CHAPTER VII

THE QUANTITY OF COOLING AIR

1. Calculation of quantity of cooling air

The portion of the total heat which must be transferred to air by cooling is determined from the thermal balance of the engine. For a preliminary calculation it may be assumed, that the heat Q removed by cooling equals

$$Q = 0{\cdot}6\text{—}0{\cdot}8\ (\text{b.h.p.}) \times A \times 75 \times 3600 = 0{\cdot}6\text{—}0.8\ (\text{b.h.p.}) \times 632 \quad (36)$$

where $A = 1/427$ kcal/kg m denotes the reciprocal mechanical equivalent

$$1\ \text{h.p.} = 1/427 \times 75 \times 3600 = 632\ (\text{kcal/h})$$

The quantity of air required for the transfer of the above quantity of heat is calculated by the equation

$$Q = V_w \cdot c_p \cdot \Delta t \cdot 3600 \quad (37)$$

where V_w denotes the mass velocity of air (kg/s)

c_p „ „ specific heat of air at constant pressure (kcal/kg °C)

Δt „ „ temperature increase of air (°C)

The increase of air temperature is larger in air-cooled engines than in water-cooled ones; it depends on the temperature difference between cylinder and air, the air velocity, the size and shape of the space between the fins, and the length of air travel between fins.

In well designed air-cooled automotive engines the temperature increase of air amounts to 60—80 °C; therefore the quantity of cooling air is about half of that required in water-cooled engines of the same performance.

In properly finned engines the quantity of cooling air is 52—104 lb/b.h.p.-h. (20—40 kg/b.h.p.-h). With too widely spaced fins and large heat transfer from the combustion products to the cylinder the quantity of cooling air may rise up to 200 lb/b.h.p.-h (75 kg/b.h.p.-h.) For rough estimates we may assume an air consumption of 2·6 lb/min b.h.p. (1 kg/min b.h.p.)

The larger the increase of air temperature during its flow along the cylinder, the smaller the quantity of cooling air required. Large temperature increase of air is beneficial as each unit of air removes a large quantity of calories. However, we must consider also the fact that the quantity of air removed is

proportional to the temperature difference Δt and coefficient of heat transfer q. Thus

$$Q \propto q \cdot \Delta t \quad (38)$$

If air temperature is increased during its passage between the fins, and if equal amounts of heat are to be removed from the unit area of the finned surface in the front and rear of the cylinder, the temperature at the rear of the cylinder is necessarily increased.

2. The influence of air velocity

The surface coefficient of heat transmission q depends largely on the air velocity. Higher air velocity means better heat transfer and better cooling.

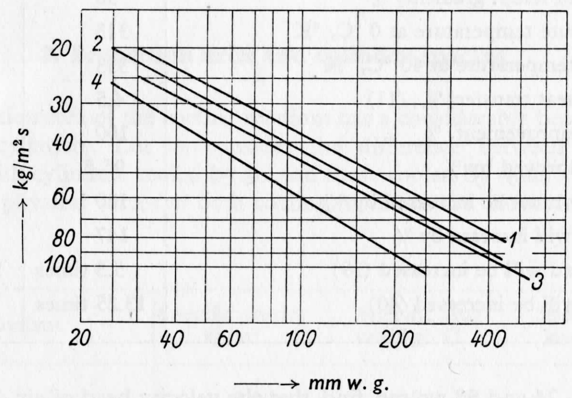

FIG. 88. Air flow in kg related to pressure head Δp.

1. Fin spacing 3 mm, bore 75 mm, without securing bolts. *2.* Fin spacing 5.5 mm, bore 110 mm, cylinder with cowling. *3.* Fin spacing 5 mm, bore 75 mm. *4.* Fin spacing 5.5 mm, bore 110 mm, cylinder without cowling and bolts.

However air velocity must be suitable to the type of finning used. With cylinders exposed to small thermal stress, widely spaced fins and low air velocity should be satisfactory. By increasing the thermal stress, cooling must not be improved by an increased air velocity only; in such case the finning must be changed as well.

The coefficient of heat transfer q increases with 0·73 power of the air velocity. Thus a double air velocity means a coefficient q improved by 66%. If 1 m³ of air removed a heat quantity of 100% Q, 2 m³ of air will remove 166% Q. Thus by increasing air velocity to its double value the quantity of heat removed by 1 m³ of air is reduced to 83% of the former value. Consequently the utilization of air becomes worse and the exit temperature is also lowered. Thus, specific air consumption is increased.

Table 17

Variation of Cooling Input for a Water- and Air-Cooled Engine, if Cooling is Improved only by a Changing Air Flow at an Increase of Ambient Temperature from 0 to 40 °C

Coolant	Water	Air
Temperature of ambient air, °C	0	0
Temperature of cooled surface, °C	80	180
Temperature gradient, °C	80	180
Temperature of ambient air, °C	40	40
Temp. gradient at this temperature, °C	40	140
Reduction of temp. gradient, %	50	22
Mean absolute temperature at 0 °C, °K	313	363
Mean abs. temperature at 40 °C, °K	333	383
Improved heat transfer, %, (11)	4.5	4
Required improvement, %	100	28.5
Must be improved by %	95.5	24.5
Air velocity must be increased by %	100	30
Air supply will increase by %	147	32.8
Velocity head will be increased (39)	5.3 times	1.725 times
Fan input will be increased (40)	13.25 times	2.33 times

From Fig. 74 and 88 we can find that the velocity head of air Δp increases with the 1·3 and 1·96 power of the air quantity according to the type of finning employed. Let us, therefore, assume that, with finnings normally used in automobile engines, the velocity head is proportional to 1·82 power of the air quantity. Table 17 illustrates the increase of power consumed by the cooling system, with unchanged type of finning, if cooling is being improved by an increase of air velocity only. For simplification we suppose that air velocity is directly proportional to the weight of air and

$$q \propto v^{0\cdot 73} \qquad \Delta p \propto Q^{1\cdot 82} \qquad Q \propto \Delta p^{0\cdot 55} \tag{39}$$

From these correlations we can establish the quantity of air, velocity head and the power consumed by the cooling system at an increased engine performance and constant finning.

