Thermal Flow Analysis of Vehicle Engine Cooling System

Kyoung Suk Park*
Department of Mechanical Engineering, Kyung Hee University, Kyunggi 449-701, Korea
Jong Phil Won, Hyung Seok Heo
Korea Automotive Technology Institute (KATECH), Chungnam 330-912, Korea

This paper deals with theoretical model developed for analyzing the heat transfer of automotive cooling systems. The model has a modular structure which links various cooling system submodels. From the model, heat transfer rate of automotive cooling systems can be predicted, providing useful information at the early stages of the design and development. The aim of the study is to develop a simulation program for automotive cooling system analysis and a performance analysis program for analyzing heat exchanger. Heat release rate from combustion gas to coolant through the cylinder wall in engine cylinder was analyzed by using an engine cycle simulation program. In this paper, details of each submodel are described together with the overall structure of the vehicle model.

Key Words: Modular Structure, Cooling System, Heat Exchanger, Computer Simulation Program

Nomenclature

A	: Surface area (m^2)
B	: Cylinder bore (m)
C_1, C_2	: Constant
C_3, C_4	: Constant
C_{bl}	: Blowby constant (1/s)
C_p	: Specific heat (J/kg·K)
C_r	: Flow stream capacity-rate ratio, C_{min}/C_{max}
D_e	: Equivalent Diameter (m)
$\dot{E}$	: Energy (W)
h	: Enthalpy (J/kg)
h_c	: Convection heat transfer coefficient ($W/m^2 \cdot K$)
i	: Overall reduction ratio
k	: Thermal conductivity (W/m·K)
l	: Connecting rod length (m)
m, m_1	: Mass (kg)
$\dot{m}$	: Mass flow rate (kg/s)
N	: Engine speed (rpm)
NTU	: Number of Transfer Units, UA/C_{min}
P	: Pressure (bar)
P_m	: Motored cylinder pressure (bar)
$\dot{Q}$	: Heat transfer rate (W)
r_c	: Compression ratio
r_D	: Effective radius of tire (m)
S	: Stroke (m)
$\bar{S}_p$	: Mean piston speed (m/s)
T	: Temperature (K)
U	: Overall heat transfer coefficient ($W/m^2 \cdot K$)
V	: Volume (m^3)
V_{af}	: Frontal air velocity (m/s)
V_d	: Displaced cylinder volume (m^3)
V_v	: Vehicle speed (km/h)
$\dot{W}$	: The work transfer rate out of the system across boundary (W)
w	: Average cylinder gas velocity (m/s)
x_b	: Mass fraction burned

Greek symbols

ε	: Effectiveness
δ	: Thickness (m)

* Corresponding Author,
E-mail: ks2507@khu.ac.kr
TEL: +82-31-201-2507; **FAX**: +82-31-202-8106
Department of Mechanical Engineering, Kyung Hee University, Kyunggi 449-701, Korea. (Manuscript **Received** August 9, 2001; **Revised** March 27, 2002)

η_o : Overall surface fin efficiency
θ : Crank Angle (deg.)
θ_1 : Start of compression (deg.)
θ_s : Start of heat release (deg.)
$\Delta\theta_b$: Burn crank angle (deg.)

Subscripts
a : Air
b : Burned gas
c : Coolant or convection
i : Tube inside of the heat exchanger
m : Mean or average
o : Tube outlet condition
p : Tube side
rad : Radiator
u : Unburned gas
w : Wall

1. Introduction

Within the automotive industry, the area of vehicle design is moving rapidly into the use of new, high technology techniques. The purpose of a motor vehicle cooling system is to ensure that the engine is maintained at its most efficient practical operating temperature. The current trend in car engine design is towards smaller more efficient engines, but this results in less waste energy being available for heating purposes and passenger comfort. Current high-efficiency engine systems create hot engine compartments, hot exhausts, hot lubricating oil, but poor heat output to the passenger compartment, at least from cold starting conditions. There is a need to look at the total heat balance and control system for the vehicle in order to search for performance optimization and cost saving. Consequently, a more detailed analysis is required by the designer to optimize the available heat distribution resources. To this end, computer simulation provided the means to optimize cooling circuit configurations and employ sensitivity analysis to components.

It will therefore be very useful to the development engineer, if a computer model can be used not only to support but also reduce the amount of testing required during development work. Such a model can simulate the thermal behavior of the engine cooling system and its components and predict the effects of various design and operating parameters. The use of computer simulation can greatly enhance development effort by predicting performance trends and trade-offs and will, therefore, result in more efficient and better optimized heating and cooling systems for the new generation of vehicles.

The main objective of this paper was to develop a computer design program which could be used by the development engineer to analyze thermofluid performance of the vehicle cooling systems. The model can be predicted engine cooling system temperatures and analyzed the effects of design parameters and operating conditions on the vehicle thermal performance. Detailed thermal analysis of the system and its components can be obtained by predicting coolant temperatures and heat transfer rates in the cooling circuits and across boundaries with other components.

