MODELLING LINERED ENGINE BLOCKS

Factors that affect heat transfer in the linered aluminium engine block are examined to determine their importance. Conduction is found to be the dominant mode of heat transfer, and the interface is characterised as imperfect contact if there are no surface manufacturing defects larger than 139 microns. A model is proposed to estimate the effective conductivity for imperfect contact. This thermal conductance depends on the area of contact, macroscopic roughness, the contact pressure and the interstitial medium. The transfer of heat and the distribution of stress in linered engine blocks are coupled, and the problem is strongly non-linear. A finite element solution procedure for solving the heat transfer problem in the linered engine block is outlined.

1. Introduction

Aluminium engines are much lighter than cast iron engines. Lighter engines allow the weight of the car to be reduced, both by the reduction in weight of the engine and a reduction in the weight of the structural supports. This, in turn, allows a smaller capacity engine to be used to give the car the same power. The overall weight saving for all factors is about double the reduction in the weight of the engine. The use of aluminium engines confers significant cost and efficiency benefits. Aluminium, however, has inferior wear properties to the conventional cast iron. Two different approaches are used in building aluminium engines. A special Comalco aluminium-silicon alloy, called 3HA, with enhanced wear properties resulting from the dissolved silicon can be used. This is more difficult to use in the manufacturing process as large silicon crystals can form if the procedure is not performed correctly. Alternatively, a cast iron liner can be inserted into each cylinder in a normal aluminium block. This second manufacturing method is easier. These engines are described as mono-block and linered block engines respectively.

The Study Group was asked to examine the problem of heat transfer in the linered engine block and also to consider the related stress problem. Aspects to be considered included the nature of the liner-block interface, its conductivity and the resulting effects on the temperature distribution. The Study Group was asked to determine which aspects of the problem were important, what assumptions could be made and how modelling of the system would be carried out. The ultimate aim of this work is to compare the heat transfer properties and performance of the mono-block engine to the linered block engine.

There are many different engine parts that affect the final performance and efficiency. These include intake and exhaust valves, the combustion chamber, the cooling system and the lubricating oil. The geometry of the block and the construction methods used to manufacture the engine are also important.

We define the following quantities and give their appropriate values.

T_0 - temperature on the inside of the cast iron liner
T_1 - temperature on the outside of the cast iron liner
T_2 - temperature on the inside edge of the aluminum cylinder
T_3 - temperature of the coolant
$h = 120mm$ - height of liner
$r = 40mm$ - inside radius of liner
$l_1 = 3mm$ - thickness of liner
$l_2 = 7mm$ - distance between cylinder wall and cooling ducts
$\alpha_{Al} = 2.7 \times 10^{-5}/K$ - coefficient of thermal expansion of aluminium
$\alpha_{Fe} = 1.44 \times 10^{-5}/K$ - coefficient of thermal expansion of cast iron
δ - width of any macroscopic gap between the cylinder and the liner
$k_{Fe} = 58W/(m.K)$ - thermal conductivity of cast iron for the liner
$k_{Al} = 144W/(m.K)$ - thermal conductivity of aluminium for the block

2. Mechanisms and steady state

In one dimension, the temperature distribution through the liner is governed by the time dependent heat transfer equation

$$\frac{\partial T}{\partial t} = \kappa_{liner} \frac{\partial^2 T}{\partial x^2}$$

where $T(x)$ is the temperature distribution through the liner with thermal diffusivity κ_{liner}. The coordinate x is the distance into the liner from the inside surface. The engine cycle begins with a compression phase where the temperature gradually rises until ignition. The temperature then rapidly reaches a peak of $2500^o\,C$. It then declines exponentially during the expansion phase and drops further during exhaust phase. The minimum gas temperature is about $1000^o\,C$. Heat is transferred to the inside liner of the wall by a mixed process of convection and conduction. The temperature on the inside of the liner is strongly time dependent. A description of the boundary condition will be given later.

By considering Fourier components of this time dependent surface temperature, we can determine the penetration depth for each component. This is the distance into the engine block which a particular frequency variation travels before its amplitude is decreased by a factor $1/e$. The higher the frequency the smaller is its penetration distance. The high frequency components are damped out first. The lowest frequency component of the surface temperature will penetrate the furthest. For this system, the lowest or fundamental frequency is the piston frequency ω. Substituting the Fourier component $T(t,x) = e^{i\omega t}T_1(x)$ into the heat equation gives an amplitude for T_1 of $\exp(-\sqrt{\omega/2k}x)$. The penetration depth is then given by $\sqrt{2k/\omega}$.

