Non-Road
Cooling System Design

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For presentation at the 2011 Agricultural Equipment Technology Conference
Atlanta, Georgia, USA 5-7 January 2011

Published by the
American Society of
Agricultural and Biological Engineers
2950 Niles Road, St. Joseph, MI 49085-9659 USA

The Lectures Series has been developed by the Power and Machinery Division Tractor Committee (PM-47) of ASABE to provide in-depth design resource information for engineers in agricultural industry. Topics shall be related to the power plant, power train, hydraulic system, and chassis components such as operator environment, tires, and electrical equipment for agricultural or industrial tractors or self-propelled agricultural equipment.

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ASABE Publication Number 913C0111

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This is not a peer-reviewed article.
Paul, Neal. 2011. *Non-Road Cooling System Design*. ASABE Distinguished Lecture #35, pp. 1-19. Agricultural Equipment Technology Conference, 5-7 January 2011, Atlanta, Georgia, USA. Copyright 2011 American Society of Agricultural and Biological Engineers. ASABE Publication Number 913C0111.

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Abstract. *Cooling systems for non-road vehicles normally consist of the heat exchanger components required to cool engine fluids such as the radiator, charge air cooler and fuel cooler. These cooling systems may also contain additional heat exchangers for cooling additional vehicle fluid circuits, such as hydraulic and transmission oil coolers and air conditioning refrigerant condensers. The cooling fan, fan drive, and shroud can also be considered to be part of the cooling system. Cooling systems must be designed so that the heat exchangers and other components in the system work together to provide sufficient cooling for all of the affected fluid circuits, often for several different operating conditions. Non-road vehicles often operate in applications where large amounts airborne particulates may be present. The cooling systems for these vehicles should be designed to prevent particulate fouling to the extent that is possible and should also allow access for debris to be cleaned from the heat exchanger fins. The mounting structure for the heat exchangers and the heat exchangers themselves must be designed to withstand the loads that will be imposed by the vehicle operation in its various applications. The heat exchangers and supporting structure must be designed to withstand stresses such as those caused by internal static and fluctuating pressure, thermal cycles, and static, shock and vibration loading. Potential internal and external corrosion problems should also be considered and prevented through proper material selection. Testing should be performed to validate each cooling system design for proper thermal performance and structural durability.*

Introduction

The design procedure for engine cooling systems for agricultural and non-road vehicles is very similar in most respects to that of on-road vehicles. The design of cooling systems for non-road vehicles does require some special considerations. In general, there is more variation in the configuration of non-road vehicles. In addition, production volumes of non-road vehicles are generally lower than on-highway vehicles. This wide variation in vehicle configurations coupled with lower production volumes require that heat exchanger designs used for non-road vehicles be flexible in size and fin density. It is generally not economically feasible to design and manufacture an entirely unique heat exchanger for individual non-road vehicle models. A more economical approach is to create a flexible design that can be scaled in length, width, and depth. As much as possible, production tooling and equipment must be flexible enough to handle this variation so that large tooling investments for each individual heat exchanger design can be minimized or avoided.

Basic Heat Exchanger Components

Radiator

The radiator cools the water/glycol solution, which has a primary purpose of cooling the engine. The radiator is generally the largest, most significant and occasionally the only component of the engine cooling system. The most common material used to manufacture radiators is currently aluminum so in general this paper will refer to brazed aluminum radiators only. Radiators are unique to other heat exchangers in the engine cooling system due to the fact that internal fins or webs are generally not included in radiator tubes. One reason for this is that water/glycol engine coolants have a low ratio of viscosity to specific heat. This combination of low viscosity and relatively high specific heat coupled with high flow rate results in a high convection coefficient inside the tubes. Non-road heat exchangers typically have lower fin densities and continuous fin surfaces as opposed to higher fin densities and louvered fin surfaces for on-road heat exchangers. The result of these factors is that the ratio of internal vs. external thermal resistance of radiators for non-road radiators is low. Due to this, it is generally not advantageous to include any sort of internal fin in the radiator. Non-road radiator designs are often deep when compared to their on-road counterparts. Low fin density coupled with the use of continuous fins instead of louvered fins also contributes to the large size of these components. These large radiators can require a large number of tubes. The radiator coolant flow is in many cases not high enough to promote fully turbulent flow in smooth tubes. When the internal flow is laminar, the heat transfer coefficient may be quite low. In order to lower the critical velocity of the coolant, dimples or other internal roughness features inside the radiator tubes may be used. Typically the use of dimpled tubes will improve the radiator performance from 1% to 3% at rated speed. However, performance can be improved from 5% to 7% at lower engine speeds often associated with peak torque operating conditions. Dimpled tubes, however, have higher internal resistance than smooth tubes.

Charge Air Cooler

The charge air cooler cools the hot, pressurized air from the turbocharger prior to the intake manifold. The charge air cooler is generally an air-air cooler and usually the second-most significant cooler in the cooling system. The purpose of the charge air cooling is to lower the source temperature of the air-fuel mixture, thus improving its thermodynamic potential. Cooling also increases the charge air density allowing a greater amount of fuel combustion to occur in the cylinder thereby increasing the engine power. The initial temperature of the combustion air/EGR mixture is also lowered allowing use of higher compression ratios before the onset of pre-ignition leading to higher thermodynamic efficiency.

Charge air coolers are generally very effective, usually >80% due in part to the fact that the internal air flow is generally 10% to 25% of the external cooling air flow and the inlet charge air temperature is quite high. The charge air circuit is an open circuit as opposed to most of the other circuits in cooling systems, which are closed circuits. The cooling goal of the charge air cooler is generally expressed as the amount of heat transfer required or, more commonly and directly, as the charge air outlet temperature desired. The desired outlet temperature is often expressed as maximum IMTD (intake manifold temperature difference), which is the temperature difference between the air supplied to the intake manifold and the ambient temperature. Use of IMTD generalizes the cooling goal for all ambient temperature conditions. An IMTD goal of 20°-25°C is typical.

Internal pressure loss of the charge air through the cooler and the associated piping must be minimized, since pressure loss adversely affects the increase in density of the charge air. At design conditions, normally 3-5 kPa of pressure loss is allowed for piping and 8-13 kPa is allowed for the charge air cooler core and tanks.

Oil Cooler(s)

Oil coolers may be used to cool transmission, hydraulic, or other oil circuits. The internal pressure in transmission and hydraulic oil circuits is quite high compared to radiator coolant pressure and charge air pressure. Burst requirements of >500 psi up to 1000 psi and operating pressures up to 200 psi are typical. Aluminum oil coolers especially must be carefully designed to withstand these pressure requirements. Otherwise, the design of oil coolers for meeting heat rejection and maximum pressure loss requirements is similar to that of radiators. Since oil circuits are generally closed circuits, the cooling goal is generally expressed as a maximum inlet temperature at a given oil flow and heat load, with air flow and temperature conditions either specified for component design or calculated as part of the system simulation. Internal pressure loss should be minimized in order to minimize the power required to pump oil through the heat exchanger, since this pumping power is a parasitic power loss for the vehicle.

Whenever possible, the operating pressure and amplitude of pressure fluctuations that oil coolers will be subjected to should be minimized. Placing an oil cooler on the low-pressure side of a hydraulic system and putting pressure relief valves in place to protect the oil cooler against sustained high pressure are two of the strategies that may be employed. Lowering the operating pressure and pressure fluctuation amplitude may increase the life of an oil cooler or allow a lower cost design to be applied.

Fuel Cooler

The fuel cooler cools the fuel returning to the fuel tank. The fuel circuit is not entirely a closed circuit or an open circuit due to the fact that some of the fuel is consumed and a significant portion of the heat gained in the fuel system can be lost from the lines and fuel tank. Thus, modeling the fuel circuit as a closed circuit may result in an overly conservative design. In some cases it is more appropriate to model the fuel circuit as an open circuit and express the cooling goal as a desired minimum heat rejection or desired outlet temperature rather than a maximum inlet temperature.

As with the other heat exchangers in the system, it is desirable to minimize the internal pressure loss. The pressure loss requirements for fuel coolers are low (<20 kPa). Use of some types of metals, such as copper and zinc, should be avoided in the construction of fuel coolers due to potentially undesirable reactions with soy-based biofuels.

Condenser

The condenser cools the refrigerant in the air conditioning circuit in order to reject heat from the cabin and the energy from the compressor. At design conditions, refrigerant enters the condenser as a superheated vapor and leaves as a sub-cooled (temperature below condensing temperature) liquid. R134a is currently the most commonly used refrigerant.

The component heat rejection calculation is more difficult for condensers than for the other heat exchangers for a few reasons. The refrigerant changes phase from a superheated vapor entering the condenser, is condensed, and leaves the condenser as a sub-cooled liquid. In order to accurately calculate the performance of the condenser, the phase and quality (ratio of vapor quantity to total quantity) of the refrigerant must tracked through the heat exchanger. Therefore, for calculation, the condenser must be broken up into smaller sections and the performance of each section calculated successively. While radiators, charge air coolers, and often oil coolers are single-pass heat exchangers, condensers are usually multi-pass heat exchangers. Normally the number of tubes per pass grows progressively smaller as the refrigerant condenses and its density increases as the refrigerant passes through the condenser. The cooling goal for condenser design conditions is often expressed in terms of a minimum required heat rejection or a minimum amount of refrigerant sub-cooling.

Fan, Fan Drive, and Shroud

The engine cooling fan, fan drive, and shroud provide the means for forcing cooling air through the heat exchanger cores. This paper focuses on the heat exchanger components themselves. An entire paper could easily be written on the design of the fan, fan drive, and shroud so this topic will not be covered in detail here.

The normal configuration is to place the cooling fan behind the cooling package and to draw air through the cooling system by suction. Theoretically higher air flow could be achieved with a fan in the pusher configuration due the higher density of the cooling air prior to being heated by the cooling system. The pusher fan configuration is favored in some types of vehicles. Care must be taken when applying a pusher fan because this configuration has a greater tendency to create non-uniform air flow and is more prone to recirculation. The cooling fan must create sufficient pressure rise to overcome the pressure loss of the cooling air flow as it travels through the grill screen, heat exchanger cores, shroud, and out of the vehicle, often through the engine compartment and past any other obstructions in the vehicle that may be present in the air flow path.

Noise is another factor that must be considered when selecting appropriate fan speeds. Fan noise levels increase as the rotational speed of the fan increases. The maximum noise level allowed by regulation for European machines is 89 dBA. Excessive noise levels degrade customer satisfaction in other markets as well. Typically axial fan noise levels are a function of the fan blade tip speed. As fan tip speeds rise to 90 m/s and above, excessive fan noise is increasingly likely.

In the past, most non-road vehicles used a fixed-ratio fan drive system that was normally pulley driven or directly mounted to the engine. This is still the case in some applications, especially if those applications have a single duty cycle. In most cooling systems today, fan drives are used to deactivate or slow down the fan during off-peak operating conditions in order to decrease parasitic power draw and improve fuel economy. Fan drives can be driven directly off of the engine or powered hydraulically or electrically. Directly driven fan drives normally use a viscous clutch. The viscous clutch can normally be activated incrementally either electronically or thermostatically. Electronic control can be very advantageous in vehicles such as agricultural tractors that operate in a wide variety of conditions and duty cycles. The various conditions and duty cycles can place more cooling requirements on one heat exchanger component than the others. Suppose an electronic fan drive can be electronically activated by radiator top tank temperature, charge air cooler outlet temperature, and oil cooler inlet temperature. In this case, the fan speed can be controlled in order to provide sufficient cooling for each of these components without unnecessary overcooling. Thermostatically actuated fan drives increase the fan speed drive ratio as the temperature at the fan drive increases. Additional cooling is supplied as the external temperature increases and as the heat exchangers in the system reject more heat. This strategy works well unless one heat exchanger requires significantly more cooling than the others at a relatively low temperature. An example of this situation is a charge air cooler that must maintain a minimum IMTD at relatively low ambient temperatures, where the other heat exchangers in the system would not be near maximum operating temperature limits.