2. Description of the Model

In order to analyze the thermofluid characteristics of the vehicle cooling system and heat transfer and fluid flow circuit, the modeling work was carried out at component and overall system. A complete model required a submodel for each of the main components that could be integrated into an overall model representing the entire vehicle thermal system. The component modeling involved the development of a number of submodels representing the major components of the engine cooling system together with their respective heat balance and fluid equations. The overall system model integrates the submodels of the components into a thermal system to predict the system performance.

Simulation submodels were developed for each component: this was based on a mathematical representation of their heat and fluid flows. This paper describes vehicle cooling system heat transfer model and its application to vehicle development programs. Earlier work involving the development of the vehicle cooling model has already been reported by several papers (Sidders et al., 1997 ; Sakai et al., 1994 ; Eichiseder et al., 1993 ; Veshagh et al., 1993 ; Assanis et al., 1986 ; Xu et al., 1984 ; Chiang et al., 1982 ; Rising, 1977).

The overall model is based on 1500cc passenger car incorporating all the major heat-transferring components such as the engine, cooling system, radiator and thermostat. The structure of the model is based on a modular arrangement of individual submodels simulating the performance of all heat-transferring components. Thermal performance of the overall system can be simulated by solving a set of differential equations representing the system mass flow continuity and energy conservation.

2.1 Vehicle model

The vehicle model integrates all the submodels described above into an overall system model representing the whole vehicle. At the core of the model are the mass and energy flow iteration loops which are repeatedly updated until a steady-state solution is reached. The main iteration loop for steady-state calculations then proceeds to determine whether any imbalance exists in the energy conservation and mass flow continuity. For each iteration, the new values of the unknown variables are recalculated to reduce the error. These calculation are repeated until all the energy and mass flow compatibility criteria are satisfied within a specified error.

A typical vehicle cooling system consists of a number of components and several fluid flow circuits as shown schematically in Fig. 1. As can be seen, various components are linked via mass flow and energy links depending on their function and performance characteristics. The whole vehicle model is made up from many individual component models, linked together such the output results from one individual model from the input boundary conditions for another.

The vehicle speed is given by:

$$V_v = \frac{2\pi \cdot r_D}{1000} \cdot \frac{60 \cdot N}{i} \tag{1}$$

2.2 Engine model

The engine model has been developed to provide a representation of the thermal characteristics. The engine model is based on a set of differential equations defining the conservation of energy and mass in a control volume representing the cylinder (Heywood, 1988). These equations

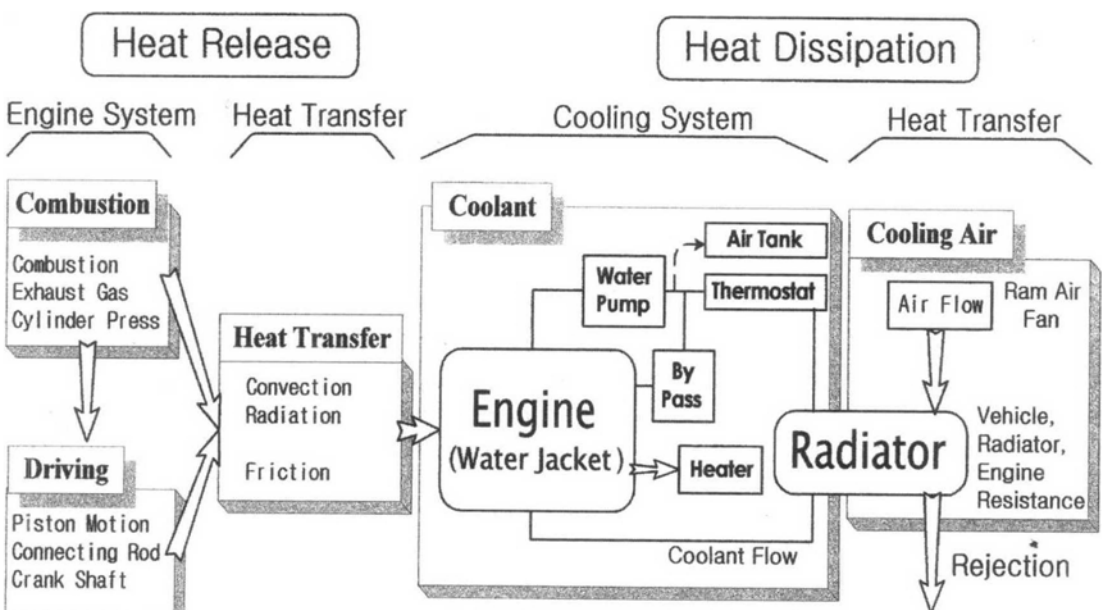

Fig. 1 Automotive engie cooling systems

are solved to calculate cycle temperature and pressure using 2-zone cycle simulation program. The combustion process is represented by a heat release process. The cycle simulation is carried out using the basic engine design parameters as input data and the iterative calculations are continued until a stable condition is achieved.

