Chapter 9

Engine cooling system

Klaus-Dieter Emmenthal

9.1 Introduction

Much of the energy available in fuel is converted into heat during the working cycle of an engine. Although this heat can be used to warm the vehicle interior, most is transferred to ambient air, either by air cooling or more often by water cooling. From a practical standpoint, water cooling is still a form of air cooling, but using water as a transfer medium. This chapter is solely concerned with water cooling systems for passenger cars.

Specifications have changed markedly in recent times for systems of this type. Traffic density demands compact, high-performance vehicles that meet safety standards for body design and structure. This poses the problem of how to transfer the relatively large amount of engine heat to ambient air by means of a radiator, for which there is generally very little room, and in a space which is further constricted by the various auxiliary power and servo units. Furthermore, aerodynamics, body styling and visibility considerations have led to steeply sloped hoods (bonnets), so that the air inlet has become smaller, thus emphasizing the importance of cooling fan design. Either the space problem or transverse mounting of the engine/transmission assembly will necessitate the use of an electric cooling fan, and the fan's performance is limited in turn by electrical system capacity.

Certain things must be considered before designing the cooling system. The available cooling air inlet cross-section and the maximum available space for the radiator and cooling fan are determined by vehicle styling and engine placement. The effect of ram air can be roughly estimated from the road performance map and heat rejection to coolant can be estimated from engine output. To minimize fuel consumption and noise, as well as electrical demand when the fan is switched on, cooling fan power consumption should be minimized, and the cost of the radiator and the fan with its drive must also be considered.

Finally, the problem of cooling fluid to ambient air heat transfer must be solved before the design of the radiator, water pump and cooling fan can be finalized: the total effect of radiator size and cooling fan power is of particular importance. These data plus, for example, coolant flow data can be found experimentally, through computation, or through a mixture of both.

Experimental testing is both time-consuming and expensive: yet calculation alone is an inadequate foundation for design. Calculation must be supported by experimental data so that the precise effects of any factor that may influence the size of the radiator, cooling fan, or water pump can be determined during the initial design stage.

The following section is concerned with steady-state heat- and flow-related problems of water cooling systems in passenger cars. Heat transfer problems during engine start up and idling, and technical details such as thermostats, hose connections, etc., will not be considered.

9.2 Cooling system requirements

The cooling system, consisting of the radiator, cooling fan and water pump, must be designed so that the cooling fluid temperature stays below boiling point, or below an upper temperature limit for the cooling fluid, during all practical operating conditions. From Fig. 9.1 it is clearly

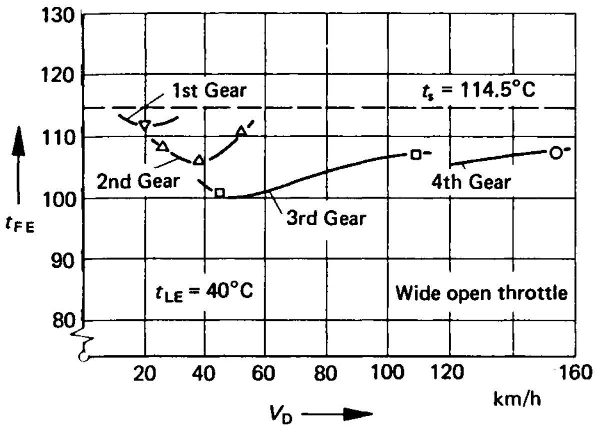


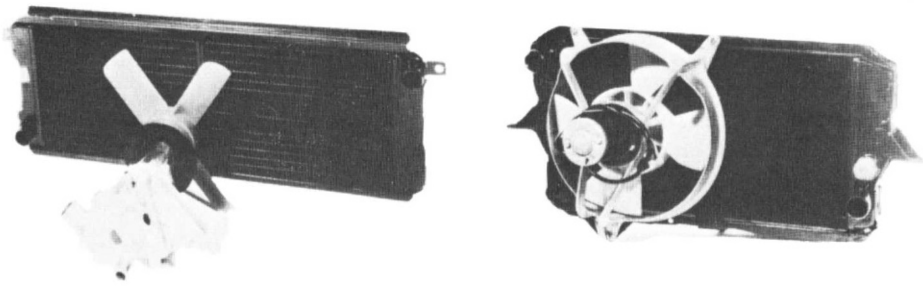
Figure 9.1 Coolant temperature map of a 1.6 litre passenger car (wind tunnel measurements)

apparent that coolant temperature decreases with increasing vehicle speed for a specific cooling system layout; it approaches boiling point in the first-gear range.

The maximum coolant temperature requirements can lead to completely different engineering approaches within a given class of vehicles. For example, the radiators and cooling fans of vehicles 'A' and 'B', both having the same engine power, are compared in Fig. 9.2. One system has a wide mesh matrix with a cooling fan driven by the crankshaft, the other a dense mesh radiator with an electric cooling fan. Both require virtually the same installation space.

Other requirements must be fulfilled along with the cooling function:

- Low total manufacturing cost
- Low overall weight
- Low operating cost (i.e. low energy consumption of fan, good drag coefficient, reliability)
- No annoying noises, from high cooling fan tip speeds, etc.



Vehicle A		Vehicle B	
Pressure drop	$\zeta_K = 3$	Pressure drop	$\zeta_K = 6$
Coefficient of heat transfer per unit of frontal area	6.5 kW/m²/°C	Coefficient of heat transfer per unit of frontal area	10 kW/m²/°C
Radiator material	Cu-Ms	Radiator material	Steel
Frontal area	0.15 m²	Frontal area	0.11 m²
Cooling fan power	1.1 kW	Cooling fan power	0.08 kW

Engine power	85-90 HP
Displacement	1.6 ℓ

Figure 9.2 Radiator and cooling fan systems for vehicles with the same engine power

For example, Fig. 9.3 shows the qualitative curves for manufacturing cost, operating cost, and fan noise against radiator frontal area. As radiator frontal area increases, cooling fan noise decreases, since ram air provides most of the heat transfer. Manufacturing cost would be minimized if an expensive cooling fan were not required, and if the radiator size were limited, thus limiting the associated technical problems of installation. Operating cost would be high if the power required to drive the fan were high. A large radiator frontal area could increase the cooling system's contribution to overall air drag and result in increased fuel consumption. The 'optimum' result can be found within the shaded area in Fig. 9.3.

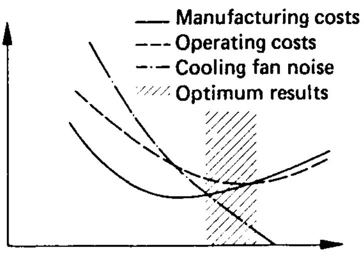


Figure 9.3 Optimizing a water cooling system

The target values for the individual components of the cooling system are established in a so-called 'Specification Catalogue' for the particular vehicle.

What follows mainly concerns cooling system aspects of thermal safety, cooling fan power consumption requirements, manufacturing cost, and weight.