Analytical Tools

Modes of Heat Transfer

There are three modes of heat transfer that contribute to heat loss from the heat exchangers in a cooling system: conduction, convection, and radiation.

Conduction is the transfer of energy through materials due to transfer of molecular kinetic energy between adjacent molecules. The rate of energy transfer is governed by Fourier's Law:

$$Q = kA\frac{\Delta T}{L}$$

where Q = heat transfer rate
k = thermal conductivity
A = heat transfer area
ΔT = temperature gradient across the material
L = length of material in the direction of heat transfer

Convection refers to energy transfer by the bulk motion of fluid adjacent to surface when the fluid and surface are at different temperatures. The rate of energy transfer due to convection is governed by Newton's Law of Cooling:

$$Q = hA(T_\infty - T_s)$$

where h = convection coefficient
T_∞ = bulk fluid temperature
T_s = surface temperature

Radiation heat transfer refers to energy transfer by electro-magnetic waves to or from a surface and its surroundings due to a temperature gradient between that surface and its surroundings. Thermal radiation is constantly being emitted and absorbed by matter. The rate of energy transfer is dependent of the rate of energy emitted vs. the rate of energy absorbed. Thermal radiation is generally not a significant portion of the heat rejected from the heat exchangers in an engine cooling system and is typically neglected in design calculations.

Internal/External Pressure Loss

Pressure loss of the fluids passing through heat exchanger cores must be calculated. Pressure loss in heat exchangers can generally be analyzed with the analytical methods for flow inside channels:

$$\Delta P_{total} = \Delta P_{entrance} + \Delta P_{core\ friction} + \Delta P_{exit} + \Delta P_{change\ in\ velocity - inlet/outlet} + \Delta P_{gravity}$$

The friction pressure loss component is expressed as:

$$\Delta P_{friction} = f \rho \frac{L}{D_h} \frac{V^2}{2}$$

where f = Darcy friction factor
ρ = fluid density
L = flow length
D_h = hydraulic diameter
V = fluid velocity

The entrance and exit pressure loss components are expressed as:

$$\Delta P_{entrance} = K_e \rho \frac{V^2}{2} \quad \text{and} \quad \Delta P_{exit} = K_e \rho \frac{V^2}{2}$$

where K_c = contraction pressure loss coefficient at core entrance
K_e = expansion pressure loss coefficient at core exit

Pressure loss due to the change in fluid momentum is expressed as:

$$\Delta P_{velocity} = \rho \frac{{V_{inlet}}^2 - {V_{exit}}^2}{2}$$

Internal pressure loss includes the above components along with pressure loss in the tanks, manifolds, and fittings. The pressure loss of each of these may be modeled as a “minor” loss. The K_L factor(s) may be determined by experimental measurements.

$$\Delta P = K_L \rho \frac{V^2}{2}$$

where K_L = pressure loss coefficient for tanks, manifolds, fittings, etc.

Pressure loss due to elevation change is expressed as:

$$\Delta P_{gravity} = \rho g \left(h_i - h_o\right)$$

where ρ = fluid density
g = gravitational constant
h_i = elevation of inlet
h_o = elevation of outlet

Heat Exchanger Modeling

There are two primary methods for modeling heat exchangers: the F-LMTD method and the effectiveness-NTU method.

The F-LMTD method is derived from the analytical method for calculating the performance of a parallel-flow heat exchanger. The product of overall heat transfer coefficient and the area is defined as the inverse of the overall thermal resistance of the heat exchanger. For heat exchangers other than parallel flow heat exchangers, the log-mean temperature difference does not perfectly describe the mean average temperature difference. For those other configurations, the formula is modified by multiplying the formula by a correction factor F. Correction factors for common heat exchanger configurations such as single and multi-pass, cross-flow heat exchangers can be found in literature. The log-mean temperature difference method has the disadvantage of requiring that both the inlet and outlet temperature of both fluids are known. Normally the outlet temperatures are not known and an iterative procedure must be used. When using the LMTD method, heat transfer is expressed as:

$$Q = F\, U\, A\, \Delta T_{LM}$$

where F = correction factor for heat exchanger configurations other than parallel flow
U = overall heat transfer coefficient
A = heat transfer area
ΔT_{LM} = log-mean temperature difference:

$$\Delta T_{LM} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$

where ΔT_1 = temperature change of the internal (hot) fluid
ΔT_2 = temperature change of the external (cold) fluid

The product of the overall heat transfer coefficient and heat transfer area is equal to the inverse of the overall thermal resistance of the heat exchanger:

$$UA = \frac{1}{R}$$

where R = overall thermal resistance.

For a concentric-tube, parallel-flow heat exchanger, overall thermal resistance is expressed as:

$$R = \frac{1}{h_i A_i} + \frac{\ln\left(\frac{d_o}{d_i}\right)}{2\pi k t} + \frac{1}{h_o A_o}$$

where h_i = convection coefficient inside the inner tube
A_i = heat transfer area inside the inner tube
d_o = outside diameter of the inner tube
d_i = inside diameter of the inner tube
k = tube wall thermal conductivity
t = tube wall thickness
h_o = convection coefficient on the outside of the inner tube
A_o = heat transfer area on the outside of the inner tube

Heat exchangers in cooling systems are generally classified as finned cross-flow heat exchangers where:

$$\frac{1}{UA} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o}$$

$$\frac{1}{U_i} = \frac{1}{h_i \eta_i} + \frac{t}{k} + \frac{A_i}{h_o \eta_o A_o}$$

$$UA = U_i A_i$$

where η = the overall surface efficiency and subscripts i and o refer to the inner and outer heat exchanger surfaces, respectively.

Another heat exchanger analysis method that avoids the iterative nature of the log-mean temperature method is the effectiveness-NTU method. NTU stands for number of transfer units and is defined below. The NTU method is based on the principle of the maximum possible heat transfer from a heat exchanger:

$$Q_{max} = C_{min}\left(T_{h,i} - T_{c,i}\right)$$

where $T_{h,i}$ = hot fluid inlet temperature
$T_{c,i}$ = cold fluid inlet temperature
C_{min} = the minimum of the hot and cold fluid heat capacities, C_h and C_c:

$$C_h = \left(\dot{m}c_p\right)_h$$

$$C_c = \left(\dot{m}c_p\right)_c$$

where $\dot{m}$ = mass flow of the hot and cold fluid streams
c_p = specific heat of the hot and cold fluid streams
Subscripts h and c refer to the hot and cold fluid streams, respectively.

The effectiveness of a heat exchanger is defined as:

$$\varepsilon = \frac{Q}{Q_{max}}$$

The heat transfer of a heat exchanger can be calculated in terms of effectiveness by the following formula:

$$Q = \varepsilon C_{min}\left(T_{h,i} - T_{c,i}\right)$$

The effectiveness of any heat exchanger is only a function of its flow arrangement (parallel flow, counterflow, cross-flow, etc), the ratio of the minimum to maximum fluid stream heat capacity rates (C_r), and the dimensionless parameter NTU defined as:

$$\text{NTU} = \frac{UA}{C_{min}}$$

$$C_r = \frac{C_{min}}{C_{max}}$$

For a single-pass cross-flow heat exchanger with both fluids unmixed, the formula for effectiveness is:

$$\varepsilon = 1 - e^{\left[\left(\frac{1}{C_r}\right)\text{NTU}^{0.22}\, e^{\left(-C_r\,\text{NTU}^{0.78}\right)} - 1\right]}$$

Expressions for effectiveness for other heat exchanger flow arrangements may be found in literature.

The steps for performing a heat exchanger heat transfer calculation are as follows:

1. Calculate the internal and external heat exchanger surface area.
2. Using test data or available Nusselt Number (Nu) vs. Reynold's Number (Re) or j-factor vs. Re correlations (see below), calculate internal and external convection coefficients. For the first calculation iteration, inlet fluid properties or an estimate of average fluid properties may be used.
3. Calculate overall internal and external surface efficiency.
4. Calculate overall U value.
5. Calculate internal and external fluid heat capacity, identify C_{min} and C_{max}, and calculate C_r.
6. Calculate NTU.
7. Calculate effectiveness.
8. Calculate heat transfer.
9. Repeat steps 2 through 8 using average fluid properties until the solution converges to an acceptable accuracy.

Fluid properties for ethylene glycol solutions and oils may often be obtained from the manufacturer of these products. REFPROP, published by the National Institute of Standards and Technology (NIST), is an excellent database for the properties of air, water, refrigerants, and other substances.

Similitude

Heat exchanger surface convection coefficients and friction factors, cooling fan performance, and vehicle pressure loss curves are represented most accurately and conveniently by the use of curves based on dimensionless parameters. For example, correlations for convection coefficient (h) at heat transfer surfaces are commonly expressed in terms of Nusselt Number as a function of Reynold's Number and Prandtl Number.

Nusselt Number: $\text{Nu} = \dfrac{hd_h}{k}$

where h = convection coefficient
d_h = hydraulic diameter
k = fluid thermal conductivity

Stanton Number: $\text{St} = \dfrac{h}{\rho V c_p}$

Reynold's Number: $\text{Re} = \dfrac{V d_h}{\upsilon}$

where υ = fluid kinematic viscosity.

Prandtl Number: $\text{Pr} = \dfrac{c_p \mu}{k}$

where μ = fluid dynamic viscosity.

Often Nusselt Number, or alternatively Stanton Number, is correlated with Prandtl Number to the 2/3 power. The Colburn j factor takes this into account and is defined as:

$$j = \text{St}\ \text{Pr}^{\frac{2}{3}}$$

Friction pressure loss coefficients are commonly expressed as a function of Reynold's Number. Friction (f) and j factor versus Reynold's Number can be found in many sources in literature. One important consideration when using data from these sources is to determine how the hydraulic diameter was calculated and whether the friction factor reported is the Fanning friction factor or the Darcy friction factor. The Fanning friction factor is equal to onefourth of the Darcy friction factor. The ratio of j to f is often used to compare the relative efficiency of different heat transfer surfaces.

Traditional fan curves show static pressure rise vs. volumetric flow rate for various fan speeds. These curves are limited in their application to the conditions (temperature, pressure, and fan speed) at which they were generated, although corrections may be made through the use of fan laws. Fan curves can be reduced to a single set of curves through the use of similitude. Physical air properties and rotational speed are accounted for through the use dimensionless parameters. Furthermore, fan curves may be scaled for size within a reasonable range (±10%) through the use of similitude. The fan curve should first be resolved into dimensionless terms as shown. The fan curve should be reconstituted into an ordinary set of curves at the actual operating conditions. This may require an iterative approach as the temperature at the fan will not be known until the flow rate of air and the heat rejection from open circuit heat exchangers can be determined.

