TECHNICAL PAPERS

Design of Air Conditioning System Using CFD Combined with Refrigeration Cycle Simulator

Toshio TAKEUCHI* Naoya KAKISHITA* Itsuhei KOHRI*

Abstract

Finding optimum layout of the condenser and other air conditioning system components in a vehicle is as essential as optimizing specifications of each system component to ensure adequate cooling performance of the system under a variety of conditions. This paper concerns a mathematical scheme that was developed for the use in finding an optimum layout of air conditioning system components. The scheme consists of an analysis model for engine compartment thermal flow that utilizes computational fluid dynamics (CFD) code and a refrigeration cycle simulator that is based on the thermal dynamic heat balance of air conditioner refrigerant. Described here are the concept of the scheme and examples of its application to development of vehicles.

Key words: *Air Conditioning, Computer Aided Engineering, Numerical Analysis*

1. Introduction

The recent widespread use of three-dimensional computer-aided design (CAD) applications has enabled automotive engineers to use computer-aided engineering (CAE) tools for predicting the performance of a vehicle from various aspects in the early stages of its development. One of the main reasons for this is that CAE, which used to be merely a qualitative prediction means, now allows even quantitative evaluations to be performed.

A vehicle's air conditioning system is one item that automotive manufactures must develop in close liaison with component makers. The design of the cooling unit and ducts, for example, is inseparably linked with the cabin equipment design and layout; the compressor and condenser greatly influence the design of the component layout in the engine compartment. The development of air conditioning system components themselves and the design of their layout in the vehicle and basic vehicle structure, therefore, would be most efficient if all of them could be carried out concurrently. Computer fluid dynamics (CFD) analysis is a recent achievement of CAE technology that is finding growing application in the air conditioning field to enable such concurrent development to be performed through simulation. To date, CFD has mainly been used by air conditioner manufactures to analyze the airflow of air conditioners in isolation, or by automotive manufactures to analyze defroster performance and interior airflows; in other words, it has been applied most frequently to in-cabin component layout and event analyses[(1) – (3)] but has rarely been used to analyze component layout in the engine compartment.

One of the challenges in the development of trucks and buses is the more stringent requirements on controlling the engine compartment thermal environment so that the products will comply with future tougher emission and noise standards. These requirements must be met by, for example, optimizing the specifications and location of the condenser. The aerodynamics and thermodynamics research team at MMC has therefore recently developed a cooling performance prediction scheme consisting of a refrigeration cycle simulator that calculates the thermal dynamic heat balance of the refrigerant in an air conditioner and a CFD-based code that simulates thermal flow in the engine compartment[(4)]. This paper will first discuss the method of calculating the dynamic characteristics of the refrigeration cycle under the system's daily operating conditions, and will then outline the analysis of engine compartment thermal flows[(5)(6)] to predict the temperatures of the cooling air flow through the condenser. Finally, a case of applying this scheme in the actual development of a vehicle, in which the scheme was shown to be valid and useful for qualitative predictions, will be described.

2. Refrigeration cycle simulator

The heat balance of refrigerant (refrigeration cycle) must be calculated from the operational characteristics of the compressor, evaporator and condenser before the air conditioning system's performance can be calculated. As the air conditioner on a vehicle is used under a variety of conditions, prediction calculations using only bench-test-based basic performance data provided by the air conditioner manufacture are inaccurate for some conditions.