9.2.1 Design goals in the road performance map

In practice, very large, and therefore expensive, radiators would result from the need to run the vehicle at steady state for all operating points in the road performance map. Extreme gradients in the order of 20 per cent, which must still be negotiable by the vehicle according to the map, are only encountered for such short distances that they are meaningless as design criteria. Studies completed by Haas[9.1] aimed to define criteria in the road performance map, so that the cooling system would handle all conditions that occur in normal vehicle operation. Accordingly the cooling system must be sized for two specific operating points that have crystallized out of long-term observations made of the traffic scenario in Europe:

1. The vehicle must be able to drive continuously up a 10 per cent gradient at a speed of 25 km/h (15.6 mile/h) while fully loaded and towing maximum trailer weight.
2. The vehicle must be able to run at top speed without constraints.

The following specifications apply to the temperature differences between the cooling air inlet and coolant fluid inlet, in addition to the conditions for the technical aspects of driving: while climbing hills and while at top speed, this difference may not be greater than 80°C or 65°C respectively. This applies for 'European' operating conditions and when pressurized systems with water and antifreeze are used,[9.2] as is common practice today.

Figure 9.4 shows the road performance map for a vehicle with a 66 kW

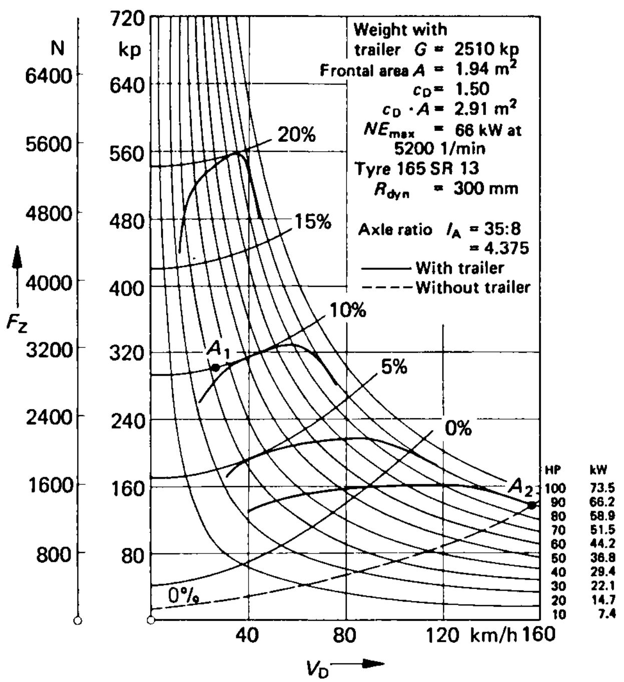


Figure 9.4 Design targets for the cooling system in the road performance map

(88 hp) engine towing a trailer. The resistance curve for level road operation without a trailer has been included: design targets are identified as A_1 and A_2. A drag coefficient was assumed (as per Beauvais,[9.3] see also section 4.3.2.13) and the towing vehicle cross-section served as the reference frontal area. It can be seen in Fig. 9.4 that a power of around 23 kW is required for hill climb A_1, using first gear with the engine under partial load. The full power level is 66 kW at design target A_2.

9.2.2 Heat rejection to coolant by the engine

The heat rejected to the coolant by the engine is the next item of interest, since the required power levels have been established in the previous section. Various attempts have been made to calculate this heat with greater precision than that provided by the raw figure of 30 per cent of the input fuel energy, which was mentioned in section 1.1.1. Thus, for example, Drucker[9.4] gave an equation for computing the heat rejected to coolant fluid. Work by Cramer[9.5] concerning the influence of engine thermal efficiency on heat flux to the coolant should also be mentioned here. However, since the amount of heat transferred to the coolant depends on such items as cylinder head design, precise test stand measurements with a sample engine are indispensable for optimum cooling system design. Such test data are compiled in Fig. 9.5 for various engines in

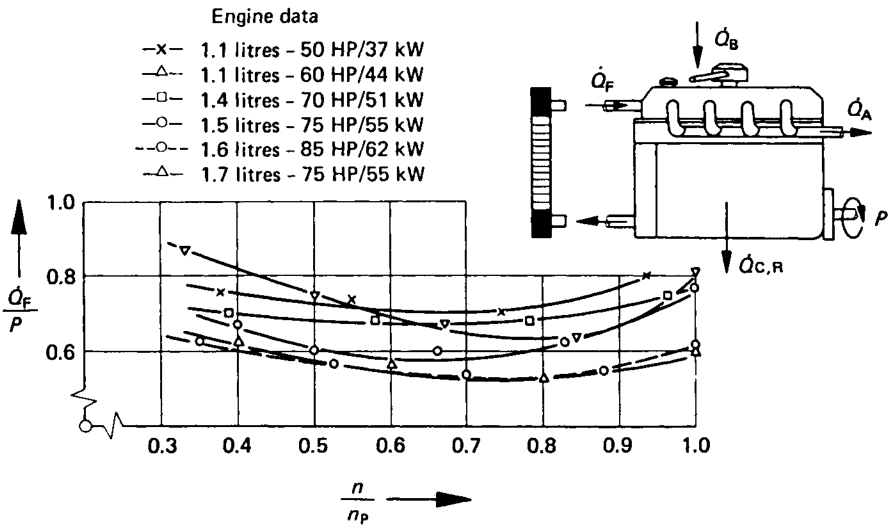


Figure 9.5 Rejected heat to coolant for various engines under full load

the form of heat rejected to the coolant at full load, plotted against engine speed as a fraction of the engine speed at peak power.

Three aspects are of special interest:

- The dependence of heat rejection upon engine speed is very slight.
- The ratio of heat rejection to output power is between values of 0.5 and 0.7, and thus is less than unity.

- The ratio of heat rejection to output power decreases with an increase in engine displacement for the same number of cylinders, due to the variations in combustion chamber volume and surface area for heat transfer.

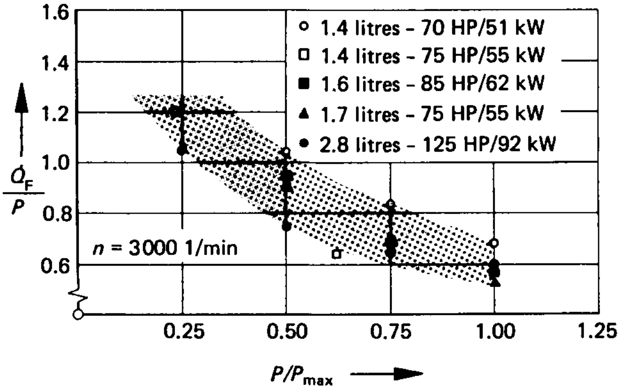


Figure 9.6 Rejected heat to coolant for different engines under part load

Engine heat flow under part load must be known for sizing a radiator for hill climbing. Figure 9.6 shows measurements for a series of 4-cylinder engines, and one 2.8 litre, 6-cylinder engine (the latter from Haas[9.1]). The relationship of heat flow to power clearly increases to values above one as the power ratio P/P_{max} decreases.