Traditional fan curves are expressed as the air pressure rise as functions of volumetric flow rate and fan rotational speed. Separate curves are normally plotted for several discrete fan speeds.

$$\Delta P = f\left(\dot{V}, n\right)$$

$$P = f\left(\dot{V}, n\right)$$

where ΔP = pressure rise

$\dot{V}$ = volumetric flow rate

n = rotational speed

P = fan power

Dimensionless fan curves are expressed as the pressure number and fan efficiency as functions of the flow number.

$$\psi = f(\varphi)$$

$$\eta = f(\varphi)$$

where ψ = pressure number = $\dfrac{2\Delta P}{\rho u^2}$

η = efficiency = $\dfrac{\dot{V}\Delta P}{P}$

φ = flow number = $\dfrac{4\dot{V}}{D^2 \pi u}$

where D = fan diameter

u = fan tip speed = $n \pi D$

Cooling air pressure loss through the heat exchanger cores and the vehicle can be broken up into the pressure loss components of various obstructions including the grill screen, shroud, and engine compartment. The overall pressure loss may be determined by combining the pressure loss curves of the various heat exchanger cores and obstructions. An electrical analogy can be used to visualize combining these resistance curves based on whether they are in series or parallel. Heat exchanger tanks and manifolds that block the air flow of other cores may be modeled as solid obstructions that allow zero or very little air flow. Normally a simplifying assumption that the air must flow through the heat exchanger cores only in the perpendicular direction to the cooling package is used. In reality, some air will flow around obstructions or curve around a heat exchanger core that is placed in front of another core but does not cover its face area entirely. By using the assumption of only orthogonal air flow relative to the heat exchanger cores, the cooling package can more easily be broken down into restrictions in series and parallel. Often, for simplicity's sake, all the restriction of the vehicle to cooling air flow outside of the heat exchanger cores themselves is lumped together into a single restriction curve in the form of:

$$\Delta P_{vehicle} = K_L \rho \frac{\dot{V}^2}{2}$$

Many times restriction curves for obstructions such as grill screens and for the air passing through the engine bay or the overall vehicle restriction curve may be known only at a single operating condition. These curves can be corrected for other conditions using similitude in a manner similar to that applied to the fan. Flow should be transformed into Reynold's Number and pressure loss should be transformed into friction factor. The choice of hydraulic diameter when determining these factors is arbitrary as long as a consistent approach is used to transform the pressure

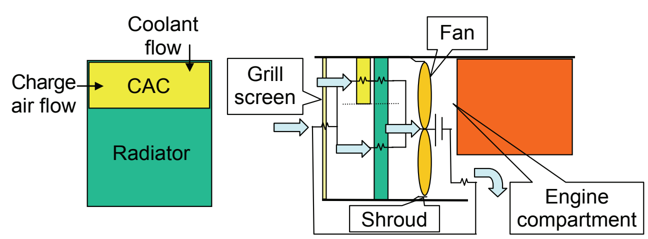


Figure 1. Schematic of the example cooling system.

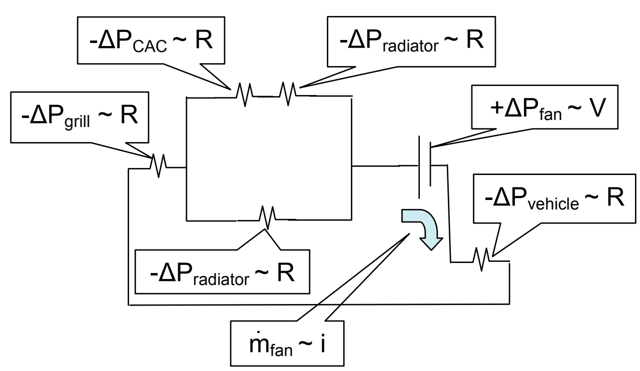


Figure 2. Electrical analogy of the air flow through the example cooling system.

loss curve factors into the dimensionless parameters and back again. The pressure loss curve can be corrected by reversing the calculation and applying the new conditions. In order to find the operating point of the fan (flow and pressure rise) and the overall pressure loss of the system, an overlay plot of the corrected pressure loss curves and the corrected fan curve in terms of mass flow vs. ΔP should be created. The point at which these curves intersect represents the mass flow and ΔP of the system. Cooling air pressure rise due to ram air speed is generally ignored for non-road vehicles because these types of vehicles are normally operated at low speeds.

Once the overall air flow through the system has been found, the air flow through each heat exchanger component can be found by finding the pressure loss across each core(s) in series. The pressure loss across cores in parallel must be equal. When the pressure loss across the cores is known, then the flow through each core can be found from the core pressure loss curve. Once the air flow is known, the heat rejection performance of each heat exchanger can be found. For closed-loop circuits (engine coolant and oil circuits), the heat load is known and the inlet temperature of the fluid must be calculated. For open-loop circuits (charge air and in some cases fuel), the heat rejection and/or the outlet temperature are calculated.

Air conditioning circuits and condensers represent a unique case. Refrigeration system circuits are more complicated to model than simple open or closed circuits. Often for the designer of engine cooling systems, it is not necessary or desirable to model the entire AC system. Usually for the purposes of condenser and engine cooling system design it is sufficient to model the AC system and condenser as a simple closed-loop circuit with the maximum design condenser heat load for the purposes of representing the heat load on the system and the cooling air pressure loss. Once the air flow is known, the condenser performance can be calculated to ensure that the condenser has sufficient capacity to reject the required heat load at design conditions.

Performing an analysis on a system of heat exchangers involves the following steps:

1. Determine operating conditions:
   a. Ambient temperature
   b. Pre-heat (re-circulation)
   c. Fan speed
   d. Closed circuit heat loads (radiator, oil coolers, condenser)
   e. Closed circuit flow rates
2. Resolve the fan curve into dimensionless format and correct the fan curve to estimated operating conditions (Figure 7).
3. Resolve the vehicle restriction curves into dimensionless format and correct to estimated operating conditions (Figure 7).
4. Calculate the pressure loss curves for heat exchangers at the estimated operating conditions (Figures 3 and 4).
5. Create a resistance circuit diagram and combine the restriction curves for series and parallel restrictions appropriately. This can be performed graphically as shown in Figures 5 and 6. Restriction curves in series can be combined by adding together the ΔP values at the same mass flow rate for each curve (Figure 5). Restriction curves in parallel can be combined by adding together the mass flow rate at the same at the ΔP for each curve (Figure 6).
6. Calculate the overall cooling air mass flow rate of the system. This can be done graphically by finding the intersection of the fan curve and the overall restriction curve (Figure 7).
7. Calculate the cooling air mass flow through each heat exchanger by working back through the restriction circuit.
8. Calculate the cooling air inlet temperature for each heat exchanger based using the air mass flows calculated and the heat rejection of the upstream heat exchangers and other sources of heat.
9. Calculate the heat transfer for open circuit heat exchangers and inlet temperatures for closed-loop circuit heat exchangers.

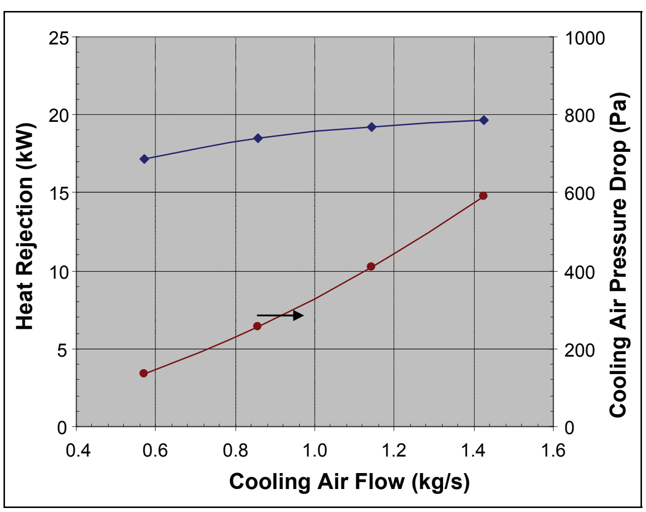


Figure 3. Charge air cooler performance curves.

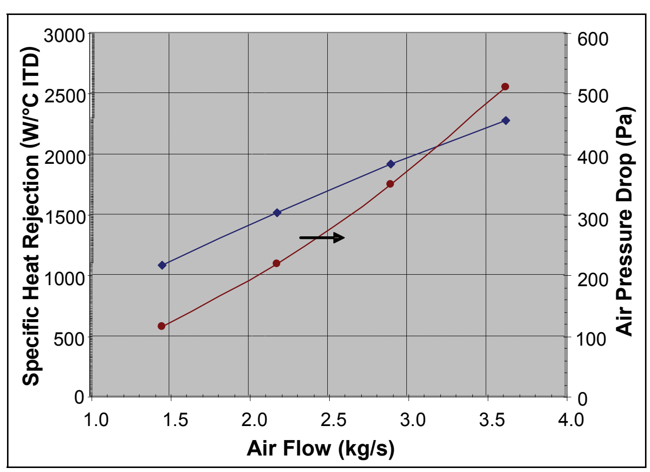


Figure 4. Radiator performance curves.

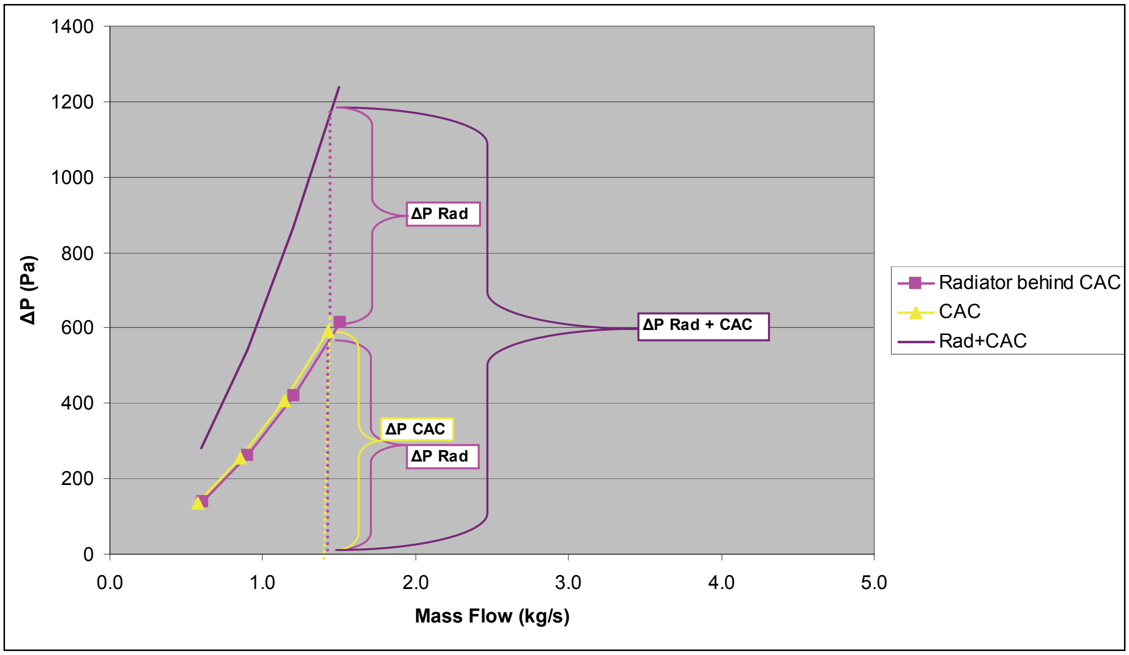


Figure 5. Radiator + CAC section restriction curve.

