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Challenges in Cooling System Design for Hybrid Electric Vehicles

N. Staunton[1], V. Pickert[2], R. Maughan[3]
[1]*N. Staunton (corresponding author) EECE, Newcastle University, UK, n.staunton@ncl.ac.uk*
[2]*EECE, Newcastle University, UK.*
[3]*Comesys Europe Ltd., UK*

Abstract

An internal combustion engine, up till now has been the only significant source of heat in commercial vehicles. A mechanical coolant pump and mechanical fan usually suffice to crudely control the operating temperature of the internal combustion engine. The introduction of power electronics and electric machines into the vehicle means that the cooling system must be able to cope with thermal loads that vary greatly from one another. Removing heat from thermal loads that operate at different temperatures is one of the main challenges imposed on the hybrid vehicle cooling system. This paper will analyse the difference between thermal loads introduced by the hybrid drivetrain. A simple vehicle dynamics model will be used to predict thermal loading on a series hybrid drivetrain. This model, coupled with cooling system component models will be used to verify the concepts laid out on this paper. It will be shown that with careful consideration of cooling system topology and control strategy, all thermal loads can be effectively managed.

Keywords: cooling, radiator, simulation, thermal management, pumps.

1 Introduction

A hybrid drivetrain, whether series or parallel, typically consists of an internal combustion engine, one or more electric machines providing traction and regeneration, power electronics for the electric machines and a battery pack. The internal combustion engine is the main source of energy. Since it is not 100% efficient, it produces heat as well as useful work. Typically, an internal combustion engine is very inefficient and approximately 60% of the combustion energy is manifested as waste heat. The internal combustion engine operates at a relatively high temperature of 90 to 100 degrees Celsius. Before the hybrid drivetrain was introduced, this single source of heat has been quite easy to deal with. A single cooling circuit and heat exchanger are employed to remove waste heat. This cooling arrangement is uncomplicated and has remained largely unchanged over the last century. However, the simplicity has its drawbacks. Such a system has no or very basic control and the mechanical pump and fan operate with a duty that is directly related to the speed of the engine and not as a function of the thermal load placed into the cooling circuit.

The hybrid drivetrain has introduced other significant sources of heat. These are electric machines used for generating electrical power when coupled to the internal combustion engine or for generating traction when coupled to the wheels of the vehicle. It is common to see traction motors being used to decelerate the vehicle and recover kinetic energy that would have been lost under friction braking. Typically, the maximum operating temperature of the power electronics is 60 degrees Celsius while the maximum operating temperature of the electric machines is in the region of 70 degrees Celsius. It is desirable to operate electric machines at as low a temperature as possible since efficiency decreases with an increase in temperature [1]. Now it can be seen that a single cooling loop will not suffice to adequately deal with the very different thermal loads in the system. It may seem logical that a cooling circuit designed to remove over 100kW of heat from an internal combustion engine under all operating conditions will easily manage the smaller heat loads imposed by the power electronics and electric machines. However, after some numerical analysis and vehicle simulation, it will be shown that it cannot be this simple. In the next section, a brief overview of the principle of vehicle cooling will be followed by some application examples that demonstrate the cooling challenges imposed by hybrid vehicle drivetrains.

2 Cooling System Overview

2.1 Basic Cooling System Layout

The basic internal combustion engine cooling circuit is shown in Figure 1. Many hybrid drivetrains still use this arrangement for cooling the internal combustion engine.

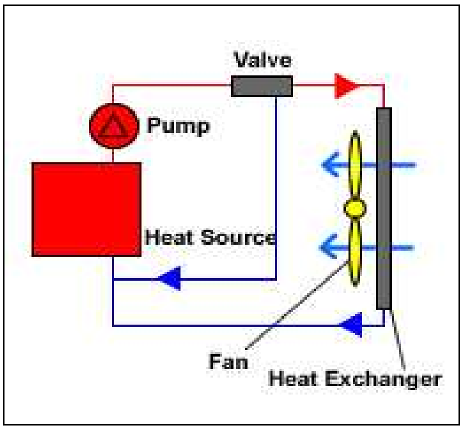

Figure 1: Basic internal combustion engine cooling circuit

The heat source generates waste heat that is transferred to a coolant fluid, usually a mixture of ethylene glycol and water. The coolant pump is usually driven directly by the engine. This moves the coolant through a wax based, thermostatic valve. The valve allows the coolant to pass over the heat exchanger when it is hot and bypasses it directly back to the engine when it is cool. When the coolant fluid is hot, it is cooled by forced convection with a mechanically driven fan. Air, at ambient temperature, is forced over the heat exchanger. The heat lost by the coolant is given by the following equation:

$$Q = \dot{m}.C_p.dT\ , \qquad (1)$$

where Q is the amount of heat energy transferred in Watts, $\dot{m}$ is the mass flow of the fluid in kg/s, C_p is the specific heat capacity of the fluid in J/kg.K and dT is the change of temperature on the fluid in degrees Celsius.