9.3 Elements of the cooling system

9.3.1 Cooling air inlet

The design-related preconditions for installing a radiator in a vehicle are unfavourable, as compared to those for designing a ducted radiator for an aircraft (see Linke[9.6]). Available space is defined and limited by cross-members, the engine, and auxiliaries, and air flow is constricted by car body design and auxiliary headlights. Figure 9.7 shows that the air

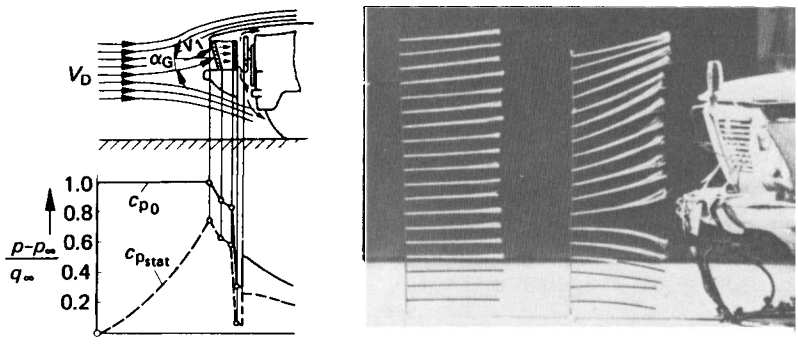


Figure 9.7 Cooling air flow and the pressure characteristic in the cooling system of a passenger car

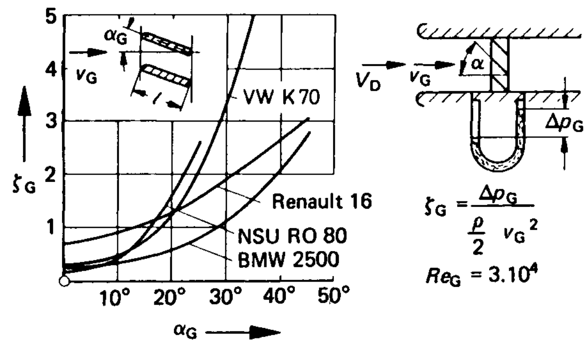


Figure 9.8 Loss coefficient of various radiator grills for angular air inflow

stream does not approach the inlet grill parallel to the road surface as is frequently assumed.[9.25] This flow pattern must be considered when the radiator grill is designed. Figure 9.8 shows that the pressure drop coefficient ζ_G of the radiator grill is largely dependent upon the angle of attack α_G of the radiator grill to the local air flow.[9.25] According to Eqn 2.53 the following definition applies for the grill:

$$\zeta_G = \frac{\Delta p_G}{\frac{\rho}{2} v_G^2} \tag{9.1}$$

The flow in front of a porous wall (analogous to a radiator) was studied by Taylor.[9.7] For a vehicle speed V_D the cooling air face velocity v_A can be calculated for two different radiator arrangements (with or without shroud baffling of the air leaving the radiator) as a function of the pressure loss coefficient ζ_t of the air duct, including the radiator. This assumes that all components that cause a pressure loss are represented as a single item by the pressure loss coefficient ζ_t. Figure 9.9 depicts curves for the functions:

$$\frac{v_A}{V_D} = \frac{1}{1 + \frac{\zeta_t}{4}} \tag{9.2}$$

$$\frac{v_A}{V_D} = \frac{1}{\sqrt{(1 + \zeta_t)}} \tag{9.3}$$

The case of an unbaffled air flow from the radiator corresponds approximately to that of a 'free-flying' oil cooler installation in a racing car ($\zeta_t = \zeta_k$), while the shroud-baffled air flow from the radiator is the general radiator arrangement in a passenger car. Individual elements such as radiator, grill, air baffling and engine compartment, have varying effects on system pressure loss. Limits are at approximately $\zeta_t = 4$ for a very porous system and $\zeta_t = 8$ for a severely constricted system. In this range of pressure loss coefficients, experiments with scale models and full-size vehicles agree well with theory; the difference between the two limiting radiator arrangements is only small. As shown in Fig. 9.9, higher cooling

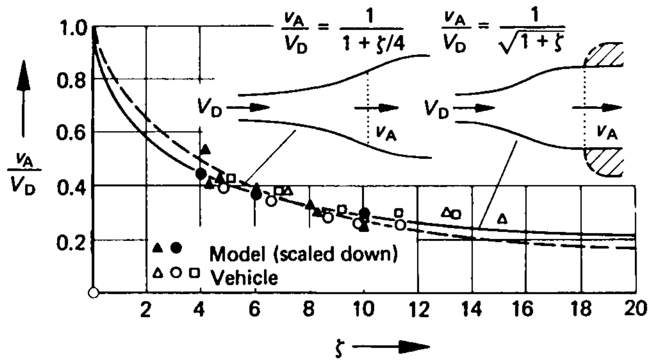


Figure 9.9 Ratio of cooling air face velocity to vehicle speed against different cooling system loss coefficients

air face velocities can be achieved with a 'free-flying' radiator arrangement only for small pressure drop coefficients ζ_t.

The cooling system should contribute as little as possible to total drag. Assuming that air flows into and out of the cooling system at the same velocity, and that static pressure at the outlet corresponds to ambient pressure,[9.8] then:

$$c_{DR} = 2\left[\frac{1}{\sqrt{(1+\zeta_t)}} - \frac{1}{1+\zeta_t}\right] \tag{9.4}$$

for the resistance due to cooling air flow through the system. The drag coefficient of the cooling system c_{DR} can be converted to the cooling system's share of the vehicle drag coefficient through the ratio of radiator frontal area A_R to vehicle frontal area A:

$$\Delta c_{DR} = c_{DR}\,\frac{A_R}{A} \tag{9.5}$$

For example, for a very porous cooling system ($\zeta_t = 3$) and a frontal area ratio of 0.1 of the vehicle cross-section, the radiator system adds to the vehicle's drag coefficient $\Delta c_{DR} = 0.05$ according to Eqns 9.4 and 9.5. Most current passenger cars have a drag coefficient in the range $0.35 \leqslant c_D \leqslant 0.45$. Thus the cooling system's share of the drag coefficient could amount to more than 10 per cent for a very porous system.

The drag due to cooling can only be approximately determined from Eqn 9.4, since it does not consider the resistance due to the change to the air flow around the nose of a vehicle caused by air through the radiator system (interference drag). Schenkel[9.9] examined the influence of front-mounted spoilers on flow relationship within the engine compartment; see also section 4.3.2.9.