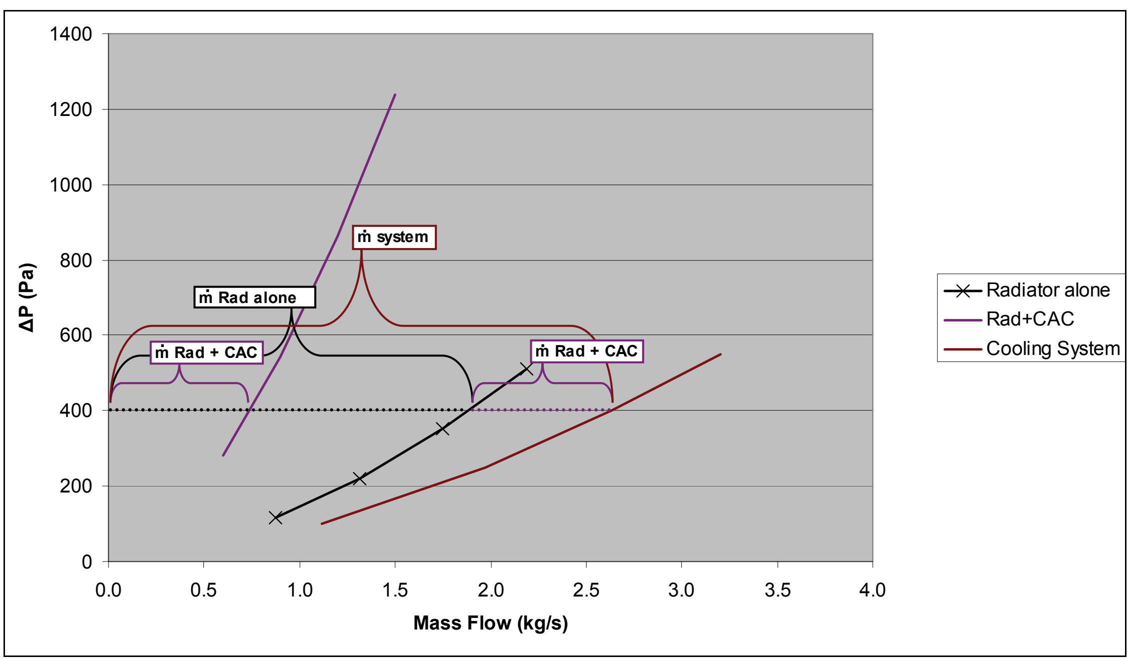


Figure 6. Cooling system restriction curve.

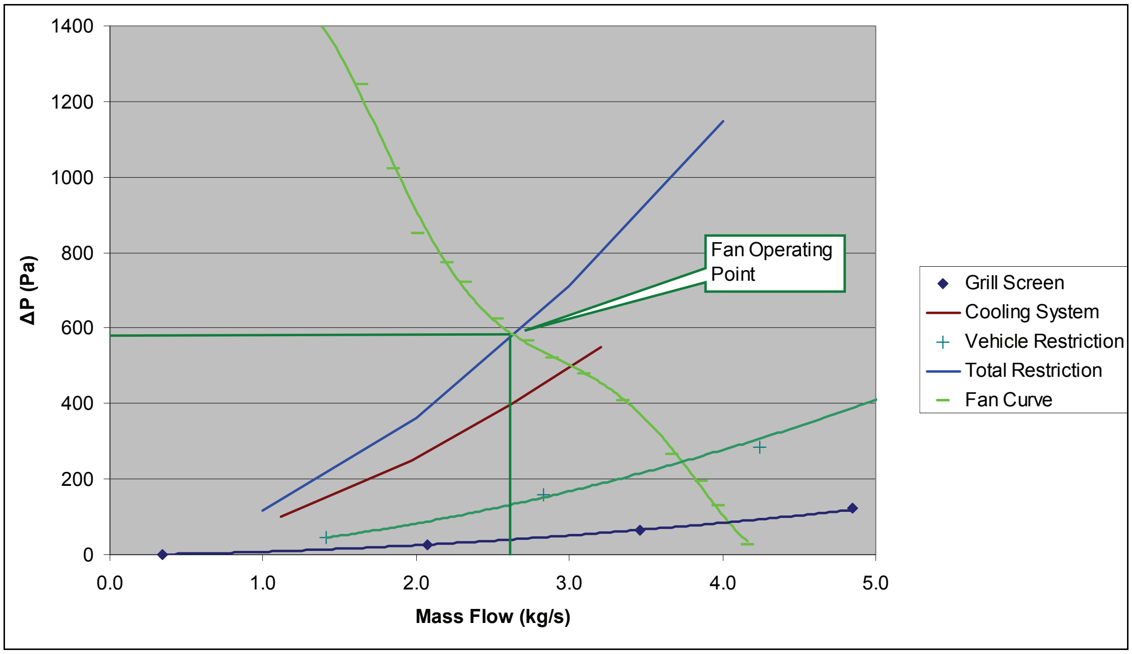


Figure 7. Fan curve operating point.

10. Calculate the heat transfer for open circuit heat exchangers and inlet temperatures for closed-loop circuit heat exchangers.
11. If applicable, calculate condenser heat rejection at design conditions and the air flow and temperature calculated.
12. Repeat steps 4 through 11 until the solution converges.
13. Calculate the internal pressure loss of each heat exchanger.
14. Compare calculated heat rejections, outlet temperatures, inlet temperatures, and pressure losses for each circuit to the specified goals and adjust the size and/or design of each heat exchanger as needed.

The process outlined above was performed as an example for a simple engine cooling system including only a radiator, charge air cooler (CAC), and cooling fan and restriction curves for the grill screen and the engine compartment. In this system, the CAC in a horizontal flow orientation is placed ahead of a down-flow radiator. The cooling fan and shroud is placed in a sucker configuration (drawing air through the cooling package) behind the radiator. The air exits the vehicle through the engine compartment and finally through openings in the engine compartment to the outside. It is assumed that a portion of the heated air will recirculate and re-enter the vehicle cooling system by being drawn into the grill screen, increasing the effective ambient cooling air temperature.

Predicted heat rejection and pressure loss curves were calculated for the radiator and CAC using the methods outlined previously (see Figures 3 and 4). Since the CAC is ahead of the radiator in the cooling system and no additional heat sources are in front of it, the cooling air inlet temperature is known for the CAC. The heat rejection and pressure loss curves can be calculated directly at the design conditions and used without correction later in the analysis. The inlet coolant temperature (top tank temperature) of the radiator is unknown prior to the analysis because the engine coolant system is a closed loop. For the purposes of calculating the heat rejection and pressure loss curves, the top tank temperature and air inlet temperatures are estimated and corrected iteratively.

The point at which the total restriction curve intersects the fan curve represents the operating point of the system. Once the total air flow is known, the air flow through each component in the cooling system can be calculated by working backward through the restriction curve combinations. The heat rejection of the CAC and the radiator top tank temperature can be found based on the CAC heat rejection curve and the radiator specific heat rejection curve, respectively. The inlet and outlet cooling air temperature for each cooling system component can now be calculated based on the heat loads of the CAC and radiator. A few iterations are required for the solution to converge so that the correct temperatures are used to correct the restriction curves.

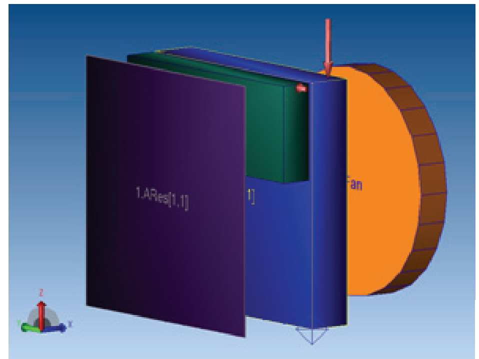
The analysis procedure above can become tedious and time consuming when it is being set up for cooling systems that have more than a few heat exchanger and obstruction components. One-dimensional fluid networking programs available on the market can be used to automate this process and greatly reduce the time and effort required for each cooling system analysis. Kuli (published by Magna Steyr) is such a program, specifically tailored for cooling system design. Flowmaster is a generalized fluid networking and thermal simulation software that may also be used for this purpose.

The cooling system described earlier was also simulated using Kuli. Figure 8 shows that the manual cooling system simulation and the Kuli simulation are in close agreement.

Cooling air flow restrictions due to grill screens, vehicle structures, obstructions and the shroud and engine compartment of non-road vehicles are often not known. These restrictions can be difficult to measure accurately, requiring sophisticated and expensive equipment. However, it is possible to estimate the vehicle restriction curve with a simple hot room or wind tunnel test. A sample cooling system should be designed using an estimated vehicle restriction curve. If, for example, no data is available from a similar vehicle, a reasonable estimate for vehicle pressure loss for many off-road vehicles is 50% to 60% of the total fan pressure rise. Once the sample cooling system has been designed, the vehicle can be tested in a wind tunnel or hot room. When testing has been completed, the cooling system simulation should be repeated to simulate the conditions of the test. The vehicle restriction factor can be adjusted until the simulation matches the test results as closely as possible. Matching the radiator top tank temperature to the test data should receive the most attention since the radiator is typically the largest and most significant heat exchanger in the system.

Often the goal of cooling system simulation is to minimize the fan power required to cool all the fluid circuits in the system to an acceptable level. The simulation must be performed in such a way that the fan speed is adjusted until all the cooling goals including maximum radiator top tank temperature, CAC outlet temperature (IMTD), maximum inlet temperature of the oil cooler(s), and fuel cooler and heat rejection capacity of the condenser are met. The goal when the simulation is performed in this manner is to minimize the fan speed and the associated fan power required. This type of simulation can be used to simulate the actual vehicle operation since many vehicles are equipped with electronically controlled fan drives that will control the fan speed in order supply the required amount of cooling under operating conditions without overcooling and consuming excess power. Often this type of fan drive can be controlled by multiple inputs. The simulation should be performed so that the fan speed is set to the minimum value required to cool the fluid circuit that reaches its cooling goal last. Only the cooling circuits that will actually be measured and used to control the fan speed on the vehicle should be used to set the fan speed in the simulation. The heat exchangers for the other circuits should be designed to meet their cooling goals prior to minimum fan speed set by the controlling inputs at the operating conditions considered.

The one-dimensional method outlined above is an appropriate engineering model for cooling system design for most non-road vehicles. However, the method does contain many assumptions and simplifications to the actual physical process. For example, in the one-dimensional methodology, air is assumed to move straight through the cooling



Input
Calculated

	Manual	Kuli
Operating Conditions		
Ambient temperature [°C]	43.00	43.00
Preheat [°C]	7.00	7.00
Barometric Pressure [kPa]	100.00	100.00
Grill Screen Restriction		
Cooling air mass flow [kg/s]	2.65	2.64
Air pressure loss [Pa]	40.81	38.93
Charge Air Cooler		
Air entry temp. [°C]	50.00	50.00
Charge air entry temp. [°C]	174.70	174.70
Charge air exit temp. [°C]	69.47	69.02
Heat Rejection [kW]	18.05	18.13
Charge air mass flow [kg/s]	0.17	0.17
Charge air entry pressure [kPa-a]	239.00	239.00
Charge air pressure loss [kPa]	6.17	6.09
Cooling air mass flow [kg/s]	0.75	0.75
Cooling air pressure loss [kPa]	208.51	212.77
Radiator behind CAC Section		
Air entry temp. [°C]	74.20	73.89
Cooling air mass flow [kg/s]	0.75	0.75
Cooling air pressure loss [Pa]	195.80	191.19
Radiator Alone Section		
Air entry temp. [°C]	50.00	50.00
Cooling air mass flow [kg/s]	1.90	1.89
Cooling air pressure loss [Pa]	404.31	403.96
Radiator		
Air entry temp. [°C]	56.85	56.79
Coolant entry temp. [°C]	113.52	113.95
Specific heat rejection [W/°C ITD]	1790.90	NA
Heat rejection [kW]	101.50	101.29
Coolant volumetric flow [l/s]	7.40	7.40
Coolant pressure loss [kPa]	20.00	19.94
Cooling air mass flow [kg/s]	2.65	2.64
Cooling air pressure loss [kPa]	345.30	319.92
Cooling Fan		
Fan diameter [mm]	559.00	559.00
Fan speed [rpm]	2100.00	2100.00
Temperature [°C]	94.76	94.72
Mass flow [kg/s]	2.65	2.64
Pressure rise [Pa]	580.49	583.53
Fan power [kW]	4.42	4.47
Vehicle Restriction		
Air entry temp. [°C]	94.71	94.72
Air pressure difference [Pa]	135.00	140.63

Figure 8. Manual to Kuli system calculation comparison.

package without flowing around any obstructions. Also, no allowance is made for non-uniform pressure gradients created by the cooling fan that affect air flow over the face of the cooling system. More of these details can be captured and more accurate results obtained by using CFD (computational fluid dynamics) analysis methods. The disadvantages of CFD include the additional time needed to set up and solve the simulation and the additional computer resources that are needed. Kuli software allows air flow distribution over the face of the cooling system obtained from CFD analysis to be used as input resulting in a hybrid calculation approach.