To be able to predict air conditioning performance under different vehicle operating conditions, the research team has constructed a refrigeration cycle simulator that uses systematically collected and simplified bench-test data for calculation[(7)]. This section describes the method used for formulating an algebraic model

* Vehicle Research Dept., Truck & Bus Research & Dev. Office

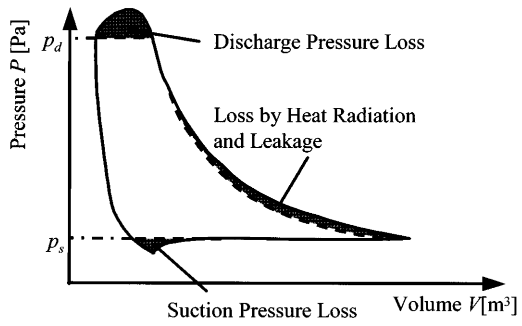

Fig. 1 P-V diagram

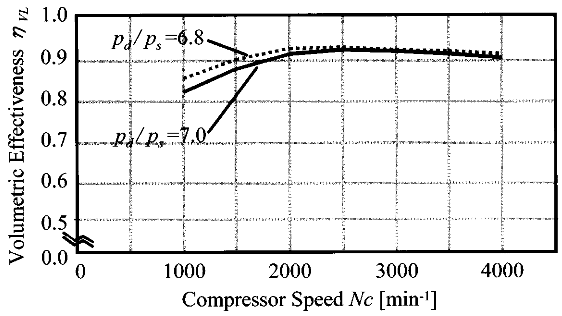

Fig. 3 Volumetric effectiveness

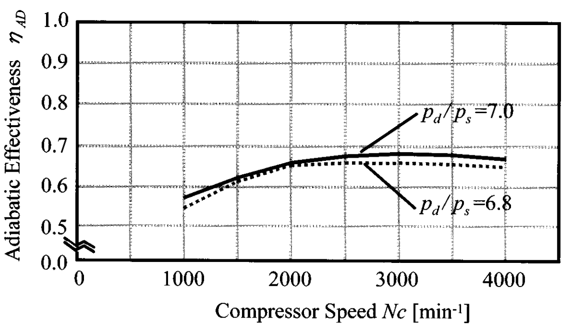

Fig. 2 Adiabatic effectiveness

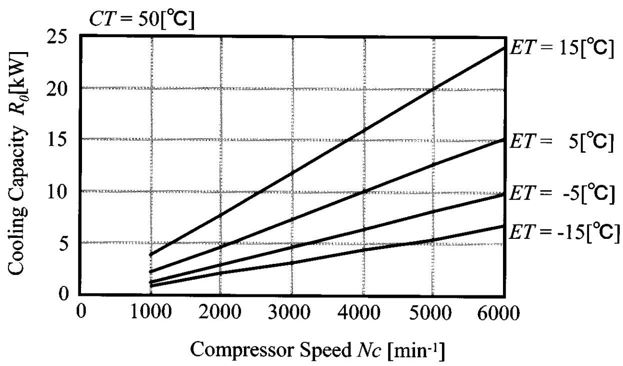

Fig. 4 Standard bench test data (cooling capacity)

that expresses air conditioning characteristics using a minimum number of parameters. The method is based on the previously established theory for calculating the thermal equilibrium point in a refrigeration cycle[8].

2.1 Formulation of compressor characteristics model

The characteristics of a compressor are represented by compression work (power) W and cooling capacity R. During compressor operation, a cycle consisting of adiabatic compression, discharge, adiabatic expansion, and suction is repeated. Each cycle involves pressure losses during the suction and discharge events and refrigerant losses due to internal leakage[9] (**Fig. 1**). These losses are represented by adiabatic compression effectiveness η_{AD} and volumetric effectiveness η_{VL}, and are quantities specific to each compressor (**Figs. 2** and **3**). The W and R characteristics are expressed using η_{AD} and η_{VL} as functions of the following four parameters: compressor inlet refrigerant temperature T_S, discharge and suction pressures P_d and P_s, and speed Nc of the compressor[1].