For a typical engine speed of 3000 rpm or 50 Hz, the penetration depth is 0.75 mm. The engine liner is 3 mm wide. Less than 2% of the variation due to the fundamental frequency penetrates to the interface between liner and block. At higher engine speed, the penetration depth is even smaller. This means that all the engine block, aside from the innermost part of the liner can be regarded as being in steady state. The temperature at the interface is critical to the overall heat transfer, but the time dependent fluctuations do not reach this far. The temperature variations in the innermost part of the liner are not important. We conclude that solving the steady state heat transfer problem is sufficient to give a good description of the engine thermodynamics.

The feature of the linered engine block that dominated the Study Group's considerations was the nature of the interface between the liner and the cylinder wall, how to model it, and what effect it has on the dynamics of the heat transfer throughout the engine.

It was decided that the interface could be characterised as either a *macroscopic gap* or as *imperfect contact.* A *macroscopic gap* is a large fluid-filled region between the liner and the cylinder wall whose width is greater than the size of the microscopic variations in surface profiles and whose extent in the other directions is large enough so that the spacing of the contacts cannot be ignored or averaged when calculating the heat transfer. *Imperfect contact* occurs when the spaces between the liner and the cylinder are of the same order as the variations in the surface profiles and the contacts are sufficiently close together that their precise spacing can be ignored. The gap would need to be modelled exactly and would require knowledge of its geometry. The imperfect contact, shown in figure 1, could be modelled as a thin uniform interface with reduced thermal conductivity. This depends on the actual physical microscopic contact area. The existence of a gap and the modelling of the conductivity for imperfect contact are discussed in detail below.

Each of the three modes of heat transfer across the interface between the liner and the cylinder were considered to determine which were important or dominant.

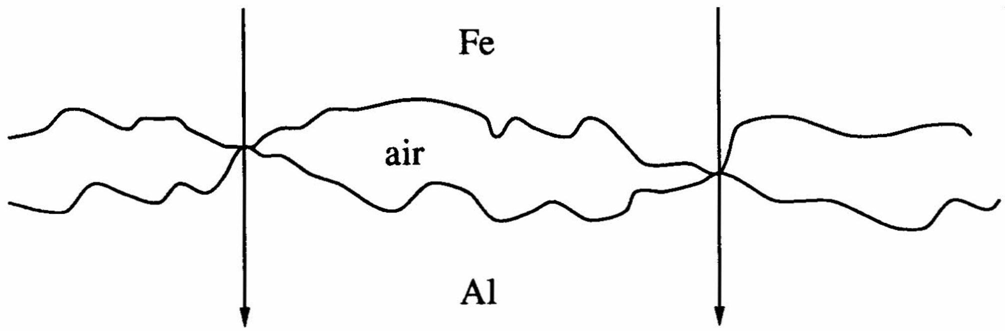

Figure 1: Imperfect contact between two smooth, flat surfaces.

For convection to occur in a gap of width δ filled with air the Grashof number $Gr = g\alpha\delta^3\Delta T/\nu^2$ must be of the order of 10^3 or greater. Here g is gravity, α is the coefficient of thermal expansion of air, ΔT is the temperature drop across the gap of width δ and ν is the kinematic viscosity of air. For a gap of $50\mu m$ filled with air, the Grashof number is of the order 10^{-3}. This indicates that the size of the gap is far too small to allow convective cells to form.

The total radiative heat flux across a macroscopic gap between the liner and the cylinder wall is $\dot{Q}_{rad} = \pi r^2 h\sigma(\epsilon_{Al}T_1^4 - \epsilon_{Fe}T_2^4)$, where $\sigma = 5.67 \times 10^{-8} W/m^2K^4$ is the Stefan-Boltzmann constant and the emissivities of oxide coated aluminium and iron are $\epsilon_{AL} = 0.3$ and $\epsilon_{Fe} = 0.64$ respectively. The maximal radiative heat transfer occurs when the entire temperature drop occurs across the gap, that is when $T_1 = 200^o C$ and $T_2 = 80^o C$. This gives a radiative heat transfer rate of about 8 Watts (W).

