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Some heat transfer studies on a diesel engine piston

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INTRODUCTION

HEAT TRANSFER studies in the cylinder–piston assembly of an internal combustion engine have gained importance because of the effect on engine performance of these components. With the present day trend for producing engine parts of an interchangeable nature it is important to study the feasibility of converting a water-cooled diesel engine into an air-cooled one. With this object in mind this article presents some results of heat transfer studies carried out on a 13-cm diam. cast-iron diesel engine piston. Results show the nature of temperature distribution, heat flow rates to the cooling media and the resulting thermal stresses in the piston body of water-cooled and air-cooled engines at different loads.

MODEL DESCRIPTION

Figure 1 shows the piston cross-section along with its various boundary conditions and contact coefficients. Two sets of problem are considered: firstly when the engine is water cooled ($T_w = 85°C$ and $h_w = 1600$ kcal m^{-2} h^{-1} °C^{-1}) and secondly when it is air cooled ($T_w = 25°C$ and $h_w = 100$ kcal m^{-2} h^{-1} °C^{-1}). For both types of cooling on the cylinder walls four engine load conditions are considered for which the resulting gas temperature T_{gr} takes the values of 1000, 800, 600 and 400°C. The mean heat transfer coefficient h_{gm} has been assumed to be constant at 250 kcal m^{-2} h^{-1} °C^{-1} because in practice the range of its change is approximately 150–300 kcal m^{-2} h^{-1} °C^{-1}.

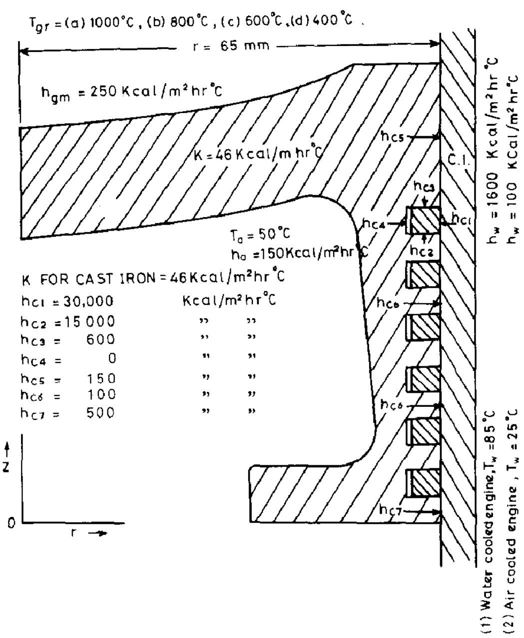

FIG. 1. Boundary conditions of a cast iron I.C. engine piston.

Due to the axisymmetry, the differential equation governing heat conduction in the piston body is given by

$$\frac{1}{r}\frac{\partial}{\partial r}\left(Kr\frac{\partial T}{\partial r}\right)+\frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right)=0, \tag{1}$$

with a convective boundary on the top face as

$$-K\frac{\partial T}{\partial z}+h_{gm}(T_g-T)=0, \tag{2a}$$

at the bottom surface exposed to the crank case as

$$k\frac{\partial T}{\partial z}+h_a(T_a-T)=0, \tag{2b}$$

and on the lateral surfaces as

$$-K_i\frac{\partial T}{\partial r}+h_{c_i}(T_{cw}-T)=0, \quad i=1,\ldots,7. \tag{2c}$$

Heat dissipation by the cylinder wall to the cooling medium—which may be air cooled or water cooled—is governed by

$$-K_{cw}\frac{\partial T}{\partial r}+h_w(T_w-T_{cw})=0. \tag{3}$$

Along the central axis of the piston (i.e. at $r = 0$), due to the axial symmetry, $\partial T/\partial r = 0$ for all z.

The solutions of the problem, as posed above, have been obtained by the heat–electricity analogy [1]. The heat transfer coefficient, h_g, on the gaseous side of the piston is obtained from the pressure–crank angle diagram using the formula due to Eichelberg [2]. For the mean value of the heat transfer coefficient, h_{gm}, and the resulting gas temperature, T_{gr}, over the complete cycle, calculations are based on the formula given in ref. [3]. The heat transfer coefficient, h_w, between the lateral surface of the cylinder wall and cooling liquid is found by using the formula [4] based on the flow velocity of the fluid (which may be water or air).

Average radial thermal stress in the piston body has been calculated using simple formulae from ref. [5].

