CHAPTER III

HEAT TRANSFER FROM HOT GASES TO CYLINDER WALLS

Heat transfer from the combustion products to the cylinder walls is the basis for the calculation of engine cooling. The state of thermal balance requires that the quantity of heat removed from the cylinder walls equals the quantity conveyed to it. As stated in the preceding chapter, heat transfer from gases to the wall is determined by the equation:

$$Q = q \,.\, \Delta t \,.\, A \text{ (kcal)}$$

Previous considerations are confined to the influence of the absolute temperature T, which affects also air density at atmospheric pressure, and the influence of the mean air velocity v_m. It has been established that owing to the dominating influence of the above two values, heat transfer can be calculated, with sufficient accuracy, in all practical cases according to equation (11), in spite of the fact that the velocity exponent 0·73 is only a compromise and its value varies with changing conditions.

Heat transfer from hot gases to cylinder walls in an internal combustion engine proceeds under conditions far more complicated than those existing with a regular air flow along a solid wall. Basically the coefficient of heat transfer depends on the specific pressure, specific weight, and specific heat of the gas as well as on the gas velocity and roughness of the wall surface. Exact calculations should, furthermore, consider also the coefficient of heat transfer by radiation q_r.

It has been established by experiments, that, in internal combustion engines, the portion of heat transmitted by radiation amounts to about 2—6% only. Therefore, in further calculations heat transferred by radiation is not considered. The coefficient of heat transfer to the cylinder walls is difficult to measure. The influence of gas velocity, being of great importance for the calculation of the coefficient, is difficult to establish, for velocity is different at various parts of the wall and cannot be measured by conventional methods. Moreover, gas pressure in the cylinder varies within a wide range, and its effect upon the density and transfer coefficient must be taken into account. The best known equations used for calculating the coefficient of heat transfer contain relations to gas pressure p (kg/cm^2), absolute gas temperature T (°K), absolute temperature of the cylinder wall T_c (°K) and mean piston velocity

c_m (m/s). The latter is, to some extent, correlated to the velocity of gas flowing along the wall.

The quantity of heat transferred to a cylinder wall in an interval of time dt, amounts to

$$dQ = q \, . \, A\,(T - T_c)\,dt$$

where, according to Nusselt,

$$q = 0{\cdot}99\,(1 + 1{\cdot}24\,c_m)\,.\,\sqrt[3]{p^2\,.\,T}$$

Nusselt was aware of the limited validity of this equation, and recommended a mean piston velocity of 16 ft/s (5 m/s) as the upper limit of velocities for which the equation should be applied. The transfer coefficient is largely influenced by the turbulence of gases in the cylinder. Nusselt, Jacklisch and others have assumed that the intensity of turbulence depends on the mean piston velocity c_m. For existing automotive engines this assumption is not proved to be correct even approximately.

Turbulence in petrol engines depends largely on the width of the surface area between cylinder head and piston. In compression-ignition engines turbulence is determined by the tangential flow of air entering the cylinder during the induction stroke and further intensified by compressing the air in the recess of the piston head. In pre-combustion chamber engines intensive turbulence is caused by the combustion products entering the cylinder by the chamber port. Thus, intensity of gas turbulence in the cylinder can vary considerably regardless of mean piston velocity.

According to Briling the influence of turbulence on the transfer coefficient can be expressed by the formula

$$q = 0{\cdot}99\,(1 + d + 0{\cdot}185\,c_m)\,.\,\sqrt[3]{p^2\,.\,T}$$

where $d = 0$ for petrol engines, and $d = 1{\cdot}45$ for compression-ignition engines. In this modification of the Nusselt formula the value d expresses the intensity of gas turbulence. It is evident that d is different for different types of combustion chamber, and, for a newly designed engine, it can be determined only roughly according to the theory of similitude. Bryzov performed experiments on various engines with the following results: $d = 3{\cdot}5$ for an engine of 140 mm bore at 800 rev/min, and $d = 4{\cdot}2$ for a 100 mm cylinder bore engine having a speed of 1,100 rev/min. These results of practical measurements clearly indicate the large degree of approximation which is bound to appear in calculations. Difficulties of calculations are caused also by the variability of the internal cylinder surface area exposed to hot gases; the different temperatures of combustion chamber walls, cylinder walls, piston faces and valves; the insulating effect of the oil film on the cylinder wall and the same effect of carbon deposited on combustion chamber walls and pistons.

