very quickly (less than one second) exceed 350°C with the lining surface temperatures being between 40°C (position 1) and 25°C (position 3) higher than the drum temperature. When the brake is released the lining temperatures drop immediately to a value not discernible above the signal from the drum exterior surface. Maximum lining surface temperatures in excess of 420°C are measured at the end of the brake application. These values are somewhat lower than those predicted in the theoretical analysis where very high temperatures are indicated at the ends of the linings, but similar to the more uniform lower temperatures predicted for the bulk of the lining. Because of the infrared spot size and the inclusion of the metal shoe in the field of inspection it is not possible to use the infrared thermometer to measure the temperatures at the ends of the linings. However, visual inspection of the linings at the end of the test shows degraded areas along both ends (Figure 8.7) demonstrating that these areas have been subjected to much higher temperatures than the rest of the lining.

Figures 8.8 and 8.9 show the predicted temperature distribution on the leading shoe at the beginning and end of the heavy duty brake application. As measured, high lining surface temperatures are predicted early in the application, increasing gradually as the stop progresses. Early in the stop the temperatures over the surfaces of the two friction material blocks on the leading shoe are, if the high leading and trailing edge temperatures are ignored, similar, with some variation across the rubbing path from the drum flange to the drum mouth. As the brake application is maintained the temperature over the middle and towards the trailing end of the shoe is increased above that in the centre of the leading block. This is in good agreement with the measured results which show the temperatures at positions 2 and 3 to increase above that at position 1 as the stop progresses.
During each brake application the drum surface temperature in the path of the infrared thermometer was measured with a rubbing thermocouple. The measured temperatures of the drum rubbing surface were seen to be typically 20 to 70°C less than those of the friction material surface. This phenomenon is one of the findings resulting from the finite element analysis although the temperature differential between drum and lining temperatures is predicted to be higher than that measured, with the drum temperatures being in good agreement.

8.2.4 Discussion of Results

The temperature of the friction material rubbing surface of a commercial vehicle drum brake has been determined using an infrared thermometer. This approach, previously used to investigate brake disc temperatures (Ref 88), has not been used before to determine the temperatures of the contacting areas during braking. The method used of viewing the lining surface through a hole in the drum rubbing path at present limits the measurement of the lining surface temperature to once per revolution at the position of the infrared sensor. A more thorough investigation could be made of the surface temperatures by constructing an array of sensors around the lining arc length, although to maintain the drum stiffness only one hole could be made in each drum. Despite this limitation the measurement of lining surface temperatures using an infrared thermometer provides a better understanding of the conditions that exist at the friction interface.

From the comparison of lining and drum surface temperatures it is seen that the lining surface temperature can be 70°C or more higher than the drum temperature in the same rubbing path. This observation supports the suggestion of Ling and Pu (Ref 89) that a high interface contact resistance exists between the friction pair.

It has already been seen how the time step approach of the thermal analysis may lead to the prediction of higher temperatures than are seen in practice. However, this may not be the only cause of these
predicted high temperatures. For the purpose of the finite element analysis the friction material is assumed to have uniform properties throughout its thickness and the brake drum to be perfectly smooth and elastic and unaffected by wear. In practice the interface and material of the friction pair are more complex than can be simulated with this finite element approach, with frictional forces that are not directly transmitted between rotor and stator but between a third body layer or transfer film. These layers can be coatings either on the drum or lining, or both, and are formed from wear debris and by the transfer of material within the friction pair. They are known to significantly affect the friction and wear performance of the friction pair and their effect on surface temperatures has been studied (Ref 90) and found to be considerable.

Further studies of the friction interface (Ref 29) have identified that degradation produced under high temperature sliding conditions is not as severe as would be expected, supporting the existence of an ablation type wear mechanism providing sacrificial protection to the subsurface material. Mechanisms such as this that exist at the friction interface may have an influence on surface temperatures that cannot readily be incorporated into the finite element analysis.

