CHAPTER VI

THE FINNED SURFACE

1. Heat transfer from fins to air

Removal of heat from the outer cylinder surface is improved by finning. Fins are variously shaped according to the requirements they have to fulfil and their method of production. The simplest fin has a rectangular cross section with a thickness b constant along its whole length h. A fin of this type is illustrated in Fig. 53. The second widely used type is the fin with a trapezoidal cross section. The thickness b_b at the base of the fin is bevelled down to b_t at the top.

The trapezoidal fin approaches most closely the ideal conditions: by the subsequent removal of heat in the base-top direction the heat flow is diminishing, and the fin thickness is gradually reduced with a better utilization of the material. The chamfering of the fin is required also for easier production of cast and machined fins. However, the bevelling is insignificant in both cases and need not to be considered in calculations.

Although a fin of triangular cross section fulfils all thermal requirements, it is difficult to produce. Thickness at the top is zero and the fin is terminated by a sharp edge. The sharp edge may cause injuries during assembling, or it may be easily bent and damaged.

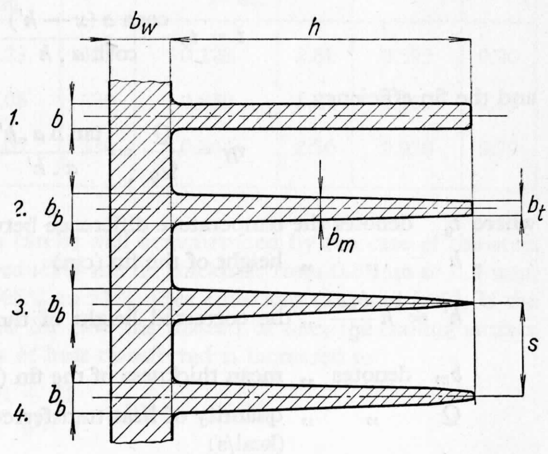


FIG. 53. Basic shapes of fins.
1 — rectangular, 2 — trapeziodal, 3 — triangular, 4 — parabolic.

From the thermal and material point of view a parabolic cross section is the best. For the removal of a given quantity of heat this cross section produces the lightest fin but owing to machining difficulties parabolic fins are not used. Rectangular and trapezoidal cross sections are more usual.

Conditions of heat transfer from fins to air have been studied by numerous authors. An exact calculation of a trapezoidal circular fin is a very complicated matter. Therefore, for practical purposes, this cross section is calculated as a rectangular one having a thickness equal to the mean thickness b_m of the trapezoidal fin. In calculations of rectangular fins we assume a constant temperature along the whole base of the fin and, further, the temperature at any point of the fin being dependent only on the distance from the base. We assume also the surface coefficient of heat transfer being, on the whole surface area, directly proportional to the temperature difference between the fin and surrounding air, and also to the flow velocity of the air. Direction of air flow is considered along the fin area, i.e. to the fin radius.

The calculation of thermal conditions on the fin surface is very complicated and must be based on the introduction of certain ideal assumptions. However, exact calculations are of small value to the designer, because the prerequisites of the calculations are not complied with anyway. We have seen in the preceding chapters, that the heat transfer coefficient q is not a constant valid for the entire surface area of the fin, because the laminar flow changes, at a certain distance, into a turbulent one, and this distance from the leading edge varies in turn with the air velocity. Air temperature will not be the same in the whole interspace between the fins, because temperature differences at the base and the top of the fin are different, and the quantity of heat transferred to air is determined by the temperature difference.

Despite these circumstances it is sometimes desirable to establish thermal conditions and fin efficiency at least approximately. In such cases Pye's formulas are the most suitable. According to Pye the temperature difference between fin surface and air, at a distance x from the base, can be calculated as

$$t = t_o \frac{\cosh a\,(x - h')}{\cosh a \,.\, h} \tag{20}$$

and the fin efficiency

$$\eta_f = \frac{Q}{Q_0} = \frac{\tanh a \,.\, h'}{a \,.\, h'} \tag{21}$$

where t_0 denotes the temperature difference between fin base and air (°C)

h „ „ height of the fin (cm)

$$h' = h \frac{b_m}{2} \quad \text{„ the corrected height of the fin} \tag{22}$$

b_m denotes „ mean thickness of the fin (cm)

Q „ „ quantity of heat transferred from the fin in unit time (kcal/s)

Q_0 „ „ quantity of heat transferred from the fin in unit time at a temperature difference of t_0 (kcal/s)

cosh, tanh are hyperbolic functions

$$a = \sqrt{\frac{2\,q}{\lambda\, b_m}}$$

λ denotes the coefficient of thermal conductivity of the material o the fin (kcal/m h °C)

Calculations with the above equations start from the basic equation of thermal equilibrium

$$\frac{d^2t}{dx^2} = \frac{2\,q\,t}{\lambda\, b_m} = a^2 \,.\, t$$

and from the conditions that at the base of the fin, i.e. for $x = 0$, $t = t_0$; and at the top of the fin, i.e. for $x = h'$, $dt/dx = 0$.

Thus the fin efficiency is the ratio of heat actually transferred to the heat which would be transferred if temperature on the entire fin surface were equal to O_0, i.e. the temperature at the fin base. The larger the fin, the worse is its efficiency. Excessively long fins have a bad efficiency and their weight is not properly utilized. Greater fin lengths are permissible with materials having a good thermal conductivity. Normally, however, production considerations and not efficiency, are the factors limiting the length of fins. Some typical types of fins used in practice are shown in Table 11. The diagram in Fig. 54 serves as a practical aid for rapid calculations of fin efficiency.

Table 11

Comparison of Different Cooling Fins

Material	Height cm	Thickness cm	λ	$a = \sqrt{\frac{2 \cdot q}{\lambda \cdot b_m}}$	h′	ah′	η_f
Aluminium	2.5	0.23	175	0.228	2.61	0.595	0.90
Steel	1.6	0.08	39	0.830	1.64	1.360	0.65
Copper	2.54	0.05	330	0.364	2.56	0.930	0.79

The role of fin thickness can be well demonstrated by the case of the steel fin given in Table 11. By reducing the fin thickness from 0.8 mm to 0.4 mm, efficiency is reduced from 65% to 51%. [See equations (21) and (22)]. If the same weight of metal is used for two fins instead of one, the cooling surface is doubled and the quantity of heat transferred is increased to

$$\frac{51 + 51}{65} \times 100 = 157\ \%,$$

i.e. the same amount of steel removes 57% more heat. The conclusion that thinner fins mean better utilization of the material is thus evident. Fin thickness is limited by production problems rather than by thermal conditions, even in the case of steel which is a material with notoriously bad thermal conductivity. A full utilization of thermal conductivity would, in such metals as aluminium or copper, lead to extremely thin and practically unusable fins. Therefore, fin thickness is chosen as small as permitted by the production process. The number of fins should be the largest possible, but the interspace between the fins must be large enough to ensure a satisfactory flow of cooling air.

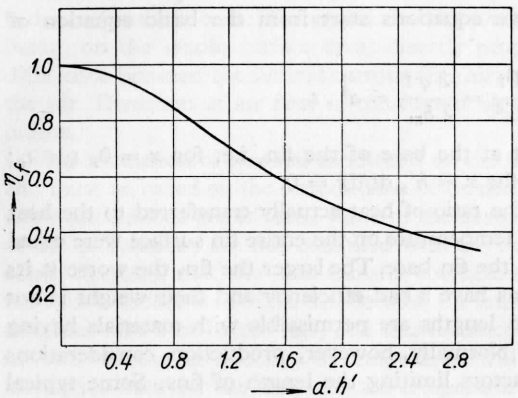


FIG. 54. Efficiency of cooling fins according to equation (21).

2. Spacing of fins

The lower limit of the interstice is determined by the cooling conditions. In the case of a motor-cycle engine with a low cooling air velocity, the spacing of fins must ensure an air flow to reach the base of the fins. Distribution of velocities in small and large interstices is shown in Fig. 55. However, it should be borne in mind that the diagram applies to small air velocities and to a freely exposed cylinder having no cowling. In this case a spacing of 8—12mm is recommended.

If a stronger air flow is available for cooling, fins are more densely spaced. Critical conditions depend on the thickness of the boundary layer at the fin surface. The influence of the boundary layer upon fin efficiency was established by tests carried out on straight fins 152 mm long, 25 mm high and 0·5 mm thick. Spacing varied from 12 to 2 mm. The frontal edges were carefully rounded and air flow direction was parallel to the fins. These experiments proved, that a rapid deterioration of fin efficiency started at spacings below 2·5 mm at varying air velocities. Thickness of boundary layers, calculated for the given test velocities and distances of transitions from laminar to turbulent

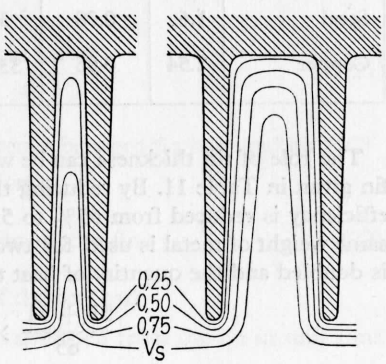


FIG. 55. Velocity of air with a small and a wide interstice between the fins.

flow, are shown in Table 12. The shape of the boundary layer is illustrated in Fig. 56. It can be seen from Table 12 that the critical interstice is smaller than twice the thickness of the turbulent layer at the end of the fin. Therefore, it can be safely assumed that a contact established between two adjacent turbulent layers has no adverse effect.

More important is the thickness of the laminar boundary layer. It must be

Table 12

Thickness of Boundary Layers for Straight Fins, 152.4 mm Long, Exposed to Air Stream of Velocities 140 and 254 km/h.

Transition of laminar into turbulent flow is assumed to take place at Reynolds number = 2 × 10⁵. (See Fig. 56). Values are given in mm.

Air velocity	140 km/h = 39 m/s	254 km/h = 70 m/s
Distance between leading edge and point of transition from laminar to turbulent flow	114.4	63.5
Thickness of turbulent boundary layer at the fin root	4.32	4.06
Thickness of laminar boundary layer at the point of transition	1.016	0.508
Thickness of turbulent boundary layer at the point of transition	3.30	1.778

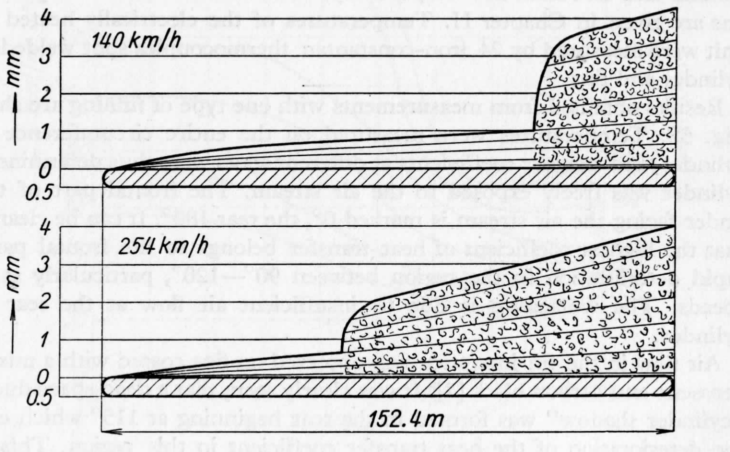


FIG. 56. Shape of boundary layer for speeds of 140 km/h and 254 km/h. Thickness of layer is drawn to a scale 10 times that of the length.

remembered that velocity in the turbulent boundary layer does not drop below 0·9 v_m as long as the distance from the surface is not less than half the thickness of the boundary layer.