For a control volume encasing the cylinder contents, the energy equation is (Heywood, 1988; Blumberg et al, 1983):

$$\dot{E}=\dot{Q}-\dot{W}+\sum\dot{m}\cdot h \tag{2}$$

where $\dot{Q}$ and $\dot{W}$ are the total heat transfer rate into the system across the boundary, and the work transfer rate out of the system across the boundary respectively.

The heat transfer to the combustion chamber surfaces in contact with the burned and unburned gas zone is given by

$$\dot{Q}_b=h_{c,b}A_b(T_b-T_w) \tag{3}$$
$$\dot{Q}_u=h_{c,u}A_u(T_u-T_w) \tag{4}$$

where A_b and A_u the areas of burned and unburned gas in contact with the cylinder walls at temperature T_w. The fraction of cylinder area contacted by burned gas is assumed to be proportional to the square root of the mass fraction burned.

$$A_b=\left(\frac{\pi B^2}{2}+\frac{4V}{B}\right)x_b^{1/2} \tag{5}$$
$$A_u=\left(\frac{\pi B^2}{2}+\frac{4V}{B}\right)(1-x_b^{1/2}) \tag{6}$$

The mass at any crank angle is

$$m=m_1\exp[-C_{bl}(\theta-\theta_1)/\omega)] \tag{7}$$

The cylinder volume is calculated from the compression ratio r_c, volume at tdc V_0, and $a=S/2l$ to be

$$V=V_0\left[\left[1+\frac{r_c-1}{2}\left\{1-\cos\theta+\frac{1}{a}[1-(1-a^2\sin^2\theta)^{1/2}]\right\}\right]\right] \tag{8}$$

The mass fraction burned is calculated from the burning law (Bumberg and Kummer, 1971)

$$x_b=0.5\{1-\cos[(\theta-\theta_s)\pi/\Delta\theta_b]\} \tag{9}$$

The heat transfer coefficient on the gas side is based on the correlation provided by as follows (Woschni, 1967, 1982):

$$h_c=110B^{-0.2}P^{0.8}T^{-0.53}w^{0.8} \tag{10}$$
$$w=\left[C_1\bar{S}_p+C_2\frac{V_dT_r}{P_rV_r}(P-P_m)\right] \tag{11}$$

For spark-ignition engine with swirl, cylinder averaged gas velocities w(m/s) were given by Woschni's correlation. Spark-ignition engine tests showed that the average cylinder gas velocities gave acceptable predictions for this type of engine (Woschni, 1982).

2.3 Cooling system model

This model links the submodels of the engine, radiator and thermostat and calculates heat flow rates, heat fluxes and temperatures for each component of the vehicle cooling system. The heat losses in the hose of the cooling system are assumed to be negligible and therefore the coolant temperature is calculated on the basis of the heat absorbed from the engine and heat dissipated by the radiator.

The energy equation is used iteratively to obtain a set of thermodynamic conditions which satisfy the overall heat balance between the heat absorption and dissipation from the engine and radiator. A numerical solution is employed to adjust the value of coolant temperature repeatedly until an equilibrium condition is found (Press et al., 1982). The iteration starts with an assumed coolant temperature from which the heat flows into and out of the coolant circuit can be calculated. The overall heat balance is then checked: if there is any imbalance in the energy equation, the coolant temperature is corrected and the loop is repeated until a heat balance is reached.

In these calculation it is assumed that the thermostat lift is proportional to the difference between the coolant flow temperature and the thermostat activation temperature. Figure 2 shows the actual characteristic of the thermostat. The speed ratio between the engine and the coolant pump is assumed to be a fixed ratio which is given

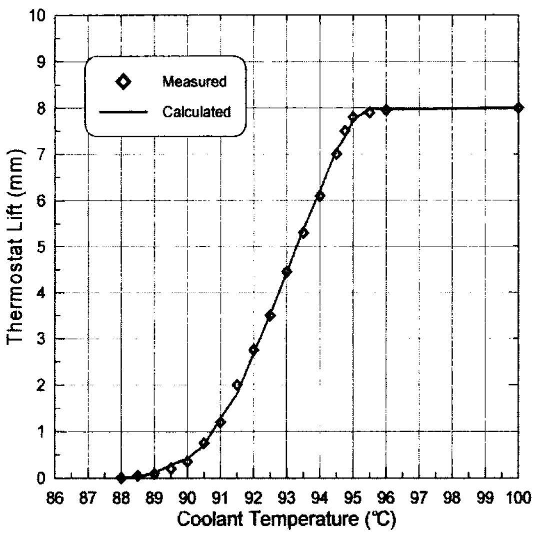

Fig. 2 Thermostat characteristics

by the manufacture.

2.4 Heat exchanger model

The heat exchanger model is based on the radiator heat dissipation characteristics and empirical correlations to calculate heat transfer coefficients on the air and coolant sides. The heat exchanger model calculates radiator heat transfer based on the effectiveness-NTU method described by Kays and London (1984).