Despite higher temperatures being predicted at the lining friction surface than those measured, there is good agreement between the trends that emerge when the two sets of results are compared. The measured values, for example, confirm the existence of a thermal contact resistance between the brake drum and linings predicted in this, and earlier finite element investigations of the friction interface. Furthermore the temperature distribution over the surface of the linings, and its variation during the stop predicted by the finite element analysis is also substantiated. This is additional to the comments made previously regarding brake performance results where good agreement between measured and calculated values was obtained.
FIGURE 8.1
EMISSITIVITY OF VARIOUS MATERIALS

<table>
<thead>
<tr>
<th>Material</th>
<th>Emissivity</th>
</tr>
</thead>
<tbody>
<tr>
<td>WATER</td>
<td>1.0</td>
</tr>
<tr>
<td>OXIDISED STEEL</td>
<td>0.9</td>
</tr>
<tr>
<td>CAST IRON (200 - 600 C)</td>
<td>0.8</td>
</tr>
<tr>
<td>CORRODED BRASS (200 - 500 C)</td>
<td>0.7</td>
</tr>
<tr>
<td>POLISHED CAST IRON (200 C)</td>
<td>0.6</td>
</tr>
<tr>
<td>NON - OXIDISED CAST IRON (200 C)</td>
<td>0.5</td>
</tr>
<tr>
<td>POLISHED ALUMINIUM (100 C)</td>
<td>0.4</td>
</tr>
<tr>
<td>POLISHED COPPER (100C)</td>
<td>0.3</td>
</tr>
</tbody>
</table>
FIGURE 8.2 - SCHEMATIC ILLUSTRATION OF THE INFRARED MEASUREMENT OF LINING TEMPERATURES

- Fibre Optic Head Position 1, 2, 3
- Amplifier Unit
- Signal Processor
- Data Storage
- Flexible Fibre Optic
- Displacement Transducer
- Rubbing Thermocouple
FIGURE 8.3

INFRARED MEASUREMENT OF LINING SURFACE TEMPERATURES
FIGURE 8.4
MEASURED LINING SURFACE TEMPERATURES @ POSITION 1

Temperature (deg C)

Time (seconds)

Drum temperature  Lining surface temp
FIGURE 8.5
MEASURED LINING SURFACE TEMPERATURES @ POSITION 2
FIGURE 8.6
MEASURED LINING SURFACE TEMPERATURES @ POSITION 3

Temperature (deg C)

0 0.5 1 1.5 2 2.5 3 3.5

Time (seconds)

Drum temperature  Lining surface temp
FIGURE 8.7
CHARRED EDGE OF LINING AFTER HEAVY DUTY BRAKE TEST
Fig. 8.6 Temperature Distribution Leading Shoe T = 1s

Fig. 8.9 Temperature Distribution Leading shoe T = 5s

Drum Mouth

Pivot

Drum Mounting Flange

Cam

1500
1000
500

Pivot

Drum Mounting Flange

Cam

2000
1500
1000
500
Over recent years the increase in speeds and weights of commercial vehicles, combined with stricter requirements due both to legislation and operator pressure (maintenance free, long life products) have placed pressure on the brake and friction material industries to produce high performance brake systems providing reliable problem free operation. In order to achieve this goal, a thorough understanding of the factors effecting brake performance is required and this has been achieved in this study using modern analysis and measurement techniques.