Design for Debris Clogging

Non-road vehicles operate in a wide variety of airborne debris laden conditions. If not managed appropriately, debris can clog the air fin passages of the heat exchanger core. This clogging increases the restriction of the heat exchanger cores reducing air flow. Excessive restriction due to clogging can lead to overheating of the engine coolant and oil and fuel circuits, and under-cooling of the CAC and refrigerant circuits. One method of managing the debris is to match the fin density of the heat exchanger cores with the primary type and quantity of debris typically seen during operation in the conditions of the vehicle applications. Harvesting machines, such as combines and windrowers, often operate in environments where large amounts long fibrous debris is present. This type of debris can easily bridge the gaps between fins, matting and clogging the core primarily on the leading face of the heat exchanger. Normally heat exchangers in vehicles in these types of applications utilize fin densities that are as low as possible, from 5 to 7 fpi. Some vehicles, such as agricultural tractors, operate in a wide variety of conditions. Fin densities in these types of vehicles are normally limited to less than 10 fpi. Construction equipment normally operates in environments where dust and dirt particles make up the majority of the airborne particles. Fin densities from 10 to 12 fpi are common in these types of vehicles. Continuous fins with unbroken surfaces are normally used in non-road vehicles to prevent fouling by airborne debris. The small openings in the surface of louvered and offset type fins normally used in heat exchangers in on-road cooling systems can easily become clogged with dirt and debris. These types of fins lose effectiveness when these openings become clogged. If louvered or offset fins are used in non-road heat exchangers, removable screens that can be easily cleaned are often placed ahead of each heat exchanger core face.

Even with strategies to prevent air-side fouling, such as limiting fin densities and using continuous fins, the air fins of the heat exchangers must be cleaned in order to maintain an acceptable level of cooling effectiveness. Cleaning wands are commonly used in harvesting machines to continuously clean the surface of the core faces or inlet screen.

Table 1. Typical fin density ranges.

Vehicle Type	Typical Fin Density Range
Harvesting	5 - 7 fpi
Agricultural tractor	9 - 10 fpi
Construction	10 - 12 fpi

Also, for most non-road vehicles, it is helpful to be able to access both the front and back side of each heat exchanger core in order for the core faces to be cleaned manually and also for the entire fin surfaces to be cleaned either by washing with water or by blowing out with high pressure air. Non-road cooling systems are often designed with hinge and latch systems to allow the cooling package to swing or fold open to allow access to each heat exchanger core. When packaging space allows, the heat exchangers can be solid mounted with sufficient space between the cores to allow access with water or air nozzles. Access panels that can be removed for cleanout may be required in order to properly force air flow through the heat exchangers during operation. Placing all the heat exchangers in the cooling package side by side in a single plane is another good way to ensure access to the heat exchanger cores.

Structural Design

In addition to their primary function of rejecting heat to the cooling air, the coolers in the cooling system must function as leak-free vessels under pressure cycle, thermal cycle, vibration and shock loading. The mounting structure of the heat exchangers to each other and to the vehicle must also be cable of withstanding the vibration and shock loads encountered during vehicle operation.

Design for Internal Pressure Cycles

Heat exchangers are subject to cyclic internal pressure during application. Large pressure spikes are created during start-up in oil coolers as the cold, low-viscosity oil is pushed out of the heat exchanger. Charge air coolers experience a significant increase in temperature and pressure during extreme load conditions such as climbing a hill at altitude. Transmission oil coolers may see millions of short duration pressure cycles over the life of the heat exchanger when shifting gears.

The first step in designing for pressure events is to translate the field loading into a set of range-mean-count data (a histogram of pressure fluctuation ranges with the mean identified for each interval). This is accomplished by monitoring the inlet and outlet pressure over a series of worst-case operating conditions. For example, pressure may be monitored on a skid loader application during digging, stockpiling, hammering, road operation, and idling operating conditions. The rain-flow-cycle counting method may be used to generate a range-mean-count from the field data. The linear damage accumulation rule (Miner's Rule) may be used to transform the range-mean-count data into an equivalent constant amplitude pressure cycle bench test (Figure 9).

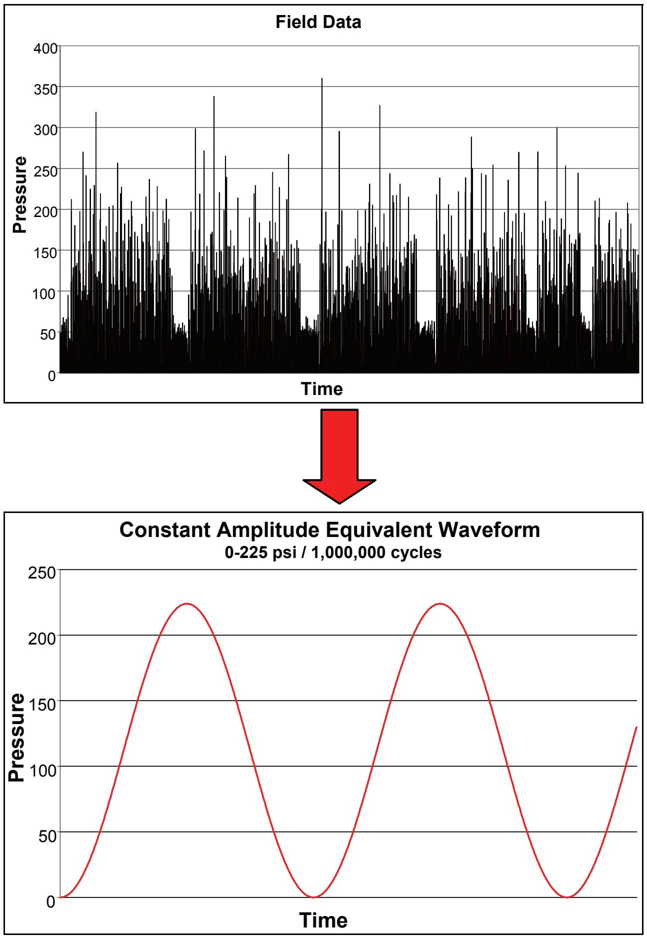


Figure 9. Field pressure data transformed into a constant amplitude pressure cycle.

The second step in the design process is to calculate the stress imposed on the structure of the heat exchanger by internal pressure. Normally this is accomplished through the use of finite element analysis (FEA). Figure 10 is an example of a Von-Mises stress plot from FEA where the color blue represents an area of low stress and the color red represents a highly stressed area.

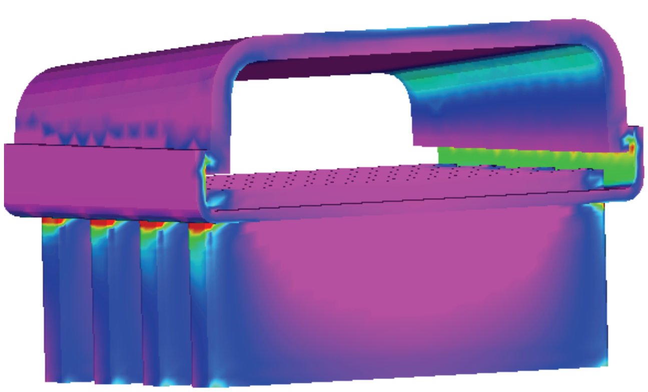


Figure 10. Example stress plot from FEA.

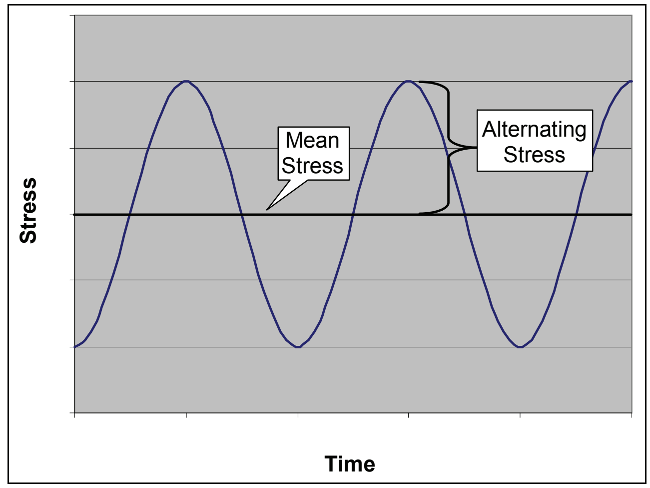


Figure 11. Mean and alternating stress plot.

The third step in the design process is to calculate the fatigue life from calculated stress results. If the number of pressure cycles over the intended life of the heat exchanger will be high (>10,000 cycles), then high cycle fatigue methods may be used to design the component for its intended life. The general procedure for designing for high cycle fatigue is as follows:

1. Determine the mean stress and alternating stress imposed on the heat exchanger structure from the mean and alternating internal pressure (Figure 11).
2. Obtain or estimate any stress concentration factors if the peak stress occurs at a sharp transition, such as a corner with a small radius or a structural joint such as a weld or braze joint. The stress concentration factor may be obtained by joint specific testing or correlation of test data with FEA. Apply the stress concentration factor to the nominal mean and alternating stress.
3. Obtain the fatigue life and tensile strength of the material at the normal operating temperature of the heat exchanger. Material fatigue life curves (stress vs. number of cycles to failure or S-N curves) may be obtained from published sources or developed from test data.
4. Plot the Goodman fatigue life line (Figure 12). The Goodman line is a plot of mean stress on the x-axis and alternating stress on the y-axis. This line is constructed by plotting the tensile strength on the x-axis (alternating stress = 0) and the fully-alternating fatigue strength of the material (mean stress = 0) on the y-axis. The fatigue strength is taken from the plot of the S-N curve (Figure 13).

Plot the mean and alternating stress values imposed by the internal pressure fluctuations on the Goodman line. Note that if the lower limit of the pressure cycle fluctuations is zero pressure, the mean and alternating pressures are the same value. Therefore, the point plotted on the Goodman line should be half of the stress calculated at the peak pressure for both the x (mean) and y (alternating) components. The stress values plotted should be scaled

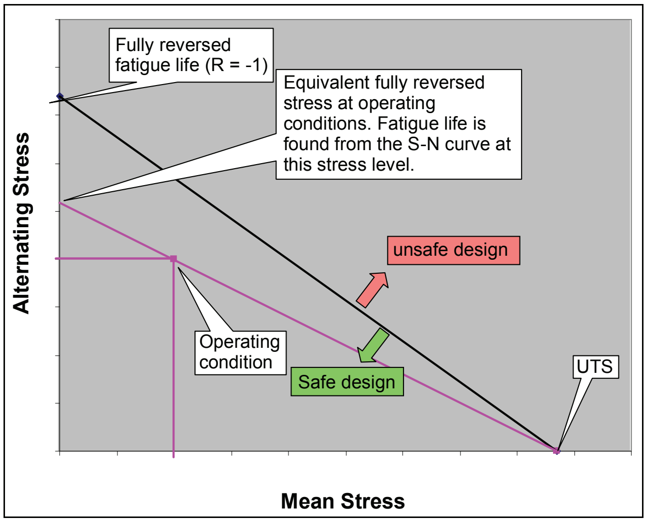


Figure 12. Goodman line plot.