The following method by Janevay is a good example of how to determine the heat transfer coefficient q of a petrol engine cylinder.

The variability of cylinder surface area is comparatively easy to determine from the known engine dimensions. Cylinder surface area plotted against crank position is shown in Fig. 35. The curve represents conditions of an engine of the following characteristics: bore 82·55 mm, stroke 108 mm, side valves, compression ratio 5·4 : 1 [27].

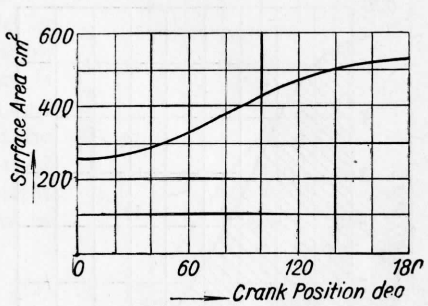

FIG. 35. Cylinder surface area exposed to hot gases plotted against crank position.

Temperature of the gases and that of the cylinder wall must be established as data for the calculation of the temperature difference. For an accurate determination of the gas temperature, we must consider the different exponents n of the polytropic compression and expansion curves. Fig. 36 shows computed temperatures for different exponents n plotted against various crank positions of the same engine.

Great difficulties are encountered in determining the mean temperature of the cylinder wall. The mean wall temperature varies in the course of a single stroke as new and colder sections of the surface area are uncovered by the piston, and the wall temperature of the combustion chamber also varies within a narrow range. An accurate calculation of the mean wall temperature is thus hardly possible. Therefore the mean temperature of the cylinder wall is determined by practical estimating and measuring. If the maximum gas temperature during combustion is 1500 °C, the corresponding mean temperature of the inner cylinder wall surface is 150 °C. A fluctuation of this value within the range of, say, ± 28 °C represents an error of ± 2% in the calculated heat transfer, and this is sufficiently accurate.

Measurements at various parts of the cylinder surface indicated a mean temperature of the cylinder wall 150 °C. This temperature can be considered as being independent on the engine revolutions. At increased revolutions the quantity of heat transmitted to the walls is increased by the amount of heat produced by friction; however, due to the intensified circulation of the cooling medium heat is removed more rapidly and so the temperature of directly cooled parts remains practically constant.

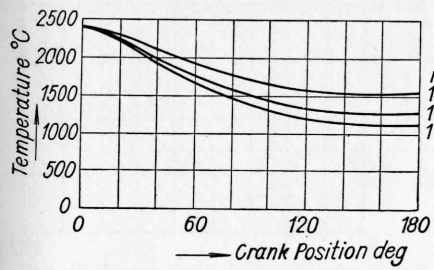

FIG. 36. Gas temperature during the expansion stroke related to crank position for various values of the exponent n.

Having determined the surface area and the temperature difference, we may enter both values into the diagram. The product of both values plotted against the crank position during the compression stroke is shown in Fig. 37. At the beginning of the compression stroke the air-fuel mixture is cooler than the walls, heat transmission is negative, i.e. heat is passed from the walls to the mixture. The overall transfer in the course of the stroke is positive and signifies the heat loss during compression.

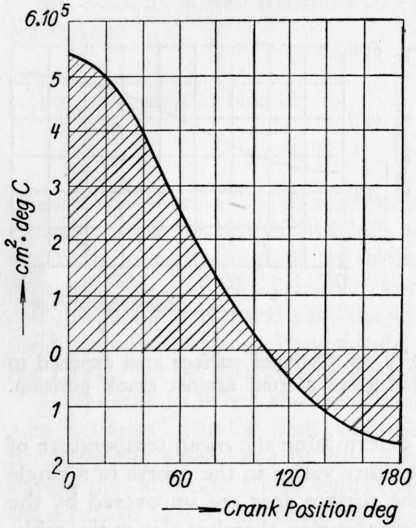

Fig. 37. Thermal losses during the compression stroke.

