

Design for Six Sigma (DFSS) for Optimization of Automotive Heat Exchanger and Underhood Air Temperature

Alaa El-Sharkawy, Asif Salahuddin, and Brian Komarisky
Chrysler Group LLC

ABSTRACT

In this paper a design methodology for automotive heat exchangers has been applied which brings robustness into the design process and helps to optimize the design goals: as to maintain an optimal coolant temperature and to limit the vehicle underhood air temperature within a tolerable limit. The most influential design factors for the heat exchangers which affect the goals have been identified with that process. The paper summarizes the optimization steps necessary to meet the optimal functional goals for the vehicle as mentioned above. Taguchi's [1] Design for Six Sigma (DFSS) methods have been employed to conduct this analysis in a robust way.

CITATION: El-Sharkawy, A., Salahuddin, A., and Komarisky, B., "Design for Six Sigma (DFSS) for Optimization of Automotive Heat Exchanger and Underhood Air Temperature," *SAE Int. J. Mater. Manf.* 7(2):2014, doi:10.4271/2014-01-0729.

INTRODUCTION

It is generally understood that, the goal of an automotive heat exchanger (namely the radiator) is to reject the engine heat in the most efficient way possible. In an automotive radiator, the hot fluid is the engine coolant and the cold fluid is the ambient/rams air that enters through the front fascia of the vehicle. To fulfill this goal of rejecting engine heat in an efficient way and as well as to help in achieving desired fuel economy from the car, it is required to design a radiator to maintain engine coolant temperature at an optimal predetermined limit under variable ambient temperatures. But, the issue that is often poorly emphasized is the necessity to keep the radiator out air temperature (referred to as fan blast air temperature also) under a certain limit to protect the under hood components from thermal stress. Therefore, the investigation in this paper focuses on the optimized design of an automotive radiator which can meet the basic requirement of cooling the engine, operate at an optimal fuel economy enabling coolant temperature and as well as maintain the underhood air temperature below a limit that is acceptable for sustaining durability and performance of underhood components.

BASIC HEAT EXCHANGER (RADIATOR) DESIGN

Tentative major geometric parameters of design interest are the water tubes and fins. The Mathematical equations for a basic heat exchanger design are described in the next section. An in-house computer FORTRAN code is used which captures this design process.

DESIGN EQUATIONS

The design equations, used in this work, are based on the heat exchanger performance calculations by Kays and London [2].

The rate of heat transfer for a heat exchanger is given by:

$$Q = C_{\min} (ETD) \epsilon \quad (1)$$

Where,

Q , is the rate of heat transfer

ETD , is the "Entering Temperature Difference", defined by:

$$ETD = T_{c_{in}} - T_{air_{in}} \quad (2)$$

Where,

$T_{c_{in}}$: is the entering temperature for engine coolant which is considered as the hot fluid, and

$T_{air_{in}}$: is the entering temperature for air which is considered as the hot fluid.

ϵ : is the effectiveness as defined by Kays and London [2] for a cross flow heat exchanger. The cooling air is considered as the mixed fluid while the engine coolant is the unmixed fluid.

The minimum and maximum mass capacity ratio, Z , is defined as follows:

$$Z = C_{\min} / C_{\max} \quad (3)$$

Where $C_{\min}$ and $C_{\max}$ are, respectively, the smaller and the larger of the two magnitudes C_{air} and C_c .

Where

$$C_{air} = \dot{m}_{air} C_{p_{air}} \quad (4)$$

$$C_c = \dot{m}_c C_{p_c} \quad (5)$$

$\dot{m}_{air}$: is the mass flow rate of air.

$C_{p_{air}}$: is the average specific heat of air

$\dot{m}_c$: is the mass flow rate of engine coolant.

C_{p_c} : is the average specific heat of coolant.

The number of heat transfer units, NTU , is calculated from:

$$NTU = (U_o A_o) / C_{\min} \quad (6)$$

The overall heat transfer coefficient U_o is given by:

$$1/(U_o A_o) = 1/(h_c A_c) + 1/(\eta_{ov} h_{air} A_o) + x_w / (k_w A_w) \quad (7)$$

A_o : is the air side total heat transfer area including fin surface area. The fin area is based on the fin density, fin dimension, number of tubes and tube length. In the above equation, the only term which is a function of air density is air-side heat transfer coefficient h_{air} .

x_w : is the tube wall thickness

k_w : is the tube wall thermal conductivity

A_w : is the conduction heat transfer area through the tube wall

Heat Transfer Coefficients

For the air side heat transfer coefficient, the following equation for Nusselt number (Holman, 1981) [3] is used:

$$Nu_{air} = 0.25 Re_{air}^{0.6} Pr_{air}^{0.38} \quad (8)$$

Re_{air} : is the Reynolds number given by the following equation:

$$Re_{air} = (\dot{m}_{air} / A_{\min})(d_{eq_{air}} / \mu_{air}) \quad (9)$$

And $\dot{m}_{air}$ is the air mass flow rate given by the following equation:

$$\dot{m}_{air} = \dot{V}_{air} \rho_{air} \quad (10)$$

Where $\dot{V}_{air}$ and ρ_{air} are the average volumetric air flow rate and average air density respectively.

$A_{\min}$: is the minimum free flow area for air

Pr_{air} : is the Prandtl number defined by the following equation:

$$Pr_{air} = C_{p_{air}} \mu_{air} / k_{air} \quad (11)$$

Where μ_{air} and k_{air} are the average viscosity and average thermal conductivity of air respectively.

For the coolant side, for fully developed flow inside tubes, the following equations are used (Perry & Chilton, 1973) [4]

$$Nu_c = 0.023 Re_c^{0.8} Pr_c^{1/3} \quad (12)$$

$$Nu_c = h_c d_{eq_c} / k_c \quad (13)$$

$$Re_c = (\dot{m}_c / A_c)(d_{eq_c} / \mu_c) \quad (14)$$

For transient flow regime:

$$Nu_c = 0.116(Re_c^{2/3} - 125) Pr_c^{1/3} [1 + (d_{eq_c} / L)^{2/3}] \quad (15)$$

For $d_{eq} \ll L$:

$$Nu_c = 0.116(Re_c^{2/3} - 125) Pr_c^{1/3} \quad (16)$$

The following equation, developed by Seider and Tate (1936) [5], applies for laminar flow:

$$Nu_c = 1.86(Re_c)(d_{eq_c} / L)^{1/3} \quad (17)$$