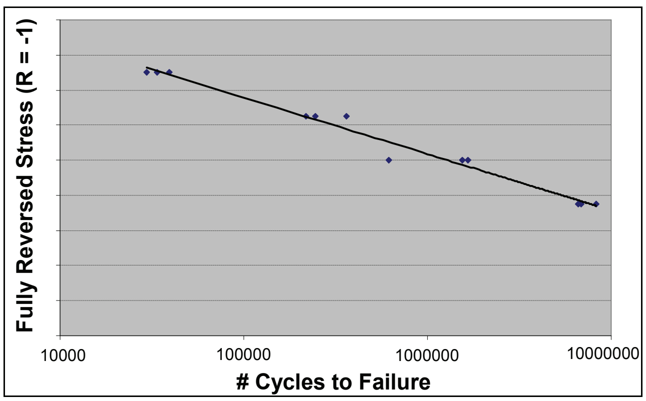


Figure 13. Stress vs. number of cycles to failure (S-N) curve.

up by imposing any stress concentration factor and factor of safety desired. If the plotted point falls above the Goodman fatigue life line, then the design should not be considered to be safe and redesign is required. If the plotted point falls below the fatigue life line, then the designer should proceed to investigating whether the stress should be combined with others and if the factor of safety is sufficient.

5. The fatigue life of the design may be found by projecting another line through the tensile strength on the x-axis and the operating stress point plotted to the y-axis. Once the fully alternating stress value is found from the y-axis, the designer may return to the S-N curve and find the expected life in number of cycles.
6. In order to combine the entire range-mean-count pressure data set or if pressure cycle fatigue must be combined with stresses imposed by other types of loading, then Minors' Rule may be used. If the sum of the ratios of number of cycles to fatigue life for all the conditions considered is less than one, then the design is considered to be safe. A ratio greater than one indicates an unsafe design. This ratio is normally called percent damage.

Minor's Rule: $$\text{Damage} = \sum\left\{\frac{n_i}{N_i}\right\}$$

where n_i = number of cycles at each operating condition
N_i = fatigue life at each operating condition

Minor's Rule may also be used to transform range-mean-count pressure data and pressure data from multiple operating conditions into an equivalent pressure cycle specification. The number of pressure cycles at a given mean and amplitude must be chosen so that the percent damage caused by the pressure cycle specification is equal to the percent damage from the operating conditions.

Design for Thermal Cycles

The design of heat exchangers to withstand thermal cycles is normally associated with the warm-up of the engine after being started. As the fluid circuits come up to their respective operating temperatures, the heat exchangers do not come up to temperature uniformly in all cases. The thermal imagining photograph shown in Figure 14 shows the temperature gradients present in a charge air cooler during a thermal cycle test. This non-uniform temperature distribution can result in large stresses. Vulnerable locations such as tube-to-header joints are of particular interest.

Designing for thermal cycles is a difficult engineering problem due to the transient nature of the warm-up. Transient thermal simulations normally require sophisticated computational fluid dynamics (CFD) software and a large amount of computing resources. Alternatives to a computer simulation are to measure temperatures with temperature probes or measure temperatures with a thermal imaging camera (see Figure 14). The thermal imaging camera is a better choice because it is difficult to take enough single-point temperature measurements to obtain a clear picture of the temperature distribution. It is a good idea to take a few single-point temperature measurements along with the thermal images to verify and calibrate the accuracy of the thermal image temperature scale. Another disadvantage of simply relying on measurements is that an initial design must be created, a sample built, and a test conducted in

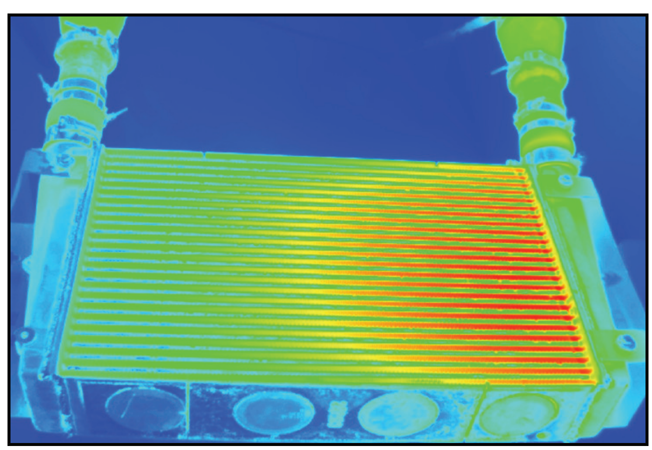


Figure 14. Thermal image from a CAC thermal cycle test.

order to find out if the temperature distribution is acceptable or not. Also, without a robust design tool it is often difficult to understand how a design should be modified in order to improve temperature distribution.

The warm-up cycle in the vehicle may last several minutes or longer. For validation test purposes, it is often desirable to develop an accelerated thermal cycle. These thermal cycle tests are normally conducted by alternately forcing hot and cold fluid through the heat exchanger core. Air is normally used as the working fluid during thermal cycle testing of charge air coolers and water or water-glycol solution is used for other heat exchangers. The temperature distribution and the associated stresses that occur in the heat exchanger due to these thermal cycles depend on the temperature difference between the hot and cold fluids, the ambient temperature surrounding the heat exchanger, the dwell time at the hot and cold temperatures, the rate of change (mixing) from hot to cold and cold to hot, and the mass flow of the hot and cold fluids. Thermal cycle testing will be discussed in more detail in another section. However, it is important to note that it may be important to take strain measurements in high-stress areas during the warm-up cycle in the vehicle and during the test thermal cycle. The test parameters can then be adjusted to match the strain and thereby inflict equivalent damage to the heat exchanger as a single warm-up-cool-down cycle. Heat exchangers are typically designed for 5,000 to 20,000 cycles. Due to the relatively low number of cycles involved in the design for thermal cycle, low-cycle strain-based fatigue methods may be required to design for thermal cycle fatigue.

Once the temperature distribution in the heat exchanger is known either from CFD simulation (Figure 15) or from measured or thermal image data, FEA may be used to predict stresses in the heat exchanger structure caused by the non-uniform temperature distribution (Figure 16).

Low-cycle fatigue methods involve correlation of constant strain versus the number of reversals to failure. The plastic strain vs. life relationship can be described by the Coffin-Manson equation:

$$\varepsilon_p = \varepsilon_f^{'}\left(2N_f\right)^c$$

where ε_p = amplitude of plastic strain
$\varepsilon_f^{'}$ = fatigue ductility coefficient (failure strain for a single reversal)
N_f = number of cycles to failure (1 cycle = 2 reversals)
c = fatigue ductility exponent (-0.7 to -0.5 for common metals)

Elastic strain life is described by the Basquin equation:

$$\varepsilon_e = \frac{\sigma_f^{'}}{E}\left(2N_f\right)^b$$

where ε_e = amplitude of elastic strain
$\sigma_f^{'}$ = fatigue strength coefficient

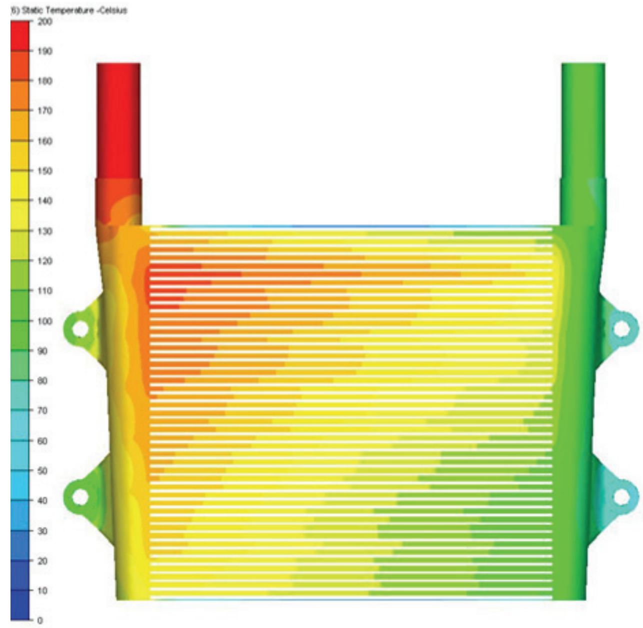


Figure 15. CAC temperature plot from CFD analysis.

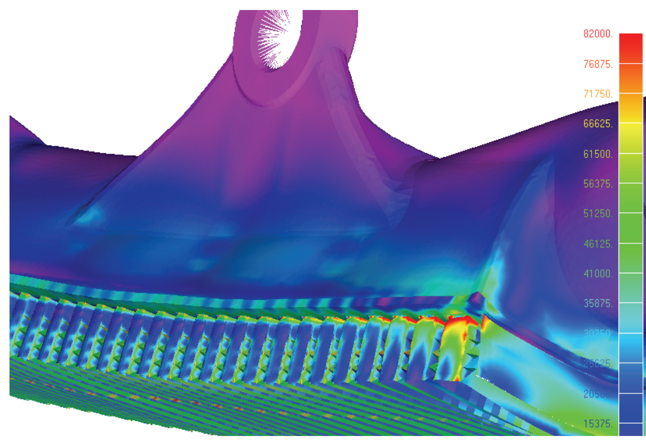


Figure 16. Stress plot, CAC during thermal cycle.

N_f = number of cycles to failure (1 cycle = 2 reversals)
b = fatigue strength exponent (-0.12 to -0.05 for common metals)

The strain life curve (Figure 17) is formed by the summation of the elastic and plastic strain components:

$$\varepsilon_t = \varepsilon_e + \varepsilon_p$$

The fatigue ductility coefficient (c) is found by plotting plastic strain vs. the number of strain reversals on a log-log scale and finding the slope. Similarly the fatigue strength exponent (b) can be finding the slope of the curve of elastic strain amplitude vs. strain reversals. The fatigue life is found from the total strain curve at a given strain amplitude. Inspection of the total strain curve shows that plastic strain is the dominating factor in low-cycle, high strain situations and elastic strain dominates in high-cycle situations. In fact, the strain-life and stress-life (S-N curve) methods will yield near the same results in high-cycle fa-

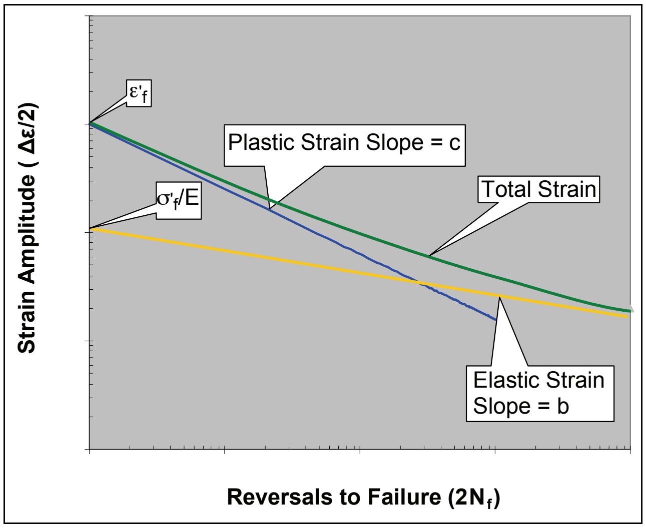


Figure 17. Strain life curve.

