A Simulation Platform for Vehicle Thermal Management System

Du Xiaoke¹, Lou Ke²

¹Technology center, China Automotive Engineering Research Institute Co. Ltd, Chongqing 401122, China;
²Technology center, Chongqing CVS Environmental Protection Technology Co. Ltd, Chongqing 401122, China;

Abstract. New challenges are arising in vehicle thermal managing systems as air-pollution and fuel consumption rules become more rigid. Therefore, automobile manufactures have to face more competitive pressure. The use of software and hardware-in-loop (HIL) simulation is essential to develop and validate new real-time applications of engine, air conditioner and battery thermal management systems. This paper presents a simulation platform developed to work in a joint-simulation approach of thermal system and control system. The joint-simulation platform functions in a closed loop integrating KULI and Matlab/Simulink, which make it possible to validate both the capacity of thermal system and new control policies without a real system controller at the same time. The steady-state performance of the simulator correlates well with the provided manufacturer’s curve. The simulation platform could provide substantial improvements to the thermal management system performance.

1. Introduction
The growing complexity of vehicle thermal management systems requires more efficient heat-exchanger, compressor, tube and novel control strategies. Thermal management systems must integrate cooling systems of engine, motors, battery with air conditioner, based on vehicle thermal analysis. As in many lab experiments, it is much more feasible to define and conduct experiments using meaningful models of thermal management systems rather than the actual thermal parts and systems. These models provide flexibility to change the characteristics of the thermal management sources.

In traditional model-based develop procedure, thermal system should be designed and developed at the first step, then controller and software. Usually, vehicle thermal system could be simulated and validated based on thermal system simulation software such as KULI, AMESim. Simultaneously, thermal management control software is designed on the Matlab/Simulink. If there is any problem found in bench-test, the system solution needs to be rectified in thermal system and control system separately. Furthermore, the optimization of control policy and parameters calibrations mainly depend on bench-test and experience. The whole V model develop procedure is time consuming and costing.

One advantage of the proposed simulation platform is that it merges the advantages of different simulation approaches allowing thermal management systems simulation in closed loop. Therefore, the real-time interface between KULI and Matlab enables more accurate and realistic simulations that consider all the transient effects of thermal management systems introduced by different equipment (loads, compressor, motors, tubes and circuit relay, etc.). Another important aspect, is the synchronization of the different modules of the platform, by which time cost of control policies optimization could be
reduced obviously. Furthermore, the existence of joint-simulation platform allows more flexibility and more control strategies for vehicle thermal management.

2. Principle of Thermal Analysis

2.1 Thermal Analysis of Gasoline Engine

The vehicle thermal management system focuses on two main research topics. The first one is how to reduce heat losses, such as piston work heat loss, combustion heat loss, friction heat loss and other heat loss of the piston mechanical work \[1\][2]. The second main goal is to develop control methodologies in order to maintain the thermal system in normal balanced operation. In order to acquire heat balance, it requires high heat conduction efficiency and low heat capacity of cylinders.

The engine unit heat dissipation is represented by the following formula:

\[ \Phi_{\text{rej}} = \eta \cdot M_{\text{fuel}} \cdot \Phi_{\text{LHV}} \]  

In the formula, \( \Phi_{\text{rej}} \) is the engine heat dissipation (w/s), \( \eta \) is the engine heat dissipation ratio factor, \( M_{\text{fuel}} \) is the engine fuel flow rate (g/s), and \( \Phi_{\text{LHV}} \) is the fuel low heat value (w/g).

The heat transferred to the cylinder head and the cylinder block by the heat loss of combustion and the frictional heat loss transmitted to the cylinder block are respectively represented by the following formula:

\[ \Phi_{\text{chead}} = \Phi_{\text{cl}} \cdot (1 - a_c/100) \]  
\[ \Phi_{\text{cblock}} = \Phi_{\text{cl}} \cdot a_c/100 \]  
\[ \Phi_{\text{fblock}} = \Phi_{\text{f}} \cdot a_f/100 \]

In the formula, \( \Phi_{\text{chead}} \) is the heat transmitted to the cylinder head by the heat loss of combustion(w), \( \Phi_{\text{cl}} \) is the heat loss of combustion (w), \( a_c \) is the heat transfer coefficient of combustion, the value is between 0-100. \( \Phi_{\text{cblock}} \) is the heat loss of combustion the heat of the cylinder block(w), \( \Phi_{\text{fblock}} \) is the friction heat loss transfer coefficient, and the value between 0 to 100.