The W and R to be evaluated here are those of the compressor mounted on a vehicle, so to obtain W and R under actual operating environments, it was decided to use the results of studies on the theoretical relationship between the effectiveness η_{AD} and η_{VL} and the amounts W and R as a basis (as proposed previously by the authors, reference[7]) and to use the characteristics W_0 and R_0 measured under a reference condition rather than using η_{AD} and η_{VL} as explicit functions. Specifically, W and R are expressed using the equations below, taking into consideration the fact that the characteristics of W and R are subject to change, under the influence of superheated temperature ΔT_{SH} and subcooled temperature ΔT_{SC}, from the reference condition (represented by W_0 and R_0 in the equations) where the density of the refrigerant is free from superheated temperature ΔT_{SH} and subcooled temperature ΔT_{SC}:

$$W = f_W\,(ET,\ \Delta T_{SH}) \cdot W_0\,(ET,\ CT,\ Nc) \tag{1}$$

$$R = f_R\,(ET,\ CT,\ \Delta T_{SH},\ \Delta T_{SC}) \cdot R_0\,(ET,\ CT,\ Nc) \tag{2}$$

Thus, W and R can be expressed without using η_{AD} and η_{VL} by using the following three parameters: evaporator temperature ET, condenser temperature CT and compressor speed Nc. W_0 and R_0 are separately determined as characteristics subordinate to the compressor speed Nc from the data obtained through systematically conducted bench tests (**Figs. 4** and **5**), and, in environments involving superheated and subcooled temperature, they are determined through prediction using equations (1) and (2) above.

2.2 Formulation of evaporator heat exchange performance model

At the outlet of the evaporator, refrigerant must be in a state of superheated vapor to prevent liquid refrigerant from entering the compressor. To ensure this, the expansion valve detects the temperature at the outlet of the evaporator and controls the flow rate of refrigerant such that ΔT_{SH} remains appropriate. As a result, the refrigerant in the evaporator is maintained in a partly two phase (gas-liquid) state and a partly single phase (superheated vapor) state (**Fig. 6**). Assume here that the whole evaporator volume is 1 and that the part occupied by superheated vapor corresponds to a ratio of α, then the value of α should change as ΔT_{SH} changes due

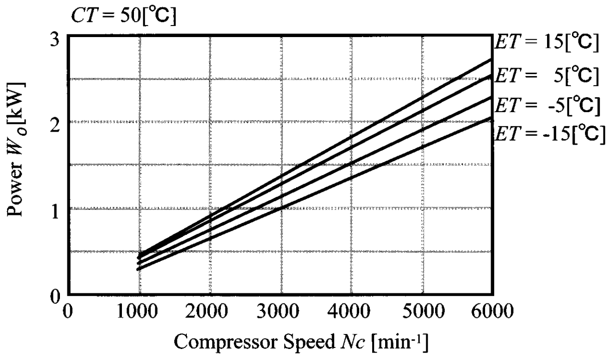

Fig. 5 Standard bench test data (power)

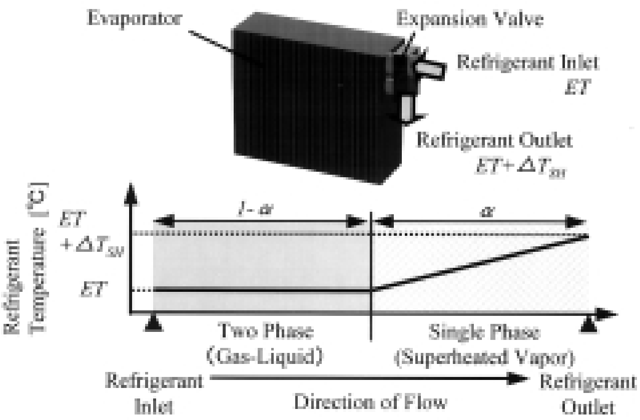

Fig. 6 Phase change of refrigerant in evaporator

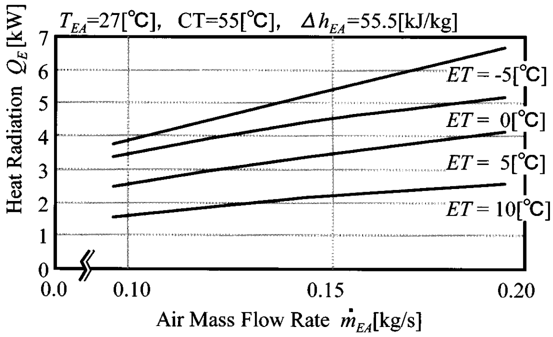

Fig. 7 Heat exchange performance (evaporator)