As already stated, this principle of cooling has been widely adopted in all automotive environments and applications.

2.2 Drawbacks of Standard Cooling Layouts

While the simplicity of the above approach to vehicle cooling makes it a desirable method to adopt for hybrid vehicle cooling, we will now see why this is not straight forward.

Take the example of an internal combustion engine that is producing 100kW of waste heat. The coolant fluid is assumed to be pure water for simplicity and this waster is flowing at a rate of 200 litres per minute. A sensible operating temperature to choose for these conditions is 90 degrees Celsius. Under these conditions, we can rearrange equation 1 to get dT :

$$dT = \frac{Q}{\dot{m}.C_p} = \frac{100kW}{2.895m^3s^{-1}.4.2kJ.Kg^{-1}.K^{-1}} = 8.2\deg.C$$

As a comparison, we will look at a hybrid drivetrain cooling problem, based on a commercially available hybrid drivetrain. In this example there are three machines, one generator and two traction motors each with a maximum waste heat rejection of 10kW. These are individually controlled by a separate power electronic inverter. Each inverter has a maximum waste heat rejection of 5kW. The major constraints imposed by this system are the maximum allowable temperature on the power electronics and the maximum flow rate of the coolant. The maximum temperature of coolant returning to the drive is 55deg.C while the maximum coolant flow rate is limited to 60 litres per minute due to the use of an electric coolant pump. This pump will deliver this flow rate when operating from a 24V power net with a pressure head of 2 bar. Figure 2 shows all of the mentioned thermal loads combined into one cooling circuit. The total required heat rejection based on the component specification outlined previously is 45kW at a maximum coolant exit temperature of 55 degrees Celsius.

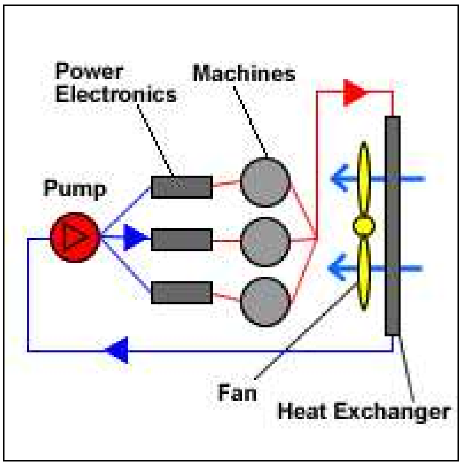

Figure 2: Hybrid electric heat loads combined into one cooling circuit

Using equation 1 again, we can find the required dT to achieve adequate cooling:

$$dT = \frac{Q}{\dot{m}.C_p} = \frac{45kW}{0.985m^3s^{-1}.4.1kJ.Kg^{-1}.K^{-1}} = 11.1\deg.C$$

We can now see that, despite the lower heat rejection required by the power electronic thermal loads, the required dT is higher due to the restriction on coolant flow rate. To make matters worse, ambient temperature and heat exchanger effectiveness must be considered. Most worst case vehicle cooling system testing procedures specify an ambient temperature of 45 degrees Celsius. If this were the case in the above example, the 45kW heat rejection target would not be reached because the maximum, theoretical dT would be 10 degrees Celsius. Another major consideration is the heat exchanger effectiveness which is the ratio of actual heat transfer to the theoretical maximum heat transfer [2]. This, in reality is in the region of 80% in the most ideal conditions. The effectiveness decreases as the temperature difference between the coolant fluid and the ambient air decreases. We can now see why the internal combustion engine cooling circuit has relatively little difficulty in rejecting large amounts of heat in worst case conditions.

2.3 A solution to the above problem

Figure 3 shows a possible solution to the above cooling problem. Splitting the cooling loop into smaller ones means that the system complexity is increased. Extra pumps are required to drive fluid through the new circuits and there are additional heat exchangers and fans. However, now the heat presented to each heat exchanger is much lower at approximately 15kW in worst case conditions.