At a constant value of the heat transfer coefficient, the resultant positive area of the diagram represents the total heat removed.

The corresponding diagram for the expansion stroke is shown in Fig. 38. It is evident that the product of surface area and temperature difference remains practically the same during the expansion stroke, because the temperature drop in the expanding gas is compensated for by the increasing surface area of the cylinder wall. The next step is the determination of the coefficient of heat transfer from the hot gases to the cylinder wall which depends on the velocity of gas flow along the wall. The correlation between transfer coefficient and velocity originates in the diffusion of molecules in the boundary layer at the cylinder wall. Heat transfer is proportional to friction losses in the boundary layer and it is represented by the number of molecules diffused in unit time or, in other words, the product of mass velocity and density. Thus in exact calculations the heat transfer coefficient q must be considered as being dependent not only on the velocity but also on the density of gas. This fact is important for combustion conditions at dead center position, where q is consequently larger than during compression or expansion.

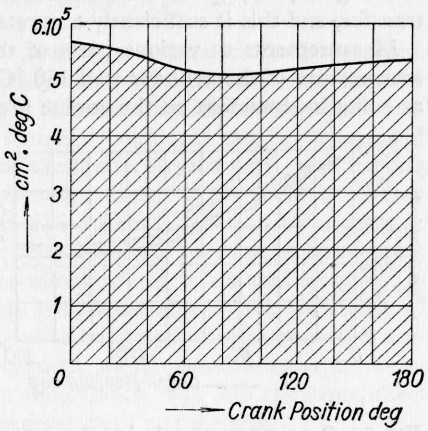

Fig. 38. Thermal losses during the expansion stroke.

The correlation between the coefficient of heat transfer q measured during expansion and speed is illustrated in Fig. 39. The curve clearly indicates how q rises with increasing speed. The exactness of the correlation is even more marked by plotting the same curve on a log-log scale. In order to include in this diagram the relation between q and gas density the diagram is drawn with the product of the speed n and the volumetric efficiency η_v entered on the horizontal axis (Fig. 40). Four computed points of the correlation lie on a straight line having a slope 0·526, and this is in accordance with the relation

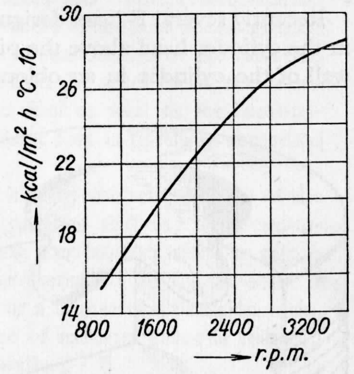

Fig. 39. Change of coefficient of heat transfer q with engine speed.

$$q = \alpha\,(n\,\eta_v)^{0\cdot526} \tag{12}$$

where n denotes the engine speed in rev/min

η_v „ „ volumetric efficiency

α „ „ constant

Measured values of q for compressed gases flowing through tubes under a pressure of 551·8 lb/in^2 (38·8 kg/cm^2) are drawn in the same diagram (Fig. 40) for the sake of comparison. Results of measurements are plotted against the product of speed and density. The resultant exponent equals 0·578. In spite of the fact that the value q is in each case plotted against a different value, the correlation is evidently of the same type. The larger q value in the engine is explained by the more intensive turbulence in the engine compared with that in the tube.

In order to reduce heat lost by cooling to a minimum, the ratio of the surface area to the volume of the combustion chamber must be as small as possible. This condition is best fulfilled by a semi-spherical combustion chamber which has the additional advantages of sufficient space for large valves, good flow conditions through the valves, etc. In flat combustion chambers, particularly with side valves, heat losses by transfer to the walls are always considerable: they reduce the overall engine efficiency and increase the specific fuel consumption. Larger

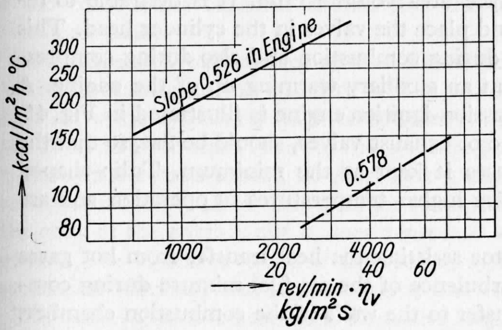

Fig. 40. Dependence of coefficient of heat transfer q on the velocity and density of gases. Full line represents tests in engine, dashed line tests with compressed air in a tube.

cooling surfaces require more power input to the cooling system. For this reason side valve designs are being abandoned in vehicle engines despite lower production cost.