9.3.2 Radiators for passenger cars

All current radiators are of the cross-flow type, though different materials are used, e.g. heavy metal radiators (iron, brass, copper) and light metal radiators (aluminium). Five different aluminium radiator designs are

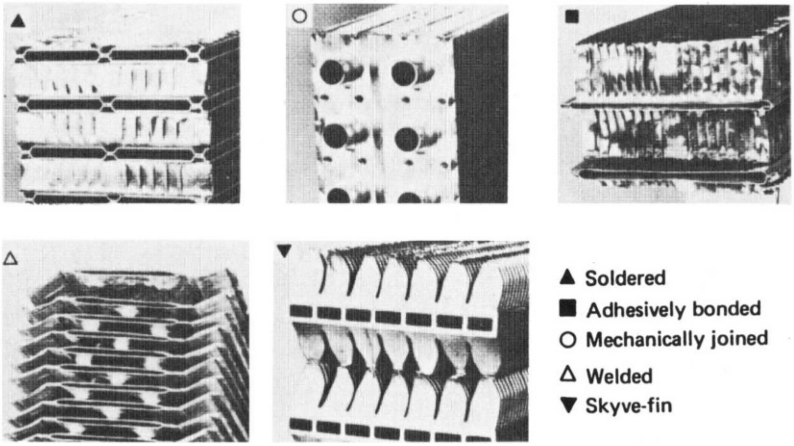


Figure 9.10 Aluminium matrix designs

shown in Fig. 9.10.[9.26] They are a soldered core of VW manufacture, the standard VW-Sofica radiator core, the adhesive bonded AED-Covrad core, the laser welded core design of Union Carbide, and the Skyve Fin core design of Schmöle.

Low manufacturing cost and minimum fan power are the most important factors in the development of car radiators. Here the work of Dehn,[9.10] Lorenz,[9.11] and Kays and London[9.12] should be mentioned.

9.3.3 Radiator cooling fan

Axial fans are used in almost all cars to increase the cooling air flow. Low cost and power consumption and low noise are the main advantages. Belt drive from the engine crankshaft is the cheapest, but optimum matching of the fan to the various load conditions is best realized by an electric fan.

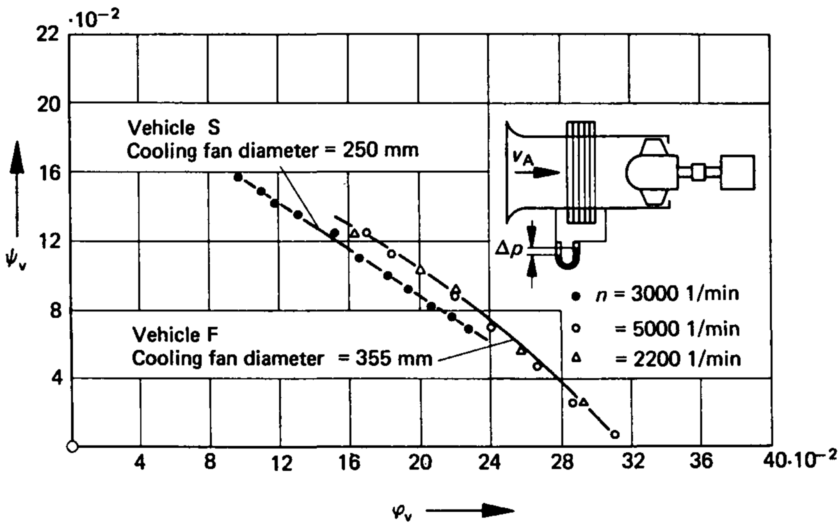


Figure 9.11 Cooling fan map

Cooling fan data must be known before designing a cooling system. Fan characteristics should be presented as the relationship between pressure coefficient and volume coefficient (Fig. 9.11), since the cooling fan map would then shrink to a single curve. Extensive work has been done in this field by Eckert[9.13–9.15] and Marcinowsky.[9.16]

9.3.4 Water pump

Centrifugal (radial) pumps are used exclusively in vehicle cooling systems to circulate the coolant, partly because of cost. They are usually crankshaft-driven.[9.17] Dimensionless presentation of the characteristics is recommended (Fig. 9.12).

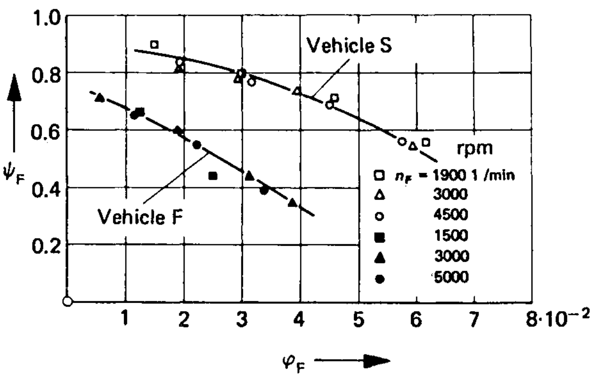


Figure 9.12 Water pump map

9.4 Designing the cooling system

To design a cooling system, the following must be known: heat rejection by the engine, vehicle road speed (road performance map), and the flow rate and composition (glycol additive) of the coolant. A procedure for designing and computing has been developed by Emmenthal.[9.18]

9.4.1 The radiator as a cross-flow heat exchanger

Nusselt[9.19] described the laws of heat transfer in cross-flow. Bošnjaković et al.[9.20] expressed the relationships in the form of radiator efficiency Φ, which is dependent upon the flow capacity rates of the fluid W_F and the air W_L, and the product of the overall coefficient of heat transfer and heat transferring surfaces. The average coefficient of heat transfer can also be referred to other surface areas since the overall coefficient of heat transfer is, by itself, not of importance in this context, but only in the previously mentioned combination with heat-transferring exchanger surface areas. The frontal area of the radiator is taken as a reference surface since it is of decisive importance in passenger car construction. Also, since it is more easily measured than surface areas within the radiator core, errors due to imprecise surface measurement become negligible when this reference is used. The drawback, that the overall coefficients of heat transfer k_A will be solely device-dependent, can be accepted in this case.

The function:

$$\Phi\left(\frac{k_A A}{W_L}, W_L/W_F\right) = \frac{t_{LE} - t_{LA}}{t_{LE} - t_{FE}} \tag{9.6}$$

is reflected in Fig. 9.13 for the case where the coolant flow will be compensated for with respect to the temperature profile for each heat exchanger element (W_F is well mixed transverse to its own flow direction).

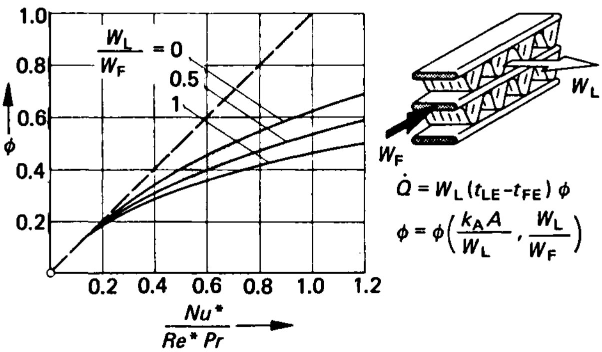


Figure 9.13 Radiator efficiency of the cross-flow heat exchanger

The radiator efficiency Φ that achieves the uppermost limit of unity if the air leaves the radiator at the coolant fluid inlet temperature, could assume values of $\Phi = 0.8$ for very densely packed radiators and low air velocities (hill climbing). The ratio of W_L/W_F is between limits of 0.05 and 0.5.