The radiative heat transfer rate of 8 W is very small compared to the measured total heat output through the cylinder wall of 1200W. Convection does not occur. So almost all the heat transfer is by conduction, even in the worst case of a large air gap between the liner and the cylinder. For cases with smaller, partial or no gap the thermal contact between the liner and the cylinder is better. This will result in increased conduction and even less radiative transfer. We conclude that the dominant heat transfer mechanism across the gas-filled gap is always conduction and that convection and radiation are negligible.

3. Existence of gap and its consequences

A gap could arise from the manufacturing process and be present before the engine is heated, or could arise from differential thermal expansion of the cast iron liner and the aluminium engine block once the engine is running. In this section, we discuss whether or not a gap exists and determine its effects on the heat transfer and ultimately for the engine.

As a first estimate, the largest gap that can be produced by differential thermal expansion when there is no initial gap and there are no residual stresses is $(\alpha_{Al} - \alpha_{Fe}) r \Delta T$. At the top of the cylinder, the typical temperature difference is $120^o C$. This would produce a gap of about 60μm. This is large compared to the size of typical surface variations which are of the order 10μm.

Once it is established that a gap can be caused under the right conditions, we must solve the heat transfer problem to determine the effect of the gap. Consider the one dimensional heat transfer along a horizontal radial line from the inside of the cylinder to the coolant region. Figure 2 shows a cross section of the relevant part of the engine. The heat flux entering the liner must equal the heat flux across the gap and the heat flux through the block. Then

$$\dot{Q} = k_{Fe}\frac{T_0 - T_1}{l_1} = k_g\frac{T_1 - T_2}{\delta} = k_{Al}\frac{T_2 - T_3}{l_2}$$

where k_g denotes the thermal conductivity of air. The coefficients of the temperature drops can be thought of in terms of thermal resistances $R_1 = k_{Fe}/l_1$, $R_g = k_g/\delta$ and $R_2 = k_{Al}/l_2$. Typical values of 3 mm and 7 mm for l_1 and l_2 give resistances of $R_1 = 19333.3$ and $R_2 = 20571.4$ for the liner and cylinder respectively. So despite the aluminium being substantially more conductive, the effective resistances of the liner and the block are about the same. The temperature drop is then about the same across both parts.

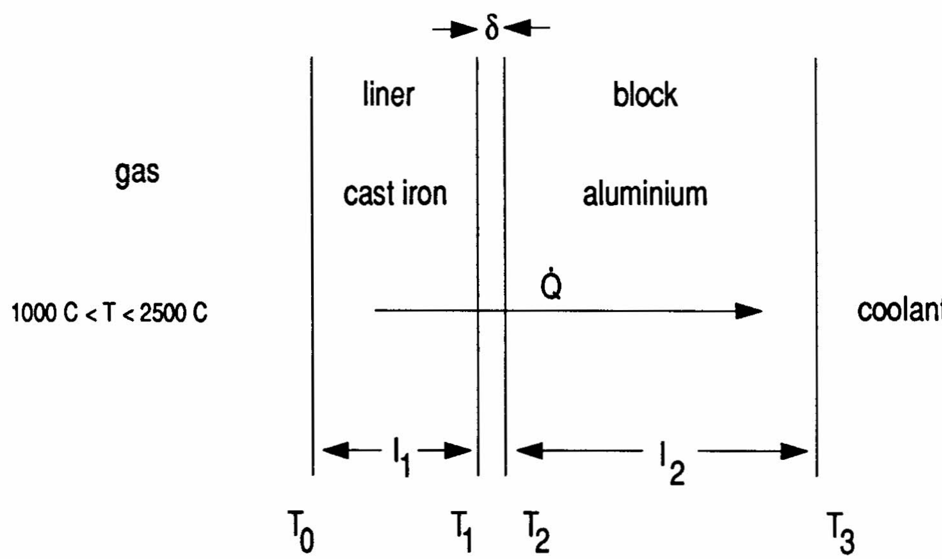

Figure 2: One dimensional representation of a cylinder.