If, for instance, engine output is to be increased by 33%, the quantity of cooling air must be increased in accordance with a transfer coefficient q increased also by 33%:

$$133\% = v^{0\cdot 73}, \text{ thus } v = 147.7\%.$$

Air velocity must be increased by 47·7 %, and the quantity of air is increased in the same proportion. The velocity head is then calculated from the relation

$$p = 147{\cdot}7^{1\cdot 82} = 300\%$$

The velocity head is increased by 103%, and the fan input, depending on the increased air quantity and velocity head, is calculated as

$$\text{fan h.p.} = 147{\cdot}7 \times 203 = 300\%.$$

If the original engine required 50 kg/b.h.p. h of cooling air, the specific air consumption of the engine with a performance increased by 33% is

$$50 \times \frac{147{\cdot}7}{133} = 55 \text{ (kg/b.h.p.-h)}$$

3. Transition area and cylinder spacing

The transition area of the cooling medium has a considerable bearing on the spacing of cylinders. Let us consider the difference between conditions prevailing with cylinders cooled by air and those cooled by water. Differences between the physical values of both cooling mediums are shown in Table 18.

Table 18

Coolant	Specific gravity kg/m³	Specific heat kcal/kg °C	Specific heat kcal/m³ °C
Air at 80 °C	1.00	0.241	0.241
Water at 80 °C	972.00	1.003	974.92

One m³ of water heated by 1 °C removes 4000 times more heat than 1 m³ of air. Therefore, a far greater volume of the cooling medium passes the cylinder of an air-cooled engine than that of a water-cooled engine.

For comparison let us determine the required flow area for both types of cooling related to a cylinder of internal combustion engines having the following parameters:

D = cylinder bore . . . . 100 mm

L = piston stroke . . . . 100 mm

p_e = mean effective pressure . . . . 6 kg/cm² (85 lb/in²)

n = engine speed . . . . 3000 rev/min

The output of the cylinder is

$$\text{b.h.p.} = \frac{V \cdot p_e \cdot n}{900} = 15{\cdot}7 \text{ (b.h.p.)}$$

where V denotes the piston displacement in litres. Let us convert the result to thermal units

$$15{\cdot}7 \text{ b.h.p.} = 15{\cdot}7 \times 632 = 9{,}920 \text{ (kcal/h)}$$

For further calculation let us assume that 70 % of this heat is removed by cooling, i.e.

$$Q = 9{,}920 \times 0{\cdot}7 = 6{,}944 \text{ (kcal/h)}$$

We shall now calculate the thermal load of a unit area of the outer cylinder surface. For simplicity we shall consider the outer cylinder surface as that limited by the bold line in Fig. 89, i.e. regardless of the valve ports and with a supposed cylindrical combustion chamber. Subject to these assumptions the area of the outer cooling surface is calculated from the equation

$$A_c = \pi (D + 2b) \cdot \left[L + \frac{L}{\varepsilon - 1} + b \right] + \frac{\pi (D + 2b)^2}{4}.$$

The thermal load is calculated by dividing the removed heat by the above area, i.e.

$$U = \frac{Q_c}{A_c}$$

b denotes the thickness of the cylinder wall (6 mm)

ε „ „ compression ratio = 16

For the cylinder discussed $A_c = 494{\cdot}5$ cm², and $U = 14$ kcal/cm²h.

During its passage through the engine cooling water is warmed by 5—6 °C. The quantity of cooling water required for the removal of the heat Q_c is

$$V_1 = \frac{Q_c}{c'_p \cdot t_1} = 1.15 \text{ (m}^3\text{/h), where } t_1 = 6 \text{ °C}$$

The quantity of cooling air required is calculated in a similar way under the assumption of the temperature increase of air being $t_2 = 60$ °C

$$V_2 = \frac{Q_c}{c_p \cdot t_2} = 482 \text{ (kg/h)}$$

which, at a mean air temperature of 50 °C and specific weight $\gamma = 1{\cdot}09$ kg/m³ amounts to

$$V_2 = 482 : 1{\cdot}09 = 442 \text{ (m}^3\text{/h)}$$

At a water velocity of 1 m/s the transition area required is

$$A_1 = \frac{1.15}{1 \times 3600} = 3{\cdot}2 \text{ (cm}^2\text{)}$$

and at an air velocity of 25 m/s the transition area required is

$$A_2 = \frac{442}{25 \times 3600} = 49 \text{ (cm}^2\text{)}$$

The transition area around the cylinder of an air-cooled engine is reduced by the cross section of the cooling fins. The area occupied by the fins amounts

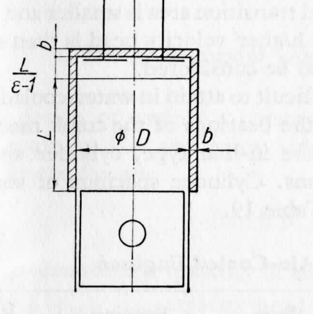

FIG. 89. The outer surface of a cylinder serving for the removal of heat.

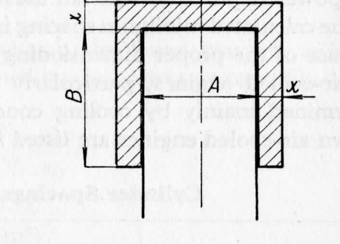

FIG. 90. Surface area around the cylinder for heat transfer to the cooling medium.

to 20—45% of the total area. Therefore the calculated transition area in an air-cooled engine must be increased by an average value of 30 % to

$$A_2 = \frac{49 \times 100}{70} = 70 \text{ (cm}^2\text{)}$$

Supposing that this transition area is uniformly distributed around the circumference of the cylinder according to Fig. 90 and limited by the bottom plane of the piston skirt at the bottom dead centre, the following relation holds:

$$A_f = 2Nx + Mx + 2x^2$$

$$2x^2 + x(M + 2N) - A_f = 0$$

$$x = \frac{-(M + 2N) + \sqrt{(M + 2N)^2 + 8A_f}}{4}$$

According to Fig. 90 $M = D + 2b = 11{\cdot}2$ cm

$$N = L + \frac{L}{\varepsilon - 1} + b = 11{\cdot}27 \text{ cm}$$

From the above quadratic equation we can calculate the width of the transition area x_1 of the water-cooled and x_2 of the air-cooled engine. Thus

$$x_1 = 0{\cdot}09425 \text{ cm, and } x_2 = 1{\cdot}868 \text{ cm}$$

The following conclusion may be drawn from the preceding analysis:

At the conditions given the minimum spacing of cylinders of a water-cooled engine is

$$S_1 = D + 2b + 2x_1 = 113{\cdot}9 \text{ mm} = 1{\cdot}138\ D,$$

and that of an air-cooled engine

$$S_2 = D + 2b + 2x_2 = 149{\cdot}4 \text{ mm} = 1{\cdot}494\ D.$$

It can be seen from this example that the minimum cylinder spacings of water- and air-cooled engines are $1{\cdot}14\ D$ and $1{\cdot}5\ D$ respectively. At air velocities higher than 25 m/s the required transition area is smaller and so is the minimum cylinder spacing. However a higher velocity head is then required and power required for the fan must also be considered.