A rapid deterioration of flow conditions occurs at an interstice smaller than two thicknesses of the laminar boundary layer. At an air speed of 87 m.p.h. (140 km/h) the double thickness of the laminar boundary layer is 2 mm. Fig. 57 illustrates the interdependence between fin efficiency and spacing of fins; efficiency drops rapidly at interstices below 2·5 mm. However, with a decreasing spacing, the cooling surface increases, and the overall quantity of heat removed begins to diminish only at spacings below 2·5 mm. This is clearly shown in Fig. 77.

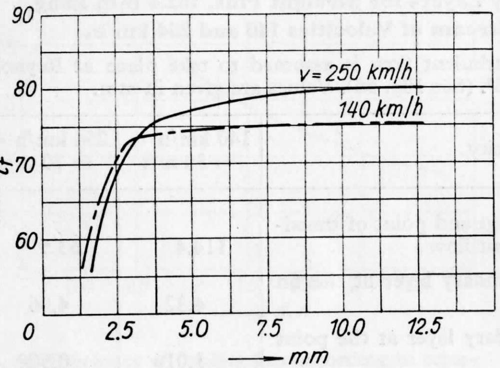


FIG. 57. Influence of interstice upon fin efficiency.

3. The surface coefficient of heat transfer

Equation 11 in Chapter II enables only an approximate calculation of the heat transfer coefficient q. The exponent of the mean air velocity v_m varies with the proportion of the surface area exposed to laminar and turbulent flow respectively. Dimensions of the test cylinder and fins are given in Chapter II. Temperatures of the electrically heated testing unit were measured by 24 iron-constantan thermocouples spot welded to the cylinder and fins.

Results obtained from measurements with one type of finning are shown in Fig. 58. Temperatures were measured on the entire circumference of the cylinder, and transfer coefficients at different spots were thus determined. The cylinder was freely exposed to the air stream. The frontal part of the cylinder facing the air stream is marked 0°, the rear 180°. It can be clearly seen that the highest coefficient of heat transfer belongs to the frontal part with rapid deterioration in the region between 90°—120°, particularly at higher speeds. The worsening is due to insufficient air flow at the rear of the cylinder.

Air flow in the fin interstice was observed on fins coated with a mixture of kerosene and carbon black [7]. It was clearly seen, that a dead space due to the "cylinder shadow" was formed in the rear beginning at 115° which explains the deterioration of the heat transfer coefficient in this region. This shows a bad utilization of fins on the rear of a freely exposed cylinder. See Fig. 59.

In our further considerations we shall use the mean surface coefficient of heat transfer

$$q = \frac{Q}{A \cdot t} \tag{23}$$

where Q denotes the total heat transferred from the cylinder
A „ „ total cooling surface area
t „ „ mean temperature difference between cylinder surface and cooling air.

The value of the mean surface coefficient of heat transfer obtained from measurements on rectangular and trapezoidal fins are given in Fig. 60, where they are plotted against fin spacing s and air velocity v_m respectively. The following conclusions can be drawn from the test results.

The influence of fin height. On a freely exposed finned cylinder with low fins the mean air velocity on the fin surface is higher, and so is the coefficient q. Tests carried out on cylinders with fins of various height and spaced 3·8 to 6.35 mm apart have proved that q deteriorates rapidly up to a height of 10 mm. A further increase of the height of the fins causes only a slight increase in the

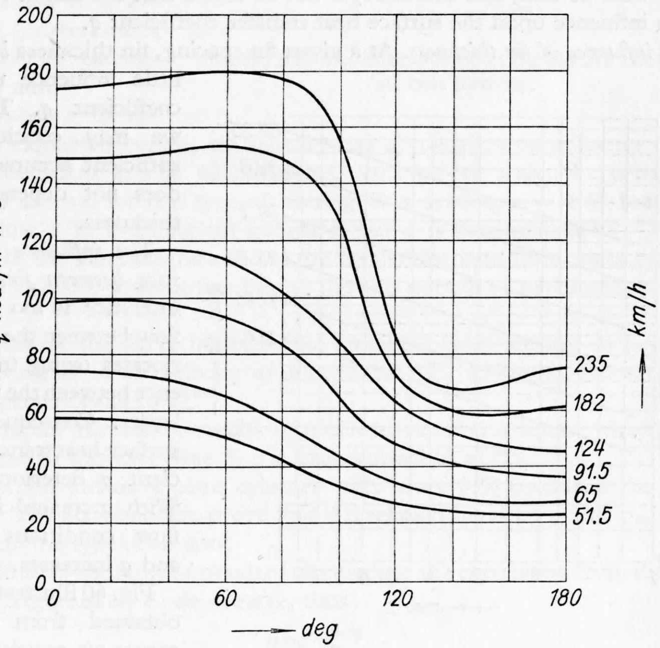


FIG. 58. Values of the coefficient of heat transfer q around the circumference of a cylinder without cowling.

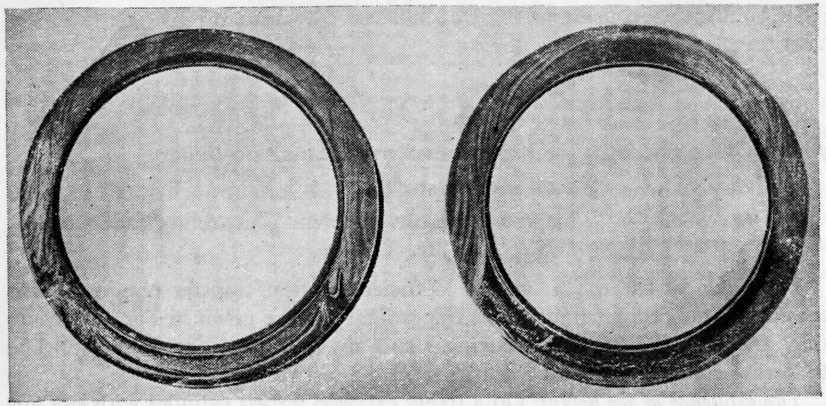


FIG. 59. Traces of air flow in the interstice between fins.

deterioration of q. In cylinders fitted with cowlings fin height had even smaller effects than in freely exposed cylinders. Fins used in practice are invariably higher than 10 mm, and therefore, it can be stated that the height of the fins has no influence upon the surface heat transfer coefficient q.

The influence of fin thickness. At a given fin spacing, fin thickness b has only little influence upon the coefficient q. Therefore, we may conclude with sufficient accuracy that q does not depend on fin thickness.

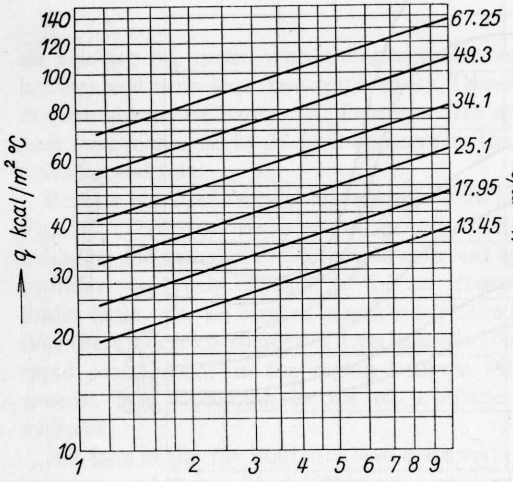


FIG. 60. Dependence of the mean surface coefficient of heat transfer on the width of the interstice and on the velocity of air.

The influence of interstice between fins. If the interstice is too small, air flow between the fins deteriorates owing to interference between the boundary layers. Consequently the surface heat transfer coefficient q deteriorates also. With increased interstice, flow conditions improve and q increases.

Fig. 60 illustrates results obtained from measurements on rectangular and trapezoidal fins of various height and thickness. The straight line showing the correlation between q and fin spacing s represents the mean values of measurements on different fins. Individual values deviated from this line by 8 % at the maximum, thus proving that the influence of height and thickness of the fin upon q is negligible. The coefficient is influenced mainly by the fin spacing. The slope of the lines in this diagram drawn in log-log scale is 0·322 signifying the relation

$$q \propto m^{0\cdot322}.$$

In cylinders with cowlings fins are spaced very closely, and it is interesting to examine the influence of spacings below 1 mm. This dependence is illustrated in Fig. 61 where q is plotted against s in a logarithmic scale. It is evident that q deteriorates considerably at spacings below 1·5 mm.

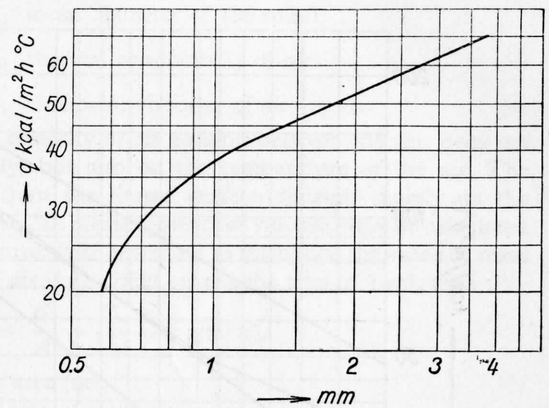


FIG. 61. Effect of interstice upon the surface coefficient of heat transfer.

The influence of air velocity. As explained before, air velocity has a considerable influence on the transfer coefficient q. Fig. 62 illustrates, on a log-log scale, the correlation between q and v_m for one type of finning at a spacing $s = 4{\cdot}45$ mm. The slope of line 4 in the diagram indicates that q is proportional to $v_m^{0\cdot796}$.

Line 3 in the same diagram shows the results obtained from measurements on a plain cylinder without fins having the axis normal to the direction of the air flow.

The break of the curve indicates the transition from laminar to turbulent flow occuring at an air velocity of about 28 m.p.h. (45 km/h). For turbulent flow q is proportional to $v_m^{0\cdot66}$.

The diagram also shows results of measurements on a square plate 152 × × 152 mm. In this case (line 2) q is proportional to $v_m^{0\cdot725}$.

Test carried out on a plain cylinder with cover [35] produced the result $q \propto v_m^{0\cdot57}$. The above results prove that the relation between q and v_m varies greatly with the type of air flow.