Recently several F-head designs have appeared with the intake valve located in the cylinder head above the piston and the exhaust valve seated in the side wall of the cylinder in an oblong chamber. With this type of valve arrangement it is possible to design the combustion chamber for knock-free combustion even at high compression ratios. A high compression ratio ensures economical operation, particularly at partial loads, and the advantageous combustion process produces a smooth operation. However, the large surface area of the combustion chamber signifies large heat losses and a large cooling radiator.

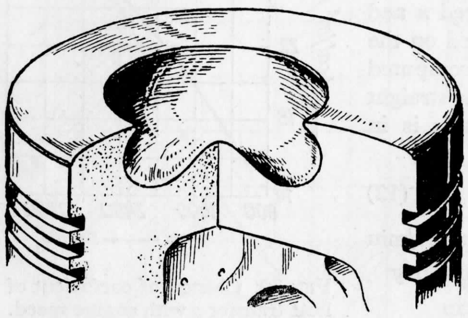

FIG. 41. Well designed combustion chamber in piston of oil engine.

The requirement of small surface area holds for pistons and valves, too. Smallest heat losses occur with flat-head pistons. A piston with a convex face or deflector transmits more heat; its temperature is higher and the piston rings are subjected to higher stresses. This is one of the reasons for the growing unpopularity of pistons with deflectors for two-stroke engines. Convex pistons with valve recesses are equally inefficient and should be regarded as an inherent evil for engines of a high compression ratio.

Pistons of compression-ignition engines must also have compact combustion chambers with the proper surface area:volume ratio. It is desirable to remove recesses from the piston and place the valves in the cylinder head. This arrangement reduces heat losses during combustion and also during compression, giving an easy start without an auxiliary warming up of the engine. A well designed piston for a compression-ignition engine is illustrated in Fig. 41.

Valve heads, particularly those of exhaust valves, should be flat, so that the surface area exposed to hot gases is kept at the minimum. Tulip-shaped valves have a low weight, but give higher temperatures in operation and are easily distorted.

Turbulence is the second factor assisting the heat transfer from hot gases to the cylinder walls. A high turbulence of the air-fuel mixture during combustion causes a larger heat transfer to the walls of the combustion chamber. Ricardo [51] recognized this fact when introducing high turbulence into combustion chamber design. Higher heat transfer has a beneficial effect during compression, temperature is reduced towards the end of the compression stroke and the application of a higher compression ratio is made possible with

subsequent advantages. On the other hand higher heat losses occur which leads to higher specific fuel consumption and to larger dimensions of radiators.

The design and shape of the combustion chamber has an important bearing upon heat loss. In the design of combustion chambers of modern spark ignition engines increased attention is paid to the course of combustion or the variation of the flame face. In compression-ignition engines, on the other hand, turbulence is required for the proper mixing of air and fuel as well as for combustion control. Here the disadvantage of higher heat loss is partially remedied by direct fuel injection.

It is evident, that heat losses are influenced also by the temperature of the combustion gases. In high compression ratio engines and in compression-ignition engines heat losses are thus higher during combustion with the piston in dead centre position. This disadvantage is compensated for by the smaller surface area of combustion chambers operating at a higher compression ratio. Due to the increased expansion the temperature of exhaust gases is reduced and so also is the heat transfer to the cylinder walls.

Thus an increasing compression ratio means an increasing ratio of the heat transferred to the cylinder head to that transferred to the cylinder walls. However, the overall quantity of heat transferred decreases with an increasing compression ratio and the thermal efficiency of the engine is higher.