The calculated minimum spacing is difficult to attain in water-cooled engines because of the proper dimensioning of the bearings of the crank mechanism. In air-cooled engines, particularly of the in-line type, cylinder spacing is determined mainly by cooling conditions. Cylinder spacings of some well known air-cooled engines are listed in Table 19.

Table 19

Cylinder Spacings of Air-Cooled Engines

Make	Fuel	Bore **D** mm	Spacing **T** mm	Ratio **T/D**
Tatra T 600	petrol	85	143	1,685+
Tatra T 87	petrol	75	108	1.44
Tatra T 603	petrol	75	100	1.34
Steyer	petrol	78	118	1.51
KdF - VW	petrol	75	110	1.47
Tatra T 111	oil	110	155	1.41
Tatra T 116	oil	120	200	1.67+
Tatra 928	oil	120	155	1·29
Deutz F4L	oil	110	150	1.36
Deutz F8L	oil	110	180	1,64+
Simmering	oil	135	215	1.59+
ČKD	oil	145	260	1,79+
Škoda	oil	140	205	1.46
SLM	oil	110	150	1.36
Robur NVD 12.5 SRL	oil	90	150	1.66
Robur NVD 12.5 SRL	oil	100	150	1.50
NATI DV 30	oil	95	140	1.47
Continental AVDS-1790	oil	146.1	228.6	1.56

Engines marked+ have a cylinder spacing determined by the crankshaft bearings

The condition of minimum cylinder spacing in air-cooled engines requires a design which fully respects the elimination of an undesirable increase of engine dimensions and weight. Air-cooled radial, horizontally opposed and V engines are being designed with complete regard to the condition of minimum cylinder spacing. The crank mechanism of these engines permits a cylinder spacing beneficial from the standpoint of air-cooling. The ratio between the transition areas of cylinder head and cylinder depends on the quantity of heat to be removed. However, the type of finning employed is also important.

4. The economical velocity head of cooling air

In the design of engines a compromise must be found between cylinder spacing, engine weight and power consumed by the cooling system. The latter is given by the formula

$$\text{fan h.p.} = \frac{V_a \,.\, \Delta p}{75 \,.\, \eta_f} \qquad (40)$$

where V_a denotes the quantity of air (m³/s)

Δp „ „ velocity head (kg/m²)

η_f „ „ overall fan efficiency.

It is evident that small air quantity and small velocity head are requirements of first importance. Small quantities of air usually entail large velocity heads. In such cases air leakage must be prevented by carefully produced ducting.

In high performance engines with large cylinders dense finning ensuring a high transfer coefficient q_b should be used. The flow area between the fins is small, and, in spite of the small specific air consumption the absolute quantity of cooling air is large. Therefore velocity heads in these engines amount to 200—400 mm of water column. The dynamic pressure head in aircraft engines is often insufficient, and fans must be used.

Vehicle engines have smaller cylinders and the ratio of cooling surface area to displacement is also favourable. Velocity heads in these engines do not exceed 100—150 mm. If the calculated velocity head is higher than 200 mm, redesigning of the engine is usually indicated. The cooling system of a properly designed air-cooled engine requires 6—8% of the b.h.p. as a maximum.

Power consumed by the cooling systems of various engines is shown in Table 20.

Motor-cycle engines with single cylinders have sufficient space for finning. The dynamic pressure head produced by the moving vehicle ensures good cooling. Small engines mounted on bicycles exhibit a particularly beneficial ratio of cooling surface to displacement, and, therefore, a velocity head of 1.5 mm of water column is sufficient. This velocity head corresponds to a travelling speed of only 11 m.p.h. (18 km/h).

Table 20

Cooling Power Required by Some Motor Engines

Make	bhp	rpm	fan hp	Ratio bhp: fan hp	Coolant
GAZ AAA [1]	50	2800	4	8 %	water
GAZ M1 [1]	50	2800	2.7	5.4 %	water
GAZ A [1]	40	2200	1.9	4.75%	water
Ford	103	3800	7	6.8 %	water
Deutz [31]	90	2300	7	7.8 %	water
V 2 in tank [56]	500	1800	42	8.4 %	water
Maybach in tank [56]	700	3200	56	8.0 %	water
Tatra T 301	160	1600	4.8	3.0 %	air
Tatra T 111	200	2000	7.6	3.8 %	air
Tatraplan T 600	52	4000	3.5	6.7 %	air
Deutz [31]	90	2300	7.0	7.8 %	air
Arsenal USA D 117 [3]	32	3200	1.6	5.0 %	air
Arsenal USA D 146 [3]	68	3000	4.1	6.0 %	air

Small power consumption for cooling results usually from a small velocity head. It increases with 2·75—2·9 power of the air velocity. However, the choice of a small velocity head is limited by the conditions of cylinder spacing and by the possibility of mounting large fin surface areas per unit of the outer cylinder surface. Owing to the small increase in temperature, use of cylinders without covers is inefficient — large quantities of air are required and, in spite of the small velocity head, power consumed by the cooling system is large.

At a low heat transmission per unit area from combustion chambers and with dense finnings only small quantities of cooling air are required. Results of measurements carried out by Löhner [35] on a supercharged air-cooled engine are shown in Fig. 91. At high specific performances a cylinder can be cooled at a velocity head of 150 mm w.c. and 36 lb/b.h.p.-h (14 kg/b.h.p.-h) air consumption corresponding to a 1% b.h.p. power consumption for cooling purposes.

Power consumed by cooling depends very much on the cylinder temperature. A 10% reduction of the cylinder temperature (say from 200 to 180 °C) means an increase of power for cooling by 150%. The rate of increase is higher with rising cylinder temperatures. The use of a properly designed finning, made

of a material having a good thermal conductivity, enables a considerable reduction of power required for cooling. Tests carried out on a cylinder of the Ranger type aircraft engine are most instructive in this respect [44]. The engine concerned is a small bore in-line engine, and, therefore, the test results should be of interest to designers of automobile engines.