This model assumes that the radiator is split up into a number of rectangular subsections and calculates the heat dissipation for each segment. The coolant outlet temperature for each element is calculated and integrated over the whole face area of the radiator. As a result, the model has the capability to deal with partially overlapping heat exchangers in a multiplayer radiator pack and with nonuniform flow velocity distribution. Cooling air flow is calculated as a function of vehicle speed. The incoming air velocity is supplied by an adjoining model which accounts for vehicle speed and cooling fan operation.

$$V_{af}=C_3V_v+C_4 \tag{12}$$

where C_3 and C_4 are constant which are determined from vehicle tests.

The rate of heat transfer is calculated using the radiator model as follows:

$$(\dot{m}C_p\Delta T)_{Rad}=(\dot{m}C_p\Delta T)_c-(\dot{m}C_p\Delta T)_a \tag{13}$$

$$(\dot{m}C_p\Delta T)_c=\varepsilon C_{\min}(\Delta T)_{\max} \tag{14}$$

where,

$$\varepsilon=1-\exp\left\{\frac{1}{C_r}(NTU)^{0.22}\cdot\left[\exp(-C_r(NTU)^{0.78})-1\right]\right\} \tag{15}$$

The overall heat transfer coefficient is given by

$$UA=\frac{1}{\dfrac{1}{h_{c,i}A_i}+\dfrac{\delta_p}{k_pA_m}+\dfrac{1}{\eta_o h_a A_o}} \tag{16}$$

The heat transfer coefficient on the coolant side is calculated from the forced convection heat transfer correlation provided by Dittus-Boelter (1992) and Hausen (1943). The heat transfer on the air side is based on the formular developed by Chang and Wang (1997) for louvered fin.

3. Analysis Program of Vehicle Cooling System

Figure 3 shows an analysis program flow chart of vehicle cooling system. Form the following diagram we know that the analysis program of cooling system is organized as analyzing cooling system by inputting conditions of these following; operating conditions of motor vehicle, engine geometry, ambient conditions, radiator geometry and thermostat characteristics. When the related gear of vehicle speed is decided, it is then organized as analyzing cooling system; by calculating engine conditions and then assuming initial coolant temperature and flow conditions.

Next, the coolant temperature which recirculates to and from the engine is calculated by the cooling system model, also with thermostat opening degree and coolant flow rate this program is organized to have a solution by recalculating until these two subtracted values which are the assuming value of cooling model and the calculated coolant temperature by engine model, comes in the tolerance. The details of the cases studies are as shown in Table 1.