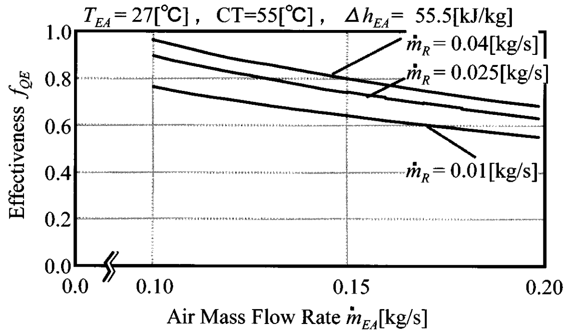

Fig. 8 Heat exchange characteristics (evaporator)

to variations in operating conditions. This necessitates consideration of the heat exchange performance which reflects changes in the value of α. In the methods of obtaining the exchanged heat quantity that have been reported by air conditioner manufactures, local heat values are calculated by applying calculations to different parts constituting the flow path of refrigerant, which are defined by detailed shapes such as fins and tubes[10][11]. On the other hand, automotive manufactures require the exchanged heat values of an evaporator on a vehicle, as in the case of a compressor. To meet this need, a simplified heat exchange performance model is formulated.

In an open circuit system, the evaporator's heat exchange performance depends on the mass flow rates $\dot{m}_{EA}$ and $\dot{m}_R$ of air and refrigerant, the temperatures of air and refrigerant (evaporation temperature) T_A and ET, and the two phase region ratio α. Knowing that the expansion valve controls ΔT_{SH} to maintain it nearly constant and that the refrigeration cycle depends on the components of the air conditioning system which forms a closed circuit, α can be defined by parameters $\dot{m}_{EA}$ and $\dot{m}_R$ provided the compressor and condenser form components of a closed-circuit system like that in a vehicle. This is because the expansion valve controls $\dot{m}_R$ depending on $\dot{m}_{EA}$ in order to maintain ΔT_{SH} constant. This is expressed by the equation given below, in which the change in the two phase region ratio is represented by the heat exchange characteristic f_{QE} which corresponds to the difference in enthalpy between air and the fin Δh_{EA}, and is considered as a function of the mass flow rates $\dot{m}_{EA}$ and $\dot{m}_R$ of the air and refrigerant, respectively.

$$Q_E \cong \dot{m}_{EA} \cdot f_{QE}\,(\dot{m}_{EA}\,,\ \dot{m}_R\,) \cdot \Delta h_{EA} \qquad (3)$$

The heat exchange characteristic f_{QE} is calculated by obtaining the evaporator's heat exchange performance Q_E from ET and CT obtained through a systematically conducted bench test (**Fig. 7**) and solving equation (3) for $\dot{m}_{EA}$ and $\dot{m}_R$ (see **Fig. 8**).

2.3 Formulation of condenser heat exchange performance model

Superheated vapor refrigerant discharged from the compressor quickly condenses at the inlet of the condenser, but the degree of sub cooled ΔT_{SC} at the outlet is not so large; this suggests that CT is constant inside the condenser and so it is appropriate to express the condenser's heat exchange performance Q_C by the heat exchange characteristic f_{QC} with regard to the temperature difference between refrigerant and air ΔT_{CA}. Provided CT is constant, the heat exchange performance is considered to be independent of the refrigerant flow rate $\dot{m}_R$; consequently, f_{QC} can be expressed as a function of the mass flow rate $\dot{m}_{CA}$. If the heat capacity flow rate of air is represented by C_{CA}, then

$$Q_{CA} = C_{CA} \cdot f_{QC}\,(\dot{m}_{CA}\,) \cdot \Delta T_{CA} \qquad (4)$$