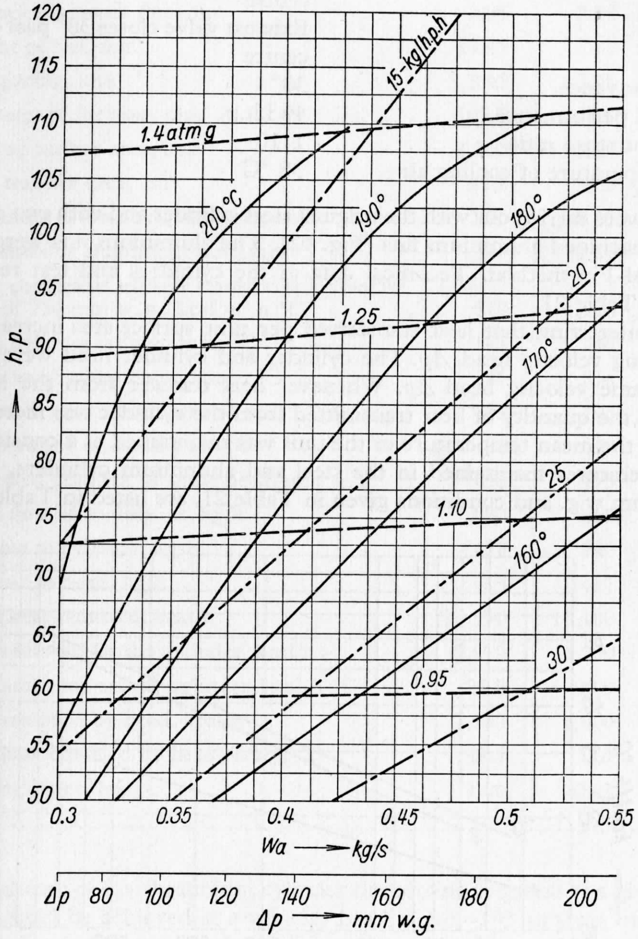

FIG. 91. Characteristics of an air-cooled supercharged engine.

Vertical scale — corrected performance of one cylinder, b.h.p. Horizontal scale — weight of cooling air, kg/s. Lower horizontal scale — static pressure head of cooling air, mm w.g. Full line — temperature of cylinder head, °C. Dash line — intake pressure from supercharger, atm. g. Dash and dot line — unit weight of air, kg/b.h.p. *h*. Bore 115.5 mm, stroke 162 mm, 2,400 rev/min, $\varepsilon = 6.5$, $\lambda = 1$, temperature of intake air 80 °C.

The Ranger experimental single-cylinder was identical with Mark V-770-12 of the following parameters:

bore and stroke	101·6 and 133·4 mm
displacement	1080 cm³
compression ratio	1:6.5
valve timing	Inlet valve opens 15° before dead centre Exhaust valve closes 30° past dead centre
ignition advance	30°
indicated performance	49 i.h.p.
fuel-air mixture ratio	1:10
inlet temperature of cooling air	38 °C

Tests were carried out with the original steel cylinder and with two cylinders having machined aluminium fins (Fig. 92). The aluminium fins were cast on by the Al-Fin method. Technical data of the cylinders and test results are given in Table 21.

It is interesting that heat transferred per unit surface area increased with a declining velocity head Δp. The cylinder and cylinder head were exposed to the same velocity head Δp. Whenever heat transfer from the head was reduced, the quantity of heat transmitted from the cylinder was increased. In this way the mean temperature of the unit was maintained at a constant level.

Temperatures maintained in the steel and aluminium cylinders, at $\Delta p =$ $= 483$ mm w.g. and conditions given in Table 21, are listed in Table 22.

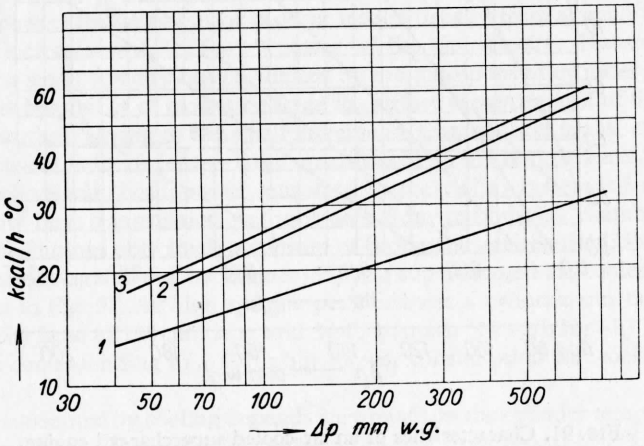

FIG. 92. Relationship between heat removed from the cylinder and pressure head Δp
1 — series cylinder with steel fins, *2* — cylinder with aluminium fins, *3* — aluminium cylinder with special fins.

Table 21

Cooling Tests on the Ranger V-770-12 Aircraft Engine

Type of cylinder	Steel	Alumi- nium	Testing
Number of fins	28	33	43
Interstice between fins, mm	2.95	2.28	1.57
Height of fins, mm	13.45	12.7	23 (13.3)
Fin spacing, mm	3.45	2.95	2.24
Diameter of fin base, mm	105.5	111.0	110.0
Cooling surface area, cm²	3,160	3,610	8,220
Heat transfer area, cm²	22.2	19.3	18.0
Thickness of steel cylinder wall, mm	2.03	2.15	2.41
Thickness of aluminium wall, mm	—	2.66	1.94
Basic coefficient of heat transfer at a velocity head of 254 mm w. g., kcal/m² h °C	810	1,090	—
Mean cylinder temperature, °C	148	146	145
Mean spark plug temperature, °C	221	254	271
Corrected static head, mm w. g.	483	193	170
Temperature of cooling air, °C	38	38	38
Corrected gradient of static head, mm w. g.	456	175	152
Total flow of cooling air, kg/h	1,090	659	559
Air flow to cylinder, kg/h	477	240	204
Air flow to head, kg/h	613	417	354
Total heat removed, kcal/h	10,130	10,100	9,830
Heat removed by the cylinder, kcal/h	2,970	3,200	3,530
Unit heat removed by cylinder, kcal/h °C	26.8	29.9	33.6
Heat removed by head, kcal/h	7,160	6,900	6,300
Unit heat removed by head, kcal/h °C	38.5	31.5	26.7
Cooling input, hp_f	1.60	0.41	0.31

In the case of the aluminium cylinder the cooling required at a given temperature could be achieved at a velocity head of $\Delta p = 193$ mm w.g. only. At this velocity head the prescribed temperature is attained in a steel cylinder at an engine output of 54% i.h.p. However, this does not mean, that an aluminium cylinder enables an increase of engine performance by 46%. Maximum performance depends, apart from cylinder temperature, also on the temperatures of the cylinder head and piston, and other factors.