For cylinders with fins of standard dimensions the correlation from equation 11 can be regarded as a safe average, thus

$$q \propto v_m^{0\cdot73}$$

If the cylinder casing bears tightly on the fin edges, the flow in the interstice between the fins proceeds under conditions similar to those prevailing in a bent tube of a rectangular cross section. The equivalent diameter of a round tube is calculated from the formula

$$d_e = \frac{4sh}{2\,(s+h)} = \frac{2sh}{s+h} \tag{24}$$

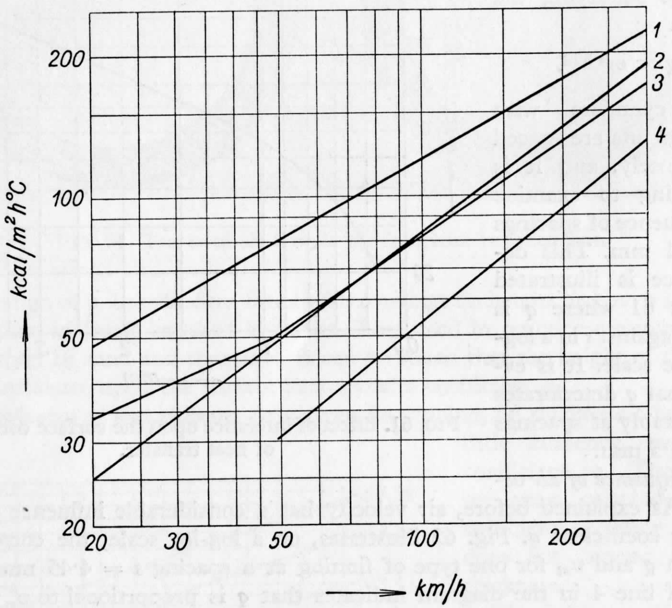


Fig. 62. Effect of air velocity upon the surface coefficient of heat transfer.

1 — measured on a plain cylinder (Löhner) of a diameter 160 mm with air flow normal to the cylinder axis, *2* — measured on an even plate 152.4 × 152.4 mm (Pye), *3* — measured on a plain cylinder to different diameters with air flow normal to the cylinder axis (Biermann), *4* — measured on a finned cylinder with an interstice between fins s = 4.4 mm (Biermann).

For a straight tube of circular cross section the transfer coefficient q is calculated, according to Nusselt, from the equation

$$q = 0.0346\,\frac{\lambda}{d}\left(\frac{x}{d}\right)^{-0.05}\left(\frac{v_m\,.\,d}{k}\right)^{0.79} \tag{25}$$

A different relation is given by Kraussold as follows:

$$q = 0{\cdot}024\,\frac{\lambda}{d}\left(\frac{v_m\,d}{\nu}\right)^{0{\cdot}8}\left(\frac{\nu}{k}\right)^{0{\cdot}37} \tag{26}$$

According to Jeschke heat transfer coefficients in spirals made of round tubes are calculated from the equation

$$q = \left(0{\cdot}0392 + 0{\cdot}139\,\frac{d}{D}\right)\frac{\lambda}{d}\left(\frac{v_m\,d}{k}\right)^{0{\cdot}76} \tag{27}$$

where d denotes the internal diameter of the tube
D „ „ mean diameter of the spiral
$k = \dfrac{\lambda}{c_p\,.\,\gamma}$ „ „ thermal conductivity of air
ν „ „ kinematic viscosity of air.

Mass velocity of air. The quantity of air passing between the fins does not depend on the velocity only, but also on the temperature of the air. The quantity of heat removed from the finned surface depends mainly on the weight of air passed. Therefore, all measurements of volume must be calculated to one comparative temperature. It is beneficial to introduce the value of mass velocity, i.e. the quantity of air (kg) which passed the area of 1 m², thus

$$v_w = \frac{G}{A} = \frac{V\,.\,\gamma}{A}\ \text{(kg/m}^2\text{ s)},$$

where A denotes the flow area (m²)
G „ „ flow rate of air (kg/s)
V „ „ volume flow rate of air (m³/s)
γ „ „ specific weight of air (kg/m³)

Fig. 63 illustrates the influence of mass velocity of air upon the surface coefficient of heat transfer q. Measurements were carried out on various types of finnings and fin dimensions according to table 13 [55]. It is evident, that q changes only very slightly between spacings 1·2—3·3 mm, it deteriorates rapidly at smaller spacings, and increases at larger ones. In this case the transfer coefficient q is calculated from the temperature difference between the *inlet* temperature of the air and the mean temperature of the fins. The resultant correlation is

$$q \propto v_m^{0{\cdot}667}.$$

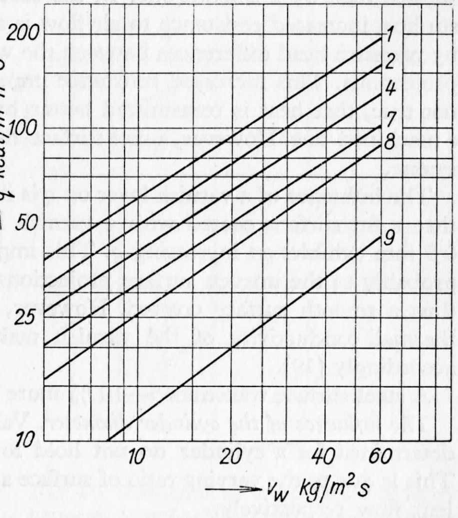


Fig. 63. Influence of "mass velocity" of air upon the surface coefficient of heat transfer q.

If we calculate with the mean air temperature, i.e.

the mean value of the inlet and exit temperature of the air, the above relation changes to

$$q \propto v_m^{0 \cdot 544}.$$

In this case the absolute value of q changes evidently too; it is larger because the same quantity of heat removed is calculated at a smaller temperature difference. For the assessment of heat transfer coefficients it is important to know if they are related to the inlet or mean temperature of the air.

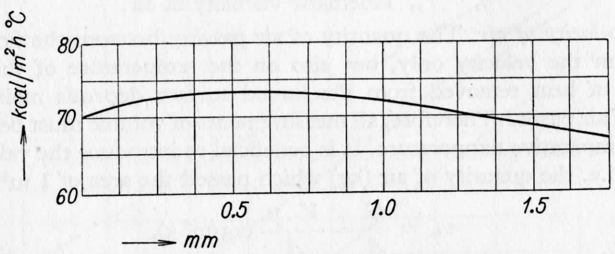


FIG. 64. Influence of the thickness of varnish layer upon the surface coefficient of heat transfer q.

The influence of surface quality. As mentioned before, the heat transfer coefficient is proportional to surface friction. Heat is better removed by a rough surface than by a smooth one. In the case of large projections of the rough surface, increased resistance to air flow is not caused by surface friction, but by pressure head differences between the windward and leeward edges of the projections. This increased resistance impedes heat transfer. Therefore, it is not true, that heat is transmitted better by a roughly cast surface than by a machined one. However, a mat surface has definite advantages over a glossy one.

The influence of a varnish layer on q is illustrated in Fig. 64. It can be seen that a fin surface coated with a varnish layer having a thickness of up to 0·5 mm exhibits an increasing q. This improvement of the coefficient is due probably to the uneven surface projections being levelled by the varnish and thus a smooth surface created. However, at thicker varnish layers the bad thermal conductivity of the varnish makes itself felt, and q deteriorates accordingly [19].

A steel surface transmits 5—10 % more heat than aluminium.

The influence of the cylinder diameter. Values of surface transfer coefficients determined for a cylinder do not hold for cylinders of different diameters. This is due to the varying ratio of surface areas exposed to laminar and turbulent flow respectively.

To date only few data are available on the influence of cylinder diameter on q. By measuring the heat removed from two similarly finned cylinders

having diameters of 93 and 161 mm respectively, the following correlation was established:

$$q \propto \frac{1}{D^{0 \cdot 25}}.$$

When measurement results obtained from one cylinder are to be applied to another cylinder, first we must establish the scale factor

$$m = \frac{D_x}{D_t},$$

where D_x denotes the diameter of the calculated cylinder and
D_t „ „ diameter of the test cylinder.

All test results are then recalculated for the new cylinder by multiplication by the scale factor, i.e.

$$D_x = D_t \cdot m,$$
$$b_x = b_t \cdot m,$$
$$h_x = h_t \cdot m,$$
$$s_x = s_t \cdot m,$$
$$v_x = v_t \cdot m,$$
$$\text{and } q_x = \frac{q_t}{m}$$

The influence of the air stream direction. If the air flow is not parallel to the plane of fins, the continuity of the air flow between the fins is disrupted. The investigation of air traces on fin surfaces coated with a mixture of kerosene and lamp black produced evidence of an intensive turbulence in the fin interstice [19]. Heat transfer is considerably augmented by this turbulence. Fig. 65 illustrates the possibility of reducing the temperature difference between fin and air (in %) for a given cylinder and a fixed quantity of heat transferred in the case of an air stream inclined by 45° from the plane of the fins. The denser the finning, the smaller is the influence of the inclined air stream.

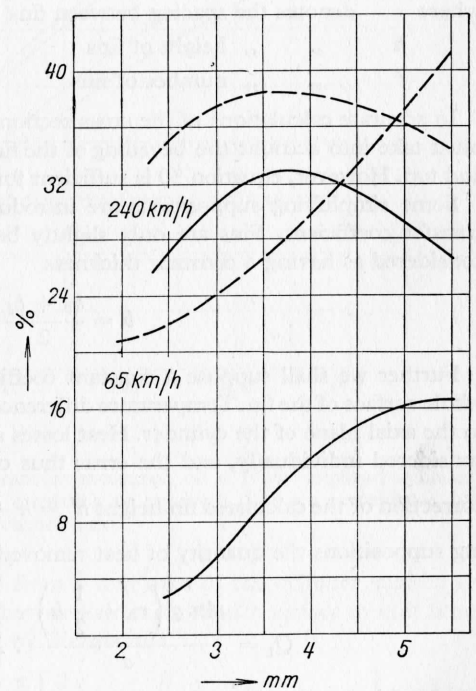


FIG. 65. Reduction of temperature difference between air and cylinder surface plotted against spacing of fins.

4. The coefficient of heat transfer from the fin base surface area

If large quantities of heat have to be removed from a limited cylinder surface area, a maximum total cooling surface must be obtained by a suitable grouping of the fins with due consideration of the high heat transfer coefficient q. Calculations are simplified and better orientation is secured by using transfer coefficients related to the fin base surface area.

The fin base surface area of a finned cylinder is

$$A_b = 2\pi \,.\, r_b \,.\, f \,.\, S \tag{29}$$

where r_b denotes the radius of the cylindrical surface of the fin base
f „ „ number of fins
S „ „ overall spacing; $S = s_m + b$.

The transition area of cooling air between the fins A_t is calculated from the equation

$$A_t = 2 \,.\, f \,.\, s \,.\, h \tag{30}$$

where s denotes the spacing between fins
h „ „ height of fins
f „ „ number of fins

In accurate calculations of the cross section of the area between the fins we must take into account the bevelling of the fin and the chamfering at the base and top. However, equation 30 is sufficient for calculations.