The heat exchange characteristic f_{QC} can be derived by obtaining the heat exchange performance of the con-

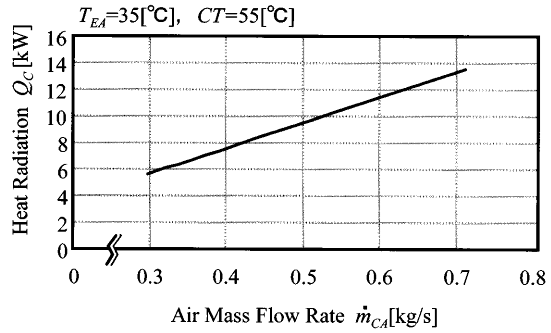

Fig. 9 Heat exchange performance (condenser)

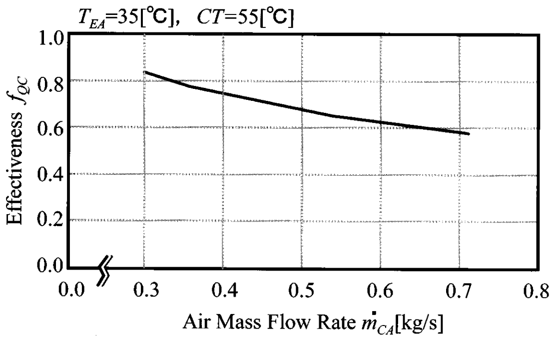

Fig. 10 Heat exchange characteristics (condenser)

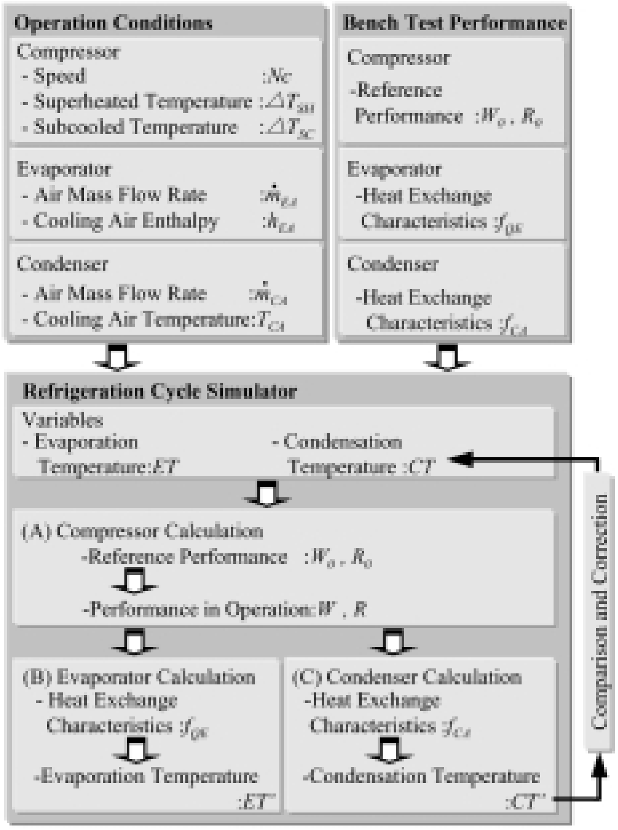

Fig. 11 Refrigeration cycle simulator

denser (**Fig. 9**) from bench test data and then solving equation (4) for the flow rate (**Fig. 10**).

2.4 Calculation process of refrigeration cycle

The thermal equilibrium point during operation of the air conditioning system is calculated by repeating the calculations of steps ① to ③ below using as parameters the abovementioned bench-test-based performance data and also the vapor and condensation temperatures (**Fig. 11**).