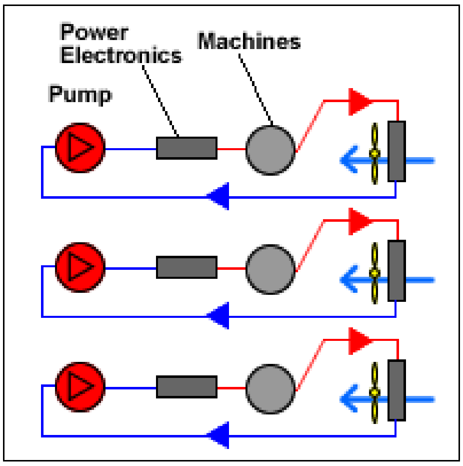

Figure 3: A solution to the power electronic thermal load cooling problem

By applying equation 1 we can now see that the required temperature drop for each circuit is a much more realistic 3.7 degrees Celsius.

$$dT = \frac{Q}{\dot{m}.C_p} = \frac{15kW}{0.985m^3s^{-1}.4.1kJ.Kg^{-1}.K^{-1}} = 3.7\deg.C$$

2.4 Airflow

The operation of the fan as described in section 2.1 is directly proportional to the speed of the engine. The series hybrid drivetrain could take advantage of this since the engine operates at a relatively steady duty cycle that varies very little under most driving conditions of the hybrid electric vehicle. So, the fan can be designed so that its peak efficiency is at the average speed of the internal combustion engine. Conversely, the parallel hybrid drivetrain will not benefit from this type of fan coupling. Ideally, the fan should be decoupled from the drivetrain and be controlled electronically so that the fan speed and hence, cooling airflow is a function of coolant temperature only. It can be shown that by controlling the fan speed so that excess power is not consumed unnecessarily, significant reductions in fuel consumption can be achieved [3].

When cooling the power electric thermal loads on the hybrid drivetrain, it is logical to employ electronically controlled, brushless DC fans. This is because the power electric loads and their heat exchangers are not necessarily placed near the mechanical drivetrain. The same benefit of controlled cooling can be achieved since the fan operation can be set as a function of coolant temperature.

Two of the fan laws are:

$$\frac{S_1}{S_2} = \frac{C_1}{C_2} \tag{2}$$

and

$$\frac{P_1}{P_2} = \left(\frac{S_1}{S_2}\right)^3, \tag{3}$$

where S is fan speed, C is fan capacity or flow rate and P is fan power. Using equation 2 and 3, it can be shown that using an array of small fans uses less energy than a single large fan to move the same amount of air. This is clearly demonstrated in [4]. With this in mind, it is clear that a large heat load imposed by the internal combustion engine in the hybrid electric drivetrain can be managed with an array of smaller, electric fans. This will reduce fuel consumption and emissions [5].

3 Vehicle Simulation

A vehicle dynamics model was constructed in Matlab/Simulink in order to predict the likely thermal loads generated in a hybrid electric bus. The governing equations for the vehicles dynamics model are given in [6] and are shown here:

Rolling Resistance

$$F_{Ro} = f.m.g.\mathrm{Cos}(\alpha), \tag{4}$$

Here F_{Ro} is the rolling resistance in N, f is the dimensionless coefficient of rolling resistance, m is the mass of the vehicle in kg, g is the acceleration due to gravity in m/s^2 and α is the gradient angle in degrees.

Aerodynamic Drag

$$F_L = 0.0386.\rho.c_d.A.\ (v - v_0)^2 \tag{5}$$

In equation (5) F_L is the aerodynamic drag resistance in N, ρ is the density of air in kg/m^3, c_d is the dimensionless drag coefficient, A is the frontal area of the vehicle in m^2, v is the vehicle speed in km/h and v_0 is the headwind speed in kmh.

Climbing Resistance

$$F_{St} = m.g.Sin(\alpha) \tag{6}$$

Where F_{St} is the climbing resistance in N.

The total running resistance power is then calculated by:

$$P_W = \frac{(F_{Ro} + F_L + F_{St})\ .v}{3600}, \tag{7}$$

where P_W is the running resistance power in kW.