Table 22

Comparative Temperatures of Steel and Aluminium Cylinders

	Mean temperature, °C	
	Cylinder	Flange
Steel Cylinder	148	158
Aluminium Cylinder	116	146
Difference	32	12

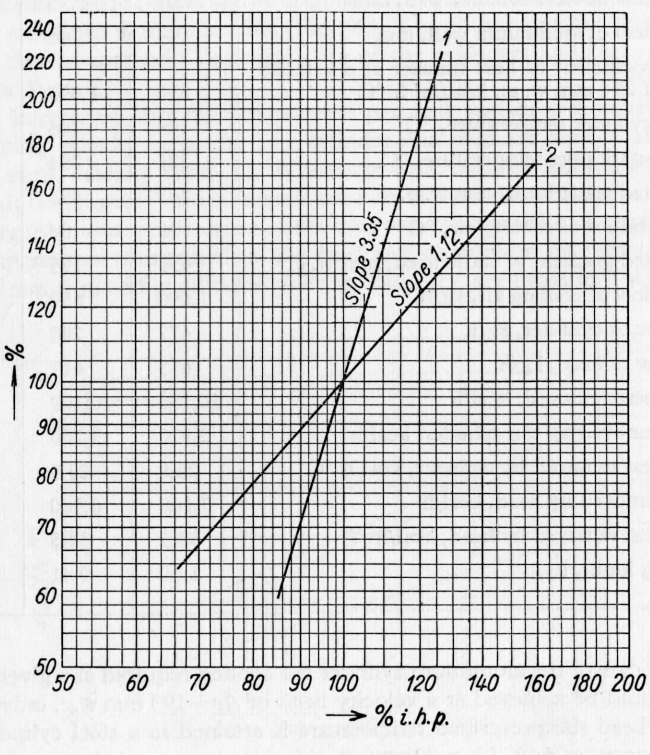

FIG. 93. Cooling input and quantity of cooling air plotted against indicated performance in % (Ranger).

1 — cooling input, *2* — quantity of cooling air. Temperature of the aluminium cylinder 145 °C, cylinder head temperature varying between 255—275 °C, constant air-fuel ratio of the mixture.

The change in the quantity of heat removed from the cylinder and cylinder head due to varying i.h.p. was investigated on an aluminium cylinder (Fig. 52). The temperature of the cylinder flange was maintained at the constant level of 146 °C. Corresponding figures of mass velocity and power required for cooling are given in Fig. 93. The experiments have shown that the temperature of the finned part of an aluminium cylinder decreases more rapidly, than that of the cylinder flange. Therefore, the conclusion seems to be justified, that flange temperatures in aluminium cylinders may be increased without involving the risk of simultaneously increasing the critical piston temperature. At a given flange temperature the temperature of the cylinder head is higher

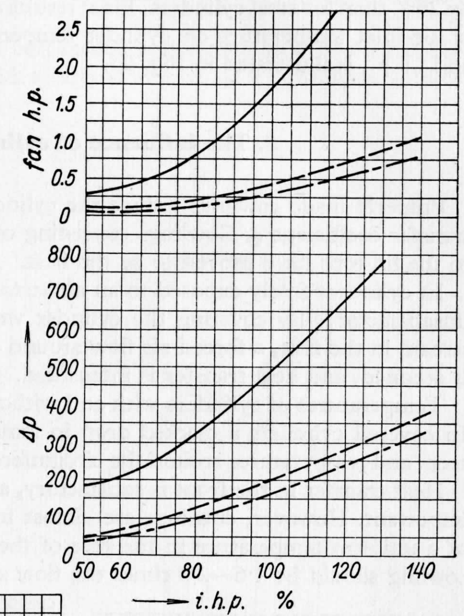

FIG. 94. Reduced cooling input and pressure head for aluminium cylinders of the Ranger aircraft engine plotted against indicated performance in %.

Full line — steel cylinder, dash line — aluminium cylinder dash and dot line — aluminium special cylinder.

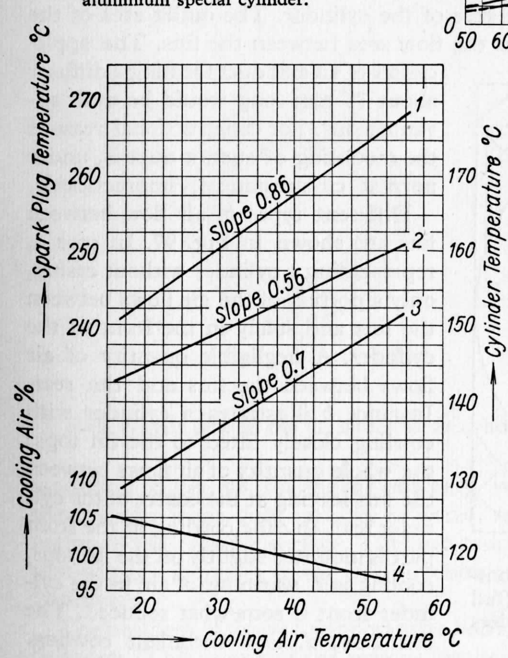

FIG. 95. Effect of inlet temperature of the cooling air upon cylinder temperature at constant indicated performance and pressure head across the cylinder.

1 — temperature of the spark plug, *2* — maximum cylinder temperature, *3* — average cylinder temperature, *4* — quantity of cooling air.

in an aluminium cylinder than in steel. These conditions are evident from Table 21.

In the case of aluminium cylinders the total power for cooling is lower by 25% than for steel cylinders. Final results are given in Fig. 94. The influence of air inlet temperature on cylinder temperatures, at constant velocity head and i.h.p., is illustrated in Fig. 95.

5. The influence of cylinder cowlings

Properly made cowlings reduce the cylinder temperature and increase the transfer coefficient q. Cowlings consisting of air guide sheets mounted closely to the fin tops have proved to be the best.

In cylinders freely exposed to an air stream, cooling of the cylinder rear is unsatisfactory. By covering the cylinder we prevent a disruption of the air stream in the rear, a forced air flow around the entire cylinder circumference is attained, and heat transfer is intensified.

Temperatures of cylinders with and without cowlings are shown in Fig. 96. In covered cylinders a marked drop in temperatures of the rear part can be seen, and temperatures around the circumference are evenly distributed.

Heat transfer in the front is satisfactory, and the use of a casing is not very important. However, cowlings are almost indispensable for the maintenance of a uniform temperature in the rear of the cylinder. The outlet area of the cowling should be 1·6—2·3 times the flow area between the fins. The application of an exit duct forming a diffuser about 75 mm long would be most advantageous. For constructional reasons the mounting of such a duct is, under normal circumstances, impracticable.

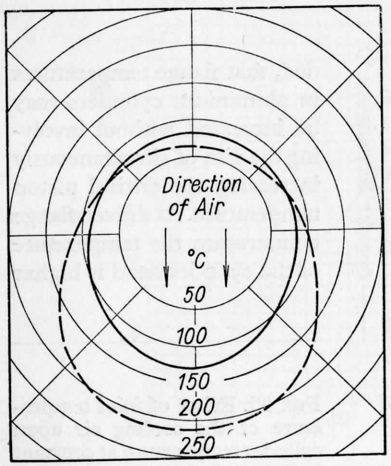

FIG. 96. Temperatures on the circumference of cylinders with cowlings (full line) compared with those of cylinders without cowlings (dash line).