Small heat losses are particularly important for compression-ignition engines. Therefore the surface area of the combustion chamber in the cylinder head should be the smallest possible. This design leads to a larger surface area of the combustion space on the piston crown, where temperatures are consequently higher. However, this is beneficial for combustion in compression-ignition engines.

In spark ignition engines it is more desirable to have a flat-crown piston and the combustion chamber in the cylinder head only. The reverse is true for compression-ignition engines. In both cases, as stated before, the shape of the combustion chamber should give the smallest possible surface area to volume ratio.

The shape and wall surface of the exhaust duct in the cylinder head have a definite bearing on the quantity of heat removed by cooling. The exhaust gases leave the cylinder at high temperatures and a considerable quantity of heat is transferred from them to the cylinder head. Therefore, exhaust ducts in the cylinder head should be short and straight. The heat transferred from the exhaust duct wall to the cylinder head does not influence the thermal efficiency of the engine, but it does represent an extra load on the cooling system. The increased work of the cooling system means a reduction in the mechanical efficiency and higher specific fuel consumption.

The removal of this heat from air-cooled engines requires robust finning on the parts adjacent to the exhaust duct, which is a design problem often difficult to solve. In water-cooled engines the radiator area must be increased, and this means additional difficulties particularly in aircraft engines.

The heat transmitted through the walls of the exhaust duct reduces the quantity of heat removed by the exhaust gases. Removal of heat by the exhaust gases is to be preferred to removal by cooling. Part of the heat energy can be recovered from the exhaust gases, while heat carried off by cooling represents a complete loss lowering the performance of the engine.

1. The effect of wall surface quality on the quantity of heat removed by cooling

It is generally known that a polished metallic surface does not absorb heat as readily as a coarse one. Therefore, the walls of the combustion chamber ought to be as smooth as possible. Machining of combustion chamber walls is being introduced and this means lower heat loss and the additional advantage of maintaining an exact compression ratio in all cylinders of the engine. In cast combustion chambers of standard spark-ignition engines up to 500 c.c. displacement the attainable tolerance is $\pm$ 1·6 cm^3, which is a permissible tolerance for compression ratios up to 8:1.

These figures hold for cases of several combustion chambers cast integrally in one common head. In air-cooled engines each cylinder head is cast separately and so higher accuracy can be attained. Combustion chambers cast in steel moulds have very exact shapes, and high accuracy in the volume of the combustion chamber can be ensured by starting the machining of the facing between cylinder-head and cylinder from the combustion chamber wall.

Combustion chambers cast in steel moulds have, apart from smooth wall surfaces, the additional advantage of permitting thermally desirable shapes without machining problems. Conditions of heat transfer to walls are valid also for induction and exhaust manifolds, where smooth walls are preferred also for lower friction losses.

A carbon layer deposited on the walls of the combustion chamber causes a worse heat transfer with a subsequent temperature increase of the inner surface. In such cases higher octane fuels are required and volumetric efficiency may be reduced due to the higher temperature of the inducted air.

Less heat is transferred through a cast iron wall than through an aluminium wall. However, this particular advantage does not compensate for all the other disadvantages of cast iron as a material for pistons and cylinder heads.

2. The effect of engine load on heat transfer to cylinder walls

Heat transfer to cylinder walls depends also on the engine load. The maximum amount of heat is released in the cylinder of engines running at full load when the temperature of combustion products is very high. In engines operating at less than full load, the quantity of heat released and temperature of the gases are lower; consequently less heat is transferred to the walls of the combustion chambers and cylinder.

Fig. 42 illustrates the dependence of the cylinder, cylinder-head, and piston

temperature on the injected quantity of fuel and charging pressure in a compression-ignition Tatra engine having a 120 mm bore, engine speed of 1200 rev/min and the quantity of cooling air remaining constant.

The cooling fan was belt driven from a pulley mounted directly on the crankshaft. A Roots blower driven by an electric motor was used for charging.