At the same time as the brake manufacturer was facing demands for increased braking performance from his products the friction material industry was undergoing an enforced change from asbestos based to asbestos free products. The change to asbestos free friction materials resulted in significant changes in friction material formulation. Asbestos had, for fifty years, provided the base of the majority of friction material compositions, its natural properties being ideally suited to meeting the demands of the friction material industry. Asbestos provided strong, flexible fibres with low thermal conductivity and good heat resistance over the operating range of friction brakes. In addition asbestos was easily processed and cheap. Initial hopes that asbestos could be replaced by an alternative fibre proved misplaced and it was necessary to re-formulate about a new range of fibres such as glass, steel, carbon and polyaramid. Of these steel and polyaramid have proved most popular. Steel, originally used as a replacement for asbestos during the second world war, can be used in large quantities in friction material (semi-metallic friction material); these materials providing stable friction and kindness to the mating surface. However, the high thermal conductivity of the steel can result in more heat being transferred to these steel based materials resulting in higher temperatures and increased wear rates. Polyaramid man made fibres can be successfully used in relatively small quantities to provide a matrix to which fillers, lubricants and abrasives can be added. Due to the high cost
of such fibres a fine balance has to be achieved between material integrity and cost. In order that the correct formulations can be achieved a thorough understanding of conditions at the friction interface is necessary and this is included in this work.

Friction material wear is dependent on local conditions of pressure, rubbing speed and interface temperature as well as the composition of the friction pair. The surface finish of the metal member as well as its composition can effect brake performance as can the physical properties of the friction material (thermal conductivity, compressibility, coefficient of thermal expansion etc). In this investigation the friction material wear characteristics are expressed as follows:

$$w = f (p, v, \theta, D, FM, ME)$$

For a given friction pair $D$, (brake design), $FM$ (friction material) and $ME$ (metal members), are fixed; wear being a function of $p$, $v$ and $\theta$. It has been shown that the wear characteristics of the friction material can be expressed as an experimental function of the third degree.

$$w = C p^a v^b \theta^c$$

The coefficients $C$, $a$, $b$ and $c$ have been determined for both asbestos based and asbestos free polyaramid fibre based friction materials fitted to a commercial vehicle brake to provide a comparison of the wear characteristics of the two materials.

The tribological properties at the friction interface are effected by a number of different parameters ($p$, $v$, $\theta$, $D$, $FM$, $ME$) and these in turn influence the performance of the brake. During braking the kinetic energy of the vehicle is dissipated by frictional heating developed at the rubbing interface between the linings and brake drum. Local temperatures can occur and their value will fluctuate depending on the size of the contacts. When the brake is applied a pressure distribution is formed over the surface of the linings which is dependent on the deformation of the brake components. These temperature and
pressure effects are interdependent giving rise to thermo-elastic instability. The thermal and mechanical distortions and friction material wear causing this instability have been investigated using the finite element method.

The friction interface is modelled using special "Gap" elements that allow dynamic friction to be developed between each pair of interface nodes. Using an iterative procedure the actuation forces are applied to the brake and the pressure distribution between the lining and drum is calculated (the normal reaction between each node pair). From the normal reaction at each node at the friction surface the drag force is determined. The pressure distribution is then re-calculated from the actuation and friction drag forces. This process is repeated until convergence to within 5% has been achieved. Early attempts to investigate the braking process using the finite element method adopted a two dimensional approach utilising shoe stiffness approximations to a curved beam of uniform section which limited investigations to the circumferential behaviour of a drum brake. A three dimensional approach has been developed here to enable the combined effects across and around the friction surface to be determined. This approach enables an accurate representation to be made of the brake components to give a more realistic picture of the complete brake in operation.

The simulation of the friction interface (known as Gapfric) has been incorporated in a fully automatic brake analysis package called "Multi". This system uses a time step approach to simulate a brake application. At the commencement of the application the actuation forces are applied to the brake shoes enabling the friction drag forces to be calculated using the Gapfric method. From the friction drag forces developed at each node at the friction surface the heat flux to that node is determined and hence the temperature distribution and thermal distortions within the brake calculated. The mechanical and thermal distortions within the brake are combined with the friction material wear at the end of each time step to define the brake geometry for the next time step. Friction characteristics of the material for each time step are determined from the local temperatures by means of a table which is incorporated in the program. This process continues until the specified number of time steps has been completed.
Using the finite element approach a number of different contact conditions at the friction interface have been investigated. For each of these contact conditions the pressure and temperature distributions over the leading and trailing shoes, together with the work done per unit area of the lining surfaces has been determined. Distortion of the brake components has been seen to influence the pressure distribution across the lining width. Under heavy duty braking conditions "bell mouthing" of the drum and torsion of the brake shoe is predicted to occur and combined with compression of the friction material in heavily loaded areas of the lining surface can result in highly non-uniform contact patterns being generated. Overheating is possible at the high pressure areas resulting in thermal expansion and high temperature wear which can lead to the formation of hot bands on the drum rubbing surface. The formation of these hot bands has been investigated experimentally and with the finite element model. In both cases the hot bands were seen to move across the rubbing path both during and between successive brake applications.