Consider the typical case where $T_0 = 200^o C$ and $T_3 = 80^o C$. The total temperature drop from liner surface to the coolant is $120^o C$. The largest gap that can be produced by thermal expansion is 60 microns. Almost the entire temperature drop is across the gap. This is shown in figure 3(a). A large gap acts as an effective thermal insulator and dominates the heat transfer. A 30 micron gap still produces a large temperature drop of $107^o C$. A gap of about 3 microns is sufficient to produce 50% of the temperature drop across the gap. It is not until the gap is only 0.4 micron that less than 10% of the temperature drop occurs across the gap as shown in figure 3(b).

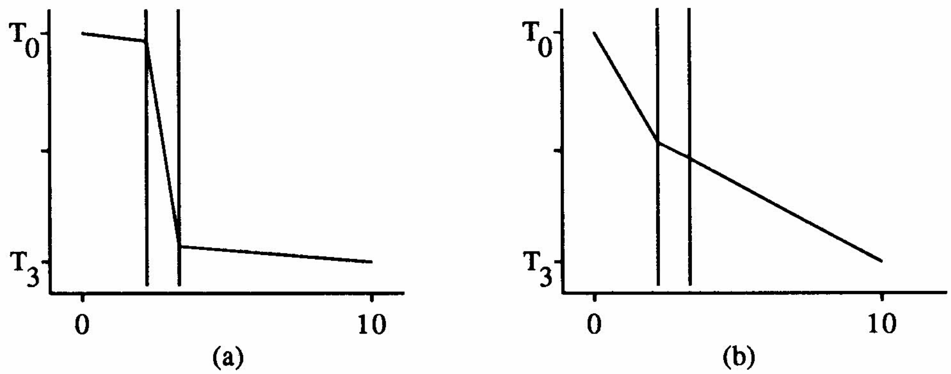

Figure 3: 1 dimensional temperature profiles for (a) 60 and (b) 0.4 micron gaps.

In our characterisation of imperfect contact, the spaces could be up to 10 microns wide. These spaces are then effectively thermally insulating. The important property for the imperfect contact is that there be many small contacts roughly uniformly spread over the interface. Any region, no matter how narrow, will act as gap if there are not many closely spaced conducting bridges through the space.

Since the piston moves up and down inside the cylinder, the upper part is exposed to the gas longer. The gas temperature also decreases as the gas expands. This produces a temperature distribution on the surface of the liner that decreases with the distance from the top of the cylinder. Typically the surface temperature at the top of the cylinder is $200^o C$ and the temperature at the bottom is $100^o C$. We assume a linear temperature profile for the surface temperature. The actual surface temperature is somewhat hyperbolic, but for the one dimensional heat transfer a linear approximation is sufficient. A more accurate initial profile would be required for the full three dimensional numerical calculation.

This temperature distribution along the surface of the liner means that the amount of differential thermal expansion between the cast iron liner and the aluminium block will vary with z. The thermal expansion gap produced by this temperature distribution, for the case of no initial stress and no initial gap, would be 60 microns at the top and 30 microns at the bottom. An important factor that has been ignored in this naive approximation is that the formation of a large gap significantly reduces the temperature of the aluminium cylinder. The thermal expansion of the block is therefore reduced and the gap becomes smaller. The smaller gap leads to more heat transfer, higher temperatures in the block and a larger gap. This is a convergent sequence leading to an equilibrium configuration. In a sense, the gap is self correcting. The final equilibrium gap can be calculated iteratively. It gives the surprising result that the gap predicted by the one dimensional heat transfer is 0.66 microns at the top of the cylinder and reaches a maximum of 18.7 microns at the bottom of the cylinder. The gap remains less than the 10 micron limit of surface smoothness until the bottom 25 mm of the cylinder. The reason for this is that, above this point, the insulating gap causes the liner to be much hotter than the cylinder. The equilibrium temperature profile is the one where the thermal expansion of the liner is equal to that of the cylinder producing a negligible gap. It is only at the bottom where the temperature drop from the liner surface to the coolant is only $20^o\,C$, that the liner and the block are at approximately the same temperature. The cylinder therefore expands twice as much as the liner and no self adjusting process is possible. The self adjusting one dimensional heat transfer solution indicates that no significant gap is produced by the differential thermal expansion except in the bottom 20% of the cylinder.