Some simplifying suppositions are introduced in calculations of the base transfer coefficient. Fins are only slightly bevelled and, therefore, they are considered as having a constant thickness

$$b = \frac{b_b + b_t}{2}$$

Further we shall suppose a constant coefficient of heat transfer q for the whole surface of the fin. Temperature difference Δt is distributed symmetrically to the axial plane of the cylinder. Heat losses at the top edge of the fin are not considered individually, and the error thus created is compensated for by a correction of the calculated fin height $h' = h + \frac{b_t}{2}$. Subject to these simplifying suppositions the quantity of heat removed from a circular fin, is

$$Q_1 = \frac{4\pi\, q \left(r_b + \frac{1}{2} h\right)}{a} \,.\, \Delta t_b \,.\, \tanh a \,.\, h',$$

where Δt_b denotes the mean temperature difference at the fin base, and

$$a = \sqrt{\frac{2\, q}{\lambda\, b_m}}$$

The quantity of heat Q_2 removed from the cylinder surface between two fins is calculated from the equation

$$Q_2 = 2\pi \,.\, r_b \,.\, s_b \,.\, q \,.\, \Delta t_b$$

provided, that q has the same value for both the cylinder and fin surface.

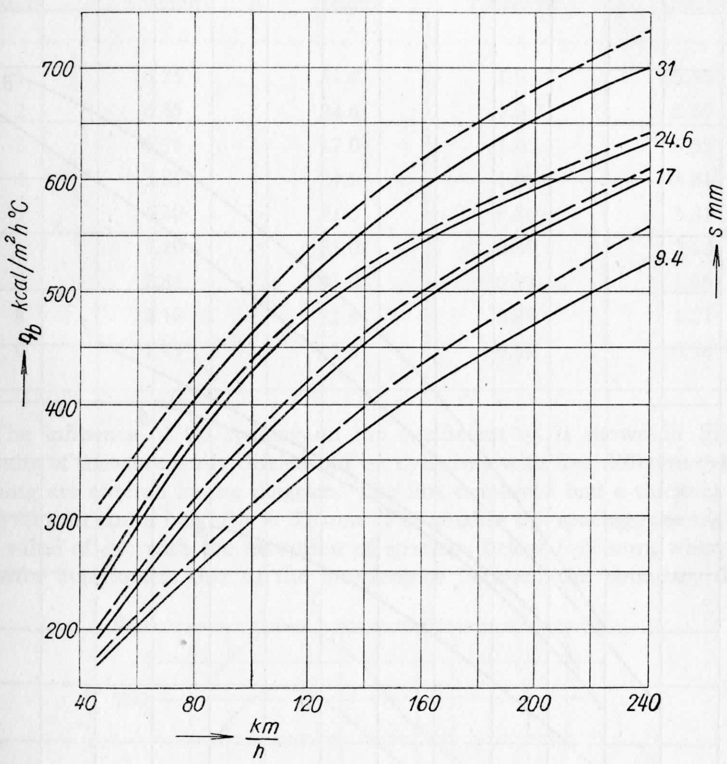


FIG. 66. Basic coefficient of heat transfer measured on a freely exposed cylinder without cowling. Values calculated according to equation (31) are represented by dashed lines.

The quantity of heat removed from a unit area of the cylinder surface at a temperature difference of 1 °C between air and cylinder surface in unit time is given by the equation (applied by Biermann):

$$q_b = \frac{Q_1 + Q_2}{2\pi \,.\, r_b \,.\, b \,.\, \Delta t_b} = \frac{q}{b}\left[\frac{2}{a}\left(1 + \frac{h}{2r_b}\right) \tanh a \,.\, h' + s_b\right] \tag{31}$$

Fig. 66 illustrates base transfer coefficients q_b measured on a finned cylinder without cowling having fins spaced at 6·35 mm and of various heights. For comparison calculated values of q_b are entered in the diagram also. It is evident that results obtained from equation 31 correspond very well with actual measurements [55].

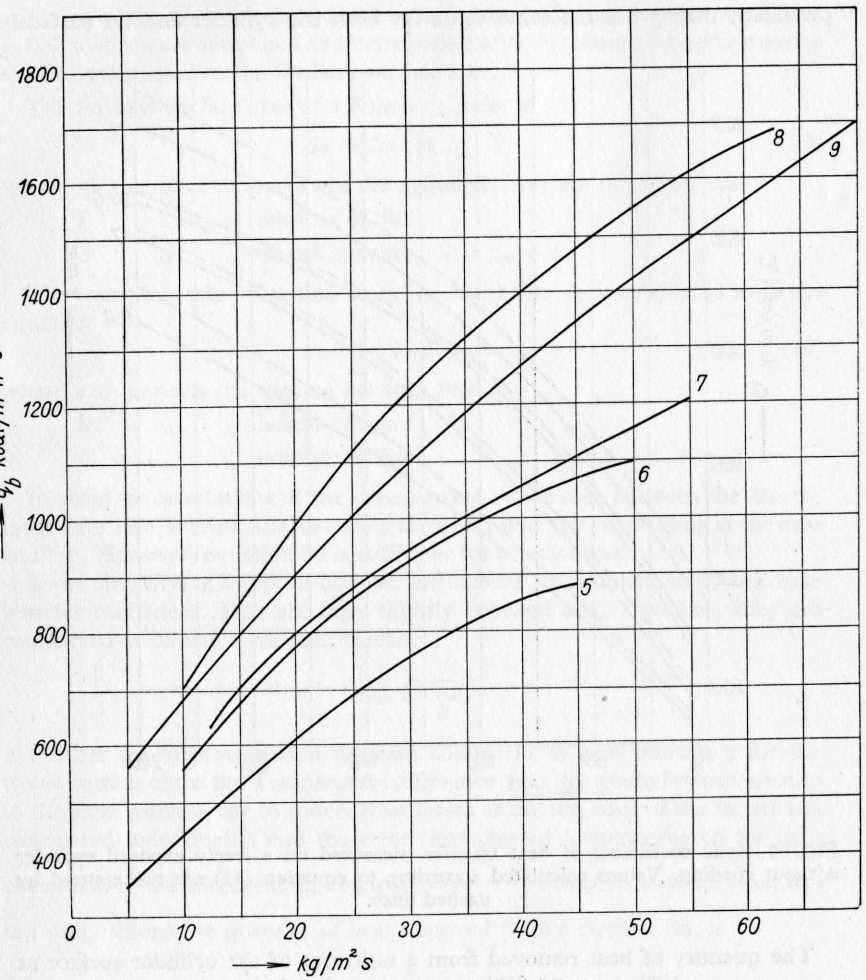


FIG. 67. Relationship between basic coefficient of heat transfer and "mass velocity" of air in a cylinder with cowling.

The correlation between coefficients q_b and mass velocity of air are illustrated in Fig. 67 for the case of a cylinder with cowling where fin spacing varies and fin height remains constant. Dimensions of fins used for the

measurements are given in Table 13. Calculated and measured values are also in agreement.

Table 13

Fin Dimensions of the NACA 587 Test Cylinder (in mm)

Mark	Spacing	Height	Thickness	Interstice
1	6.35	31.0	1.0	5.35
2	6.35	24.6	1.0	5.35
3	6.35	17.0	1.0	5.35
4	3.81	24.6	1.0	2.81
5	4.20	31.0	0.89	3.31
6	3.49	31.0	0.89	2.60
7	2.84	31.0	0.89	1.95
8	2.10	31.0	0.89	1.21
9	1.45	31.0	0.89	0.56

The influence of fin spacing on the coefficient q_b is shown in Fig. 68. Results of measurements carried out on cylinders with five different types of finning are entered in the diagram. The fins employed had a thickness $b = 0{\cdot}89$ mm and a height $h = 31$ mm. The smaller the spacing, the higher is the value of q_b; with the exception of spacings below 1·45 mm, where heat transfer deteriorates due to the interference between the boundary layers.

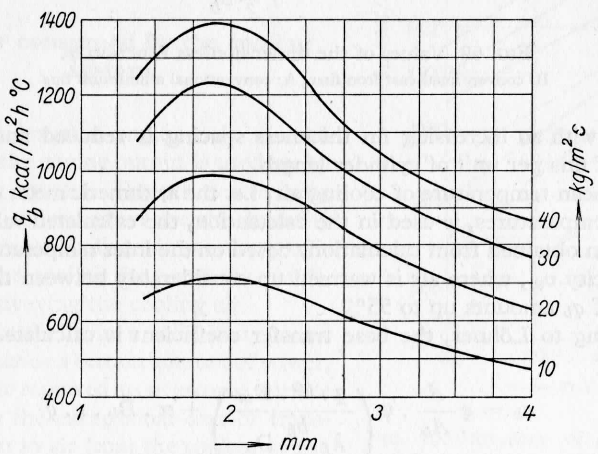


FIG. 68. Influence of fin spacing upon the basic coefficient of heat transfer at a given "mass velocity" with finning according to Table 13, h = 31 mm, b = 0.89 mm.

These results are in full agreement with conclusions arrived at in the preceding discussions. Dimensions of fins used in tests are again given in Table 13. In this case the base transfer coefficient q_b is calculated to the inlet temperature of the cooling air.

Fig. 68 illustrates the correlation between fin spacing and q_b at a particular mass velocity v_w. Maximum q_b is obtained at an optimal spacing of about 2 mm. The optimum distance between fins depends on the fin thickness.

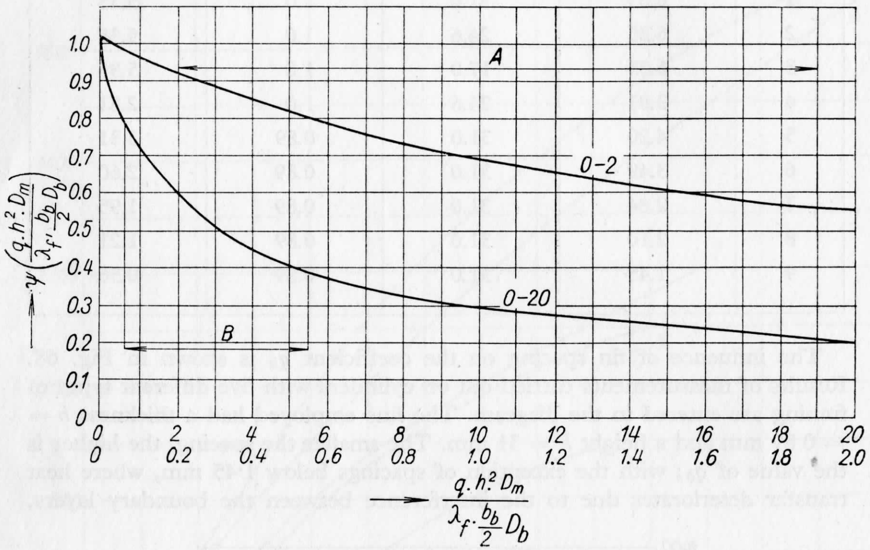


FIG. 69. Values of the dimensionless function ψ.
B: conventional cast iron fins. A: conventional aluminium fins

However, with an increasing fin thickness spacing is reduced and so is the number of fins per unit of cylinder length.

