

Various Impacting factors on a Radiator Top Tank Temperature

Ying Tang, Nicolas Grall and John Savage

Valeo, Inc., Engine Cooling Division

Copyright © 2008 SAE International

ABSTRACT

The objective of this paper is to quantify variations of radiator top tank or coolant inlet temperature. Radiator top tank temperature (radiator coolant inlet temperature) can be predicted using standard heat transfer calculations assuming pre-defined cooling module architecture for required vehicle road or wind tunnel test conditions. Variation of top tank temperature can easily be calculated using formulas derived in this paper for any changes to the influencing parameters or test conditions, such as ambient temperature, coolant flow rate etc.

Variation of top tank temperature can be expressed as a function of independent parameters and conditions for a range of assumed engine cooling module architectures. Independent parameters for consideration are air flow rates, coolant flow rate, engine to coolant and engine oil heat rejection rates, etc.

One dimensional thermal fluid simulation runs were conducted with Kuli™ software to validate the above derivations and concepts. Discussions of this mathematical expressions and its application are both presented in this paper.

INTRODUCTION

Standard vehicle power-train cooling system design practices require any production intent engine cooling system (ECM) to be validated by a series vehicle road tests or equivalent vehicle wind tunnel tests. However, vehicle road or wind tunnel tests often do not occur as planned. In addition some test conditions and/or vehicle test parameters can not be achieved as exactly as required. One classic example of this is when ambient temperatures for the road test period do not reach the design target values. These situations result in this question:

- How to interpret such “non perfect” test results for vehicle road tests?

The questions outlined below often come up during vehicle power-train cooling design and development phase:

- What is the impact to radiator top tank temperature if intended radiator coolant flow is varies slightly from its design target value?
- What happens when engine oil heat loads vary from original design numbers?
- What impact does air conditioning (A/C) system operation have on radiator top tank temperature?
- What is impact on top tank temperature if fan provided airflow varies from its target value?

This is only a short list of questions that often are raised during real world design phases.

Another important aspect of today's vehicle power-train cooling design is the consideration of design reliability vs. production variation of performance of sub-components within the ECM system. In other words, the engine cooling module design needs to perform well in its nominal condition, but is also must provide satisfactory performance at the extreme ends of its production tolerance variation.

It is not a trivial and easy task to provide complete and satisfying answers to the aforementioned questions in today's engine cooling module (ECM) design practice. In

Rahman, Sun [1], a statistical approach was used to analyze various design parameters and testing conditions influence on power-train or engine cooling designs. In Savage [2], the combined impacts of multiple parameters and conditions on such design were discussed and developed further.

This paper is a continuing discussion of same topic as in [1] and [2]. In this paper, the approach will be to establish boundaries or extreme variations of design parameters and vehicle test conditions. Instead of statistical analysis based on traditional heat transfer theory, a variation of an engine cooling modules' top tank temperature vs. the variations from design parameters and testing conditions can be derived and formulated in a mathematical expression.

A detailed derivation is given in the first part along with assumptions and conditions. Then, the derived mathematical expressions were used to evaluate influence of various design parameters and test conditions. For comparison proposes, KULI based 1-D simulations were performed for identify the influences of these parameters and test conditions. Detailed discussions and applications are provided to further explain the derived mathematical expressions.

A summary of all the notations and coefficients used in the following paragraphs is attached in nomenclature and appendixes sections of this paper.

GENERAL EXPRESSION

Typically, radiator thermal performance can be expressed as:

$$\dot{Q} = \varepsilon C_{\min} (T_{c,i} - T_{a,i}) = \varepsilon C_{\min} (ITD) \quad (1)$$

The air side mass based heat transfer capacity C is usually smaller than the coolant side behavior for typical vehicle engine cooling test conditions. Thus, temperature of top tank or coolant side inlet temperature of a radiator can be simplified as:

$$T_c = T_a + \frac{\dot{Q}}{\varepsilon C_a} = T_a + \frac{\dot{Q}}{\varepsilon \rho_a C_{p_a} V_a} \quad (2)$$

Radiator top tank temperature can vary and drift away from nominal or specified design points when influenced by fluctuations such as manufacturing tolerances, airflow rates, coolant flow rate, and air side inlet temperature. An error equation of the radiator top tank temperature can be derived from above expression and expressed as:

$$\Delta T_c = T_c^* - T_c^0 = (T_a^* - T_a^0) + \left(\frac{\dot{Q}}{\varepsilon C_a^*} - \frac{\dot{Q}}{\varepsilon C_a^0} \right) = \Delta T_a + \frac{\dot{Q}}{\varepsilon C_a^0} \left(\frac{\dot{Q}^* \varepsilon C_a^0}{\dot{Q} \varepsilon C_a^*} - 1 \right) \quad (3)$$

$$\Delta T_c = \Delta T_a + \frac{\dot{Q}^0}{\varepsilon^0 C_a^0} \left\{ \left(1 + \frac{\Delta \dot{Q}}{\dot{Q}^0} \right) \left[1 + \frac{\Delta(\varepsilon C_a)}{\varepsilon^0 C_a^0} \right]^{-1} - 1 \right\} \quad (4)$$