Until now we have assumed that there is no initial gap and no initial stresses between the liner and the block. In order to fully understand the gap, we need to take account of the method used to insert the liner into the engine block. The engine block is heated to a uniform temperature of $200^o\,C$. A room temperature liner is then inserted into the cylinder. An initial gap δ_0 exists between the liner and the cylinder. When the block cools, it shrinks. If the initial gap is not too large, this thermal shrinkage should ensure that the liner is clamped by the block. The engine head is then bolted on. If the top of the liner is initially higher than the top of the cylinder so that when the head is attached the liner is compressed vertically. This causes deformation in the liner shape. The compressed liner is then bored out to ensure that the walls are parallel. The pattern of the deformation is dependent on the position of the cylinder in the block and the order in which the bolts are tightened. Similar distortions occur in a mono-block when the head is attached. The deformation in the liner case represents the plastic deformation of the liner and the block. The liner expands radially as it and the block are compressed by bolting on the head. This will either reduce the size of any gap or will increase the initial radial stress. The Study Group concluded that the deformation was not a buckling phenomenon and would not produce any gap by the liner being distorted away from the cylinder wall. If the liner is shorter than the cylinder bore, there will be no vertical compression of the liner.

The gap size and residual stress distribution resulting from the manufacturing process can be calculated. Let a, b and c be the inner and outer radii of the liner and the inner radius of the cylinder respectively at room temperature T_r. Then the inner radius of the cylinder and the outer radius of the liner at temperature T are given by

$$c(T) = c[1 + \alpha_{Al}(T - T_r)]$$

and

$$b(T) = b[1 + \alpha_{Fe}(T - T_r)].$$

If $b < c$ then there is a macroscopic gap. If $b > c$, there is firm contact and a radial stress is produced. The cylinder radius is $c(T_h) = c[1 + \alpha_{Al}(T_h - T_r)]$ when the block is heated to the temperature T_h. The biggest possible room temperature liner is then inserted. The initial gap between the heated cylinder and the cool liner is δ_0. This is typically of the order of $10 - 20\,\mu$m. The outer radius of the liner is given by $b = c(T_h) - \delta_0$. The final gap at temperature T is $c(T) - b(T)$. The amount of elastic deformation in the liner and block produced when the heated block cools to room temperature and contracts is $b(T_r) - c(T_r)$. When the engine is operating and the temperature at the interface is T, the elastic deformation caused by the contraction of the block from T_h to T and from the expansion of the liner from T_r to T is

$$\delta_t(T) = \{c[1 + \alpha_{Al}(T_h - T_r)] - \delta_0\}[1 + \alpha_{Fe}(T - T_r)] - c[1 + \alpha_{Al}(T - T_r)] \quad (1)$$

If $\delta_t(T) < 0$ then there will be a gap of size $-\delta_t$ at this temperature. If the initial gap is less than 209 microns then there will be no gap remaining when the block has cooled to room temperature. If the initial gap is less than 139 microns then there will be no gap remaining when the block has cooled to an average operating temperature of $150^o\,C$.

The stress distribution in a cylinder subjected to internal and external pressures are given by Phelan (1957). The contact pressure at the interface is also given as a function of the deformation for the case of a cylinder thermally shrink fitted into another cylinder of the same material. Repeating the analysis with two different materials and assuming that the outer radius of the outer cylinder is infinite, to represent the entire engine block, gives a contact pressure

$$p(T) = \delta(T) / \left[\frac{1 + \mu_{Al}}{cE_{Al}} + \frac{1}{bE_{Fe}} \left(\frac{b^2 + a^2}{b^2 - a^2} - \mu_{Fe} \right) \right] \quad (2)$$

where E and μ are the Young's modulus and Poisson's ratio for each material. Substituting the expression (1) for the deformation into (2) gives the contact pressure. Since thermal expansion is linear with respect to temperature, then the deformation and consequently the contact pressure are also linear functions of the temperature.

The vertical stress σ_v associated with bolting the head to the engine block produces increased radial stress. If the liner is considered in isolation. Then the vertical stress would produce an additional radial stress of $\sigma_r = \mu_{Fe}\sigma_v$. This would cause the liner to expand and require additional deformation in the liner and block of

$$\delta_v(\sigma_v) = \frac{\mu_{Fe}\sigma_v b}{E_{Fe}} \tag{3}$$

The existence of a gap can radically alter the heat transfer properties of an engine. Depending on where the gap is, the engine could have inverted isotherms, hot spots or a number of other problems. There would be substantial variation in the behaviour of different engines because of the extreme sensitivity of the heat transfer to the precise gap geometry. The gap could also lead to vibration and metal fatigue. This would lead to decreased performance and decreased lifespan. The Study Group concluded that the presence of a macroscopic gap would be somewhere between highly undesirable and disastrous. It was also concluded as long as there are no surface defects larger than 139 microns at manufacturing time, then there would be no gap between the engine block and the liner at operating temperature. It is then extremely unlikely that any gap will exist and the interface could best be characterised as imperfect contact.