The heat of combustion and the loss of friction heat transferred to the cooling system via the coolant are expressed by the following formula:

\[ \Phi = k \cdot A \cdot [T_{\text{eng}} - (T_{\text{out}} + T_{\text{in}})/2] \]  

In formula[5], \( \Phi \) is the heat exchange heat (w), \( k \) is the heat exchange coefficient (w / m2 / k), \( A \) is the engine and coolant heat exchange area (mm2), \( T_{\text{eng}} \) is the engine block temperature (k), \( T_{\text{in}} \) is the engine cooling liquid inlet temperature (k), \( T_{\text{out}} \) is the engine coolant outlet temperature (k).

2.2 Thermal Analysis of Motor

Motor is one of the heat source in Low-carbon vehicles, Equation (6) describes the thermal transmission result. \( dQ_{\text{air}} \) is heat mass rate; \( dA_{\text{air-side}} \) is air mass rate.

\[ dQ_{\text{air}} = \alpha_a(t - t_{\text{surface}})dA_{\text{air-side}} \]  

Equation (7) reveals thermal exchange between refrigerant \[4\]. \( \sigma \) is condense coefficient of vapor condensing; \( x \) is the air humidity set by evaporator income gate temperature; \( x_{S_{\text{surface}}} \) is air humidity decided by lower surface temperature of evaporator.

\[ d\dot{m}_K = \sigma(x - x_{S_{\text{surface}}})dA_{\text{air-side}} \]  

When surface temperature is saturated, vapor stream disappeared and air humidity reduces, it could be described as equation (8).

\[ d\dot{m}_K = -\dot{m}_{t_{\text{air}}}dx \]
3. Modelling

The vehicle thermal management system chosen for simulation was selected because of the wide variety of information available. In particular, it provides many details regarding the specifics of how to model particular engine, motor and air conditioner.

Figure 1 System Structure

System structure is shown in Figure 1. This policy increases the heat source with the utilization of engine waste heat, which obviously increases the input temperature of indoor condenser. Therefore, in order to keep target cab temperature, the system consumes less compressor power.

Thermal model of engine compartment is built with KULI simulation software and system controller model is built with Matlab/Simulink. The two models communicate through an interface based on COM (Component Object Mode) module which provide a synchronized simulation time. The close-loop structure makes it possible to simulate transient working conditions under NEDC (New European Driving Cycle) with different control methodologies.

According to the actual operating condition of the vehicle and driving demand, the performance analysis and the parameter matching of the engine cooling system ensure sufficient preheating and cooling ability. Air conditioner thermal model is shown as Figure 2.

Figure 2 Model of the air conditioner

Controller model based on Matlab/Simulink integrates control logic units and control algorithm units. Real-time control logic units give switch output signals to thermal model as well as computational units contribute to stable dynamic response and low energy consumption level through PID and adaptive algorithms. [4]

As shown in Figure 3, KULI and MATLAB are running based on unified synchronized simulation environment. The outputs of thermal system are sent to controller module as input signals. Therefore, controller unit model calculates the output values and send back to KULI thermal system as real-time control signals. The mechanism of synchronized simulation is the foundation of flexible simulation environment. After model alignment with bench test data, it could be used for optimization of control strategy to work in high efficiency area.
4. Experiment

Table I gives an overview of the specifications relevant for the simulation of engine cooling operation. The validation of the thermal management simulation platform was done in two stages: a comparison during steady-state conditions and a comparison of operation using data from an existing 4 strokes engine with 4 cylinders. The average part simulation error is below 5%.

| TABLE I. Engine Specifications |
|--------------------------------|
| Part Name | Parameters |
|-----------------------------|------------|
| Cylinder head coolant volume (L) | 0.6 |
| Coolant cross-sectional area (mm²) | 500 |
| Friction heat loss coefficient | 80% |
| Combustion heat loss coefficient | 50% |
| Pump outlet diameter (mm) | 10 |
| Pump head (m) | 10 |
| Thermostat maximum opening area (mm²) | 270 |
| Compensation tank volume (L) | 1.5 |
| Thermostat opening temperature range (°C) | 90-97 |
| Thermostat off temperature range (°C) | 88-95 |
| Radiator length (m) | 0.54 |
| Radiator height (m) | 0.45 |
| Radiator surface efficiency coefficient | 0.98 |
| Radiator coolant flow volume (L) | 1.4 |
| Fan inner diameter (m) | 0.14 |
| Fan outer diameter (m) | 0.37 |
| Engine outlet pipe inner diameter (mm) | 33 |
Figure 4 Model of the engine cooling system

Because ambient temperature is one of the main influence parameters [5], the simulation is running under NEDC working condition, at different ambient temperatures from -7°C, 20°C to 40°C. In Figure 4, heat losses dynamic response to different conditions.