The effect of modifying brake components has been investigated as a possible cause of variations in brake torque output. The stiffness of the brake shoes, brake drum and friction material were all increased and decreased individually by a factor of two and the effect on brake performance determined. Increasing the friction material stiffness was predicted to increase brake output by promoting a more non-uniform contact pressure distribution as was reducing the drum stiffness. Increasing the drum stiffness and reducing the friction material compressibility were both predicted to reduce brake output by providing a more uniform pressure distribution. However, the variations in brake output resulting from these changes were, for normal contact conditions (assuming well bedded linings), predicted to be relatively small. Therefore it is concluded that the performance of friction drum brake is primarily determined by changes in the friction coefficient at each friction pair in response to the local surface temperatures. When a node at the friction surface is subjected to a heat flux as a result of work done at that node, the friction coefficient at the node will be modified according to the \( \mu \) temperature characteristics of the friction material. This will cause the friction coefficients of areas of lining receiving high heat fluxes to fall.
and hence the work done over the surface of the linings will tend towards uniformity.

Experimental verification of the predicted results was undertaken with the brake mounted on an inertia dynamometer. Careful attention was paid to the setting up of the brake prior to each test to enable the conditions used in the theoretical analysis to be repeated as closely as possible. Comparison of predicted brake output from the two and three dimensional finite element analyses with the measured values demonstrated the improved accuracy of the three dimensional approach; this method predicting the brake torque output with a high degree of accuracy. The temperature of the brake drum was recorded with a rubbing thermocouple throughout each brake application and was seen to be in good agreement with predicted values.

Although brake torque and drum surface temperature are easily monitored, direct observation of the friction interface is very difficult. To enable lining surface temperatures to be measured during a brake application a different approach was adopted whereby a high response infrared thermometer was aimed at the lining surface through a hole in the brake drum. Using this method the temperature at the friction surface could be measured once per revolution. Because of limitations in available test resource the predicted and measured temperatures are not directly comparable, however the infrared technique provides a new insight into conditions at lining surface. Using this technique the existence of a thermal contact resistance between the friction material and brake drum rubbing surfaces has been confirmed and the in-stop temperature characteristics predicted from the finite element analysis verified.

The work presented has shown that the three dimensional finite element method provides a powerful tool with which to calculate the performance of commercial vehicle drum brakes. Using this method a better understanding of the combined thermal and mechanical effects at the friction interface has been obtained providing a new insight to the friction process occurring during braking. Specific findings and results have been discussed in context and general conclusions are drawn in the following section.
9.2 CONCLUSIONS

9.2.1 A finite element model has been developed to investigate the combined mechanical and thermal effects at the friction interface of a S cam brake. In order to investigate effects both across and around the rubbing path a three dimensional approach has been advanced which provides both a better physical insight into conditions within the brake and gives improved accuracy over previous two dimensional analyses. Thermal and mechanical models have been produced and are combined in a single fully automatic brake analysis package. This package, incorporating friction material wear together with mechanical and thermal effects, utilises a time step approach to analyse a brake application. Any initial liner to drum contact condition can be specified, enabling a range of different operating conditions to be investigated.