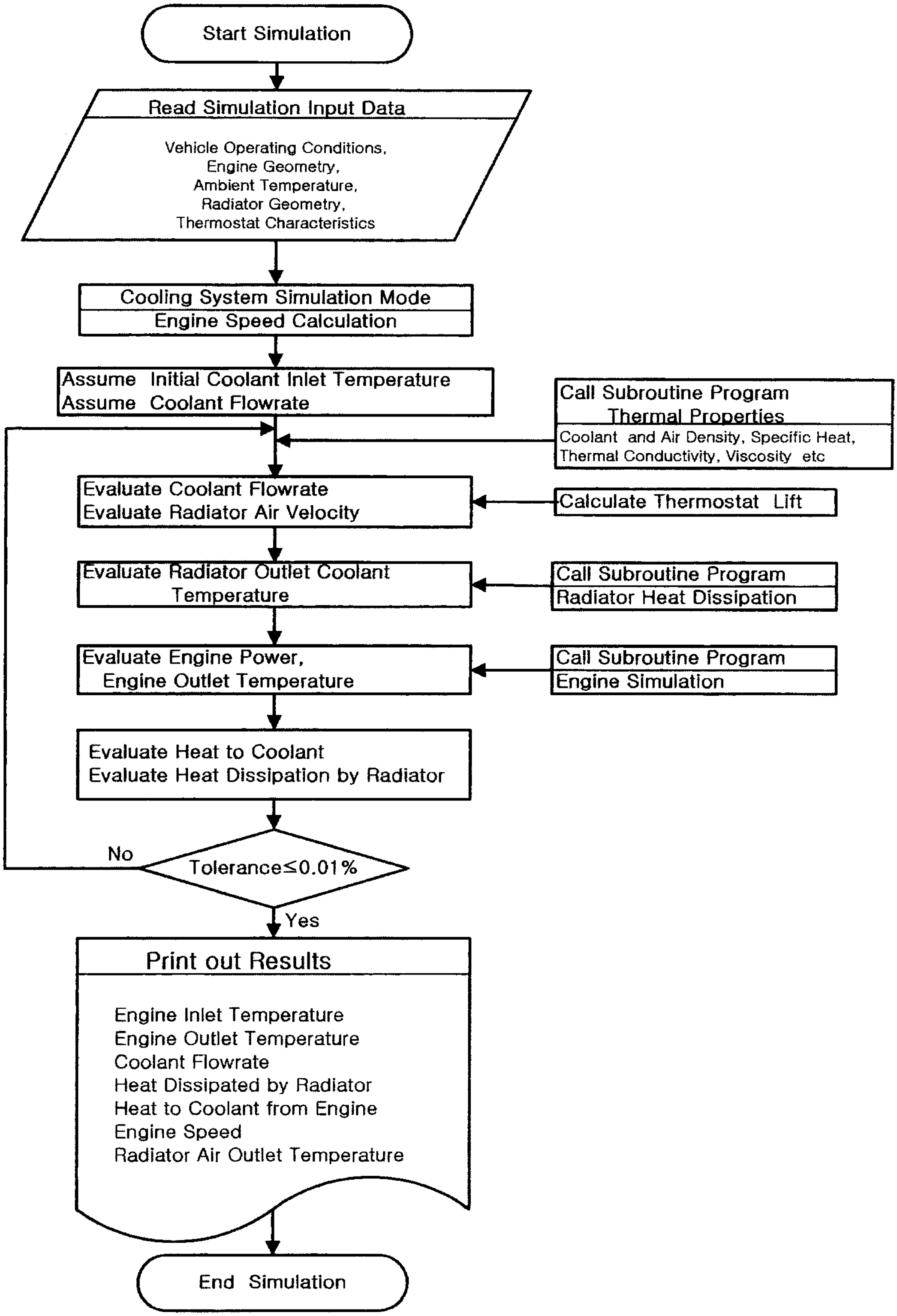

Fig. 3 Flow chart of vehicle engine cooling system simulation program

Table 1 Simulation input data

Vehicle speed	10~140km/h
Ambient temperature	35℃
Thermostat open temperature	88℃
Thermostat max. open temperature	100℃
Thermostat opening size	8mm
Height of radiator	334.5mm
Width of radiator	603mm
No. of radiator tube rows	54
Major diameter of radiator tube	2mm
Radiator tube depth	23.5mm
Radiator tube thickness	0.2mm
Radiator tube pitch	11mm
Radiator fin height	9mm
Radiator fin pitch	1.5mm
Water pump speed ratio	1.16 : 1
Gear position.	1, 2, 3, 4, 5

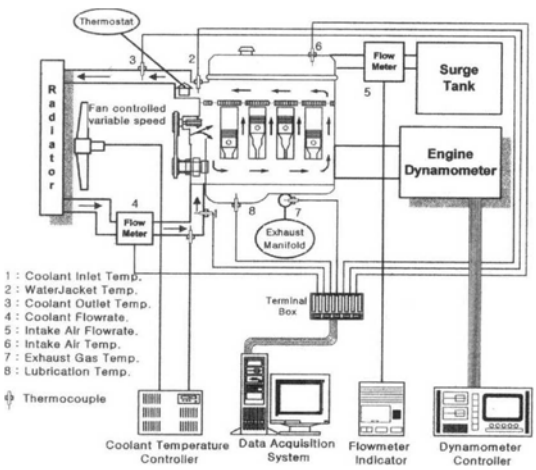

Fig. 4 Schematic diagram of cooling system experimental apparatus

Table 2 Specification of experimental engine

Type			Water cooled 4 cycle 4 cylinder gasoline engine
Displacement (cc)			1,495
Bore×Stroke (mm)			75.5×83.5
Compression ratio			10.0
Valve clearance	Intake (mm)		0.18
	Exhaust (mm)		0.24
Ignition order			1-3-4-2
No. of cylinder			4
Position of valve			Over head valve with single camshaft 3 valves per cylinder
Valve timing	Intake	Open	BTDC 14°
		Close	ABDC 42°
	Exhaust	Open	BBDC 52°
		Close	ATDC 8°
Compression pressure			13.24 bar
Spark timing			BTDC 10°±1°/ 800rpm

4. Experiments

In order to check the validity of model predictions a series of tests was carried out on a radiator wind tunnel and engine dynamometer. Coolant flow rates and coolant temperatures were measured for a series engine speed and full load conditions.

A schematic diagram of the cooling system used for the experiments is shown in Fig. 4. The test experiment system are consist of engine, dynamometer, temperature controller and flow meter as shown in the figure. Table 2 shows the specification of the test engine.

All the data of temperature and flow rate were read with PC based data acquisition system with resolving accuracy of 0.2%. Coolant flow meter used was the magnetic type with 0.2% accuracy. Heat Dissipation test of the radiator was carried out in combination of air and coolant flow to simulate the thermal performance.