4. Modelling imperfect contact

The second possible type of interface is one involving imperfect contact. The spaces between the liner and the block are all small with many small contact points between them. These bridges are sufficiently close together that their fine detail does not influence the heat flow. The isotherms near the interface can be approximated as smooth, straight and parallel to the interface. Figure 1 is a typical example of imperfect contact between two relatively smooth surfaces. Such an interface can be modelled as a uniform interface with reduced conductivity k_i.

In order to calculate such a reduced thermal conductivity for the imperfect contact, consider the idealised interface shown in figure 4. The domain is a unit square with cast iron on the left of the interface and aluminium on the right. The parameter β gives the ratio of the length of the actual contact region to the physical length of the interface. In general, β will be very small. Since the contact is now perfect in the part where the two materials touch the temperature is continuous across this part of the interface. The other part of the interface of length $1-\beta$ is filled with air and is effectively thermally insulating. There will

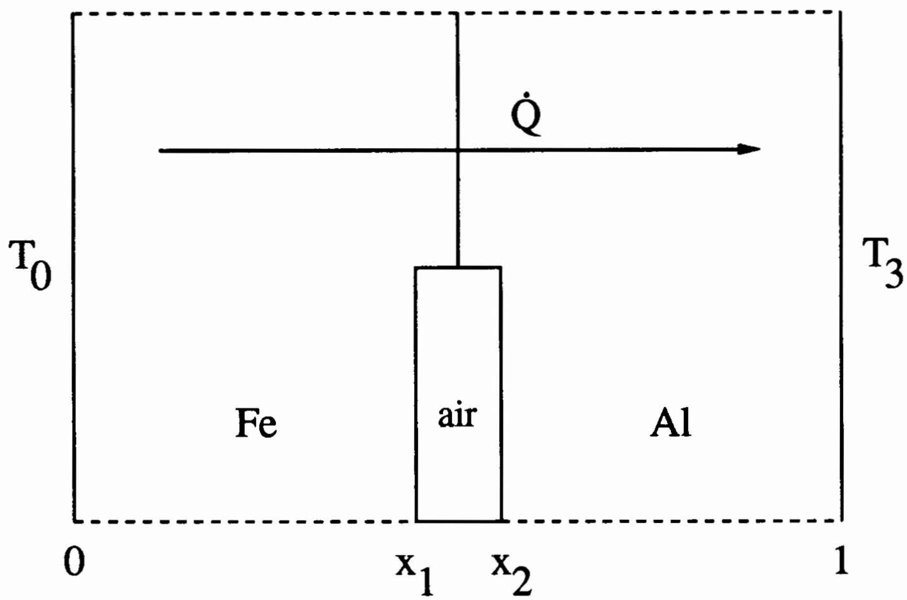

Figure 4: Integration domain for imperfect contact conductivity.

be almost no heat transfer across this air filled gap. Almost all the heat flux will be channeled through the perfect contact section. The calculated conductivity must give the same heat transfer rate and the same temperature distribution as the real interface in figure 1. The left boundary of the domain is kept at the fixed temperature T_0 and the right boundary at temperature T_3. The top and bottom boundaries are periodic. This represents infinitely many copies of the control volume stacked upon each other. This simulates a long interface and produces a thermal conductivity that is averaged to eliminate the end effects in the vertical direction. The actual geometric details of contact between two surfaces are not important for this situation. The thermal conductivity is determined only by the effective contact area.

The steady state temperature distribution is governed by the equation

$$\nabla . (k(x,y)\, \nabla T) = 0$$

where $k(x,y)$ is the thermal conductivity at position (x,y). k will take the value k_{Fe} on the left of the domain, k_{Al} on the right and k_{air} in the gap. Once the temperature distribution is known, the total heat flux can be calculated by integrating the local heat flux along a curve C from the top of the domain to the bottom. This flux is then given by the line integral

$$\dot{Q} = \oint_C k(x,y) \nabla T \mathbf{ds}$$

It is also given by

$$\dot{Q} = R_t(T_3 - T_0) \tag{4}$$

where R_t is the total thermal resistance and is the sum of the resistances of the individual components

$$R_t = \frac{k_{Fe}}{x_1} + \frac{k_{Al}}{1 - x_2} + R_i(\beta)$$

This allows the average interface thermal resistance R_i to be determined for each value of the effective contact area β.