9.2.2 The temperature and pressure distributions at the friction interface of the S cam drum brake have been predicted and the work done over the surface of the linings calculated. A number of different contact conditions have been investigated varying from the condition of perfect initial liner to drum contact to simulate the situation of well bedded linings, to the condition of band or strip braking that occurs when hot bands are formed on the drum rubbing surface. Both normal and heavy duty operating conditions have been considered and the predicted results have been validated by comparison with measured effects. From this work the brake, as tested, is predicted to have an efficiency of 87%.

9.2.3 When a brake is applied the actuation forces are reacted by the brake linings bearing against the brake drum. At the surface of the linings a tangential drag force is developed that is the product of the normal force and the local coefficient of friction. This force opposes the relative motion of the drum and linings causing energy to be dissipated in the form of frictional heating. Consequently areas of lining
subjected to high frictional forces will receive large heat inputs causing the temperature of these areas to rise. As most friction materials exhibit a reduction in friction coefficient with temperature (brake fade) the work done by these areas will fall. Therefore, under stable operating conditions, the $\mu$-temperature characteristic of most friction materials will tend to produce conditions of uniform work over the rubbing surface of the friction material as illustrated in this study.

9.2.4 Altering the stiffness of the brake drum, brake shoes and friction material to give an increase and decrease on their normal values by a factor of two showed that none of the changes had a significant effect on brake performance. It can therefore be concluded that, for a well designed brake under normal operating conditions, mechanical distortions do not significantly alter brake performance.

9.2.5 During a brake application the temperature of any path on the brake surface is determined by the work done in that path, whereas the surface temperature of the friction material is determined by the recent heat flux to each local area. When an area of lining is subjected to heat input the temperature of the area increases and hence the work done by that portion of lining decreases (conclusion 9.2.3). However, due to the low thermal conductivity of the friction material its temperature is seen not to decrease immediately with the rate of working but declines gradually. Furthermore due to the high thermal conductivity of the drum cast iron, the temperature of the path on the drum surface coincident with this area of lining will fall if there is no increase in work around this path at some other area of lining. In this manner hot bands may move across the rubbing surface of the brake drum.

9.2.6 Brake fade under heavy duty operating conditions is shown to be predominantly caused by a reduction in friction level over the surface of the leading shoe due to the high rate of working and consequent heat input. Under conditions of non-uniform liner to drum contact (band
contact for example) the frictional characteristics of the lining material are modified possibly causing mechanical distortions of the brake shoes as a result of the actuation forces being reacted unequally over the lining surface. This can persist until sufficient wear has occurred to produce an even contact pressure distribution over the lining surface and may lead to uneven or taper wear of the friction material.

9.2.7 Under non uniform liner to drum contact the friction material compressibility and brake drum and brake shoe stiffnesses can significantly effect brake performance. A low modulus friction material is shown to promote improved liner to drum conformability thus producing a more even pressure distribution and in most cases reducing brake output by reducing heel and toe pressure variations. Conversely under a high non uniform loading a flexible drum will distort giving rise to heel and toe pressure variation over the linings, hence increasing brake performance. Brake drum and brake shoe stiffness can also significantly effect cyclical torque variations arising from thermal distortions of the brake.

9.2.8 Friction material wear is a function of pressure, temperature and rubbing speed which can be expressed as

\[ w = C p^n v^\theta_0 \]

The action of wear is to produce an even contact pressure distribution under constant operating conditions (conclusion 9.2.6). However brake operating conditions are very rarely constant in which case the contact distribution will be dependent on the recent usage history of the brake. Under conditions of variable brake usage, the action of friction material wear will result in variations in contact pressure and hence brake output.

9.2.9 A new approach has been developed to investigate thermal conditions at the friction interface. Using a high response infrared thermometer
focused through a window in the brake drum, friction material surface temperatures have been measured during a brake application. During the brake stop the lining surface temperatures were measured to be up to 70°C hotter than the brake drum surface supporting the presence of a thermal contact resistance between the brake drum and friction material. When the brake comes to rest the heat in the surface layers of the friction material is dissipated rapidly into the brake drum until a common temperature is achieved.
9.3 RECOMMENDATIONS FOR FUTURE WORK

9.3.1 A three dimensional finite element brake analysis package has been developed to enable thermal and related effects at the friction interface to be studied. The complexity of the models and hence the time step used in the transient thermal analysis, has been dictated by the available computer resources. As more powerful computers become readily available the element sizes used for the thermal analysis could be reduced enabling a smaller time step to be used. This would reduce the magnitude of the error resulting from a time step approach and improve the accuracy at the edges of the lining.