If the mean temperature of cooling air, i.e. the arithmetic mean of the inlet and exit temperatures, is used in the calculation, the calculated value of q_b is higher than obtained from calculations based on the inlet temperature. At low mass velocity v_w, where air is warmed up considerably between the fins, the increase of q_b amounts up to 55%.

According to Löhner, the base transfer coefficient is calculated from the formula:

$$q_b = q \frac{A}{A_b} \cdot \psi \left(\frac{q \cdot h^2 \cdot D_m}{\lambda_f \dfrac{b_b}{2} D_b} \right) + \pi \cdot D_b \cdot s \cdot q \qquad (32)$$

where A denotes the surface of one fin

A_b denotes the cross section area of a fin base, $A_b = \pi \cdot D_b \cdot b_b$
D_m „ „ mean fin diameter; $D_m = D_b + h$
D_b „ „ diameter of the fin base
λ_f „ „ coefficient of thermal conductivity of the fin

The function ψ of the dimensionless term $\left[\frac{q \cdot h^2 \cdot D_m}{\lambda_f \dfrac{b_b}{2} D_b}\right]$ is plotted in Fig. 69.

This function expresses the fin efficiency η_f, by which we must multiply all results obtained from calculations at which the assumption of equal temperature of base and fin surface was made. A small thermal conductivy λ_f will suffice in cases of small values of q and h.

The second term in equation 32 represents the quantity of heat transferred by the cylinder surface in the fin interspace. With densely spaced, high fins this term may be omitted without a grave error being committed. For the calculation of the transfer coefficient q Löhner uses the temperature difference between the mean temperature of the fin and the mean temperature of cooling air.

Values of the base transfer coefficient q_b measured on a cylinder with cover are plotted against air velocities v_m in Fig. 70. Fins employed for this test were triangular, i.e. with a sharp edge at the top. Dimensions and markings of the fins employed are given in Table 14. The cylinder diameter at the fin base was D_b = = 160 mm.

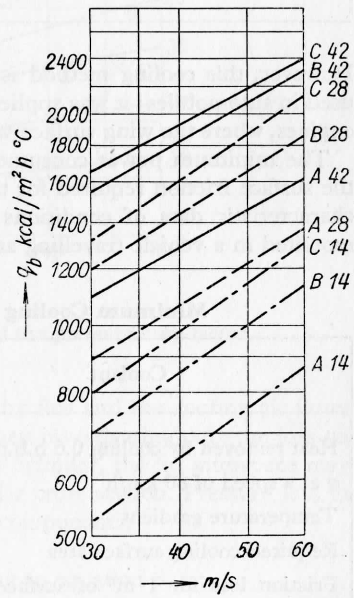


FIG. 70. Influence of air velocity and fin height upon the coefficient of heat transfer in a cylinder with cowling.

5. Power consumed by the cooling system

In a well designed engine only a small portion of the engine output is used up by the cooling system. Factors determining this power consumption are the quantity and pressure of cooling air required, efficiency of the cooling fan, and the conditions of conveying the cooling air.

Heat transfer from a solid body to air always demands a certain amount of power, at least that required to overcome surface friction. In the exceptional case of transmitting heat to air from the surface of the car body, friction becomes an unavoidable part of the air resistance of the vehicle.

Table 14

Fin Dimensions of the Löhner Test Cylinder (in mm)

Mark	Spacing	Height	Thickness at the root
A 42	8	42	4
A 28	8	28	4
A 14	8	14	4
A 7	8	7	4
B 42	5	42	4
B 28	5	28	4
B 14	5	14	4
B 7	5	7	4
C 42	4	42	2
C 28	4	28	2
C 14	4	14	2
C 7	4	7	2

However, this cooling method is complex and unpractical and it was never used in automobiles; it was applied in certain racing aircraft with water-cooled engines, where the wing surface was used as a radiator.

The minimum power consumed by the cooling system is calculated from the surface friction required for the transfer of heat to air. Table 15 includes characteristic data of conditions prevailing in cooling a 100 b.h.p. engine mounted in a vehicle travelling at 50 m.p.h. (80 km/h).

Table 15

Minimum Cooling Input for a 100 b.h.p. Engine

Coolant	Units	Air	Water
Heat removed by cooling 0.6 b.h.p.	kcal/h	38,000	38,000
q at a speed of 80 km/h	kcal/m² h °C	100	100
Temperature gradient	°C	160	60
Required cooling surface area	m²	2.38	6.33
Friction loss for 1 m² of surface area at 80 km/h	kg/m²	0.1	0.1
Loss by friction of air	b.h.p.	0.74	1.97
Minimum input for cooling	%	0.74	1.97

Supposing the value of the surface transfer coefficient q being 100 kcal/m²h °C at a travelling speed of 50 m.p.h. (80 km/h), the part of engine performance required to overcome surface friction at a temperature difference of 60 °C amounts to 1·97% of the b.h.p.; at a temperature difference of 120 °C the figure would be reduced to 0·74 % of the b.h.p. However, a power consumption as low as the latter is never attained in vehicles.

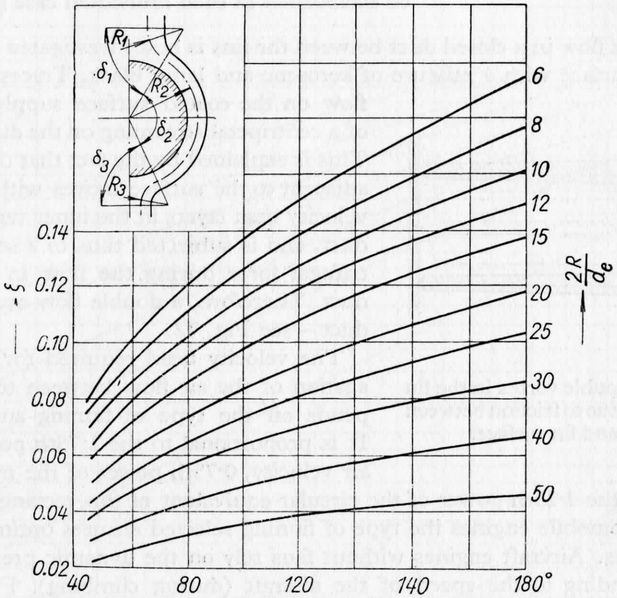


FIG. 71. Chart for the determination of the pressure drop factor.

In practice the cooling surface is formed by fins and the cooling air flows through the narrow space between them. When investigating only the loss on pressure head caused by the flow around the cylinder, the fin interspace may be considered to be a bent tube of rectangular cross section. Pressure loss in the bent tube is composed by the following components:

a) the component Δp_1 calculated according to equation 8, i.e.

$$\Delta p_1 = \frac{\lambda \cdot l}{d_e} \frac{\varrho}{2} v_m^2,$$

where d_e is the equivalent diameter calculated from equation 24,

b) the component Δp_2 representing the head loss in the bend:

$$\Delta p_2 = \gamma \xi \frac{v_m^2}{2g} \tag{33}$$

where v_m denotes the mean air velocity in the duct (m/s)

ξ „ „ factor of pressure drop in the bend, plotted in Fig. 71. The respective angles and radii must be established in each individual case [5].

The air flow in a closed duct between the fins is best investigated by coating the fin surface with a mixture of kerosene and lamp black. Traces of the air flow on the coated surface supply evidence of a centripetal streaming on the duct surface. This is explained by the fact that the air layer adjacent to the surface moves with a smaller velocity than layers in the inner region of the duct, and is subjected thus to a smaller centrifugal force during the flow in a circular duct. Therefore, a double flow occurs in the duct – see Fig. 72.

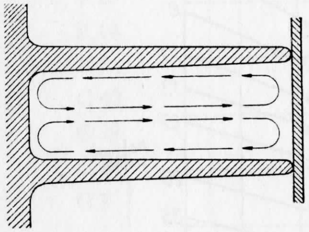


FIG. 72. Double vortex in the fin interstice due to friction between air and fin surface.

The velocity head required for the propagation of the air flow between the fins depends on the type of finning and spacing. It is proportional to the 1·75th power of the air velocity, 0·75th power of the mass of the air, and the 1·25th power of the circular equivalent of the rectangular duct.

In automobile engines the type of finning selected ensures optimal cooling conditions. Aircraft engines without fans rely on the dynamic pressure head corresponding to the speed of the aircraft (during climbing). Fin spacing and type of finning must be determined in accordance with the dynamic pressure available and without regard to the resultant weight of fins. Were an aircraft engine equipped with a fan, an optimal fin spacing could be applied independently of the aircraft speed.

The correlation between the surface transfer coefficient q and the velocity head in the fin interspace Δp is shown in Fig. 73. Minor losses occuring at the inlet and exit of air are disregarded in this diagram. A rapid deterioration of cooling conditions in interspaces below 0·5 mm is evident. Measurements were carried out on a cylinder having a fin base diameter of 118 mm and equipped with a cowling in close contact with the fin edges [6].

In Fig. 74 mass velocity of air is plotted against velocity head. It is evident that at a given velocity head the quantity of air passed between the fins decreases with diminished spacing.

Small fin thickness and small interspace are prerequisites of economical cooling. This means a smaller quantity of cooling air but higher velocity heads. Therefore, it is imperative to prevent air losses through leakages in the cowling.

With a flat plate exposed to turbulent flow the friction factor is proportional to the 1·8th power of the air velocity [35], [55]. For flows through a finned surface the value of the exponent varies from 1·75 to 1·96 according to the spacing and height of the fins. The friction factor is directly proportional to

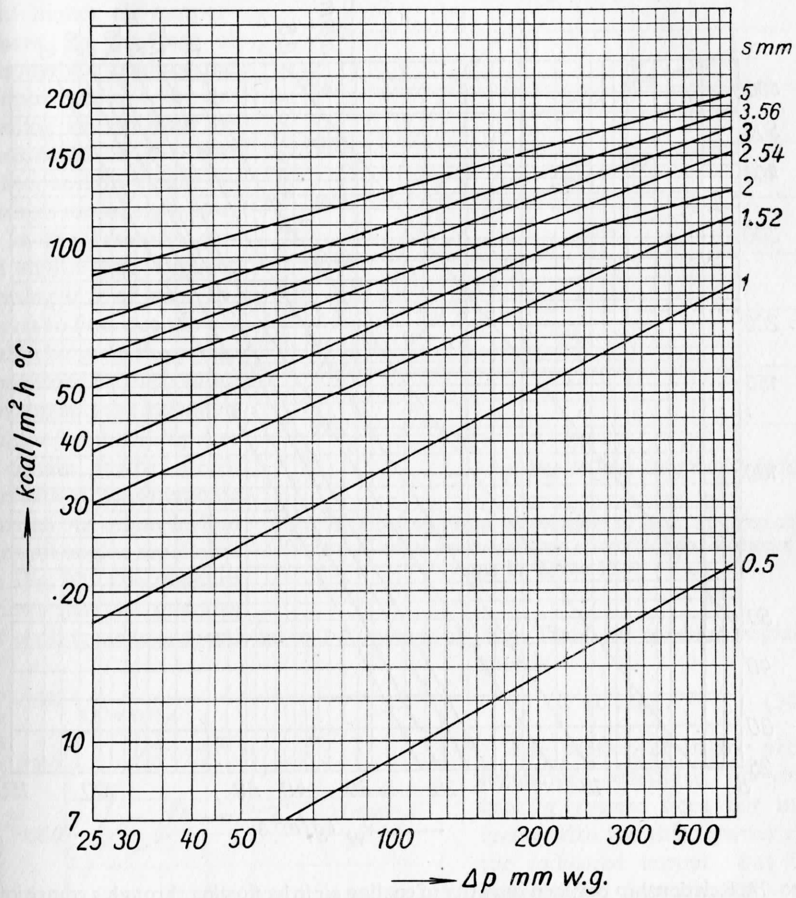


FIG. 73. Surface coefficient of heat transfer q plotted against pressure head Δp in the interstice.

the velocity head and subject to the same physical relations. Power consumed by the cooling system is the product of velocity head and quantity of air passed, i.e. theoretically it varies with the 2·8th power of the air velocity.