The heat generation of the internal combustion engine and power electronics are a function of the overall running resistance power.

$$Q_{Gen} = f(P_W) \tag{8}$$

Heat transfer from the engine to the coolant, from the coolant to the engine block, from the machine windings to the machine chassis and from the machine chassis to the cooling medium is given by:

$$Q_{Conduction} = hA.dT, \tag{9}$$

where $Q_{Conduction}$ is the rate of heat transfer between the two mediums in kW, h is the heat transfer coefficient in W/m^2.K, A is the surface area of contact between the two mediums in m^2 and dT is the temperature difference between the two mediums in degrees Celsius.

The internal combustion and electric machines are modelled as lumped thermal masses as described in [7]. The change in temperature of the internal combustion engine and electric machine masses are given by:

$$\frac{dT}{dt} = \frac{Q_{Conduction}}{m.c_p}, \tag{10}$$

where m is the mass of each component in kg and c_p is the specific heat capacity of the component material in J/kg.K.

The thermostatic valve in the standard internal combustion engine cooling circuit was modelled with a lookup table in Matlab/Simulink.

The heat exchanger was modelled using a two dimensional, interpolated lookup table using bench test data gathered for an automotive grade water to air heat exchanger.

The pump and fan were modelled in a similar manner using lookup tables of experimental data. Electric fans were also modelled for cooling the power electronic cooling loops.

The vehicle weight was taken to be 18000kg and the ambient temperature was taken to be 45 degrees Celsius.

The simulation was run with a steady state vehicle speed of 70 kmh.

4 Simulation Results

Firstly, the simulation was run for the cooling system topology shown in Figure 1. The resulting warm-up profile is shown in Figure 4 and the temperature drop across the water to air heat exchanger is shown in Figure 5.

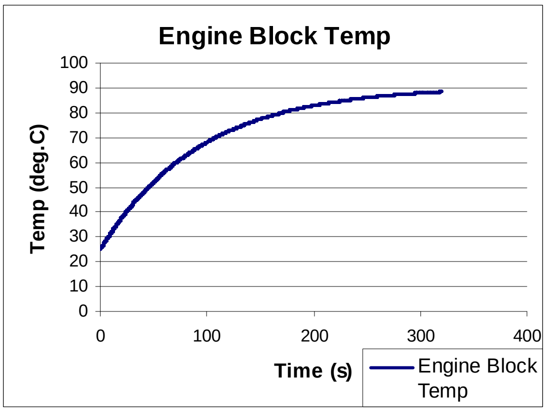

Figure 4: Simulated engine warm-up cycle for fully loaded vehicle travelling at 70kmh

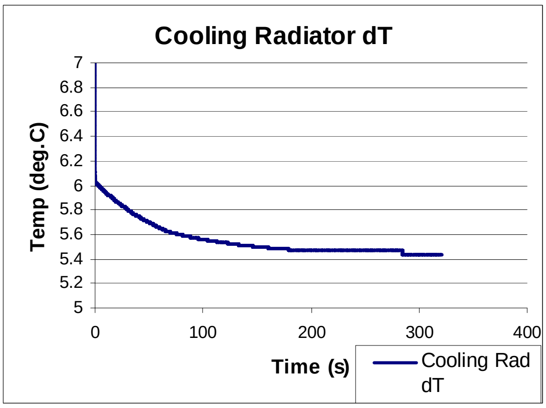

Figure 5: Simulated temperature drop across water to air heat exchanger

The simulation was run again, this time employing two electric machines to drive the vehicle in a series hybrid configuration. The cooling circuit modelled is shown in Figure 3. The warm-up profile for the machine is shown in Figure 6. The temperature drop across the water to air heat exchanger is shown in Figure 7.

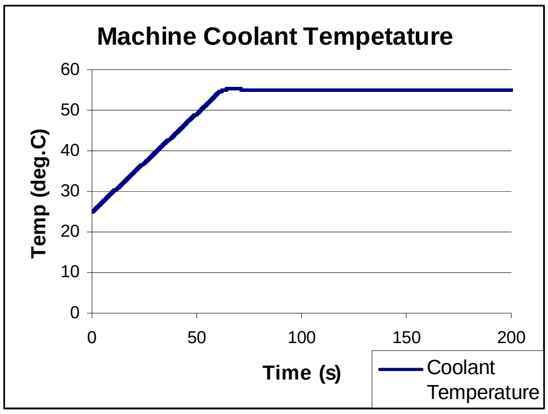

Figure 6: Simulated machine warm-up cycle for fully loaded vehicle travelling at 70kmh