Different types of air flow between fins are shown in Fig. 97. In case a, representing a cylinder without casing, only a portion of the air flows between the fins and solely in the front of the cylinder. A negligible quantity of air flows between the fins and the rear. Instance b illustrates a cylinder with cowling closely fitted to the fin tops; the whole quantity of air flows between the fins leaving at the centre of the cylinder rear. In c the cowling in the front part is mounted slightly off the fin tops, cooling and warming of air in the cylinder front is somewhat reduced. The last case, d, is a turbulent cowling.

At each entry of air a new thin boundary layer is formed, and an increased heat transfer attained. Turbulent cowlings are usually employed at the cylinder rear; their resistance to air is rather high, and, therefore, b type cowling is used for dense finnings.

A suitable cowling is one of the best means to maintain an even distribution of temperatures round the cylinder circumference. If a cowling is applied,

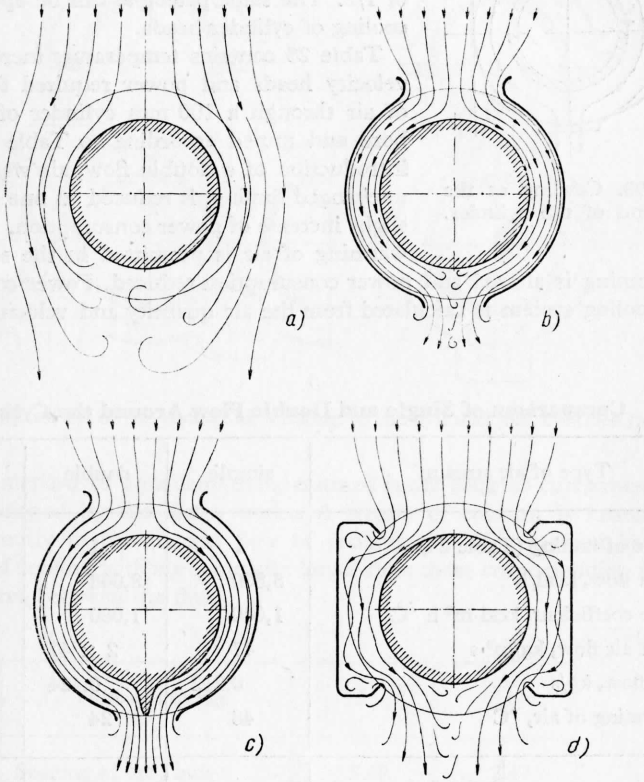

FIG. 97. Air flow in the fin interstice in cylinders with and without cowlings.

according to Fig. 98, for the air exit only, velocity drops between the fins of the cylinder front, heat transfer is worse, and temperature increases in this region of the cylinder surface. The rear casing can be declined at the entry so that velocity increases in the exit direction and cooling of the cylinder rear is improved.

The problem of uneven temperatures becomes more difficult with large bore cylinders. This is one of the reasons for the reduced application of air-cooling in large engines. Today, air-cooling is practical for cylinders up to a 160 mm bore also for engines running at heavy loads (aircraft engines). Air temperature is greatly increased in the long fin interstice particularly in finnings having a high efficiency. A good method of equalizing temperatures is illustrated in Fig. 99, where a double air stream is introduced with air flowing round 1/4 of the cylinder circumference instead of 1/2. The same principle can be applied to the cooling of cylinder heads.

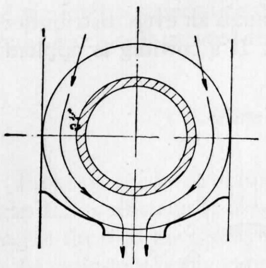

FIG. 98. Cowling at the rear end of the cylinder.

Table 23 contains temperature increases of air, velocity heads and power required for the flow of air through a 100 mm cylinder of a 200 mm bore and finned according to Table 24. By the introduction of a double flow, air warming, with unchanged finning, is reduced to one half with a slight increase of power consumption. If the same warming of air is permitted in the second case, the finning is altered and power consumption reduced. Power consumed by the cooling system is calculated from the air quantity and velocity head and

Table 23

Comparison of Single and Double Flow Around the Cylinder

Type of air stream		simple	double	double
Type of finning (Table 24)		I	I	II
Heat flow, kcal/h		8,800	8,008	8,800
Base coefficient, kcal/m² h °C		1,080	1,080	1,080
Unit air flow, kg/m² s		47	2 × 47	2 × 29.5
Air flow, kg/s		0.212	0.424	0.231
Warming of air, °C		48	24	44
	by friction	242	115	91.6
Loss in	bend-entrance	8.32	8.32	2.66
velocity head	bend-exit	8.53	8.53	2.41
mm w.g.	bend-centre	3.85	2.30	0.20
	total	262.70	134.15	97.17
Cooling input for 100% efficiency, hp		0.683	0.700	0.276

assuming the same fan efficiency in all cases. Power consumed by a single flow is 2·5 times larger than that by a double flow, with the same warming of air. Air resistances were calculated according to Berndorfer whose method is in full agreement with results obtained in practice.

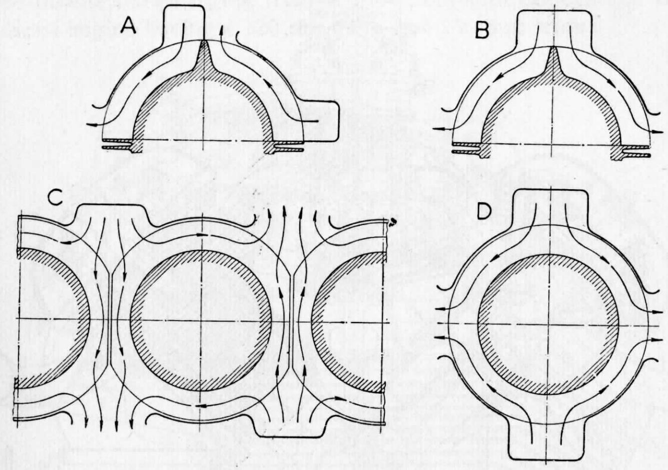

FIG. 99. Illustration of the double air ducting to the cylinder and cylinder head.

Another method of equalizing temperatures is the counter-current cooling system developed by the Tatra works. A stream of cold air is introduced countercurrently to the normal flow of cooling air. This method has the advantage of cooling with air quantities larger than those corresponding to the transition area between the fins.