Tests carried out on finnings listed in Table 13 have shown, that the power consumed by the cooling system is proportional to 2·69 power of the mass velocity of air v_w. In Fig. 75 the transfer coefficient q_b is plotted against power consumed for cooling. The latter is expressed in h.p. per 100 mm cylinder length. At a given power consumption the highest value of q_b is

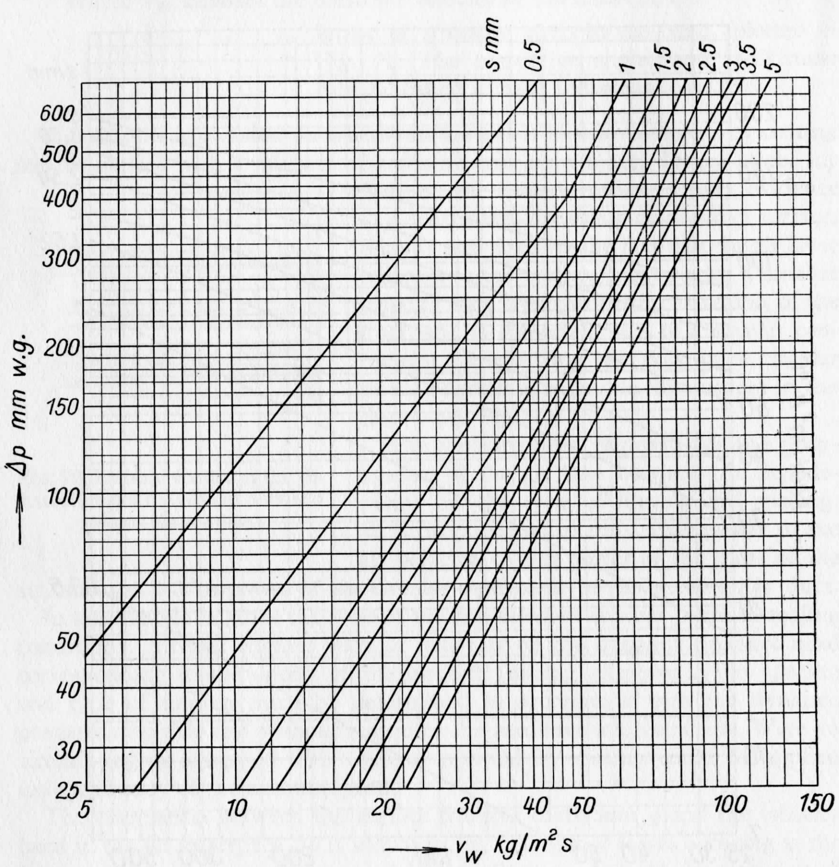


FIG. 74. Relationship between quantity of cooling air in kg flowing through a transition area of 1 sq. m per second and the pressure head Δp for different fin spacings s.

obtained in finnings with an optimal spacing, i.e. q_b gradually increases with diminishing spacing up to an optimal value. Beyond this limit a further reduction of the fin spacing does not result in an increase of the coefficient q_b, on the contrary, q_b decreases. Optimal values were obtained with finnings listed in Table 13. The optimal spacing was 2·1 mm, and by its reduction to 1·45 mm a considerable deterioration of q_b took place. A comparison of Fig. 67

and 75 shows that the correlation between optimal fin spacing and highest q_b corresponds to the correlation between optimal mass velocity v_w and smallest power consumed by the cooling system.

Small fin spacing means increased friction and higher air temperatures. In the case of automobile engines the output consumed by the cooling system must be considered in the first place, and fin weight remains a secondary factor.

In the case of a given engine and type of finning it is of some interest to find out the relation between the power consumed by the cooling system and the indicated engine output (i.h.p.) at a certain engine speed. Results of measurements carried out on an in-line aircraft engine are shown in Fig. 93. The diagram shows that the quantity of cooling air is proportional to 1·12 power of i.h.p. Thus, for constant engine speeds

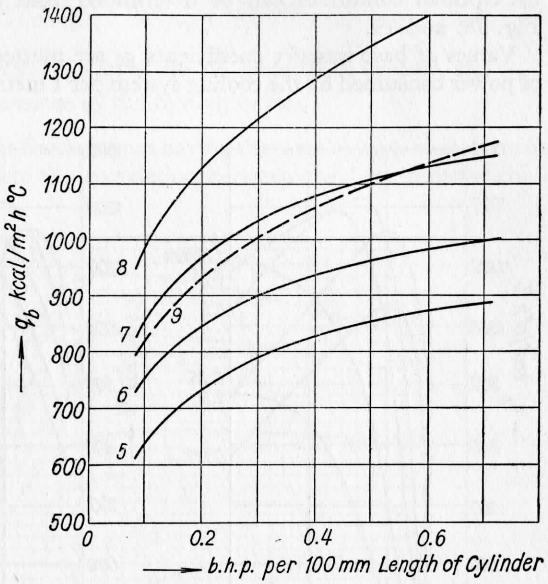


FIG. 75. Cooling input in relation to heat transfer coefficient. Curves are numbered according to fin dimensions in Table 13.

$$W \propto \text{i.h.p.}^{1\cdot 12} \qquad (34)$$

The portion of engine performance consumed by the cooling system does not increase with the third power of the indicated output, but is proportional approximately to 3·35 power of i.h.p. Thus, the conclusion is that it is uneconomical to improve cooling at higher engine outputs by increasing the quantity of cooling air.

From this it is evident, that

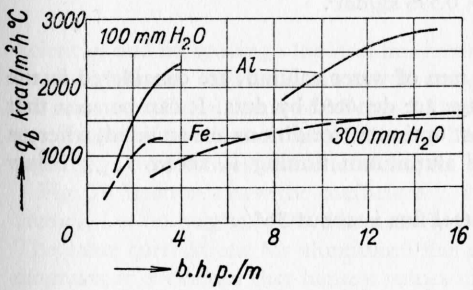


FIG. 76. Coefficient of heat transfer related to cooling input b. h. p. per 1 metre of finned cylinder length for pressure heads 100 mm w.c. (left) and 300 mm w.c. (right) for aluminium (Al) and steel (Fe) fins.

minimum power consumption by the cooling system is attained with a finning which, at a given velocity head, presents the highest base transfer coefficient q_b. Optimal conditions can be determined from the diagrams illustrated in Fig. 78. and 79.

Values of base transfer coefficients q_b are plotted in Fig. 76 against values of power consumed by the cooling system per 1 metre of finned cylinder length.

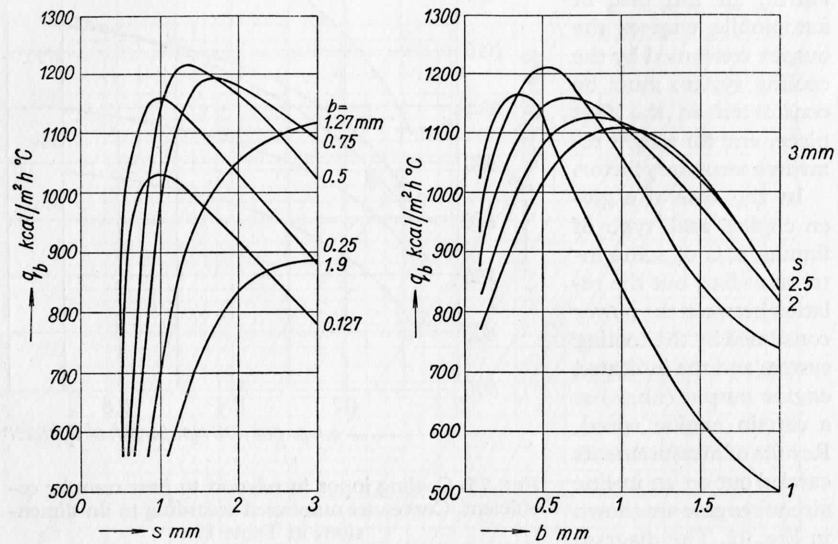


FIG. 77. Dependence of the coefficient of heat transfer upon fin spacing s and fin thickness b for different types of finning with the same unit weight of fins $m = 0.595$ kg/dm².

Two velocity heads, 100 and 300 mm of water column, are considered in the diagram. Standard types of finnings are denoted by dots. It can be seen that for a 100 mm velocity head almost optimum conditions are attained, whereas for 300 mm the performance of aluminium finning is about 33 % below optimum.

The corresponding figure for steel fins is about 36%.

6. The utilization of fin weight

It is of great importance to designers of aircraft engines to remove a given quantity of heat by using the smallest possible weight of material. It is useful, therefore, to introduce a new factor m representing fin weight in kg per

1 dm² of fin base surface area. This weight is calculated from the equation

$$m = \gamma \frac{hb}{S}\left(1 + \frac{h}{2r_b}\right) \text{ (kg/dm}^2\text{)} \tag{35}$$

where γ denotes the specific weight of the fin material in kg/dm³, and
h, b, δ, r_b are dimensions of the finning in dm.

A comparison of different finning types having the same m leads to interesting conclusions. Fig. 77 depicts the correlation between the base transfer coef-

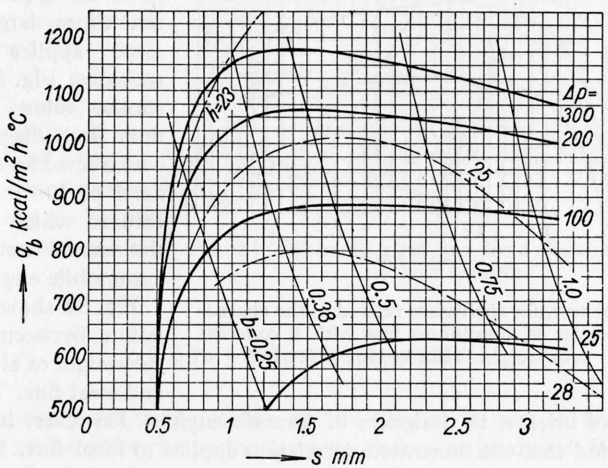


FIG. 78. Heat transfer coefficient related to the specific properties of finning for steel fins of a constant unit weight $= 0.595$ kg/dm².

ficient q_b and fin spacing s for steel fins having $m = 0{\cdot}595$ kg/dm². At a velocity head of 300 mm water column maximum q_b is attained at a spacing of about 1·5 mm and a fin thickness of 0·5 mm. However, this type of finning approaches the limits of production possibilities. The diagram clearly illustrates how rapidly q_b deteriorates with increasing fin thickness at constant m.