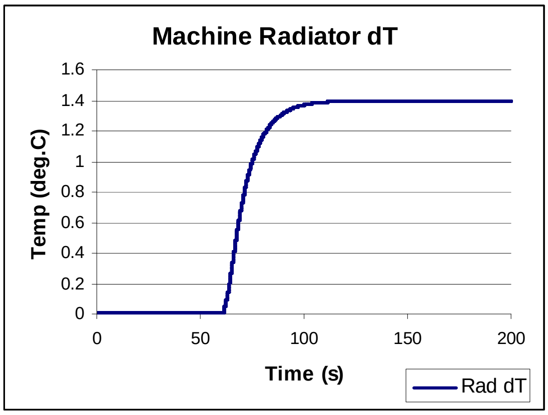

Figure 7: Simulated temperature drop across radiator

5 Discussion

The simulation results are in good agreement with the initial numerical analysis for the heat performance of the internal combustion engine and power electronic loads in a hybrid electric drivetrain. The steady state temperature drops across the heat exchanger are lower than shown initially estimated for both the electric machines and internal combustion engine. The reason for this is the difference in heat energy input to the cooling circuits and cooling flow rate since these are a function of internal combustion engine speed.

The sharp roll off for the temperature drop across the heat exchanger for the electric machine is due to a PID controller commanding the electric fan that is cooling the device.

To justify the use of three cooling loops for the electric machines, the simulation was run again with all the hybrid electric thermal loads combined into one cooling loop as shown in Figure 2. Under the standard driving cycle as in previous tests, the coolant temperature was correctly controlled. A more demanding thermal load was imposed that delivered an overall thermal load of 45kW by increasing the driving gradient. The resultant coolant temperature profile is shown in Figure 8.

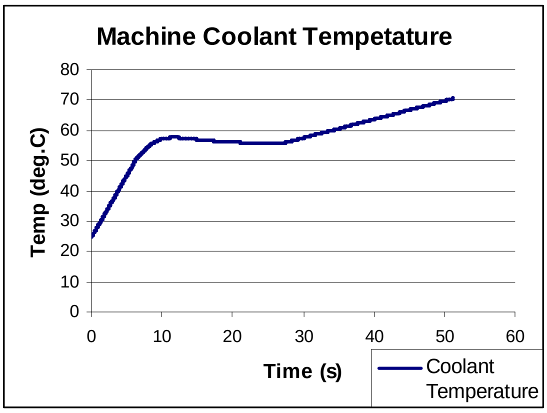

Figure 8: Simulated temperature drop across radiator with combined cooling circuit for hybrid electric thermal loads

It can be seen that the coolant temperature cannot be correctly controlled with this topology and diverges from the set point of 55 degrees Celsius.

It should be noted that the split cooling topology as shown in Figure 3 maintained control of the thermal load with the increased gradient simulation.

6 Conclusion

No simple method exists that will allow adequate cooling of all thermal loads in one cooling circuit. Table 1 shows the operating temperatures and relative magnitude of the thermal loads that must be controlled in a HEV cooling system. It also shows that the cooling system complexity increases as the operating temperature of each thermal load gets smaller.

Table 1: Thermal loads in a HEV cooling system

Device	Temperature	Thermal load	Complexity
IC	90-100	High	Low
Machines	70-80	Medium	Medium
Drives	50-60	Low	High

It has been identified that a new approach to cooling system design must be taken when dealing with the HEV drivetrain. The cooling challenges will increase as more heat loads are introduced that operate at low temperatures. It has also been shown that there is a potential to save energy by optimising the cooling fan arrangement and duty cycle.

Acknowledgments

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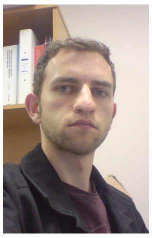

Nikolaos Staunton received a B.Sc(Hons) in Applied Physics and Electronics from the National University of Ireland, Galway in 2006. He is currently studying for an Engineering Doctorate in Power Electronics, Drives and Machines at Newcastle University. His research interest are thermal management, hybrid powertrains, automotive networks, embedded systems and electric drives.

Volker Pickert received his Dipl.-Ing. in Electrical and Electronic Engineering from the RWTH Aachen, Germany and the University of Cambridge, UK in 1994. He received his PhD from Newcastle University in 1997. His research interests are power electronics for automotive applications, thermal management, fault tolerant converters and nonlinear controllers.