DESIGN IMPROVEMENTS OF ENGINE COOLING SYSTEM USING CFD AND 1D THERMO-FLUID MODEL: MEDIUM DRIVING SPEED AND KEYED-OFF CONDITIONS

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ABSTRACT
Underhood air flow when approaching radiator is highly non-uniform, thus 3D computational fluid dynamics is required to study the dynamic pattern of cooling air flow at vehicles’ underhood. The velocity field, total pressure field and temperature distribution at underhood are obtained and analyzed. Uniform air flow approaching radiator could increase radiator heat transfer. While coolant circuit consists of numerous components, 1D thermal-fluid simulation is utilized to study temperature behavior of the coolant. Automotive designers should design a robust engine cooling system which is working well in both normal and severe driving conditions. In this paper, the alternatives to improve engine cooling system during medium driving and keyed-off condition are described. CFD air flows under different driving conditions are observed and the effects of air flow distribution on radiator heat dissipation are studied. In order to reduce coolant temperature during keyed-off condition, the practical modification plans to include electrical water pump as main and auxiliary pump are also briefed.

Keywords: Engine cooling system; CFD; thermo-fluid

1.0 INTRODUCTION
In evaluation of the performance of vehicle heat exchangers, it requires 2 main portions of job, the study of cooling air aerodynamic and heat exchanger system analysis. Since the cooling air flow will pass through the front bumper, grille, other heat exchangers, the velocity distribution at the face of radiator is highly non-uniform, especially in low speed driving.

Van Zyl (2006) developed a CFD simulation using StarCD, for the cooling air flow. The boundary conditions for 2 scenarios moving and stationary were experimental determined; the boundary conditions are the surface temperatures of heat sources (i.e. engine, reservoir, gearbox, exhaust manifold and heat exchanger). The numerical models were validated with experiment at wind tunnel. Ecer et al. (1995) calculated the velocity over the radiator and air- to-boil temperature by CFD using PASSAGE. Ning (2009) utilized AMESim as a system analysis for engine cooling system. Eichlseder et al. (1997) simulated using KULI for both cooling air circuits and coolant circuit. The flow models are based on network flow theory, which permits the construction of complex cooling circuits.

The methodology to couple the 3D CFD (Fluent, StarCD, Vectis) with 1D thermal fluid system model (KULI, Flowmaster, AMESim) is fairly interesting. 3D CFD could reflect the non-uniform velocity distribution of cooling air. While the 1D system model is utilized as a performance calculation for the heat exchangers systems, either condenser for air conditioning system or radiator for engine cooling system. Basically, it involves a process which the models exchange boundary conditions until they are converge with each other. 3D CFD will feed the 1D thermal-fluid model with boundary conditions, such as convection coefficient, velocity and fluid temperature. On the other hand, 1D thermal fluid model will feed CFD with boundary conditions, such as heat rejection to air.

automobile air conditioning analysis. Reitbauer and Hager (2000) described a procedure to couple CFD with KULI. He emphasized 1D thermo hydraulic model gives the possibility to analyze split flow patterns or non-uniform air velocity distribution. In the 1D model (KULI), the heat exchanger will be modeled in such a way that it is partitioned into a number of rectangular segments. In each segment, a fictive flow resistance will be determined according to CFD distribution. All the formers illustrated above are 3D CFD models while the latter are 1D system models. After the coupling, a fruitful and complete cooling/air conditioning system analysis could be performed with different scenarios.