Fig. 78 illustrates how the coefficient q_b for steel fins is influenced by various factors, i.e. fin height h, fin thickness b, velocity head Δp, and fin spacing s. The same correlations for aluminium fins are plotted in Fig. 79. From these diagrams it is evident that highest values of q_b at lower velocity heads Δp are attained with larger spacings s and larger thickness b. At a given fin thickness the q_b curve has a distinct peak and declines rapidly with increased spacing s.

The comparison of both diagrams shows the superiority of thin aluminium fins over steel. Unfortunately this advantage cannot be fully exploited in practice, because thin aluminium fins are very difficult to produce. The same holds for copper fins which, produced in paper thickness, could well replace aluminium fins. However, thicker copper fins have no marked advantages over aluminium.

Optimum dimensions of aluminium fins plotted against q_b are shown in Fig. 80. We can see that e.g. the best dimensions for $q_b = 1000$ kcal/m² h °C are $b = 0{\cdot}21$ mm and $h = 16$ mm. Fins of these dimensions are impossible for production reasons; practical dimensions are approximately ten times larger. The same applies to fin spacings. Fig. 81 shows similar values for steel fins. Both diagrams are constructed for a velocity head of 100 mm water column which is about the normal head used in automobile engines.

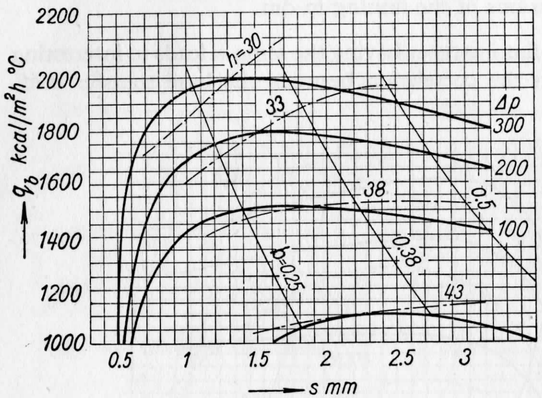


FIG. 79. Heat transfer coefficient related to the characteristic properties of aluminium fins with a constant unit weight 0.232 kg/dm².

Fig. 82 shows the relation between q_b and the weight of aluminium and steel fins. This correlation is of interest to designers of aircraft engines. However, it must be borne in mind that the illustrated correlation applies to ideal fins. Steel fins approach this theoretical diagram quite closely, but aluminium fins corresponding to the diagram are too thin for practical purposes. Aluminium fins could transfer 23% more heat, or could be 50% lighter for a given heat transfer, if their weight would be fully utilized for the transfer of heat. Some further interesting conclusions can be drawn from Fig. 82. At the same unit weight m and a velocity head of 100 mm, aluminium fins can transfer 2·25 times more heat than steel fins. However, in practice the difference between aluminium and steel fins is far smaller, because optimum spacing and thickness of aluminium fins is not practicable.

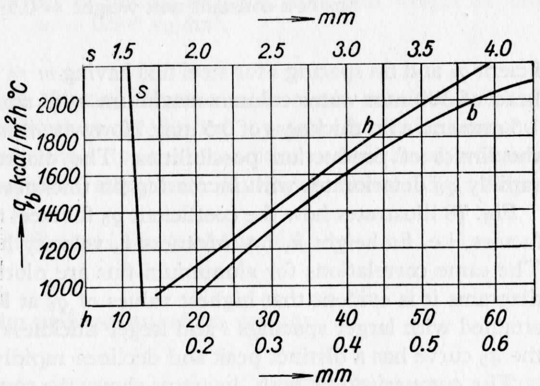


FIG. 80. Optimum aluminium fin dimensions for a desired coefficient of heat transfer.

Fig. 82 shows also that the quantity of heat removed can be considerably increased by increasing the unit weight of fins. For aluminium fins at $q_b = 1600$ a double weight of the fin means heat transfer increased by 24%. For steel fins at $q_b = 600$ an increase of m by 100% results in a 21% increase of the quantity of heat removed. Better cooling achieved by an increased m is paid for by an overall increase of the cylinder weight. Therefore, it must be calculated, wether the use of a fan, i.e. improvement of cooling by increasing the air velocity, is not more economical.

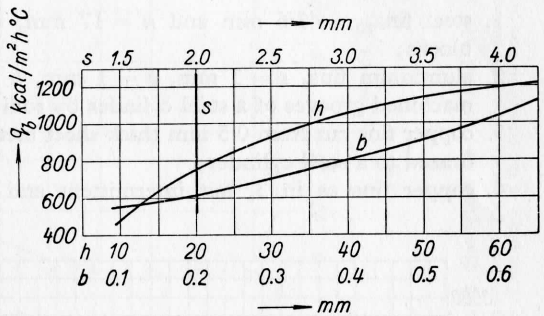


FIG. 81. Optimum steel fins for a desired coefficient of heat transfer.

7. Fin production methods

Before making the final decision on the type of finning to be used, the designer must consider the methods by which fins are produced. Individually mounted cylinders without excessive thermal stresses can be fitted with cast fins such as those for heads of automobile engines. At a mean thickness of 2 mm and height of 40 mm fin spacing should never be less than 6 mm. Castings of smaller spacings present great difficulties.

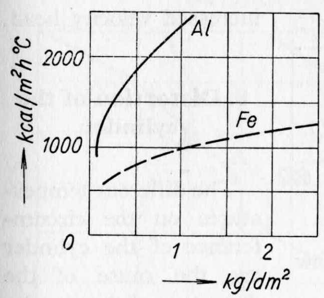


FIG. 82. Relation between heat transfer coefficient and unit weight of fins at a pressure head of 100 mm w.g.

Machined fins are used for cylinders subjected to high thermal stresses. In these cases cylinders made of cast iron can have spacings of 3·5—4 mm, thickness of 1 mm and heights of 15—20 mm. Fin dimensions for steel cylinders can be further reduced to $s = 1{\cdot}5$ mm, $b = 0{\cdot}5$ mm, $h = 25$ mm. However, these figures should be regarded as limits of production possibilities closely approaching theoretical optimum values.

Theoretically derived fin dimensions cannot be realized with materials having good thermal conductivities (aluminium, copper). When using machined fins it is necessary to calculate if the increased production cost is compensated for by the cooling system. In cast iron cylinders with cast-on fins the high percentage of waste must be considered.

Fig. 83 shows the correlation between the base transfer coefficient q_b and quantity of cooling air for various types of fins. The black dots on the curves indicate positions with a velocity head of $\Delta p = 200$ mm w.c. The following types of fins are represented in the diagram:

1. steel fins, $s = 3{\cdot}6$ mm and $h = 17$ mm, turned from compact steel blocks;
2. aluminium fins, $s = 3$ mm, $b = 1$ mm, $h = 20$ mm mounted in the machined grooves of a steel cylinder by rolling;
3. copper fins cut from 0·5 mm thick sheet metal, $s = 2$ mm, $h = 22$ mm, brazed to a steel cylinder;
4. copper fins as in 3, but intermittent and off set.

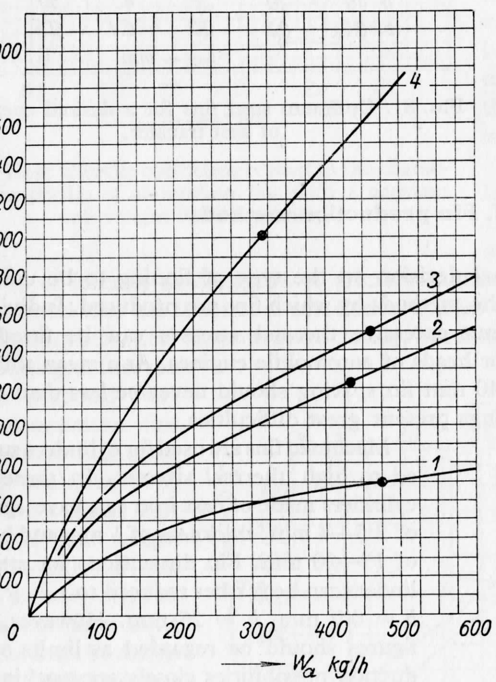


FIG. 83. Coefficient of heat transfer plotted against flow rate of cooling air.

1 — machined steel fins, $h = 17$ mm, spacing $s = 3.6$ mm. *2* — aluminium fins rolled into machined grooves of a steel cylinder, $h = 20$ mm, $s = 3$ mm, $b = 1$ mm. *3* — copper fins, brazed to steel cylinder, $h = 22$ m, $s = 2$ mm, $b = 0.5$. *4* — copper fins as above, but intermittent and off-set. Marked points represent values for a pressure head of 200 mm w.g.

It can be clearly seen from the diagram how cooling can be improved for a given quantity of air by the application of suitably produced fins. The intermittent copper fins are the best, because at each intermittance similar conditions exist as at the inlet edge, i.e. a small thickness of the boundary layer. Cooling conditions are improved on account of a higher overall resistance to air flow and this means an increased velocity head.

8. Distortion of the cylinder

The different temperatures on the circumference of the cylinder are the cause of the distortion of the circular shape of the cylinder surface. In such cases piston rings have no regular contact with the inner surface of the cylinder and blow-by occurs, the engine performance is reduced, oil consumption increased, the piston is overheated and even a seizure of piston rings may occur.