An enclosed term is given in ref. 9.20 for the function $\Phi(k_A A/W_L, W_L/W_F)$. This is used in the following for the calculation of heat transfer.

$$\Phi = \frac{W_F}{W_L}\left[1 - \exp\left\{-\frac{W_L}{W_F}\left(1 - \exp\left[-\frac{k_A A}{W_L}\right]\right)\right\}\right] \tag{9.7}$$

The coefficient of heat transfer k_A of the radiator must be known for evaluation of Eqn 9.7. With a metal radiator, the thermal resistance of the material between air and coolant fluid will have subordinate importance: the overall coefficient of heat transfer is primarily a function of the air and coolant-related heat transfer coefficients α_L and α_F.

Various approaches have been used to compute the overall air-related coefficient of heat transfer. First of all the flow within the matrix must be examined (see Beauvais[9.21] and Paish[9.22] and more recent results from Wong and Smith[9.23]).

None of the methods of computing coefficients for heat transfer and pressure loss is precise enough for radiator sizing and for predicting the operating behaviour. Also, since turbulators are sometimes used in the coolant tubes to improve heat transfer per unit of frontal area in compact radiators, knowledge of the coolant related coefficient of heat transfer α_F is also required: this, as will be shown later, is a primary co-determinant of the value of the overall coefficient of heat transfer during vehicle operation. Hence precise measurements of the radiator characteristic data are used for calculating heat transfer.

9.4.2 Measurement of radiator data

Radiator data can be determined at a test stand, as in Fig. 9.14. This involves use of a small wind tunnel. Measurements are made of air temperature rise (air side Δt), due to heat rejection by the heated core, and coolant temperature drop (coolant Δt), for different coolant flows. Air and

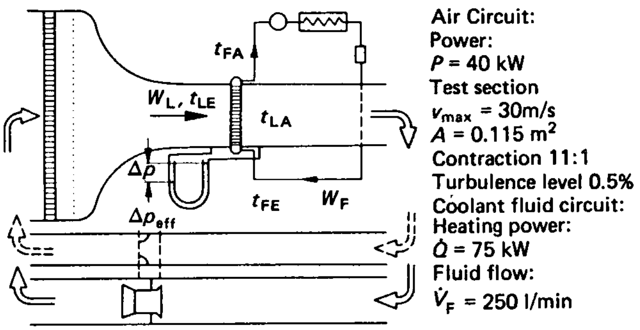


Figure 9.14 Schematic of a test stand for measuring radiator data

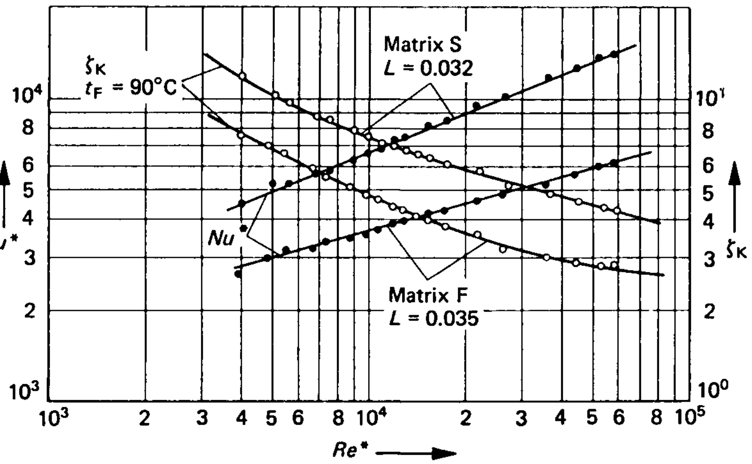


Figure 9.15 Heat transfer and air-side pressure loss of two different matrix designs

coolant flow-related pressure losses are also determined. Results are plotted in non-dimensional form. Heat transfer and air-related pressure losses of two different designs are shown versus Reynolds number, based on face velocity and radiator matrix depth, in Fig. 9.15.

Figure 9.16 shows the dependence of heat transfer upon coolant Reynolds number within the tubes of the radiator. From the characteristics of the curves it can be seen that heat transfer is strongly dependent upon coolant velocity, particularly at high air velocities.

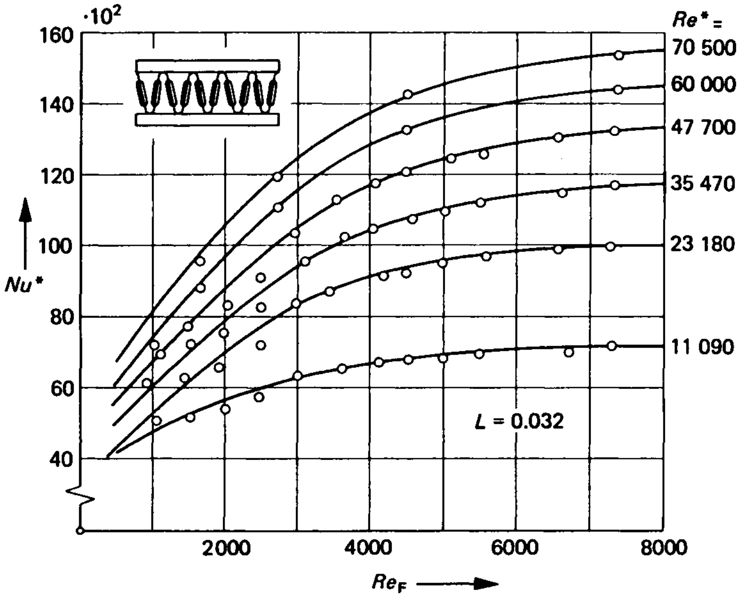


Figure 9.16 Heat transfer of a radiator matrix

9.4.3 Sizing the radiator

Calculations for radiator layout must take into consideration the fact that in practice air flow through a radiator is not uniform over its surface. Since parts of the radiator are frequently obstructed by bumpers or bodywork, the velocity profile of the incoming air is non-uniform. The cooling fan may act on the whole radiator or only part of it, so that its share in the velocity profile may also be non-uniform.

The heat flow to be rejected to the coolant by the engine is known from measurements for the previously mentioned engine loads: this can be expressed as:

$$\dot{Q}_F = W_F\,(t_{FE} - t_{FA}) \tag{9.8}$$

Therefore, the coolant flow capacity rate W_F is established for a chosen reduction in coolant temperature ($t_{FE} - t_{FA}$) at the radiator and the coolant flow of the pump is established for a known specific heat c_F of the coolant, or the reverse.

In practice, one would aim for the lowest possible temperature difference to achieve higher values of radiator efficiency (see Fig. 9.13), good coolant fluid-related heat transfer coefficients in the radiator, cylinder head and engine block, and low thermal stress levels within the engine. However, water pump power consumption will increase markedly with increasing coolant fluid flow, and so the advantages gained in heat transfer have to be set in relation to increased power expenditures.