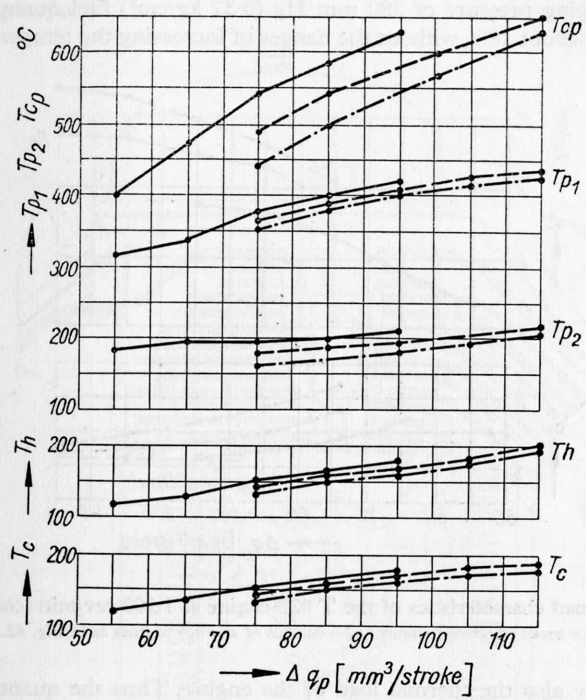

FIG. 42. Load characteristics of the T 928 engine at 1,200 rev/min.

Tcp – temperature of exhaust gases; Tp_1 – temperature of piston at point indicated in Fig. 44; Tp_2 – temperature of gases at point indicated in Fig. 44; *Th* – temperature of cylinder head; T_c – temperature of cylinder. Full line – without supercharging; dashed line – 140 mm w.g. supercharging; dash-and-dot line – 280 mm w.g. supercharging.

Quantity and temperature of air was measured separately for the cylinder and cylinder head. Temperatures of the cylinder head and cylinder were measured by iron-constantan thermocouples located 2 mm from the inner walls at the hottest spot of the cylinder head and cylinder. Piston temperature was measured by iron-constantan thermocouples by the compensating method in positions shown in Fig. 44. When charging at atmospheric pressure the temperature curves of the cylinder head and cylinder show that temperature, and thus the quantity of heat transferred, rises with increasing load more rapidly in the cylinder head, than in the cylinder. In this case the temperature-fuel quantity correlation represents an approximately linear function.

If supercharging is used for an increase of output at constant fuel quantity, the temperature is directly proportional to the charging pressure owing to the increased supply of air. Temperatures of the cylinder, cylinder head, piston, and exhaust gases are reduced owing to the internal cooling effect of scavenging.

At a charging pressure of 280 mm Hg (0·37 kg/cm^2) fuel quantity can be increased by about 18% without the danger of increasing the temperature, and

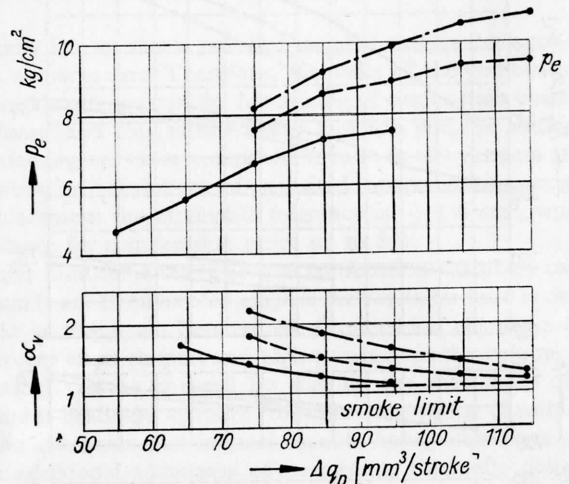

FIG. 43. Load characteristics of the T 928 engine at 1,200 rev/min (continued).
p_e – mean effective pressure; α_v – surplus of air; key to lines as in Fig. 42.

consequently also the thermal load of the engine. Thus the quantity of heat removed from the cylinder by cooling is not increased, but more heat is carried off by the exhaust gases. However, the temperature of the exhaust gases is not necessarily increased, because, due to the volume of scavenging air, the quantity is greater.

The above experiment clearly illustrates the fact, that engine performance can be considerably increased by supercharging without the necessity of improved cooling. However, with a very large performance increase cooling must be intensified.