9.3.2 This investigation has contained a comprehensive investigation of the factors influencing the performance of large commercial vehicle drum brakes. Although not in production yet, there are now several brake manufacturers developing air disc brakes intended to replace drum brakes on commercial vehicles. These brakes will provide another range of operational problems such as disc cracking, pressure sensitivity etc that could be investigated using the finite element method.

9.3.3 A new approach to the study of temperatures at the friction interface has been developed. Using a specially adapted infrared thermometer this investigation has shown that a new insight can be obtained of conditions present at the interface during a brake application. This technique should be further developed and applied to other brake types to enable a more comprehensive understanding of the complex interactions between friction material and mating surface to be realised.
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Test Schedule FM1/2

Object: Preliminary assessment of friction/temperature characteristics of friction materials.

Test Parameters

1. Prepare test sample as per standard procedure.
2. Prepare drum surface as per "Preparation of Drum Surface Procedure".
3. Speed: Constant rubbing 434 RPM (6.34 ms⁻¹).
   Pressure: (Constant) FM1 Test 6.9 bar (100 lbs/in²).
   FM2 Test 13.8 bar (200 lbs/in²).

Schedule

Initial test sample weight and measurement.

Bedding: 100 Applications
- Cycle = 10 seconds ON, 20 seconds OFF.
- Initial drum temperature = 100°C.
- Weigh and measure test sample

Run 1 6 Applications
- Cycle = 10 seconds ON, 30 seconds OFF.
- Initial drum temperature = 100°C

Run 2 1 Application at each 25°C step from 100 to 300°C
- Cycle = 10 seconds ON.
- Initial drum temperature = 100°C

Run 3 1 Application at each 25°C step from 300 to 100°C
- Cycle = 10 seconds ON.
- Initial drum temperature = 300°C

Run 4 1 Application at each 25°C step from 100 to 350°C
- Cycle = 10 seconds ON.
- Initial drum temperature = 100°C

Run 5 1 Application at each 25°C step from 350 to 100°C
- Cycle = 10 seconds ON.
- Initial drum temperature = 350°C

Run 6 1 Application at each 25°C step from 100 to 400°C
- Cycle = 10 seconds ON.
- Initial drum temperature = 100°C
Run 7
1 Application at each 25°C step from 400 to 1000°C
Cycle = 10 seconds ON.
Initial drum temperature = 400°C

Run 8
6 Applications
Cycle = 10 seconds ON, 30 seconds OFF.
Initial temperature = 50°C

Weigh and measure test sample

This test to be followed by an FM3 Schedule.
Test Schedule FM3

Object: Preliminary assessment of friction/temperature characteristics of friction materials.

Test Parameters

1. All test samples to have been subjected to either FM1 or FM2 schedule before commencing test.
2. Clear out debris from FM1/2 test with dry brush.

Schedule

Run 1
Reseating
20 applications
Cycle = 10 seconds ON, 20 seconds OFF.
Initial drum temperature = 100°C.
Weigh and measure test sample.

Run 2
200 applications
Cycle = 10 seconds ON, 20 seconds OFF.
Initial drum temperature = 100°C.
Weigh and measure test sample.

Run 3
200 applications
Cycle = 10 seconds ON, 20 seconds OFF.
Initial drum temperature = 200°C
Weigh and measure test sample.

Run 4
50 applications
Cycle = 10 seconds ON, 10 seconds OFF
Initial drum temperature = 300°C
Weigh and measure test sample.