2.0 METHODOLOGY
As air flow at underhood is highly complex and dynamic, a 3D CFD model was required to analyse underhood air flow pattern. The air which entered underhood passed through varied components like grille, frontal opening, heat exchangers, fan, should, exhaust and engine body. In CFD model, porous media method was used to define porosity of heat exchangers. Viscous resistance and inertia resistance were derived from characteristic chart of pressure drop and flow rates, as in Equation (1), with linear regression. Radiator cores were modelled as dual stream heat exchanger where two types of fluids exchange heat with each other. Condenser was modelled as a single stream heat exchanger where only one type of fluid is participating. Fan was modelled as a rotating reference frame, as actual fan blade drawing was available. As fan rotates, it can provide momentum to underhood air flow, especially during low speed driving. Heat exchangers’ temperature (condenser and radiator) and solid which provides constant heat flux (engine body and exhaust pipe) will affect underhood thermal environment.

\[ \Delta P = C U^2 + D U \]  

(1)

From the CFD underhood air flow model, we can obtain radiator air flow rates at different driving speed, at different fan rotation speed or both. By coupling of CFD air flow input and industry data of engine heat and coolant flow rates at different driving speed and conditions, transient coolant temperature (especially after keyed-off) was studied in 1D thermo-fluid simulation. As it was a transient simulation, heat flow, coolant flow and air flow varied with time horizon. It can be observed how these factors are interacting with each other and affecting coolant temperature (time behaviour). When the air flow rates and coolant flow rates are at their minimum, it is defined as keyed-off time. Keyed-off time is fixed at 4500s. The integration of one dimensional and three dimensional models are best illustrated by Figure 1.

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Fig. 1 Integration of 1D thermo-fluid model and 3D CFD model
3.0 RESULTS

3.1 CFD Air Flow under Different Driving Conditions

Figures 2(a) and 2(b) plotted air velocity contour for vehicle at 110kph driving, side and top cross sectional view. It could be observed how the ram air flows from outside into underhood. Furthermore, it showed how the air moves inside underhood. It can be determined which spot with accelerated air and which spot with stagnant air. The air velocity is accelerated at the rear side of radiator and exit the underhood at beneath. While Figures 2(c) and 2(d) plotted air temperature contour for vehicle at 110kph driving, side and top cross sectional view. It could be observed that the air temperature is higher at stagnant flow location and after passed through heat source (i.e. heat exchanger).

Fig. 2 Flow field and scalar field of vehicle underhood at 110kph driving

3.2 Effect of Air Flow Distribution on Radiator Heat Dissipation

In CFD model, the contour result of air flow distribution (at frontal surface of heat exchangers) driven by ram air and cooling fan are analyzed. The air flow driven by ram air (due to vehicle motion) is always affected by the position of components/ parts in front of heat exchangers, Figure 4 (a). The bumper, front end opening and mechanical structure are obstacles to ram air flow from front side. The ram air passed through these components will create adverse pressure and thus flow separation. While the position of frontal components/ parts pose less impact onto air flow driven by cooling fan, Figure 4 (b). The air flow is driven by the suction of fan from rear side of radiator.

Similarly, Figures 3(a) and 3(b) plotted air velocity contour for idling vehicle with 6000rpm fan, side and top cross sectional view. Since the air flow start with a low pressure area created by fan suction, the air flow initiated inside underhood. When pressure difference exists, the air started to flow from high pressure area to low pressure area (pressure drop). Also, a pressure jump could be observed at cooling fan. The pressure drop is balanced off by pressure jump at cooling fan. Figures 3(c) and 3(d) displayed temperature map of underhood for idling vehicle with 6000rpm fan, side and top cross sectional view. It could be analyzed that the underhood temperature is high at all of the enclosed area. This is attributed to air flow at underhood is limited and the heat emitted by heat exchangers cannot be carried away fast enough. The air flow also recirculates to frontal part of underhood; as a result the air temperature there is higher too.
Though total air flow rates through the radiator is similar, however total heat dissipated at radiator is varied for ram air and cooling fan scenarios. This phenomenon can be explained by the reason which highly uniform air flow distribution (in ram air scenario) participated most of the radiator frontal surface into heat dissipation. While cooling fan accelerates the air to only a restricted area of radiator, the remaining frontal area of radiator does not participate in heat dissipation. This has underutilized full potential of the radiator and resulted lower radiator heat dissipation.

Figure 5 summarized the heat dissipated at radiator by ram air and cooling fan over varied total air flow rates. It could be observed that heat dissipated by ram air is around 70% higher than heat dissipated by cooling fan.
3.3 Prolonging Fan and Pump Operation.