5. Results and Discussion

5.1 Engine Simulation Results

Figure 5 shows a comparison between the predicted and measured cylinder pressure for 1.5-liter four-cylinder spark ignition engine with 3-valve pentroof chamber. As shown in the figure, the simulated result is in good agreement with the experimental data for the case inves-

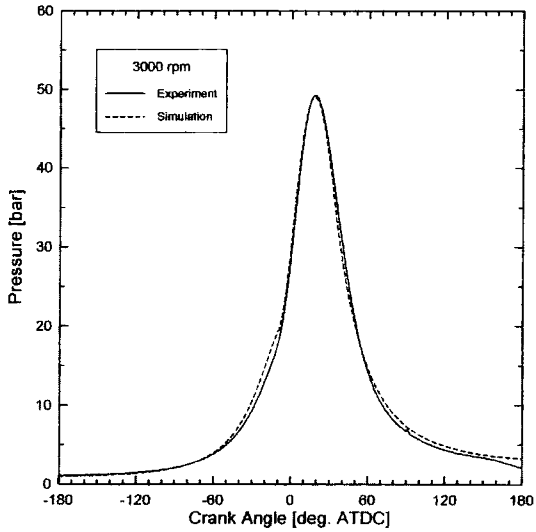

Fig. 5 Cylinder pressure diagram at 3000rpm and W.O.T

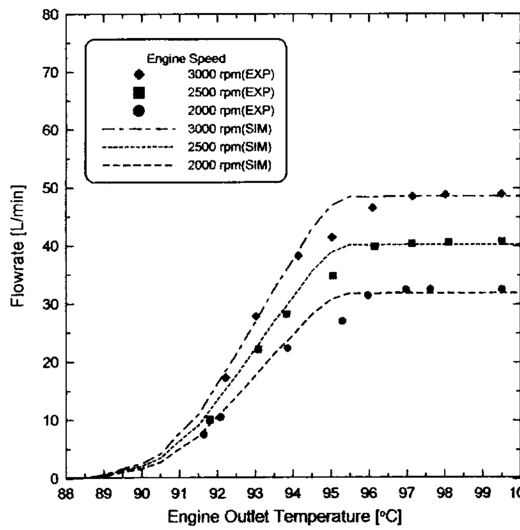

Fig. 7 Variation of coolant flowrate with engine outlet coolant temperature

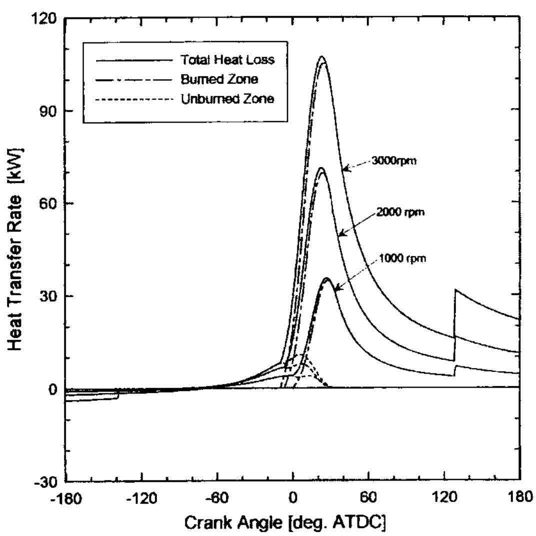

Fig. 6 Overall in-cylinder heat transfer rate

tigated. The deviation of the simulated results from experimental was within ±2.6% in the entire crank angle. Figure 6 shows the predicted overall in-cylinder heat transfer rate with various engine speed.

5.2 Cooling system analysis results

The coolant flow rate was calculated and compared with the measured data over the wide operating range of the coolant pump speed. Figure 7 shows the comparison between the flow characteristics of the cooling system obtained from rig tests and those predicted using the cooling system model. As can be seen, the results are given over a wide range of coolant pump speed and thermostat opening degrees. There is a good agreement between the predicted and measured coolant flow rates.

The standard test practice is to measure heat dissipation for a range of coolant and air flow rates. The radiator curves show that the heat dissipation varies nonlinearly with coolant flow rate. This is attributed to the transition from laminar to fully turbulent flow over the measured range. The heat dissipation tests were carried out from the coolant side reynolds number of 2125 up to 6368 in the wind tunnel test.

A comparison between the calculated and measured heat dissipation is shown in Figure 8 illustrates a comparison between the predicted and measured heat dissipation rates for various coolant and air flow rates. They show good agreement and indicate that the theoretical model based on the effectiveness-NTU method can give good prediction provided that the heat transfer coefficients on the air and coolant sides are properly represented.