Run 5
20 applications
Cycle = 10 seconds ON, 10 seconds OFF
Initial drum temperatures = 400°C
Weigh and measure test sample.
Run 6

20 applications

Cycle = 10 seconds ON, 20 seconds OFF
Initial drum temperature = 100°C

Weigh and measure test sample.
APPENDIX 2 - WEAR TEST SCHEDULE

1. **Installation**
   
   Brake: Lucas 13"x 5" HLSS  
   Rolling Radius: 0.417 m  
   Inertia: 435 kgm²  
   Wheel Cylinder Diameter: 35 mm

2. **Preparation**
   
   New brake drum and lining material to be used. Radius grind linings to drum diameter - 0.25 mm.

3. **Bedding**
   
   Speed: 60 km/h  
   Deceleration: 30% g  
   Initial Temperature: 100°C  
   No of stops: 500 or until 95% bedded

4. **Inspection**
   
   Remove wear debris from brake drum and liners using a vacuum cleaner.  
   Record brake drum and lining condition.  
   Measure drum diameter ± 0.001 mm.  
   Weigh brake drum ± 1 g.  
   Measure lining thickness ± 0.001 mm.  
   Weigh brake shoes and linings ± 1 g.

5. **Wear Test** - 500 stops for each
   
   5.1 Speed: 60 km/h  
   Deceleration: 10% g  
   Initial temperature: 100°C  
   Repeat paragraph 4.
   
   5.2 Speed: 60 km/h  
   Deceleration: 20% g  
   Initial temperature: 100°C  
   Repeat paragraph 4.
   
   5.3 Speed: 60 km/h  
   Deceleration: 30% g  
   Initial temperature: 100°C  
   Repeat paragraph 4.
   
   5.4 Speed: 40 km/h  
   Deceleration: 30% g  
   Initial temperature: 100°C  
   Repeat paragraph 4.
   
   5.5 Speed: 80 km/h  
   Deceleration: 30% g  
   Initial temperature: 100°C  
   Repeat paragraph 4.
5.6 Speed: 60 km/h
Deceleration: 30% g
Initial temperature: 200°C
Repeat paragraph 4.

5.7 Speed: 60 km/h
Deceleration: 30% g
Initial temperature: 300°C
Repeat paragraph 4.
1. **Land Fibroptic Radiation Thermometer**  
   **Model:** FP20/2/A1 Z2037

   For use over the range 300 to 750°C.  
   Optical head type: A1.  
   Output 10 volts at full scale temperature.  
   Response time 1 millisecond to 98% (special feature Z2037).  
   Performances per standard FP20/2/A1 plus some extra noise as follows:

<table>
<thead>
<tr>
<th>Target Temp</th>
<th>Noise Deg C</th>
</tr>
</thead>
<tbody>
<tr>
<td>300°C</td>
<td>1.3 - emissivity of target</td>
</tr>
<tr>
<td>350°C</td>
<td>0.4 - emissivity of target</td>
</tr>
<tr>
<td>400°C</td>
<td>0.2 - emissivity of target</td>
</tr>
</tbody>
</table>

   **Fixing Requirements**

   Optical head: External M16 thread with two clamping nuts.  
   Amplifier unit: Four fixing holes 4 mm dia on 45 mm and 148 mm centres.  
   Power supply: Supplied by Landmark Processor.
2. Landmark Thermometer Signal Processing Unit

Model: Landmark 2

Note: The Landmark 2 limits the temperature range from 350 to 750 °C.

Features:

Compatible with GP(s) and Fibroptic thermometers.
Linear outputs 0 to 20 mA or 4 to 20 mA (field selectable) and 1mV/°C.
Time functions (field selectable).
Emissivity compensation range 0.3 to 1.0.

Fixing Requirements

Power supply 100 to 240 v 50/60 Hz.
Dimensions (unpacked) 144 x 144 x 298.5 mm.
Weight (unpacked) 2.8 kg.
Panel cut out 138 x 138 mm ± 1 mm.
Depth behind panel 273.5 mm.