To simplify further by using Δ symbols and ignoring 2nd or higher order terms in above expression, the error equation of radiator top tank temperature may be expressed as:

$$\Delta T_c \approx \Delta T_a + \frac{\dot{Q}}{\varepsilon^0 C_a^0} \left(\frac{\Delta \dot{Q}}{\dot{Q}} - \frac{\Delta \varepsilon}{\varepsilon^0} - \frac{\Delta C_a}{C_a^0} \right) = \Delta T_a + (ITD)^0 \left(\frac{\Delta \dot{Q}}{\dot{Q}} - \frac{\Delta \varepsilon}{\varepsilon^0} - \frac{\Delta C_a}{C_a^0} \right) \quad (5)$$

While inside the above expression, top tank temperature will vary with: inlet air temperature variation, inlet coolant temperature and air temperature difference (or ITD), variation of heat rejection, relative coolant mass rated heat transfer capacity, and relative thermal effectiveness.

Thermal effectiveness of automotive radiators may be further impacted by many other factors such as: the design itself which is more related to Number of its heat Transfer Units (NTU), air side thermal capacity (C_a), and coolant side thermal capacity (C_c).

Linearizing the general expression for thermal effectiveness allows this to be shown as:

$$\Delta \varepsilon = \left(\frac{\partial \varepsilon}{\partial N_{tu}} \right)_{C_a, C_c} \Delta N_{tu} + \left(\frac{\partial \varepsilon}{\partial C_a} \right)_{C_c, N_{tu}} \Delta C_a + \left(\frac{\partial \varepsilon}{\partial C_c} \right)_{C_a, N_{tu}} \Delta C_c \quad (6)$$

Combining the above expression with the error ratio equation (5), the following equation can be shown as:

$$\frac{\Delta T_c}{T_c^0} \approx A \frac{\Delta T_a}{T_a^0} + B \frac{\Delta \dot{Q}}{\dot{Q}^0} - B_1 \frac{\Delta N_{tu}}{N_{tu}^0} - B_2 \frac{\Delta C_c}{C_c^0} - B_3 \frac{\Delta C_{pa}}{C_{pa}^0} - B_4 \frac{\Delta \rho_a}{\rho_a^0} - B_5 \frac{\Delta V_a}{V_a^0} \quad (7)$$

Where

$$A = \left(\frac{T_a^0}{T_c^0} \right), \quad B = \left(\frac{ITD^0}{T_c^0} \right),$$

$$B_1 = B \left(\frac{\partial \varepsilon}{\partial N_{tu}} \right)_{C_a, C_c} \frac{N_{tu}^0}{\varepsilon_0} \quad B_2 = B \left(\frac{\partial \varepsilon}{\partial C_c} \right)_{C_a, N_{tu}} \frac{C_c^0}{\varepsilon_0},$$

$$B_3 = B \left[1 + \left(\frac{\partial \varepsilon}{\partial C_a} \right)_{C_c, N_{tu}} \frac{C_a^0}{\varepsilon_0} \right]$$

EQUATION SIMPLIFICATIONS

Condenser Load

If the A/C heat load for the condenser is assumed to be constant for the vehicle test condition, and if air seals are present between condenser and radiator, then the radiator inlet air temperature term-1st one on the right side of equation (7) may be simplified as:

$$\frac{\Delta T_a}{T_a^0} \approx \frac{T_{amb}^0}{T_a^0} \frac{\Delta T_{amb}}{T_{amb}^0} - \frac{T_a^0 - T_{amb}^0}{T_a^0} \frac{\Delta V_{a-cond}}{V_{a-cond}^0} = a_1 \frac{\Delta T_{amb}}{T_{amb}^0} + a_2 \frac{\Delta V_a}{V_a^0} \quad (8)$$

Where

$$a_1 = \frac{T_{amb}^0}{T_a^0},$$

$$a_2 = -\left(1 - \frac{T_{amb}^0}{T_a^0}\right) \frac{\left(\frac{\Delta V_{a-cond}}{V_{a-cond}^0}\right)}{\left(\frac{\Delta V_a}{V_a^0}\right)} = -\left(1 - \frac{T_{amb}^0}{T_a^0}\right) \frac{V_{a-cond}^*}{V_a^*}$$