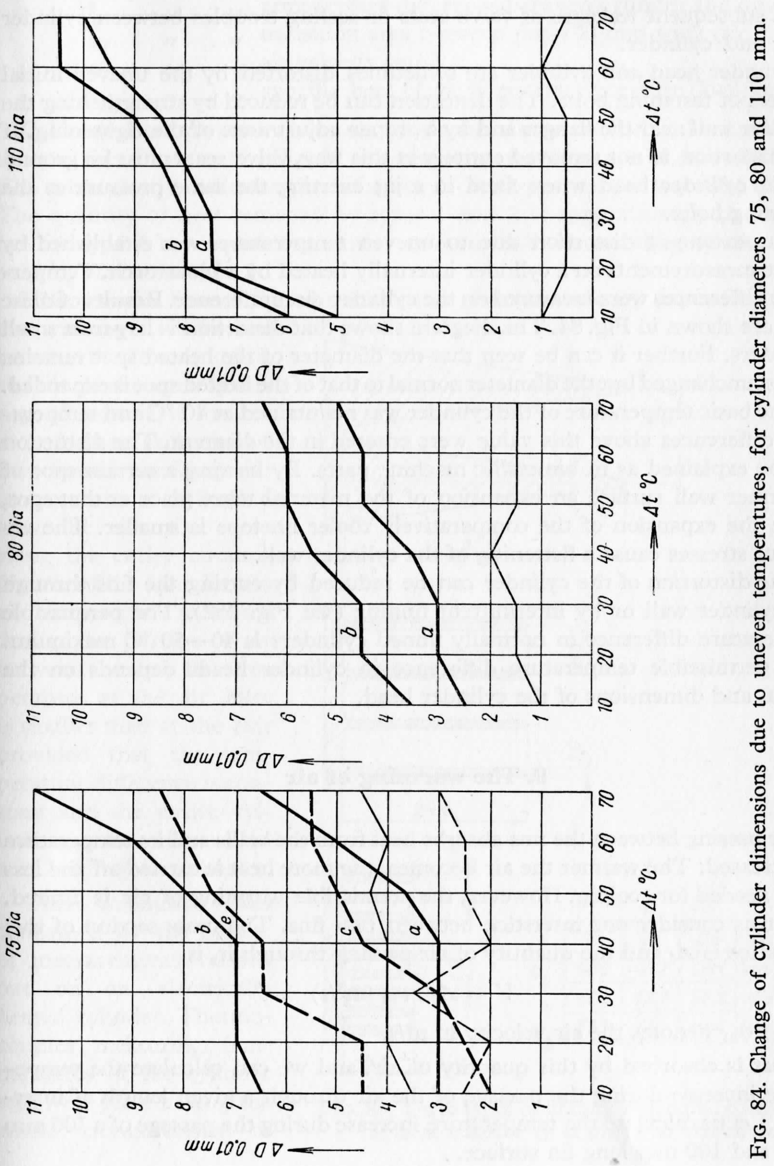


FIG. 84. Change of cylinder dimensions due to uneven temperatures, for cylinder diameters 75, 80 and 110 mm.
Arrows signify direction of heating in a plane normal to the axis of the crankshaft. Temperatures in t °C were measured at the front and rear of the cylinder. Change in dimensions at the 75 mm cylinder measured also for heating in the direction of the plane of the crankshaft axis (dashed c, e lines). Deviation is given by the thin curve $b - a$.

The causes of uneven temperatures in the cylinder may be either an increased heat transmission at certain zones of the inner surface or a reduced heat removal from some regions of the outer surface of the cylinder.

The same deficiencies can cause a thermal distortion of the cylinder head with subsequent leakages of valve seats or sealing troubles between cylinder head and cylinder.

Cylinder head and cylinder are sometimes distorted by the uneven initial stresses of fastening bolts. The distortion can be reduced by strengthening the cylinder wall near the flanges and by a proper adjustment of the tightening. If the distortion is not removed entirely in this way, valve seats must be ground in the cylinder head when fixed in a jig exerting the same pressure as the fastening bolts.

The extent of distortion due to uneven temperatures was established by direct measurements on a cylinder internally heated by a blow torch. Temperature differences were measured on the cylinder circumference. Results of these tests are shown in Fig. 84. The diagram shows that distortion is larger in small cylinders. Further it can be seen that the diameter of the heated spot remains almost unchanged but the diameter normal to that of the heated spot is expanded.

The basic temperature of the cylinder was maintained at 40 °C and temperature differences above this value were entered in the diagram. The distortion can be explained as in bimetallic machine parts. By heating a certain spot of the inner wall surface an expansion of the material takes place at that spot, while the expansion of the comparatively cooler fin tops is smaller. The resultant stresses cause a flattening of the cylinder wall.

The distortion of the cylinder can be reduced by cutting the fins through the cylinder wall or by intermittent finning (see Fig. 252). The permissible temperature difference in normally finned cylinders is 40—50 °C maximum. The permissible temperature difference in cylinder heads depends on the design and dimensions of the cylinder head.

9. The warming of air

Air passing between the fins absorbs heat from the walls and its temperature is increased. The warmer the air becomes, the more heat is carried off and less air is needed for cooling. However, the permissible warming of air is limited.

Let us consider one interstice between two fins. The cross section of this interstice is A_t and the quantity of air passing through it, is

$$V = A_t \cdot v_m \ (\mathrm{m^3/s})$$

where v_m denotes the air velocity in m/s.

Heat is absorbed by this quantity of air, and we can calculate the temperature increase during the passage of the air through a given length of interstice. Let us calculate the temperature increase during the passage of a 100 mm wide and 100 mm long fin surface.

Thus

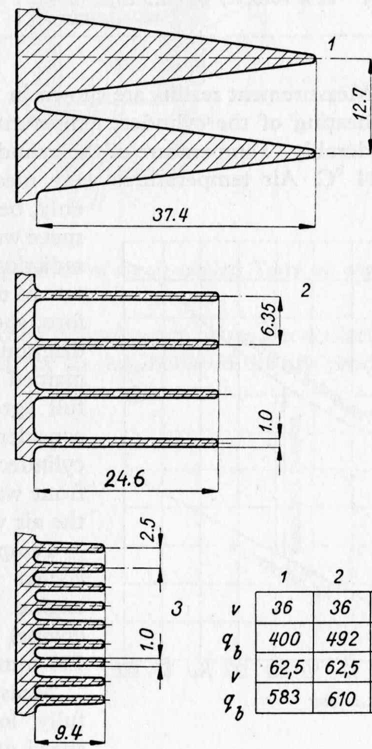
$$t_{100} = \frac{q_b \cdot \Delta t}{100 \cdot c_p \cdot A_{100} \cdot v_m \cdot 3600} \ (^\circ\mathrm{C})$$

where q_b denotes the base transfer coefficient (kcal/m² h °C)
Δt „ „ temperature difference between cylinder and air (°C)
A_{100} „ „ transition area between fins 100 mm wide (m²)
v_m „ „ air velocity (m/s)
cp „ „ specific heat of air at eonstant pressure (kcal/kg °C)

Calculated temperature increases of air are shown in Table 16 for three types of fins illustrated in Fig. 85 at a temperature difference of $\Delta t = 120$ °C. The table clearly indicates the influence of fin types on the warming of air. The quantity of heat removed by types 1 and 2 is approximately the same, but for type 1 we require four times more air and the warming of the air is only $^1/_4$ of that in type 2.

Heat transfer from a finned surface at a given air velocity depends on the temperature difference between the finned surface and air. If the same quantity of heat is to be removed from each cm² of the finned surface, the temperature difference must be also the same along the entire circumference of the cylinder. As air temperature increases during the passage of air between the fins, fin temperature at the air inlet is smaller than at the exit provided that the temperature difference is constant and the entire cylinder surface is cooled with an equal intensity.

This conclusion was confirmed by the results of measurements carried out on an electrically heated cylinder. Thermocouples measuring temperatures of the cylinder wall were placed along the whole circumference of



	1	2	3	
v	36	36	36	m/s
q_b	400	492	394	kcal/m²h°C
v	62,5	62,5	62,5	m/s
q_b	583	610	604	kcal/m²h°C

FIG. 85. Three types of finning with equal coefficients of heat transfer q_b at given air velocity v.

Table 16

Comparison of Three Types of Finning

Type of Finning		1	2	3
Transition area of a 100 mm wide strip of the cylinder surface, (cm²)		21.63	20.72	5.64
Air flow at a velocity of 36 m/s, (m³/s)		0.07788	0.07459	0.02034
Air flow at a velocity of 62.5 m/s, (m³/s)		0.13519	0.12950	0.03525
Heat removed from 1 dm² of base area at a temperature gradient of 120 °C, (kcal/s)	at an air velocity of 36 m/s	0.13333	0.16400	0.13140
	at an air velocity of 62.5 m/s	0.1949	0.20333	0.20134
Warming of air, (°C)	at a velocity of 36 m/s	5.92	7.60	22.36
	at a velocity of 62.5 m/s	4.96	5.42	19.74

the cylinder. Measurement results are shown in Fig. 86. In spite of a regular and uniform heating of the cylinder, temperatures along the circumference differed considerably. Temperature of the cylinder at the air inlet was 76 °C, at the exit 124 °C. Air temperatures were measured at the inlet and exit only, because measuring in the interspace was rendered impossible by the radiation from the fin surface. The rate of temperature increase is, therefore, shown by a straight line in the diagram. Temperature difference remained almost constant which is in full agreement with our theoretical considerations. The rear wall of the cylinder was 48 °C warmer than the front wall, and the total warming of the air was also about 48 °C.

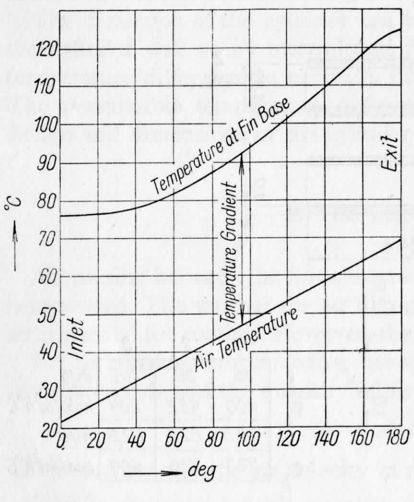


Fig. 86. Distribution of temperatures on the circumference of a cylinder with cowlings.

Temperatures measured on an air-cooled compression-ignition Tatra engine are shown in Fig. 87. The engine had a 110 mm bore and 130 mm stroke.

Measurements were taken on a fully loaded engine running at a speed of 2000 rev/min. Temperature of the cylinder wall closely below the cylinder head was 158 °C on the

air inlet side and 184 °C at the exit, i.e. a temperature difference of 26 °C.

The warming of air may thus greatly influence temperatures on the cylinder circumference and consequently the cylinder distortion. The danger of large

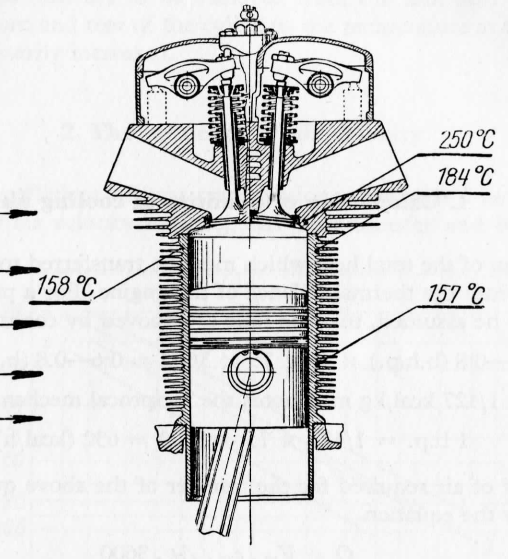


Fig. 87. Cylinder temperature of an air-cooled Tatra oil engine.

temperature fluctuations on the circumference arises particularly with dense finning where a long travel of the air flow between the fins produces a considerable warming of the air.