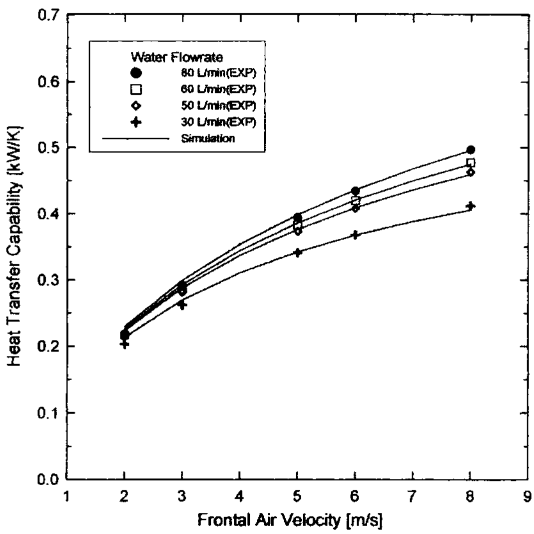

Fig. 8 Radiator Heat Dissipation Characteristics

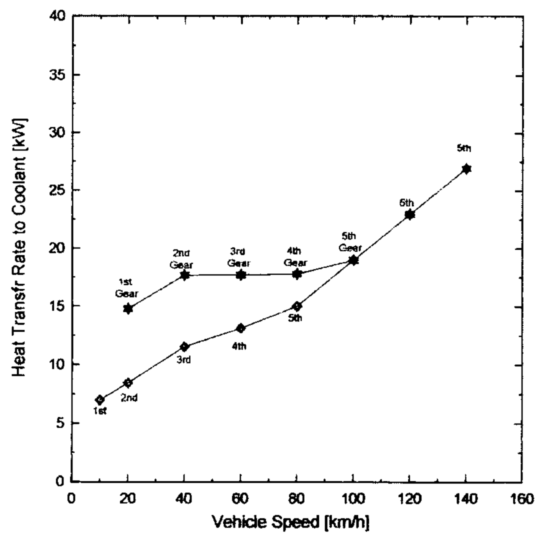

Fig. 9 Variation of heat transfer rate to coolant with vehicle speed

5.3 Overall cooling system analysis results

The vehicle thermal flow model can simulate the performance all the major thermal components of the vehicle and will thus be able to predict the thermal behavior of each component and the overall system.

The model can be used to predict the variation of heat rejected to coolant for a wide range of vehicle speed conditions. Figure 9 shows the calculated heat rejection in each gear over the full range of vehicle speed on a full road.

The engine heat transfer rate is calculated using the engine cycle simulation program described above. The air flow rate and the coolant flow rate are predicted using the coolant system model. The heat transfer calculations for the radiator are based on the effectiveness-NTU method described earlier. The vehicle speed cover a range from 10 to 140 km/h, in fifth gear, for fully loaded vehicle.

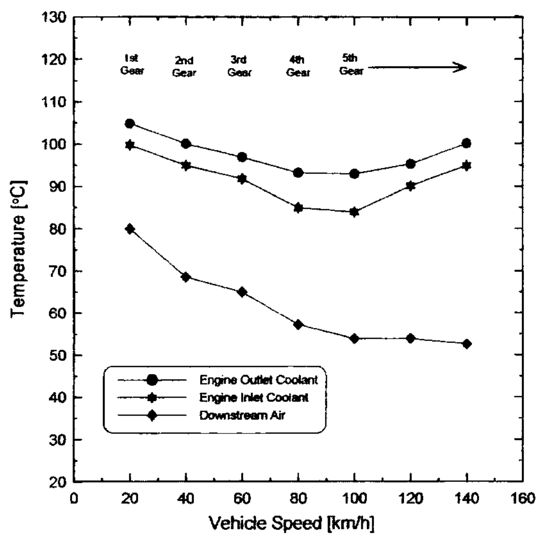

Fig. 10 Variation of coolant temperature with vehicle speed (CASE 1)

The predicted temperatures for the high speed conditions, Fig. 10, show that the engine outlet coolant temperature decreases very slowly from 104.9℃ at 20 km/h to 97℃ at 60 km/h, while the predicted radiator downstream air temperature decreases steadily 80℃ at 20 km/h to 65℃ at 60 km/h. The predicted temperatures for the low speed conditions showed that the engine outlet coolant temperature falls with increasing vehicle speed up to 80 km/h and then rises steadily, while the radiator downstream air temperature decreases steadily with increasing vehicle speed, Fig. 11. Under these conditions, the thermostat is partially open across the whole speed range, the cooling fan is on and off step changes occur.

5.4 Model applications

The vehicle thermal flow model described in

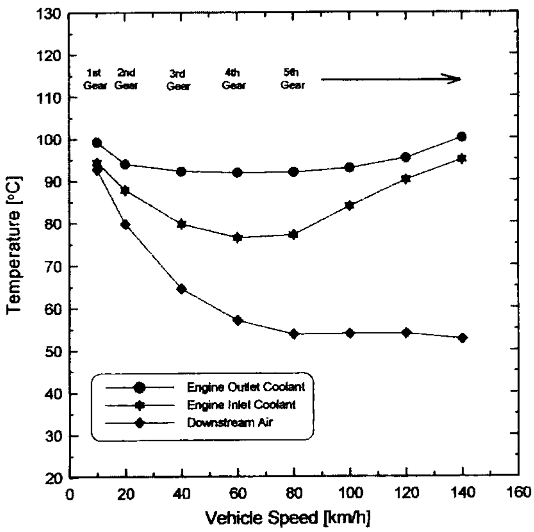

Fig. 11 Variation of coolant temperature with vehicle speed (CASE 2)

this paper can simulate the thermal behaviour of each component and the overall system and predict system performance. The model predictions consist of steady-state operating temperatures, mass flow and heat transfer rates of the cooling system. The model currently comprises the engine cooling system, radiator and can be extended to include other components such as the engine lubrication circuit, oil cooler and the air conditioning system.