RESULTS AND DISCUSSION

Results have been presented in Figs. 2–6 in terms of the non-dimensional variables $r^* = r/R$, $z^* = z/R$ and $T^* = (T-T_a)/T_a$. Figure 2 shows the plots of the isothermic distribution in the piston body for water-cooled and the air-cooled engines at $T_{gr} = 1000$ and $600°C$ ($T^*_{gr} = 19$ and 11); h_{gm} being constant and equal to 250 kcal m^{-2} h^{-1} °C^{-1}. It is seen that the highest temperature in the piston crown of the air-cooled engine is 475°C while for the water-cooled engine this is only 427°C. Similarly, the first piston ring has a temperature of 286°C in the case of the air-cooled engine and only 160°C for the water-cooled engine. In fact, as can be seen from Fig. 2, all the piston rings in the air-cooled engine have higher

NOMENCLATURE

E	Young's modulus [kgf cm^{-2}]	Q_g	total heat received on gaseous face [kcal h^{-1}]
h_{gm}, h_a, h_w	gaseous side, air side and water side heat transfer coefficients [kcal m^{-2} h^{-1} °C^{-1}]	R	radius of piston at outer periphery
h_{c_i}	contact coefficients ($i = 1, \ldots, 7$) [kcal m^{-2} h^{-1} °C^{-1}]	r, r^*	absolute and dimensionless radii
K, K_{cw}	coefficient of thermal conductivity of piston body and cylinder wall [kcal m^{-2} h^{-1} °C^{-1}]	T_{gr}, T_a, T_w	temperatures of fluids on gaseous, air and cooling water faces [°C]
Q_{cw}, Q_{PR}	heat transferred to cooling medium and through piston rings as percentage of Q_g	T_{cw}	temperature of cylinder wall [°C]
		T^*_{gr}, T^*_w	dimensionless temperatures on gaseous and water faces
		z, z^*	absolute and dimensionless z-direction coordinates.

temperature levels than those of a water-cooled engine. This shows that, from thermal creep point of view, piston rings are comparatively safer in the water-cooled engine than in the air-cooled one.

Figure 3 shows the nature of radial temperature variation from piston crown to periphery for the water-cooled and the air-cooled engines. It is seen that at all four loads (T^*_{gr}) the temperature drop in the water-cooled engine, as shown by dotted lines, is steeper than in the case of the air-cooled engine. This is responsible for the higher thermal stresses in the water-cooled engine as shown in Fig. 6. Figure 4 shows the temperature changes at the three characteristic points 1, 2 and 3 as a function of hot gas temperature T^*_{gr}. The rate of temperature rise with increase in T^*_{gr} is found to be higher in the case of air-cooled engine (shown by full lines) at all three points. This is expected because in a water-cooled engine the rate of heat conduction to the cooling media is faster and hence the temperature rise is slow. However, what is important to note is that there is not much of a difference in the rate of temperature rise of the two engines with increasing T^*_{gr}, at the points which are relatively nearer to the piston crown (hot gases) like 1 and 2. Only at the points which are quite far from the piston crown (like point 3) are the rates of temperature change going to be significantly different for the water-cooled and the air-cooled engines. In the case of the water-cooled engine the temperature at point 3 is practically constant over the entire range of T^*_{gr}.

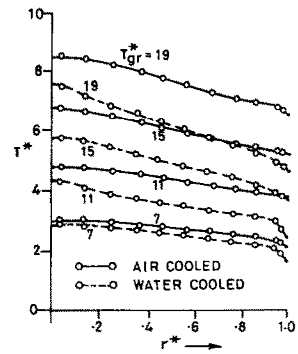

FIG. 3. Variation of radial temperatures from piston crown to periphery at different T^*_{gr}.

Figure 5 shows the variation of percentage heat transfer to the cooling medium [$(Q_{cw}/Q_g) \times 100$] and the percentage of heat transfer through the piston rings [$(Q_{PR}/Q_g) \times 100$] at different load conditions (T^*_{gr}) of the two engines. Both these percentages of heat transfer (i.e. through cooling medium and the piston rings) are higher in the water-cooled engine than the air-cooled one. With respect to the changes in the hot gas temperature (T^*_{gr}) these percentages are found to vary over