① Certain *ET* and *CT* temperatures are assumed for a compressor speed of *Nc* which is to be studied, corrections corresponding to superheated temperature ΔT_{SH} and sub cooled ΔT_{SC} are made to W_0 and R_0, then the compression work (power) *W* and cooling capacity *R* of the compressor in operation are derived (equations (1) and (2); part **(A)** of **Fig. 11**).

② With the evaporator, the evaporation temperature *ET'* at which Q_E balances with *R* is derived through inverse operation from the thermal properties of the cooling air flow and refrigerant and the heat exchange characteristics of evaporator f_{QE} (equation (3)). If the obtained *ET'* is different from the initially assumed *ET*, then *ET* is corrected and then the process returns to step ① (parts **(A)** and **(B)** of **Fig. 11**).

③ With the condenser, the condensation temperature *CT'* at which the condensation capacity *Q* reaches equilibrium ($Q = R + W$) is calculated through inverse operation from the thermal properties of the cooling air flow and refrigerant and the heat exchange characteristics of condenser f_{QC} (equation (4)). Like the calculation with the evaporator, if *CT'* is different from *CT*, the *CT* is corrected and calculation is performed again starting with step ① (parts **(A)** and **(C)** of **Fig. 11**).

2.5 Verification of refrigeration cycle simulator

Fig. 12 compares the calculated cooling capacity *R*, compressor work (power) *W*, and low and high refrigerant pressures P_H and P_L with the experimentally obtained ones for an air conditioning system on a light-duty truck running at a speed of *Vh* (compressor speed of *Nc*). In the calculation carried out for this comparison, the inputs of air-related conditions – the conditions not related to the heat exchanger (i.e., air flow rate, air temperature and humidity) – are those obtained through experiments in order to evaluate the accuracy of outputs of the refrigeration cycle simulator itself. The calculated results show a slight increase in the difference from the experimentally obtained results in higher *Vh* ranges, but the accuracy of all the calculation results is acceptable for practical purposes, proving that the refrigeration cycle simulator can adequately predict the performance of an air conditioning system, provided air-related conditions are given. The cause of the increase in error in higher *Vh* ranges may be attributable to pressure losses that may have occurred due to lubricants in the refrigerant tube and compressor.