In the previous section, we calculated the residual radial stress at the interface. While the engine is operating, the stress will have several components including the residual radial and vertical manufacturing stress, the thermal expansion stress and the stresses generated by the explosions in the cylinder. In order to determine the effect of the explosions on the interface, consider the cylindrical liner in isolation. Since the explosions occur at 50 Hz or higher, the pressure variations will not penetrate to the interface, in the same way as the temperature variations did not. So even though the explosions are time dependent, the effect on the liner will be that of a constant pressure P_e. Such a pressure will cause an increase in the radius of the liner of

$$\delta_e(P_e) = \frac{r^2 P_e}{E_{Fe} l_1} \tag{5}$$

For a typical average pressure in the cylinder of 300 kPa, the increase in the liner radius will be 33 microns. This is of the same order as the increase due to thermal expansion and cannot be neglected. Since the liner and the block are in contact at the interface, this requires an additional elastic deformation of δ_e in the the liner and block to absorb the explosions. The total deformation required is then

$$\delta = \delta_t(T) + \delta_e(P_e) + \delta_v(\sigma_v) \tag{6}$$

resulting from thermal expansion and the initial shrink fitting, the explosions and the vertical compression associated with bolting the engine head to the block. This total deformation can be substituted into equation (2) to give the total contact pressure $p(T, P_e, \sigma_v)$ at the interface.

Earlier we showed that the interface conductivity depends on the area of contact. The contact pressure at the interface is important because it determines this area. The product of this stress p and the physical area is equal to the product of the yield stress of a single aluminium-cast iron contact and the microscopic area of such contacts. That is, as the radial stress applied by thermal expansion and the explosions increases, the aluminium and cast iron in the point contacts elastically and plastically deform to increase the actual contact area until equilibrium is reached. The fraction of the total area β in which point contacts are made is then a function of the radial stress. This relationship between β and p is neither simple nor straightforward and must be determined empirically. This requires either a successful literature search or experimentation. The thermal conductivity is therefore also a function of the stress. That is $k_i(\beta(p(T, P_e, \sigma_v)))$. The thermal conductivity across the interface is vital to the entire heat transfer problem. The interface is the dominant thermal resistance and will determine the size of the heat flux through the cylinder. As the temperature increases, so does the radial stress. This increases the contact area and therefore the thermal conductivity. The heat flux then increases and the liner temperature then decreases in a self adjusting process until equilibrium is reached. The stress and heat transfer are non-linear and coupled. It is not really possible to solve them independently. It may not be necessary to solve the stress distribution for the entire engine. The stress only feeds into the heat transfer at the interface and expressions of the form (2) and (6) may be adequate for these purposes.

The effective thermal conductivity of the interface depends on the area of mutual contact, macroscopic waviness, the contact pressure, the interstitial medium and the average temperature – see Rohsenow et al. (1985) for details. Additionally, the plastic deformations of the point contacts between the aluminium and the cast iron are not reversible. A significant hysteresis effect occurs with the repeated heating and cooling of such interfaces. The effects of the lubricant and corrosion between the liner and the block are also impossible to predict. Therefore, it is important that the interface thermal conductivity be examined experimentally.

Rohsenow *et al.* confirm that the thermal conductance increases with increasing contact pressure. They also give a method for estimating the interface conductivity. It is based on experimental data for a range of ferrous and nonferrous metal pairs. The conditions for which the method is valid are unknown. The method should thus be used with extreme caution and its applicability to this problem and its sensitivity to various errors examined before use.

It is recommended that some experimental measurement of the interface conductivity be performed for samples in as close to the operating conditions in an engine as possible. This could then be used as input to numerical finite element simulation and to check the validity of the method presented by Rohsenow *et al.*

5. Numerical solution of the heat transfer problem

In previous sections, we established that the heat transfer and stress analysis problems are coupled and may need to be solved simultaneously if the analytic expressions for the contact pressure are not adequate. It is also expected that under normal circumstances there will be no macroscopic gap between liner and the cylinder. The interface is best characterised as an imperfect contact. The problem is strongly non-linear with the conductivities of the two metals being temperature dependent and the interface conductivity being a function of the temperature, the interface pressure and the vertical stress. The problem requires some type of finite element solution.