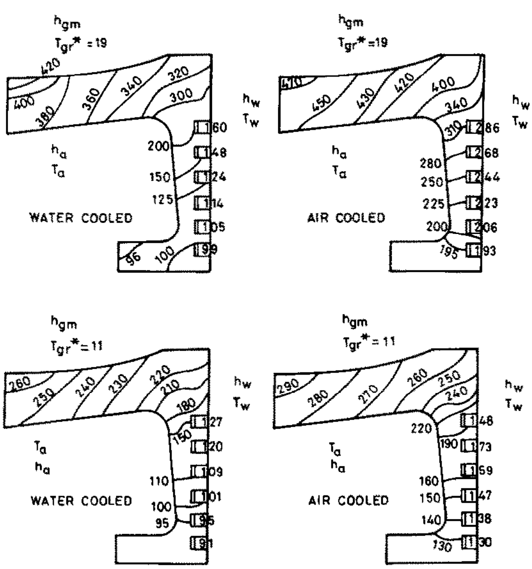

FIG. 2. Plots of isothermic distribution in the piston body.

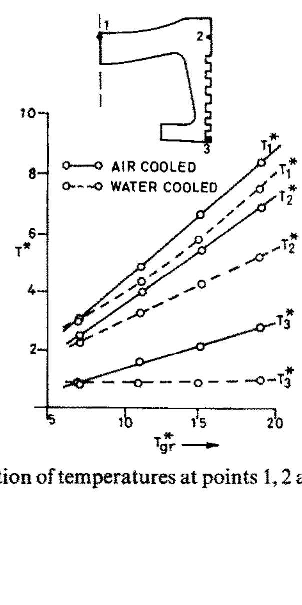

FIG. 4. Variation of temperatures at points 1, 2 and 3 with T^*_{gr}.

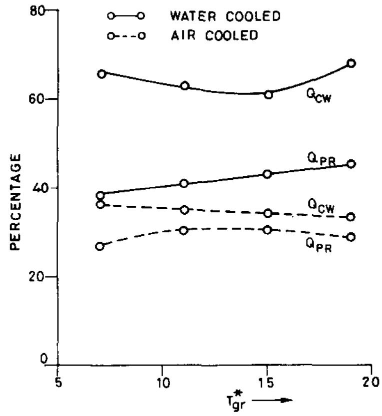

FIG. 5. Percentage heat transfer to cooling medium and through piston rings.

only a small range. Also, it is interesting to note that in the case of the air-cooled engine the percentage of heat transferred through the piston rings is of nearly the same order as through the cooling medium.

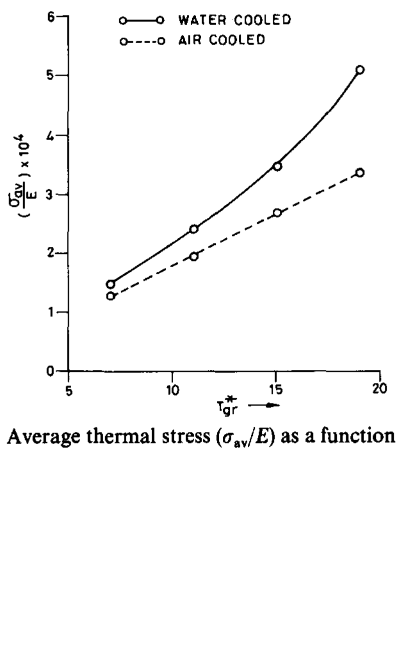

FIG. 6. Average thermal stress (σ_{av}/E) as a function of T^*_{gr}.

Finally, the plots of dimensionless average thermal stress (σ_{av}/E) distribution with respect to the hot gas temperature T^*_{gr} are shown in Fig. 6. It is seen that the thermal stresses are higher in the water-cooled engine as compared to the air-cooled one. At the lower values of T^*_{gr} the difference is, of course, not much but as T^*_{gr} increases the difference keeps increasing and becomes very significant beyond $T^*_{gr} = 15$. This happens because in the air-cooled engine the rate of change of thermal stress with T^*_{gr} is almost constant (linear variation) whereas in case of the water-cooled engine it keeps increasing (non-linear variation) with the increase in T^*_{gr} value.

CONCLUSIONS

From the above results the following conclusions may be derived:

1. Piston rings are more liable to creep failure in an air-cooled engine than a water-cooled engine.
2. Thermal stresses are higher in the water-cooled engine piston than the air-cooled one.
3. Percentage heat loss to the cooling medium is much higher in a water-cooled engine in comparison to an air-cooled engine. Therefore, in an air-cooled engine excessive heating of air is likely to occur in the crank case.
4. In case of an air-cooled engine nearly as much heat is transferred through the piston rings as through the cooling media.

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