Table 24

Type of finning	I	II
Spacing of fins, mm	3.49	2.47
Thickness of fins, mm	0.89	0.89
Height of fins, mm	31	31

Thus small fin spacing can be used. The countercurrent cold air is best introduced at the thermally most stressed region underneath the exhaust valve.

Fig. 100 illustrates this method applied in the Tatra T 603 car. A stream of cold air is led through a separate duct directly underneath the exhaust valve.

One part of the current passes through the fins to both sides of the cylinder, the other part flows to the finning of the cylinder head beneath the exhaust duct. The irregularity of temperatures around the cylinder circumference

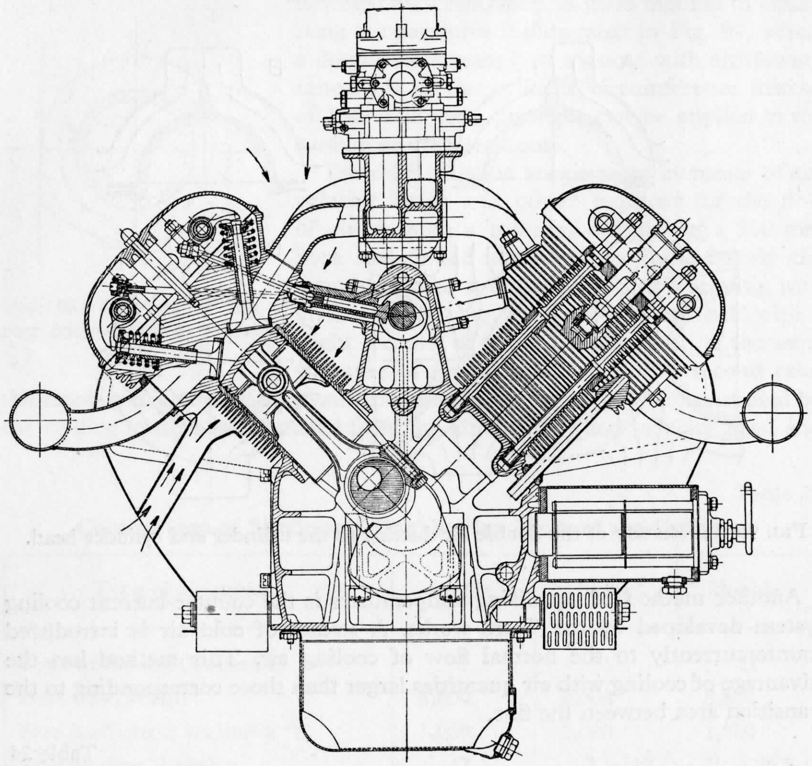

FIG. 100. On the Tatra 603 engine an auxiliary stream of cooling air is led directly underneath the exhaust valve. Arrows show direction of air flow.

was reduced to 5 °C. This cooling system is easy to assemble; the suction fan is located directly in the exit stream and the mouth of the supplementary air duct lies in the free atmosphere.

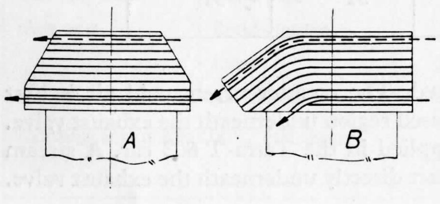

FIG. 101. Good and bad design for utilization of cooling air.
A — Air path through cold fins (dash line) is short, through hot fins (full line) long. Uneven heating of air. *B* — Air path through cold fins is long, through hot fins short. Uniform heating of the cooling air.

Cooling air should be fully utilized and should never pass any part of the cylinder without absorbing the maximum amount of heat. Air flowing along a cooler wall should be led by a longer path to have time for warming and absorbing the desired quantity of heat. Correct and incorrect directions of air-flow between fins are illustrated in Fig. 101. The application of correct principles of air flow is shown in Fig. 102 on the example of the cylinder head of a Tatra diesel engine having a 120 mm bore and 130 mm stroke.

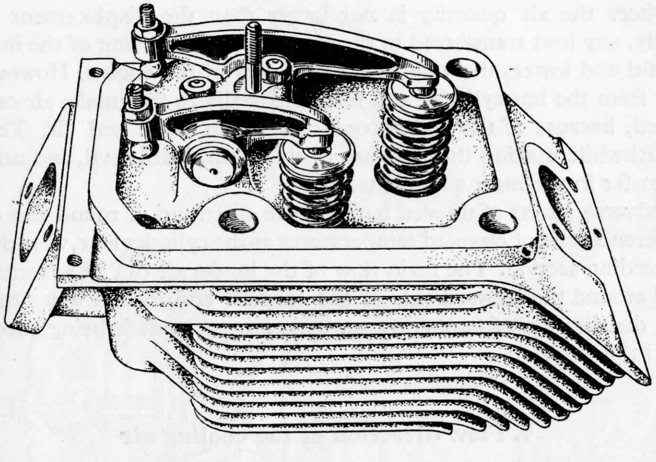

FIG. 102. Cylinder head of a compression-ignition engine. Air is directed by the curved fins to give better cooling in the region of the exhaust valve.

6. Internal cooling

This is the cooling effect exerted by the air passing through the cylinder upon cylinder walls, valves or inner walls of manifolds. The quantity of air passing through the cylinder is far smaller than that of the cooling air. Therefore its cooling capacity is also far smaller. Nevertheless, internal cooling is very effective, because the air comes into direct contact with wall surfaces which, shortly before, have been exposed to the glowing combustion products.

Internal cooling is brought about either by the combustion air solely or by surplus air. Outer cooling can never be replaced by inner cooling. Quantities of air larger than those corresponding to the rated displacement can be conveyed through the cylinders of engines without superchargers only by utilizing to some extent the pressure oscillations in the intake and exhaust manifolds. Quantities of air considerably exceeding the displacement volume can be put through the cylinder with the aid of blowers. Air is compressed in blowers to pressures considerably higher than the velocity heads produced by cooling fans. However, the compression of a unit weight of air is very expensive and is uneconomical in spite of the high heat absorption that can be achieved.

Apart from the surplus air being beneficially used for scavenging the combustion chamber, it is used with advantage for cooling the exhaust valve. Particularly in supercharged engines internal cooling of the piston crown near the exhaust valve became a necessity. A further advantage of supercharged engines is the reduced temperature of exhaust gases due to their dilution by the surplus scavenging air.