When the engine load is increased by supercharging, the overall quantity of heat removed by cooling increases, but not in direct proportion to the mean effective pressure. Therefore the percentage loss of heat transferred to the walls is lower. More heat is removed by the exhaust gases, but if exhaust gases are used as a fuel for a supercharger turbo-blower, a part of the heat energy is recovered in the form of useful work and the overall thermal efficiency of the engine is improved.

The correlation between heat removed by cooling and engine load is shown in Fig. 52. According to the diagram the quantity of heat transferred to the cylinder head is directly proportional to the engine load, while the heat transferred to the cylinder walls increases only by the 0·8 th power of the load. The overall quantity of heat removed by cooling increases with the 0·95th power of the engine load.

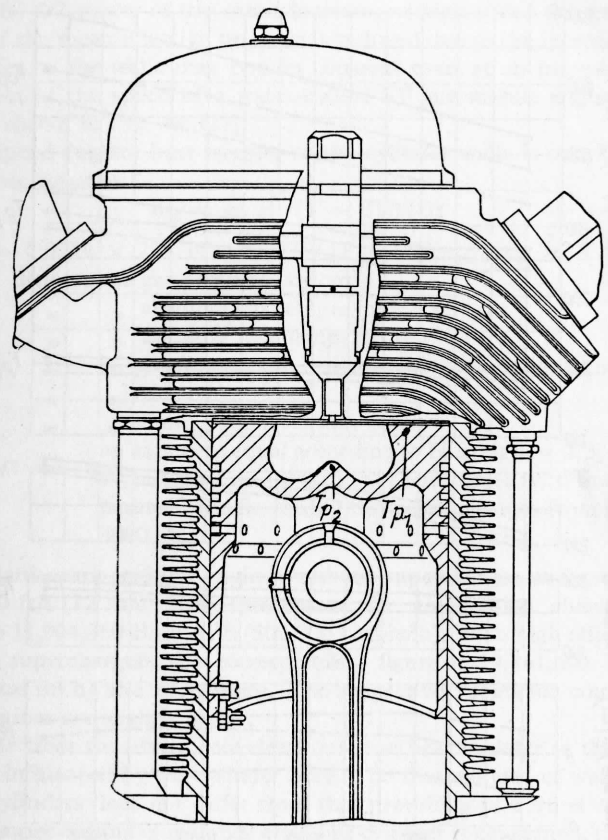

FIG. 44. Sectional view of cylinder and piston of the T 928 four-stroke experimental engine.

The above relation was established by measurements carried out on an air-cooled Ranger cylinder having a bore of 101·6 mm and a stroke 133·4 mm. Displacement was 1·08 litres and compression ratio 6·5 : 1. The cylinder head was made of aluminium and the steel cylinder was fitted with a cast-on aluminium jacket with turned fins. The mean temperature of the cylinder was kept at 146 °C, the temperature of the cooling air at 38 °C.

3. The effect of engine speed on the quantity of heat removed by cooling

Fig. 45 illustrates the dependence of the cylinder head, cylinder, piston, and exhaust gas temperatures on the engine speed at constant fuel quantity