Using a partial separation of variables technique, one can reduce the problem to a sequence of two dimensional solutions. The complicated geometry in the horizontal plane suggests separating out the vertical component of the solution which varies more slowly than the horizontal ones. The temperature is written in Taylor series form

$$T(x,y,z) = T_0(x,y) + T_1(x,y)z + \frac{1}{2}T_2(x,y)z^2 + \dots$$

All the boundary conditions separate likewise, except for the prescribed temperature distribution $T'(z)$ on the cylinder walls. A two dimensional finite element solution is generated for $T_0(x,y)$ and for the successive corrections starting with $T_1(x,y)$. These solutions are summed, to the required accuracy, to give the $T(x,y,z)$. This solution will be valid when the variation of $T'(z)$ with respect to z is relatively slow.

If the variation of $T'(z)$ is not slow or this solution is not adequate then a full three dimensional finite element solution is required. No attempt should be made to model the interface with the finite element scheme. The interface should be modelled as an internal boundary condition that preserves the heat flux and causes a temperature discontinuity whose size is governed by the effective interface conductivity k_i. Thus, the boundary conditions are

$$k_{Al}\frac{\partial T_{Al}}{\partial n_{Al}} = k_i(T_{Fe} - T_{Al})$$

$$k_{Al}\frac{\partial T_{Al}}{\partial n} = k_{Fe}\frac{\partial T_{Fe}}{\partial n}$$

where n denotes the common normal to the contact surface. The subscripts Al and Fe denote the cylinder and liner sides of the interface respectively. The solution technique will be iterative.

The stress problem was not considered in detail. At this stage, it would suffice to solve the heat transfer problem fully using the finite element approach while using the approximate analytic solution, given in equations (1), (2), (4) and (5) for the stress across the interface. The ultimate solution would consist of solving for both the temperature distribution and the stress field using the finite elements. This may not be necessary since the main point of coupling between the problems is at the interface and our knowledge of the contact pressure is reasonably complete.

There are four sets of external boundary conditions for the heat transfer problem. On the inside of the cylinder, the time averaged gas temperature at each point needs to be specified. The wall temperature $T_w(z)$ is then calculated from the gas temperature using the convective boundary condition

$$k_{Fe}\frac{dT_w}{dn} = h_c(T_g - T_w)$$

The convective heat transfer coefficient h_c is given, for example, by Heywood (1988):

$$h_c = 3.26B^{-0.2}p_g^{0.8}T_g^{-0.55}w_g^{0.8}.$$

Here B is the inside diameter of the liner, T_g, p_g and w_g are the gas temperature, pressure and velocity respectively. The wall temperature distribution is then the required boundary condition for the finite element solution. The other horizontal boundary condition is the fixed coolant temperature of $80^o\,C$. The Study Group did not consider the boundary condition near the top or the bottom of the cylinder. These are complex and need further consideration. We restricted our examination to a central plane.

6. Concluding remarks

The Study Group considered various aspects of heat transfer and the related stress problem for a linered engine block. The temperature fluctuations in the gas were found to penetrate a negligible distance into the engine block. A steady state solution is therefore satisfactory. The interface between the liner and the block was studied in detail. It can be characterised as either a macroscopic gap or as an imperfect contact. Consideration of the manufacturing process led the Study Group to conclude that large gaps between the liner and the block were extremely unlikely and would be disastrous for the engine. Heat transfer across the interface is almost entirely by conduction. Progress was made on determining an effective conductivity for the interface. This was found to depend on the interface temperature, the contact pressure between liner and block and the vertical stress caused by bolting the head to the block. The heat transfer and stress distribution problems are therefore coupled. Expressions were derived for the contact pressure resulting from the initial manufacturing process, thermal expansion and the explosions in the cylinders. These can be used to give the interface conductivity. Two and three dimensional finite element solution techniques were outlined. Experimental measurement of the interface conductivity and measurement of other liner and block properties are needed both as input to any numerical model and to confirm various models and assumptions that have been made.

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