Some engine parts can be advantageously cooled by internal air even in cases where the air quantity is not larger than the displacement volume. Evidently, any heat transferred to the air prior to the closing of the inlet valve is harmful and lowers the volumetric efficiency of the engine. However, heat transfer from the hot cylinder and manifold walls to the intake air cannot be prevented, because of the direct contact between walls and air. Therefore, it is worthwhile making the best use of this unavoidable evil, and utilize this heat transfer in the most suitable way.

The adverse effects of uneven temperature distribution round the cylinder circumference, and of elevated temperatures at the cylinder rear, were discussed in a preceding section. The main flow of the intake air can be advantageously directed around the leeward side of the cylinder and in this way utilized for a better distribution of temperatures. This arrangement is being successfully applied in practice.

7. Flow direction of the cooling air

In air-cooled engines the inlet of cooling air can be arranged either on the side of the inlet valve or on that of the exhaust valve. Either arrangement has its advantages and shortcomings.

Cold air entering the hottest zone near the exhaust valve gives the highest temperature difference between cooling fins and air. Cooling is here very intensive and a maximum lowering of the cylinder head temperature is attained. Consequently, temperatures on the intake and exhaust side of the cylinder are equalized to a large extent, and possible distortions of cylinder head and valve seats are suitably affected. Therefore, this cooling arrangement should be applied in cases of cylinder heads bursting because of excessive thermal loads due to uneven temperatures. The high temperature of the exhaust valve region can sometimes cause a permanent distortion of the cylinder sealing surface with subsequent leakages or even burning of the cylinder head.

By arranging the cooling air inlet at the intake side, an intensive cooling of the inlet valve region is attained, heat transfer to the intake air is small and higher volumetric efficiency results.

Thus engine performances attained with this arrangement are better than those achieved by the cooling air entering the exhaust side. A properly designed cylinder head should ensure a sufficient cooling of the exhaust valve region. In this case the increased engine output is a clear point in favour of the second

arrangement. Also it must be borne in mind that particularly with this arrangement the major portion of the intake air is brought to the leeward side of the cylinder contributing to a better cooling of this region. Therefore, the second arrangement is generally preferred.

Sometimes, the arrangement of cooling air inlet is determined already by the design of the engine. Fig. 103 illustrates this point quite clearly. In the flat

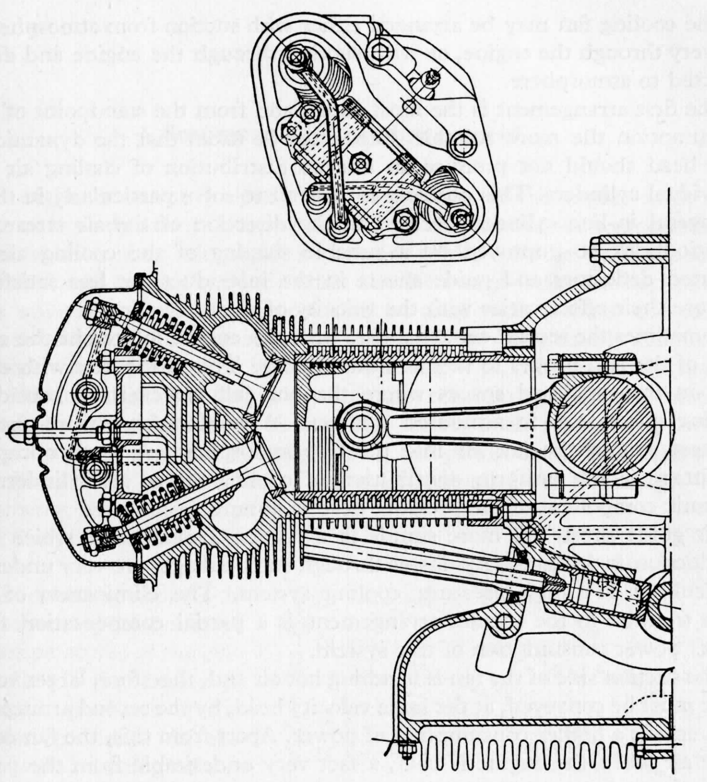

FIG. 103. Cylinder of the air-cooled Tatraplan engine. The inclined plane of the valves improves the passage of cooling air.

four-cylinder Tatraplan engine the obvious arrangement is represented by the exhaust manifold leading downwards and the intake manifold directed upwards. Cooling air inlet from below would be difficult to arrange, the air would be warmed by the exhaust pipe prior to its reaching the cylinder surface, and its cooling capacity would be greatly reduced. In this engine cooling air must be brought to the region of the inlet valve and obviously from above.

In all described arrangements it is supposed that valves are placed roughly in the direction of the air flow so that large flow areas of cooling air are obtained even with small cylinder spacings.

8. Arrangement of the cooling fan

The cooling fan may be arranged either with suction from atmosphere and delivery through the engine, or with suction through the engine and delivery directed to atmosphere.

The first arrangement is the most usual, and from the standpoint of power consumption the more suitable. Care must be taken that the dynamic pressure head should not produce an uneven distribution of cooling air to the individual cylinders. This problem is difficult to solve particularly in the case of several in-line cylinders located in the direction of the air stream. The situation can be improved by a suitable shaping of the cooling air duct. Inserted deflectors and guide sheets in the inlet duct are less satisfactory, because their effect varies with the velocity of air.

Sometimes the second arrangement, with the engine located in the suction zone of the fan, proves to be more satisfactory. This is the case with engines built-in into enclosed spaces where the hot exit air exerts a considerable heating effect, e.g. in armoured cars, etc. A suction fan ensures here the shortest exit of the hot air into the free atmosphere. A further important advantage is the uniform distribution of cooling air to all cylinders. The dynamic component of the pressure head is absent.

Air guide sheets are more simple in a suction arrangement which means a reduction in leakages. Air losses through such leakages are very undesirable particularly in a high pressure cooling system. The elimination of guide sheet troubles in the suction arrangement is a partial compensation for the higher power consumption of this system.

The suction side of the fan is handling hot air and, therefore, larger volumes of air must be conveyed, at the same velocity head, by the second arrangement. This entails a higher consumption of power. Apart from this, the fan conveys more air when the engine is cold, a fact very undesirable from the point of view of regulation. Another advantage is the possibility of locating on the delivery side a diffuser by which a part of the energy of the exit air, otherwise lost, can be recovered.

It is hard to state which of the two systems is the better. According to actual conditions either of them can be the more suitable one. In the T 603 engine the suction arrangement proved to be more satisfactory for the following reasons: movement of air through the engine space was simplified, engine performance was increased due to the flow of cold air along the carburetter to the inlet valve, cooling air was evenly distributed to all cylinders, and the control of the cooling system was facilitated. Apart from this a countercurrent cooling of the exhaust valve region was easy to arrange (see Fig. 100).