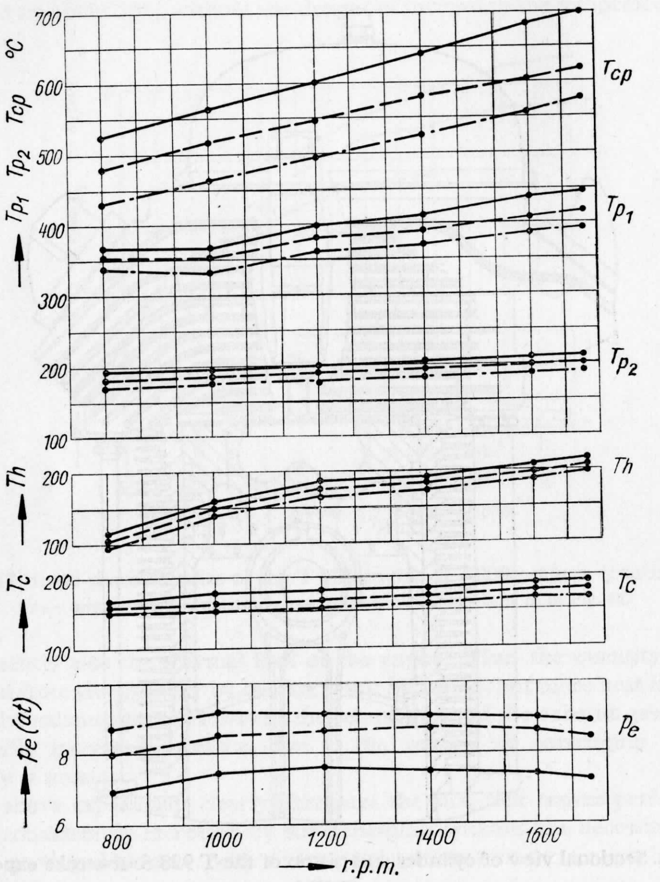

FIG. 45. Speed characteristics of the T 928 engine.
Symbols and lines as in Figs. 42 and 43. Fuel input 88 mm³ per stroke.

and various charging pressures. The temperature of the cylinder head rises more rapidly with an increasing engine speed, than that of the cylinder. Heat removed by the exhaust gases increases with higher speed, while the heat removed by cooling decreases. This fact is explained by the shorter time of

contact between the hot gases and cylinder wall at higher speeds. The total quantity of heat removed does not increase in direct proportion with the increasing speed, but approximately in proportion to the 0·65 power of the speed increase only.

The quantity of heat removed by cooling increases with growing speed, but not in a direct proportion. At lower speed ranges the heat transfer is proportional to 0·8—0·7 power of the speed increase; at high speed this exponent is reduced. If the mean effective pressure is reduced due to the increased speed, heat transfer to the walls may remain constant even at an increased speed.

The effect of the speed of a water-cooled SV automobile engine on heat transfer is shown in Fig. 39. [27].

In low speed engines heat transfer to the cylinder walls is calculated from the following equation:

$$\mathrm{d}Q = q \cdot A \cdot (T - T_w) \cdot \mathrm{d}t$$

where $q = 0{\cdot}0224 \times 228{\cdot}3^n \cdot p^n \cdot T^{1-n}\,(1 + 1{\cdot}24\, c_m)$ (kcal/m² h °C)

- p denotes the gas pressure (kg/cm²)
- T „ „ absolute gas temperature (°C)
- T_w „ „ absolute wall temperature (°C)
- A „ „ surface area in contact with the hot gases (m²)
- t „ „ time (h)
- c_m „ „ mean piston velocity (m/s)
- n „ an exponent equal according to Nusselt $n = 2/3$, according to Jacklisch $n = 0{\cdot}9$ to $0{\cdot}44$ and varies with the gas temperature which, during the cycle, changes from 273 °K to 3000 °K.

In standard spark-ignition engines without supercharger and mean piston velocity 39 ft/s (12 m/s) heat transfer to the combustion chamber walls amounts to 11,904,960 B.t.u./ft²h (300 000 kcal/m²h). With high efficiency engines with superchargers the corresponding figure is 19,841,600 B.t.u./ft²h (500 000 kcal/m² h) and even more. The respective values for compression-ignition engines are slightly lower.

It is clear from the above considerations that heat transfer to the cylinder wall is not influenced by the cylinder bore. The thermal load of walls of large diameter cylinders does not differ from that prevailing at normal conditions. As far as proper cooling of cylinder head and cylinder is concerned, the cooling of large engines depends on proper direction of the cooling air stream and the mastering of the thermal deformation of the cylinder and cylinder head. The cooling of pistons is a more complex problem. Should the total heat transferred to the piston face be removed via the piston rings to the cylinder wall, the rings would be subjected to a very heavy thermal stress and the danger of sticking and seizure would arise. Therefore, part of the heat from the piston crown must be removed directly by the lubricating oil, the temperature difference between the centre and periphery of the piston face must be reduced, so as to obtain thermal relief of the piston rings.