In Figure 5, the engine outlet temperature fluctuant is controlled according to system requirement. Under most of NEDC condition, the outlet temperature is kept at around 90°C except the hush accelerating condition. Average error is about 5% at the full range of working conditions. The platform can be easily running on normal personal computers and acquire real-time output, which is a low-expense solution.

Figure 5 Engine outlet coolant temperature

In another experiment, an integration modelling approach is presented, in which the system model is expanded to motor model and a heat pump air conditioner system as shown Figure 6. In this model, compressor motor is SV-IG5A, rated power is 12 KW. Temperature, pressure and mass flow rate is tested in experiment to evaluate the heat pump performance. Ambient temperature is 0-20.0°C.

Figure 6. Overall heating model of air condition heat pump system
Fig. 7 shows a typical evolution of the heat pump air conditioner’s COP (Coefficient of Performance) at different ambient temperature. The figure shows us one of the disadvantages of heat pump air conditioner, the lower COP at low temperature. With the system temperature increasing, a better COP performance is achieved. In order to keep the whole thermal management system working at a relatively high COP area, compressor should be controlled to running at high efficiency area.

When ambient temperature decreases from 20°C to 0°C, COP reduced about 24%. COP is about 3.28 when ambient temperature is 0°C in the system. As a comparison, the highest COP of traditional heat pump system in this condition is about 2.5. The performance of proposed system improved about 31.2%.

During the whole working condition, the max COP value is 4.32, which is above class 1 level of national standard 3.6. According to the simulation result at ambient temperature of 10°C, heat output mass is about 3.77Kw, which beyond the system heating requirement.

As a result of the discrete event simulation, the length of the time step between two subsequent simulation steps depends on the actual events taking place. Compared to approaches using fixed timesteps, this platform supports variable steps, which obviously reduce the simulation time, enable better performance at full range of running condition.

5. Conclusion

This paper presents a joint simulation system of thermal and control unit. A synchronization interface provides accurate and real-time simulation of vehicle thermal management system. The simulation platform allows different control algorithms to be evaluated in closed loop and in real time. The performance and the transient effects can be compared on it, which dedicates to high efficiency of system optimization and control algorithm assessment.

Furthermore, power motor, air conditioner and passenger compartment thermal model were added in to simulate hybrid vehicle model and electrical vehicle thermal management system models. Experiment results indicate that outside temperature of condenser, evaporator and air mass rate increases when ambient temperature and ambient condenser air mass flow increase. COP increases when temperature of condenser lifts. But when ambient temperature drops to 0°C, the performance becomes worse. In that condition, PTC should be added, or another refrigerant should be selected.

One of the most important future work is to pursue a fully validated hardware-in-loop simulator. The noncorrelation between KULI model and hardware-in-loop simulator seems to be a particularly important. Additional data regarding the state of the vehicle such as speed, torque would be revealing as they could be compared to the emulator’s own signals.

Acknowledgment

The work was supported by General Technology Project of Key Industrial in Chongqing No. cstc2015zdcy-ztzx60001 and Young Faculty Scholarship of China Scholarship Council CSC No. 201407845010.
References

[1] M. Kim, S.H Yoon, W.V Payne, “Development of the reference model for a residential heat pump system for cooling mode fault detection and diagnosis”, Journal of Mechanical Science & Technology, 2010, vol.24(7), pp.1481-1489.

[2] A. Beghi, R. Brignoli, L Cecchinato, “Data-driven Fault Detection and Diagnosis for HVAC water chillers,” Control Engineering Practice, 2016, VOL 53(4), pp.79-91.

[3] H.M Nahim, R. Younes, H. Shraim, “Modeling with Fault Integration of the Cooling and the Lubricating Systems in Marine Diesel Engine: Experimental validation”, Ifac Papersonline, 2016, vol 49(11), pp.570-575.

[4] M. Nemati, S. Seyedtabaii, “On-board diagnosis of vehicles cooling system”, Electrical Engineering. Tehran, Iran: IEEE Press, 2015, pp. 1317-1322.

[5] A. Beghi, L. Cecchinato, F. Peterle, “Model-based fault detection and diagnosis for centrifugal chillers”, Control and Fault-Tolerant Systems. Barcelona, Spain: IEEE Press, 2016, pp.158-163.