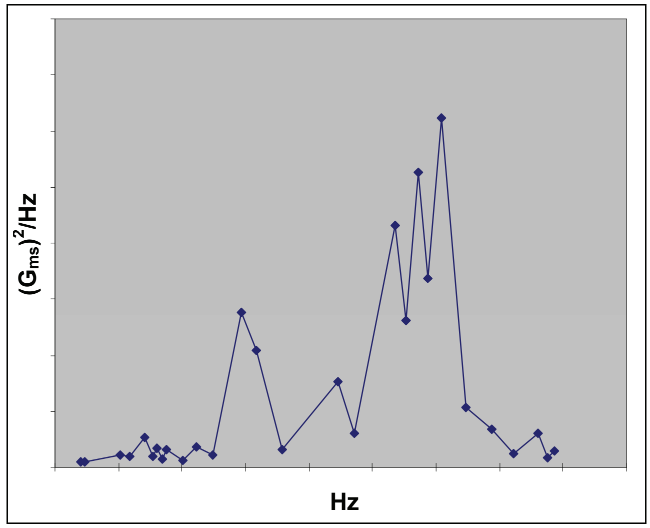


Figure 18. Example PSD curve.

tigue. There are additional corrections that can be applied to the strain life theory based on the specific situation including notch correction factors and mean stress correction that can be found in literature. Any method used to calculate fatigue life should be correlated against physical specimen test data. Another challenge in using strain life analysis is that strain life data for the materials used for heat exchanger construction may be difficult to find, especially in the post-brazed condition. Strain-life data may have to be measured experimentally for these materials or conservative estimates used.

The method for designing for fatigue life with low-cycle strain life methods is much the same as designing for high-cycle fatigue with the stress-life method. In this case the strain-life curve replaces the S-N curve. Minor's Rule is also used in this case for damage accumulation.

Design for Vibration

The design for vibration for heat exchangers lends heavily to a semi-empirical based approach. The first step is to collect acceleration data off the heat exchanger module mounting structure during field operation. This data is random in nature and is commonly expressed in a power spectral density plot (PSD). The field data is typically amplified to create an accelerated lab test PSD on the order of 10-20 hours. Figure 18 is an example of a PSD curve.

A number of methods ranging from the more simplified and heavily assumption-based G-loading analysis to the more accurate random response model can be used to aid the engineer in design against vibration.

The G-loading approach is commonly used due to its speed of calculation and readily available resources. In its most primitive form, hand calculations can be made for a specified G-load, for example to determine the safety factor on a pinned joint. FEA can be used to evaluate the stresses of more complex structures such as formed sheet metal mounting brackets. The drawback to the G-loading method is the loading assumption. Since the response of the structure is not calculated, an assumption on the response amplification factor must be made. This requires a heavy reliance on pre-existing test data of similar structures as well as very conservative assumptions on responses.

FEA models used for G-loading are usually adaptable to modal analysis. If the natural frequencies of the structure lie in the peak energy portion of the PSD curve, the structure may be stiffened to move the natural frequency to a lower energy frequency. For instance, if the structure to be tested to the example PSD curve has a natural frequency at 100 Hz, it would be beneficial to redesign the structure so this natural frequency occurs at 200 Hz or higher. In most cases the required PSD inputs for off-road equipment are fairly broad, so there is little benefit to adding or subtracting stiffness from a structure because the random response spike will still occur in the same PSD value.

Although modal analysis can identify the "excited" frequency, it does not provide any insight into the damage of the resonating structure. This requires random response analysis. More advanced FEA codes contain the functionality to input a random input curve (PSD) and calculate the random response and stresses of the structure. This is the most accurate type of analysis. However, it has a number of drawbacks. First, the random response analysis is linear. Heat exchanger mounting structures can include pertinent non-linear features such as isolators, contact joints, latches, and slots. Second, the random response analysis is a computer resource intensive problem. It requires extensive geometric simplification and very long solver times. In comparison, a static G-loading FEA can incorporate significantly more geometry content while still enabling fast solve times. Third, the random response analysis requires more advanced computer software and user knowledge.

The foregoing presents a trade-off in design methods. The more thorough and advanced FEA is typically used on designs requiring narrow safety factors or to evaluate out of the ordinary vibration specifications.

Material Selection for Corrosion Resistance

The corrosion resistance of non-road vehicles heat exchangers must provide sufficient protection against the conditions encountered in each vehicle's application(s). Internal corrosion is normally an issue only for radiators and oil-to-coolant heat exchangers where the engine coolant is used as a working fluid. Oil, fuel, charge air, and refrigerant are not highly corrosive so the heat exchangers associated with these fluids do not have an issue with internal corrosion (Figure 19).

Aluminum is currently the most common material used for the construction of heat exchangers for non-road vehicles. Aluminum is highly reactive metal which forms an oxide film very quickly in the presence of oxygen. The corrosion resistance of aluminum is derived from the strong bond between this oxide film and the surface. As long as this oxide layer is kept intact, further oxidation of the aluminum is prevented. Corrosion of aluminum is generally associated with the presence of some material or condition that destroys the oxide layer.

Aluminum corrosion can be broken up into several categories including general, galvanic, pitting, inter-granular, and erosion corrosion. These corrosion mechanisms can occur alone or simultaneously with one another.

Figure 19. Internal corrosion site in an aluminum heat exchanger (magnification 10X).

General corrosion is not commonly encountered. General corrosion can occur when aluminum is in uniform contact with a liquid with excessively high or low pH. Aluminum is passive between pH values of 4 and 8.5. As pH levels rise above 12 or fall below 4, the oxide layer may become soluble, causing corrosion to occur.

Galvanic corrosion is caused by the presence of a more-noble (cathodic) metal adjacent to a less-noble (anodic) metal in the presence of an electrolytic solution. The presence of a galvanic couple in a strong electrolyte such as seawater can cause rapid galvanic corrosion of aluminum. Aluminum is anodic to, and will corrode preferentially to, all metals except zinc and magnesium.

Pitting corrosion (see Figure 20) of aluminum is most commonly encountered in the presence of halide ions such as chloride. Pits will normally be initiated at localized weak spots in the oxide layer. In these areas, chloride ions in sufficient quantity can cause the breakdown of the oxide layer. Aluminum chloride forms locally decreasing the pH of the solution at the bottom of the pit and preventing the oxide layer from reforming. The growth of the pit can become self-sustaining at this point provided oxygen and the electrolytic solution remain present.

Inter-granular corrosion (see Figure 21) occurs due to the difference in solution potential between the grain boundaries and the grains of the material. This difference in potential is caused primarily by the tendency for copper present in aluminum alloys to be concentrated in the grains. The inter-granular regions with less copper content become anodic to the grains themselves causing corrosion to proceed along grain boundaries.

Erosion corrosion (see Figures 22 and 23) of aluminum occurs when the localized fluid velocity of water or coolant becomes high enough to strip away the protective oxide layer from the surface of the metal. Over time, material lost in a localized area can result in pitting that can eventually cause a leak. Erosion can also act in conjunction with other corrosion mechanisms by stripping away corrosion prod-

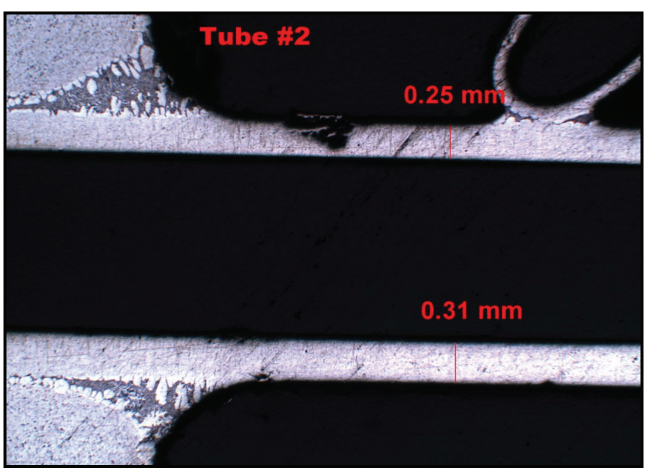


Figure 20. External corrosion failure of a tube wall after sea water acetic acid testing (SWAAT).

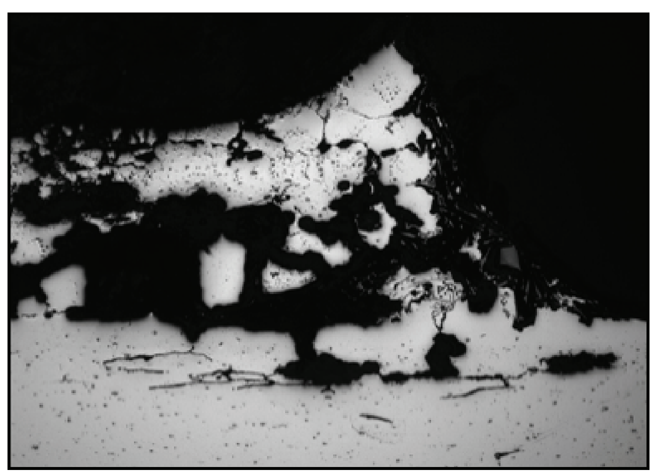


Figure 21. Cross section of the corrosion site from Figure 19 showing inter-granular corrosion.

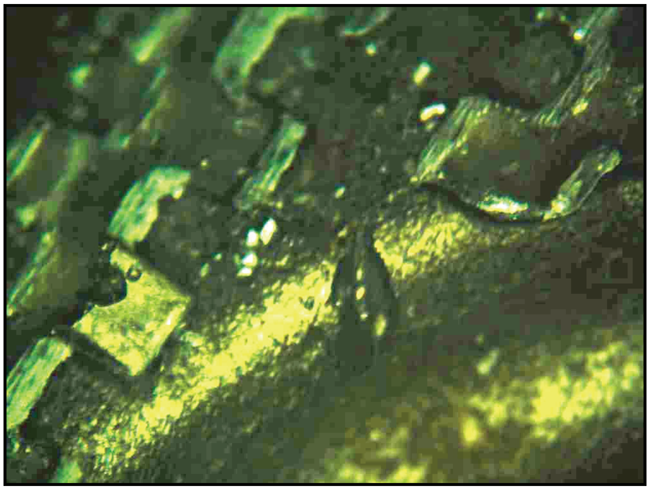


Figure 22. Erosion pit in an aluminum heat exchanger.

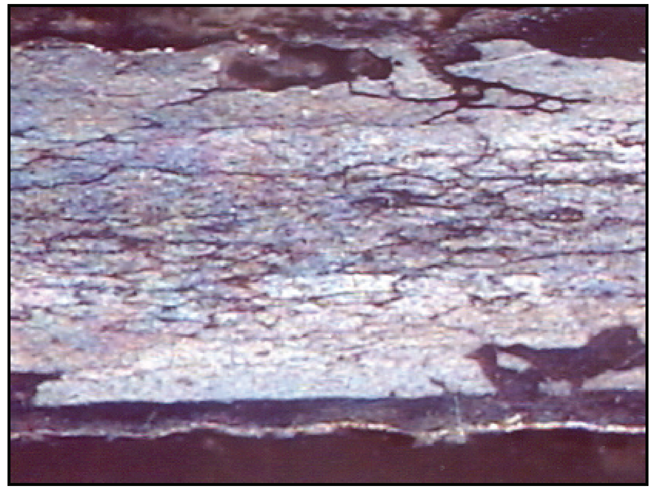


Figure 24. Example of the elongated grain structure of a long-life aluminum alloy.