The following applies for relationships between the heat flow, the radiator efficiency and the air and coolant inlet temperatures:

$$\dot{Q}_F = W_L\,(t_{LE} - t_{FE})\Phi \tag{9.9}$$

From Eqns 9.9 and 9.7, one obtains:

$$\Phi = \frac{t_{LE} - t_{LA}}{t_{LE} - t_{FE}} = \frac{W_F}{W_L}\left[1 - \exp\left\{-\frac{W_L}{W_F}\left(1 - \exp\left[-\frac{k_A A}{W_L}\right]\right)\right\}\right] \tag{9.10}$$

In Eqn 9.10 the temperature t_{LA} of the air leaving the radiator is a disruptive factor in further use of the equation. This quantity can be eliminated by combining Eqns 9.6, 9.7, 9.8 and 9.9.

$$\frac{t_{LE} - t_{FA}}{t_{LE} - t_{FE}} = \exp\left[-\frac{W_L}{W_F}\left\{1 - \exp\left(-\frac{k_A A}{W_L}\right)\right\}\right] \tag{9.11}$$

In solving Eqn 9.11 for coolant outlet temperature t_{FA} at the radiator, one obtains:

$$t_{FA} = t_{LE} - (t_{LE} - t_{FE}) \exp\left[-\frac{W_L}{W_F}\left\{1 - \exp\left(-\frac{k_A A}{W_L}\right)\right\}\right] \tag{9.12}$$

The following quantities and interdependencies are known in both Eqns 9.8 and 9.12:

Eqn 9.8

$\dot{Q}_F$	Known from measurements at the engine; see Figs 9.5 and 9.6
W_F	Defined by the coolant fluid flow and the dependence of the data of the fluid upon coolant temperature: achievable at a defined temperature difference $(t_{FE} - t_{FA})$
$t_{FE} - t_{FA}$	Defined by design guidelines: calculated at a given coolant flow rate

Eqn 9.12

t_{LE}	Defined by the maximum ambient temperature or the temperature of the air leaving from a front-mounted condenser for an air conditioner
t_{FE}	Defined by design criteria
W_F	Known from Eqn 9.8
k_A	Known from measurements of the dependence of the coefficients of heat transfer upon coolant and air flow

Equation 9.12 is evaluated by increasing the air flow capacity rate W_L for the given radiator frontal area A of a known matrix design, until the coolant fluid outlet temperature according to Eqn 9.8 sets in. At least a portion of the air side capacity rate will be provided by available ram air in a front-mounted radiator arrangement; the remainder must be provided by the radiator cooling fan.

There is a set procedure for radiator sizing. The radiator is calculated, section by section, as shown in Fig. 9.17. For a non-uniform fan velocity, the sections are broken down further into, for example, square radiator elements. The core depth L is established when the matrix design is selected. In the example cited, the core height H is governed by design limitations. Calculation starts with the first section: this is treated initially as if it had to effect the entire heat transfer by itself. Cooling air velocity v_A

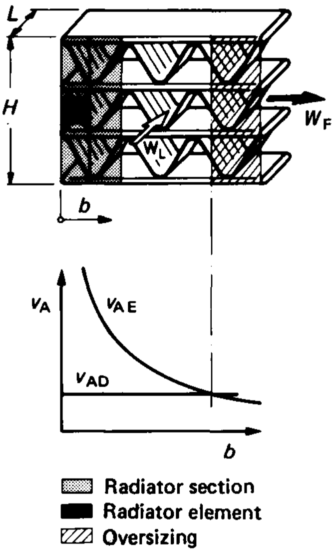


Figure 9.17 Subdividing the radiator frontal area for radiator sizing

is increased step by step until the coolant fluid temperature t_{FA} required by Eqn 9.8 is achieved. The required cooling air velocity is designated v_{AE}. A second section is added to the first. Both of these together now represent the radiator. A smaller v_{AE} velocity is now sufficient for heat transfer. This procedure continues until the width b_{max} that was prespecified by design is reached. Heat cross-transfer within the matrix is not considered.

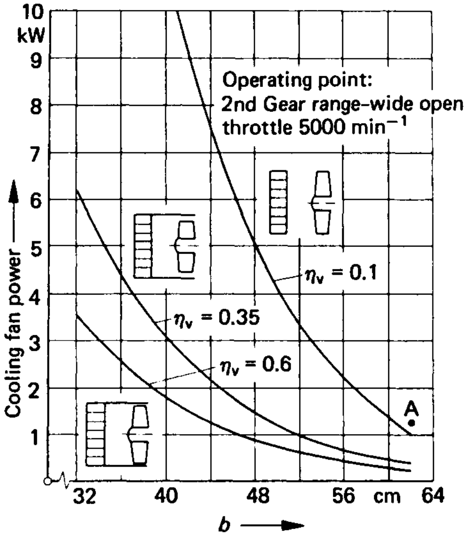
Cooling air velocity v_{AD} due to ram air can be determined for a prescribed driving speed and for the previously measured resistance characteristic ζ_t (v_A) of the grill, radiator, and engine compartment as air ducting elements, by specifying a starting value for cooling air velocity v_A, followed by iterative calculation of the function:

$$\frac{v_A}{V_D} = \frac{1}{\sqrt{[1 + \zeta_t(v_A)]}} \tag{9.13}$$

The consistent decrease in required cooling air velocity v_{AE} is entered over the radiator width b in Fig. 9.17; at a specific width, it equals that of the velocity v_{AD} supplied by ram air.

The radiator is over-sized to the right-hand side of this abscissa value; on the left-hand side, a cooling fan must be provided to bring the air velocity from v_{AD} to $v_{AE}(b)$, in order to handle cooling tasks.

The air flow that the cooling fan must supply can be calculated, since the width b and height H constitute the radiator's frontal area: the pressure loss that has to be overcome by the cooling fan can then be determined from the velocities and resistance characteristics of the individual components. For example, required cooling fan power has been entered over the radiator width in Fig. 9.18 for different degrees of fan efficiency. These computations for the cooling system of a Volkswagen K 70 passenger car show that a reduction in radiator width leads to very high levels of required cooling fan power, particularly with an unshrouded cooling fan.



Point A: VW-K70

Figure 9.18 Cooling fan power versus radiator width, for different degrees of fan efficiency

Segmenting the radiator frontal area in individual elements can also respond to a very non-uniform face velocity profile due to car body components in front of the radiator. Velocity distributions at the radiator frontal area for a given vehicle are shown in Fig. 9.19 for a driving speed of 170 km/h (106 mile/h). Calculation of the coolant fluid temperature reduction for the mean velocity and for the given profile shows that averaging the face velocity leads to incorrect results (Fig. 9.20).