Incompressible Airflow

The airflow or volumetric flow term impact can be evaluated by its pressure forces balancing with engine cooling module behavior in an isothermal under-hood airflow condition if incompressible under-hood airflow behavior is assumed. As we understand in presentation [3] and [4], for an identical vehicle, the radiator face velocity of a particularly designed engine cooling module can be influenced by its resistance co-efficiencies, pressure gain of fan system, under-hood resistances, and vehicle driving speed impacts.

$$\frac{\Delta V_a}{V_a^0} = \frac{\Delta U_R}{U_R^0}$$

$$\Delta U_R = \left(\frac{\partial U_R}{\partial U_V}\right)_{k_{hex}, k_{fan}, k_{uh}} \Delta U_V + \left(\frac{\partial U_R}{\partial k_{hex}}\right)_{k_{fan}, U_V, k_{uh}} \Delta k_{hex}$$

$$+ \left(\frac{\partial U_R}{\partial k_{fan}}\right)_{k_{hex}, U_V, k_{uh}} \Delta k_{fan} + \left(\frac{\partial U_R}{\partial k_{uh}}\right)_{k_{hex}, k_{fan}, U_V} \Delta k_{uh} \quad (9)$$

Where

$$U_R^0 = \frac{V_a^0}{A_{R-face}}$$

General speaking, polynomial relationships for all terms in the right side of (9) will exist with variables, such as vehicle velocity and radiator core face area speed:

$$U_V = U_V(U_R, k_{hex}, k_{fan}, k_{uh}), k_{hex} = k_{hex}(U_R),$$

$$k_{fan} = k_{fan}(U_R), k_{uh} = k_{uh}(U_V, U_R)$$

And

$$\frac{\Delta V_a}{V_a^0} = \left(\frac{\partial U_R}{\partial U_V}\right)_{k_{hex}, k_{fan}, k_{uh}} \frac{\Delta U_V}{U_R^0} + \left(\frac{\partial U_R}{\partial k_{hex}}\right)_{k_{fan}, U_V, k_{uh}} \frac{\Delta k_{hex}}{U_R^0}$$

$$+ \left(\frac{\partial U_R}{\partial k_{fan}}\right)_{k_{hex}, U_V, k_{uh}} \frac{\Delta k_{fan}}{U_R^0} + \left(\frac{\partial U_R}{\partial k_{uh}}\right)_{k_{hex}, k_{fan}, U_V} \frac{\Delta k_{uh}}{U_R^0} \quad (10)$$

By summarizing all the above derivations, a more compact form of ratio of error for the radiator top tank temperature can be shown as following:

$$\frac{\Delta T_c}{T_c^0} \approx A_1 \frac{\Delta T_{amb}}{T_{amb}^0} + B \frac{\Delta Q}{Q^0} - B_1 \frac{\Delta N_{tu}}{N_{tu}^0} - B_2 \frac{\Delta C_c}{C_c^0}$$

$$- B_3 \frac{\Delta C_{pa}}{C_{pa}^0} - B_3 \frac{\Delta \rho_a}{\rho_a^0} - (B_3 + A_2) \frac{\Delta V_a}{V_a^0} \quad (11)$$

Where

$$A_1 = Aa_1 = \left(\frac{T_a^0}{T_c^0}\right) \frac{T_{amb}^0}{T_a^0},$$

$$A_2 = -Aa_2 = \left(\frac{T_a^0}{T_c^0}\right) \left(1 - \frac{T_{amb}^0}{T_a^0}\right) \frac{V_{a-cond}^*}{V_a^*},$$

In practical application, such as vehicle engine cooling road test, some terms on the right side only have secondary or minor influences on the top tank temperature variation of a radiator. In this case, an approximation from of the equation may be written as:

$$\frac{\Delta T_c}{T_c^0} \approx A_1 \frac{\Delta T_{amb}}{T_{amb}^0} + B \frac{\Delta Q}{Q^0} - B_1 \frac{\Delta N_{tu}}{N_{tu}^0} - B_2 \frac{\Delta C_c}{C_c^0} - (B_3 + A_2) \frac{\Delta V_a}{V_a^0} \quad (12)$$

And its non-dimensional form can be expressed as:

$$\Delta T_c^* \approx A_1 \Delta T_{amb}^* + B \Delta Q^* - B_1 \Delta N_{tu}^* - B_2 \Delta C_c^* - (B_3 + A_2) \Delta V_a^* \quad (13)$$