Figure 23. Cross section of the erosion pit from Figure 21.

Table 2. Solution potentials of aluminum alloys and other metals (ASTM G69).

Metal or Alloy	Solution Potential, mV
Copper (99.999%)	0.00
321 Stainless steel	-300
Iron	-545
Zinc	-980
Magnesium	-1640
Aluminum (99.999%)	-750
1100	-750
3003	-725
6063	-755
Typical long-life aluminum tube alloy	-700
3003 + 1.5% Zn	-880
7072	-880
Typical braze clad (4004, 4045, 4343) layer	-710

ucts and the corroded surface of the metal. Erosion of aluminum is likely to occur when the localized velocity of water or coolant adjacent to the aluminum surface exceeds 3 m/s. The combination of erosion and corrosion can accelerate the time to failure. An increase in the surface hardness of a material will increase its resistance to erosion.

Radiators and liquid-to-liquid heat exchangers which use engine coolant as a working fluid are susceptible to internal corrosion in the case where coolant is poorly maintained or raw water is used as the coolant. Excessively high or low pH of the coolant and high concentrations of corrosive elements such as sulfates and chlorides can promote pitting and inter-granular corrosion. Aluminum radiator headers and tubes often have a sacrificial clad layer on the internal (coolant side) surface that is anodic (lower electrical potential) to the core alloy. For example, 1100 and 7072 aluminum alloys are often chosen as liner alloys. The presence of a foreign metallic particle such as iron or copper can set up a galvanic corrosion cell that can eventually lead to failure.

External corrosion can be an issue for all heat exchangers in the cooling system. If a vehicle is continuously exposed to a coastal or other sea water environment, external corrosion can proceed rapidly due to a continuous presence of moisture and salt to acting as an electrolyte. Localized galvanic corrosion cells can set up even between different aluminum alloys used for heat exchanger construction. Thus, heat exchanger alloys should be carefully chosen. If at all possible, aluminum alloys specifically engineered for long life should be used, especially for critical and thin components such as tube walls. Long-life alloys have a flat grain structure (see Figure 24). For these alloys, if intergranular corrosion does occur, the corrosion will progress primarily in a parallel direction relative to the surface of the material. A braze clad layer, if present, should corrode preferentially to the aluminum core alloy. Many modern long-life engineered aluminum alloys contain carefully selected combinations of alloying elements so that a dense, “brown band” layer forms during brazing due to the migration of Si from the clad layer into the core at the interface of the clad and core layers. In a thin region along this interface, small particles of $Al_x(MnFe)_ySi$ precipitate. This band has a lower solution potential than the core layer causing it to corrode sacrificially to the core. In addition, the designer should obtain the solution potential of each aluminum alloy to be used and select the alloys so that less-critical components of the structure, such as air fins, are anodic to critical components such as the tube core alloy. Selection of an aluminum air fin alloy with an addition of a small percentage of zinc is one way to accomplish this. A zinc coating may be used on extruded aluminum tubes to form a sacrificial surface layer.

Heat exchangers containing dissimilar metals such as mechanically expanded round-tube heat exchangers with steel or copper tubes and aluminum fins must be painted in order to prevent oxidation of the steel or copper surfaces and to prevent galvanic corrosion between the copper or steel tubes and the aluminum fins. Paint is a way to prevent aluminum heat exchanger corrosion as well. However, painting aluminum coolers is an added cost that is normally not necessary.

Validation Testing

Full-vehicle validation testing of cooling systems is a critical step in the development of any vehicle cooling system. The components and the cooling system itself, if it is to be supplied as a single unit, should be validated by component testing, ideally prior to vehicle testing. This will help to ensure that vehicle testing proceeds smoothly without failures that could have been detected by appropriate bench testing. Vehicle testing is often required to measure installed cooling air flow and to measure cooling performance.

Thermal Performance Validation Testing

The thermal and pressure loss performance of heat exchangers should be validated in a component wind tunnel. Inlet and outlet temperatures of the internal fluid and the cooling air, internal and external flow rates, and inlet-to-outlet differential pressure measurements are required for component wind tunnel testing. Specific heat transfer and pressure loss curves are normally reported.

Burst Testing

Burst testing is useful to ensure that a heat exchanger is capable of withstanding the maximum possible pressure it might see in application. These pressure spikes may occur in an oil cooler after a cold weather start-up, for example. Burst testing is also useful as a quick structural test to ensure that a heat exchanger is structurally sound and within the limits of normal production variation.

Pressure-Cycle Validation Testing

Internal pressure-cycle testing is used to validate the ability of a heat exchanger to survive the internal static and fluctuating pressure of the internal fluid over the life of the vehicle. The heat exchanger is filled with the internal working fluid or an appropriate substitute and connected to a test stand with the ability to create and control the fluid pressure fluctuations between a minimum and maximum level for a specified number of cycles. Often pressure-cycle testing is accelerated using equivalent damage techniques so that pressures higher than operating pressures are used for a reduced number of cycles to validate the component. Pressure-cycle testing should be conducted at the maximum expected operating temperature or the pressure should be increased to compensate.

Thermal-Cycle Validation Testing

Thermal-cycle testing is conducted to ensure that a heat exchanger can withstand a sufficient number of warm-up and cool-down cycles to survive for the intended life of the vehicle. The internal working fluid or an appropriate substitute should be circulated thought the cooler. Normally, thermal-cycle testing is accelerated. The temperature difference between hot and cold, the fluid flow rate, the cycle rate, and the number of cycles should be carefully chosen to simulate the amount of damage caused by the vehicle warm-up and cool-down cycles.

Vibration Testing

Vibration testing should be used to ensure that the heat exchangers and the structure that they are mounted to can withstand the shock and vibration expected over the life of the vehicle. Vehicle data is often collected and amplified to simulate vibration loading over the life of the vehicle in a short amount of time. Single- or multi-axis shaker tables are normally used for validation testing.

There are four main categories of vibration testing: shock, sine sweep, operational load, and random.

Shock testing is used to simulate the loads on a structure and the subsequent response generated by a collision or a sudden drop. In the past, simple drop-test tables were used to repeatedly apply a single instantaneous G-load on a heat exchanger or cooling system. Often this was the only type of vibration testing performed.

Sine sweep testing is used to find the natural frequencies of a heat exchanger or system. For this type of testing, a constant-amplitude sinusoidal input load is generated on a vibration table and the frequency is swept from a minimum to a maximum value. The response of the heat exchanger or system is measured and any response peaks are identified as potential resonance points. A variation of this type of testing is to dwell at the resonance frequencies at for a prescribed amount of time and input amplitude.

Operational load testing measures the vibration signal during actual operating conditions as reproduced on a vibration table. Normally, this type of testing is considered to be too time-consuming for bench test validation. However, useful structural response data may be measured by performing this type of test.

Random vibration testing is the most useful type of vibration validation testing. In order to perform random vibration testing, the vibration PSD is measured in each axis during vehicle operation. This data often simplified and amplified based on the fatigue properties of the material and the desired test time. The goal of the lab test is to cause equivalent damage to the structure as would be experienced during the intended service life of the vehicle operation.

The following formula may be used amplify the vibration PSD curve for short-duration laboratory testing:

$$\frac{t_2}{t_1} = \left(\frac{W(f)_1}{W(f)_2} \right)^{\frac{m}{2}}$$

where t_2 = test time

t_1 = design life

$W(f)_1$ = amplified test PSD ($G2$/Hz)

$W(f)_2$ = operational PSD at design conditions (G^2/Hz)

m = fatigue damage exponent

The fatigue damage exponent (m) depends on the fatigue properties and surface finish of the material and on the loading waveshape. The value of m is inversely proportional to the slope of the S-N curve (b). MIL-STD-810G recommends reducing this ratio to 80% for random vibration.

$$m = 80\% \left(-\frac{1}{b} \right)$$

where b = fatigue strength exponent.

Installation (Torque) Testing

Installation of each heat exchanger should be conducted at worst-case conditions to ensure that installation, including mounting and connecting the internal fluid lines, will not cause damage and potentially a premature failure.

Corrosion Testing

Radiators commonly require internal corrosion testing validation. The internal working fluid for charge air coolers, oil coolers, fuel coolers, and condensers is not typically very corrosive so internal corrosion testing is not usually performed on these components. Radiator internal corrosion testing is conducted by circulating a pre-determined corrosive coolant solution through the radiator core for a predetermined number of hours. The flow rate and temperature of the coolant should be selected to simulate operating conditions.

External corrosion testing is normally conducted in standardized salt spray chambers according to ASTM or other specifications. Common specifications for salt spray testing include ASTM G85, Annex A3 sea water acetic acid test (SWAAT), and ASTM B117 neutral salt spray test.

Summary and Conclusions

Non-road cooling systems typically consist of the heat exchanger components required for engine cooling, including the radiator, charge air cooler, and fuel cooler. Heat exchangers required to cool additional fluid circuits are often included as well, including hydraulic and transmission oil coolers and air conditioning condensers. Often the heat exchangers are supplied mounted together as a single unit. Normally, the heat exchangers are air-cooled as a group by an engine-driven cooling fan. Heat exchanger thermal performance and internal and external pressure loss can be calculated using standard engineering methods. System air flow and temperature rise calculations are performed by idealizing vehicle and heat exchanger core restrictions as being either in series or parallel and correcting pressure loss and fan performance curves using similitude. Special consideration must be made in the design of non-road cooling systems for airborne particulate clogging. Continuous (non-louvered) fins should be used and fin densities should be limited depending on the application. Stacked (multi-plane) cooling systems must be designed to allow access to each heat exchanger core in order for debris to be cleaned from the core periodically. The heat exchangers and supporting structures must be structurally designed to withstand internal pressure, thermal cycle, and vibration loads. Heat exchanger materials must be selected to resist internal and external corrosion. Validation testing should be conducted to validate thermal, structural, and corrosion performance of the cooling system.

References

Davis, Joseph R. 1999. *Corrosion of Aluminum and Aluminum Alloys*. ASM International.

Decker, Matthias, and Steffen Rödling. 2009. FatiRAN–Optimized Generation of Spectra for Random Vibration Testing. *MP Materials Testing* 51(2009): 7-8.

eFunda, Inc. 2010. Low-Cycle Fatigue. http://www.efunda.com/formulae/solid_mechanics/fatigue/fatigue_lowcycle.cfm.

Engrasp, Inc. 2008. *Strain-Life Fatigue Analysis, An Engineers Toolbox Calculation Module.* http://www.fea-optimization.com/ETBX/strainlife_help.html.

Incropera, Frank P., and David P. DeWitt.1996. *Fundamentals of Heat and Mass Transfer,* 4th ed. John Wiley & Sons, Inc.

Kays, W. M. and A. L. London. 1998. *Compact Heat Exchangers,* 3rd ed. Krieger Publishing Co.

MIL-STD-810G. 2008. *Department of Defense Test Method Standard: Environmental Engineering Considerations and Laboratory Tests.* Department of Defense, USA.

Munson, Bruce R. 1990. *Fundamentals of Fluid Mechanics*. John Wiley & Sons, Inc.

Simmons, Ryan. 1997 *FEMCI, The Book: Calculating G_{rms} (Root-Mean-Square Acceleration).* http://femci.gsfc.nasa.gov/randomgrms.html.