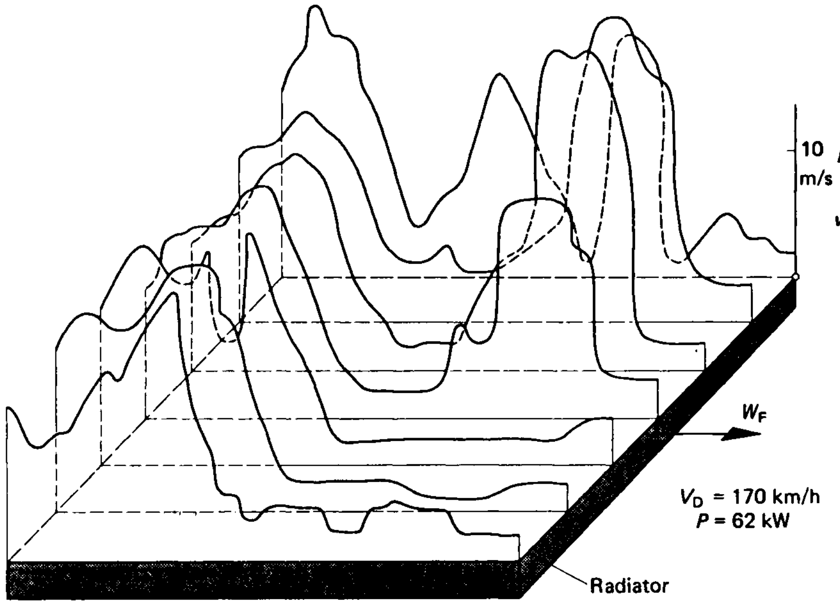


Figure 9.19 Cooling air flow profile of a passenger car with a 1.5 litre engine

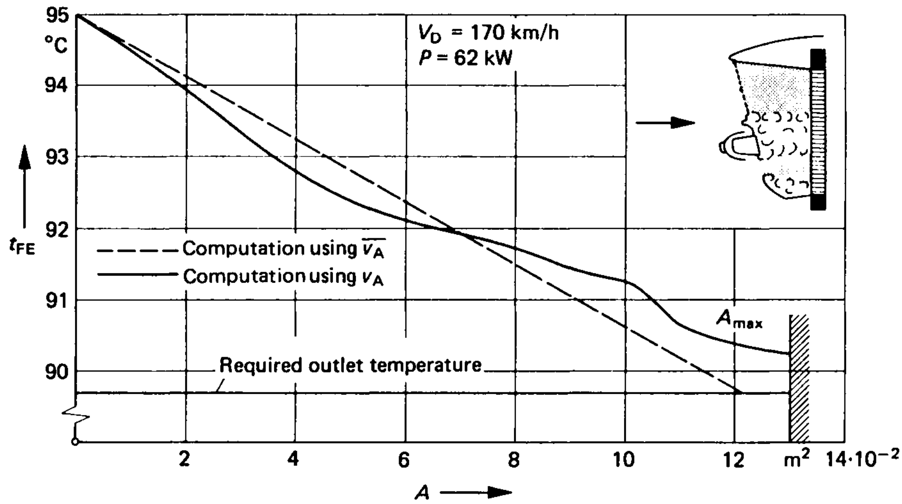


Figure 9.20 Computed coolant temperature curve of a passenger car with a 1.5 litre engine with mean air flow value and with differentiated face velocity

Since heat transfer is proportional to $v_A{}^{\varepsilon}$, where ε assumes values <1, heat transfer is poorer in the velocity 'valleys', which, compared to the mean value v_A, are not compensated for by the higher excess velocities. Accordingly, for a non-uniformly shaped face velocity profile, less heat is transferred than at the average velocity v_A computed from volume flow and frontal area.

9.4.4 Recomputing the system

The temperature curve should be computed for various speeds, otherwise the size would be valid for only one point in the road performance map. The coolant fluid temperatures in the 'off-design' points are also of

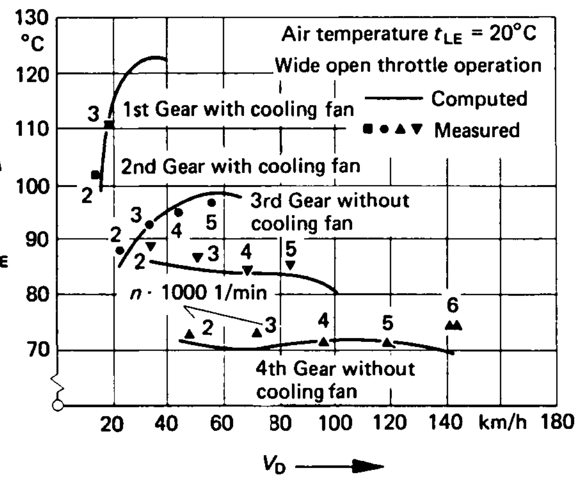


Figure 9.21 Measured and computed coolant temperatures

interest. For example, prior to the final, costly tests on the vehicle, a picture of the operating behaviour of the system can be obtained with the aid of computation procedures, as described in ref. 9.18. The results of computation and measurements are compared in Fig. 9.21. The data predicted by the method of Emmenthal[9.18] are in good agreement with the experimental results.

9.5 Experimental testing of the cooling system

The vehicle should be run at different loads on a chassis dynamometer in a full-scale wind tunnel for these measurements. The heating system should be disconnected; the thermostat is locked open. Depending upon the engine load, it takes about 20 minutes to reach thermal equilibrium. Results obtained from two different cooling systems in the same vehicle are given in Fig. 9.22.[9.25] The coolant temperature curve shows the marked

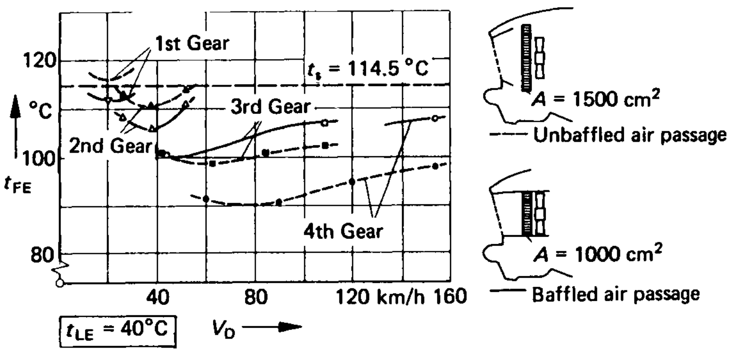


Figure 9.22 Comparison of coolant temperatures of two systems

dependence of the temperature on vehicle speed for the system with unbaffled air inlet and larger radiator frontal area, compared with the cooling system with a baffled air inlet and smaller radiator frontal area, which gives a more uniform temperature versus speed characteristic.