The vehicle thermal flow simulation model can play a key role in cooling system design analysis. It can support and enhance the development effort and also complement the thermal flow analysis or CFD calculations, thus leading to a fully optimized and highly efficient cooling systems.

6. Conclusion

A computer simulation program has been developed for predicting the thermal performance of vehicle cooling system. It has been shown that there is good agreement between experimental and predicted results, thus providing confidence to use the model as an analytical tool for new vehicle. The radiator result shows very good agreement with the experimental data.

Parametric studies using the system model, provide insight into the qualitative effects of component characteristics on system performance such as thermostat characteristics, radiator size and pump flow delivery. The use of fluid flow simulation programs to predict coolant flow rates will alleviate the dependence upon rig tests and will permit better representation of the effect of the thermostat on coolant flow. The simulations give reliable qualitative agreement with observed behavior, having shown up a number of important system characteristics.

Acknowledgment

The authors acknowledge the support of the Korean Ministry of Commerce, Industry and Energy under the G7-Next Generation Vehicle Development Program.

References

Assanis, D. N. and Heywood, J. B., 1986, "Development and Use of a Computer Simulation of the Turbocompounded Diesel System for Engine Performance and Component Heat Transfer Studies," *SAE 860329*.

Beretta, G. P. and Keck, J. C., 1983, "Energy and Entropy Balances in a Combustion Chamber: Analytical Solution," *Combustion Science and Technology*, Vol. 30, pp. 19~29.

Blumberg, P. N. and Kummer, J. T., 1971, "Prediction of NO Formation in Spark-Ignited Engines-Analysis of Methods of Control," *Combustion Science and Technology*, Vol. 4, pp. 73~95.

Chang, Y. J. and Wang, C. C., 1997, "A Generalized Heat Transfer Correlations for Louver Fin Geometry," *Int. J. Heat and Mass Transfer*, Vol. 40, No. 3, pp. 533~544.

Chiang, E. C., Ursini, V. J. and Johnson, J. H., 1982, "Development and Evaluation of a Diesel Powered Truck Cooling System Computer Simulation Program," *SAE 821048*.

Chiang, E. C., Chellaiah, S. and John, J. H., 1985, "Modelling of the Convective Heat Flow in Radiator for Coolant Temperature Prediction," ASME Paper 85-WA/HT-22.

Eichiseder, W. and Raab, G., 1993, "Calculation and Design of Cooling Systems," *SAE 931088*.

Hausen, H., 1943, "Darstellung des Wärmeuberganges in Rohren durch Verallgemeinerte Potenzbeziehungen," *VDI Z*., No. 4, pp. 91.

Heywood, J. B., 1988, *Internal Combustion Engine Fundamentals*, Automotive Technology Series, McGraw-Hill.

Holman, J. P., 1992, *Heat Transfer*, seventh ed., McGraw-Hill.

Kays, W. M. and London, A. L., 1984, *Compact Heat Exchangers*, 3rd edition, McGraw Hill, New York.

Press, W. H., Flannery, B. P., Teukolsky, S. A. and Vetterling, W. T., 1989, *Numerical Recipes*, Cambridge University Press.

Rising, F. G., 1977, "Engine Cooling System Design for Heavy Duty Trucks," *SAE 770023*.

Sakai, T., Ishiguro, S., and Sudoh, Y., Raab, G. and Hager, J., 1994, "The Optimum Design of Engine Cooling System by Computer Simulation," *SAE 942270*.

Sidders, J. A. and Tilley, D. G., 1997, "Optimising Cooling System Performance Using Computer Simulation," *SAE 971802*.

Veshagh, A. and Chen, C., 1993, "A Computer Model for Thermofluid Analysis of Engine Warm-Up Process," *SAE 931157*.

Won, J. P., 1999, "Thermal Flow Analysis of Automotive Compact Heat Exchangers," PhD. Thesis, Kyung Hee University.

Woschni, G. and Fieger, J., 1982, "Experimental Investigation of the Heat Transfer at Normal and Knocking Combustion in Spark Ignition Engines," *MTZ*, Vol. 43, pp. 63~67.

Woschni, G., 1967, "A Universally Applicable Equation for the Instantaneous Heat Transfer Coefficient in the Internal Combustion Engine," *SAE 670931*.

Xu, Z., Johnson, J. H. and Chiang, E. C., 1984, "A Simulation Study of a Computer Controlled Cooling System for a Diesel Powered Truck," *SAE 841711*.