Figure 6 (a) shown the base case, it could be observed that temperature of coolant increased to 120˚C after sudden keyed-off. This was because sudden fan and pump stoppage do not dissipate soaking heat in coolant circuit. In Figure 6(b), it could be observed that coolant temperature improve to 110 °C by prolong fan and pump for 1.5mins. In Figures 6(c) and 6(d), it shown coolant temperature reduces to 105°C and 101°C by prolong fan and pump for 3.0mins and 4.5mins.

Besides, if we merely prolong either fan or pump, it does not improve the situation significantly. Pump ensures coolant flow from engine to radiator, while fan promotes air flow through radiator. If solely prolong pump operation, heat transfer rate at radiator remains low with little air flow. If solely prolong fan operation, soaking heat at engine could not be transferred to radiator with little coolant flow rate.

Fig. 5 Comparison between heat dissipated by ram air and cooling fan

Fig. 6 Improvement in coolant temperature with prolonging of fan and water pump
4.0 DISCUSSION

4.1 Implementation of Electrical Water Pump as a Main Pump.

With this option, mechanical water pump is removed and its housing is kept. Electrical Water Pump (EWP) which weighted 900grams is mounted to radiator bottom hoses. Operation of water pump and electrical fan will be controlled by digital controller. We are allowed to set the target temperature (75, 80, 85 or 90) for digital controller. Digital controller will read signal from thermal sensor resides at radiator upper hoses. Once target temperature is reached, digital controller will give signal to electrical water pump to step in. The electrical fan will step in at target temperature plus 3°C. Conventional belt-driven water pump will sap power from engine (8-10kW, ~5%). By installing electrical water pump, could eliminate this parasite power (as parasite power is equivalent to cubed engine speed). Electrical water pump consumes power at the range of 36W-120W only. Conventional mechanical water pump run at a speed proportional to engine speed, while electrical pump run at a desired speed proportional to coolant temperature. In this option, thermostat and bypass circuit is removed. The EWP is wired directly to battery, so that it could continue to run after keyed-off (normally 2 minutes). Figure 7 and Figure 8 displayed modification of current circuit to replace mechanical pump with EWP as main pump.

4.2 Implementation of Electrical Water Pump as an Auxiliary Pump.

In this option, there are no major changes but an additional electrical water pump and thermal switch. The EWP is acting as an auxiliary pump which assisting main mechanical pump. Temperature bulb is placed inside radiator upper hoses. Temperature bulb will trigger thermal switch and turn on electrical water pump. Coolant temperature will decide the operation of electrical water pump. It may be observed that there are two thermal switches, one for electrical fan and one electrical water pump. This provides flexibility to control fan with fin air temperature and to control water pump with coolant temperature. But, this option does not remove parasite power of mechanical pump towards engine. The EWP is wired to battery in order to ensure its operation after keyed-off. Fig. 9 showed schematic drawing of adding a EWP as auxiliary pump. Secondly, EWP consumes power in the range of 36W-120W. While release belt-driven mechanical pump, it could save us another 8-10kW (5%) of power and torque.

Fig. 7 Installation of electrical water pump as main pump
5.0 CONCLUSION
In this study, half body car geometry is utilized to simulate the air flow pattern at underhood under different driving conditions. The CFD air flow pattern varied when it is driven by ram air and cooling fan. The air flow distribution at heat exchanger is more uniform for air driven by vehicle motion. Thus radiator heat dissipation for those cases driven by ram air is also higher. Later, the air flow rates obtained from CFD is fed into one dimensional transient coolant temperature simulation. Temperature result obtained from one dimensional model mimicked to industry result. However, the numerical model is further tested for other scenarios with prolonged both fan and pump operation. The numerical output is satisfying and awaiting to be validated by experiment result. Physical implementation and benefits obtained from electrical water pump is discussed in detail. Implementation of electrical water pump can eliminate the “parasite power” sapped by mechanical pump and provide higher torque to vehicle motion.

REFERENCE