9.6 Evaluating the cooling system

The development of a radiator matrix for a specific application is relatively expensive, mainly because of the high cost of constructing and testing the prototype. It is therefore desirable to have an evaluation procedure that allows preliminary selection from the many types of possible radiator matrix designs, and thus reduce expenditure on prototypes and testing. Equations listed in Fig. 9.23 describe the performance of the radiator in a vehicle during driving. The engine rejects heat to the coolant fluid which must be transferred to ambient by the radiator. For a specific project, it can be assumed that this heat flow $\dot{Q}$ as a function of engine output power and speed is known. Even so, the temperature of the ambient air t_{LE} and that of the coolant entering the radiator t_{WE} must be viewed as 'given' by the operating conditions. The flow stream capacity rate W_L of the air passing through the radiator, meaning the volume of cooling air flow, and the efficiency Φ of the radiator (both quantities are also defined in Fig. 9.23) must be adjusted to satisfy the equation for heat flow.

The velocity of the inflowing air v_A is dependent upon the speed V_D at which the vehicle is being driven, and upon the radiator cooling-fan operating data. The evaluation process is clearer in the case without cooling-fan operation. The aerodynamic efficiency η, meaning the ratio of face velocity v_A to driving speed V_D (see Fig. 9.23), is, for its part, a function of the pressure drop coefficient ζ of the cooling air path.

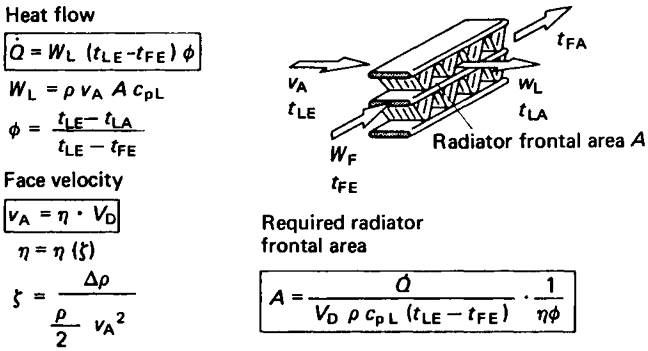


Figure 9.23 Heat flow, cooling air velocity, and required radiator frontal area

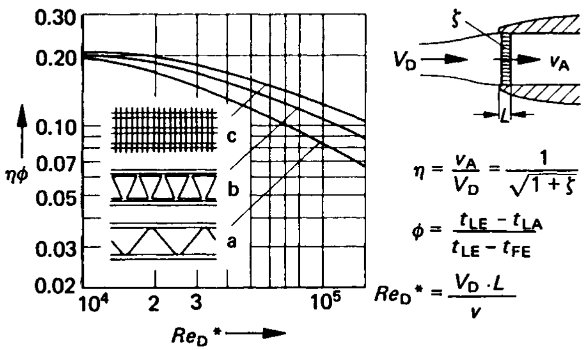


Figure 9.24 Evaluation of different radiator matrix designs, after ref. 9.25

From the equations listed in Fig. 9.23 for a specific radiator design, which is characterized by its aerodynamic and heat transfer properties in the form of $1/\eta\Phi$, the magnitude of the necessary frontal area can be derived. This frontal area is

$$A \sim \frac{1}{\eta\Phi} \tag{9.14}$$

This relation has been used to compare and evaluate different radiator matrix designs by Hucho.[9.25] The higher the product $\eta\Phi$, the better the radiator performance.

An evaluation of different radiator designs is shown in Fig. 9.24. The product $\eta\Phi$ has been plotted against Re_D, the Reynolds number at the driving speed. It is evident from Fig. 9.24 that pronounced differences in heat transfer occur at higher vehicle speeds ($Re_D \approx 10^5$).

This evaluation method indicates some of the targets for future radiator development, such as lower air-related pressure losses and improved heat transfer. Improvements to heat transfer alone could too easily be nullified by higher air resistance.

Since the only item of real importance is cost per unit of heat transferred, the selection of material and manufacturing process is of predominant interest. The radiator of the future must therefore provide increased heat transfer with a cheaper material and be capable of manufacture by an even more cost-effective process.

9.7 Notation

Geometric variables

A	frontal area
B	core width
D_F	diameter of the water pump impeller
D_V	diameter of the cooling fan
H	core height
L	core depth
b	linear width
d_{hyd}	hydraulic diameter
α	angle of incidence of the radiator

Engine/vehicle variables

F_Z	tractive force
P, P_{max}	engine power output (*PS = DIN horsepower)
n	speed in revolutions per minute
n_p	speed at rated output power

Heat flow and air flow fluid variables

$\dot{Q}_{C,R,A,B}$	heat flow from convection (C), radiation (R), exhaust (A), and fuel (B)
$\dot{Q}_F$	heat flow
U_F	tip speed, water pump impeller
U_V	tip speed, cooling fan
V_D	vehicle speed
V_F	coolant flow rate
W_L	flow stream capacity rate
c_F	specific heat of the fluid
c_{pL}	specific heat of the air
k_A	coefficient of heat transfer as referenced to the radiator frontal area
p_o	ambient pressure
t_{FE}	fluid temperature, radiator inlet
t_{FA}	fluid temperature, radiator outlet
t_{LE}	air temperature, radiator inlet
t_{LA}	air temperature, radiator outlet

t_s	boiling temperature
$\bar{v}_A$	$\bar{v}_A = V_L/A$ mean cooling air velocity
v_A	velocity of cooling air flow (face velocity)
v_{AE}	required cooling air velocity
v_{AD}	velocity of cooling air flow due to ram air
v_F	velocity in the radiator tube
v_G	air velocity in front of the radiator grill
Δp	static pressure loss
λ_L	thermal conductivity of ambient air inflow
ρ and ρ_F	density of the air and of the fluid
ν	kinematic viscosity of ambient air inflow
ν_F	kinematic viscosity of coolent fluid

Dimensionless expressions

c_D	air drag coefficient
$Re^* = v_A L/\nu$	Reynolds number, air side
$Re_G = v_G L_G/\nu$	Reynolds number, radiator grill
$Re_F = v_F d_{hydr}/\nu_F$	Reynolds number, coolant (watertube)
$Re_D{}^* = V_D L/\nu = Re_G \sqrt{(1 + \zeta_t)}$	Reynolds number of the operating condition
$Nu^* = k_A L/\lambda_L$	Nusselt number
$\zeta_k = 2\Delta p/\rho v_A{}^2$	pressure drop coefficient, radiator, air side
$\zeta_G = 2\Delta p/\rho v_G{}^2$	pressure drop coefficient, grill
ζ_t	pressure drop coefficient, total air passage, as referenced to v_A
η_V	fan efficiency
$\varphi_F = 4V_F/U_F\pi D^2{}_F$	volume coefficient, water pump
$\varphi_V = 4V_L/U_V\pi D^2{}_V$	volume coefficient, cooling fan
$\Phi = (t_{LE} - t_{LA})/(t_{LE} - t_{FE})$	radiator efficiency
$\psi_F = 2\Delta p/\rho_F U_F{}^2$	pressure coefficient, water pump
$\psi_V = 2\Delta p/\rho_F U_V{}^2$	pressure coefficient, cooling fan