3. CFD analysis of airflow and thermal fields around condenser

This section discusses the computational tool devel-

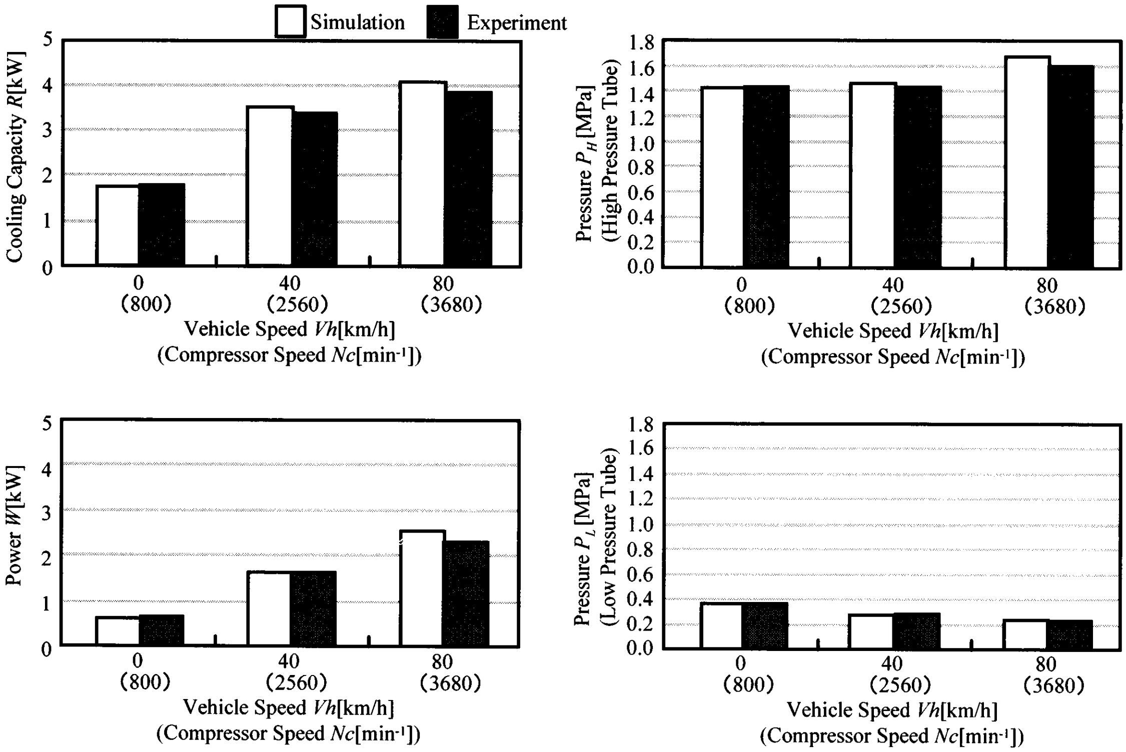

Fig. 12 Comparison between Simulator and Experiment

oped for simulating the airflow and thermal fields around the condenser for use in studies of component layout in the engine compartment. The heat exchange performance of the condenser is significantly affected by the turbulence involved in the air flow through and over components arranged in complicated configurations, as well as the heat released from engine cooling system components in the engine compartment. Therefore, the simulation of the condition of air around the condenser must consider the layout of components and the heat balance with regard to the engine cooling system. In developing the computational tool, the engine cooling performance prediction scheme that was previously reported by the authors[5][6] is used for calculating the heat balance in the engine cooling system.

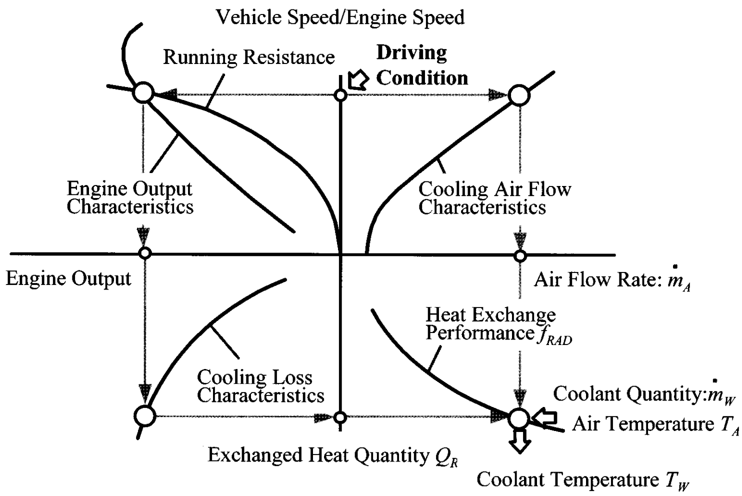

Fig. 13 Heat balance in engine cooling system

3.1 Heat balance in engine cooling system

The concept of the scheme used for calculating the heat balance in the engine cooling system will be described below, referring to **Fig. 13**. First, the engine output is obtained by giving necessary driving conditions (second quadrant). The curve in the third quadrant shows the characteristic of the engine power loss resulting from cooling which is necessary for the engine to operate normally and is represented by the heat quantity corresponding to each engine output. Reference values for the engine power loss characteristic are separately obtained from bench test measurements. Once the reference characteristic values have been established, the necessary heat quantity to be exchanged by the radiator can be determined by giving an engine output. Meanwhile, the performance of the cooling fan of a truck's engine depends on the engine speed as the fan is directly driven by the engine. If, in addition to the fan performance, the pressure loss characteristic of the engine compartment ventilation system and the temperature rise characteristic of the incoming air are given, the cooling air flow rate and air temperature can be determined (first quadrant). The exchanged heat quantity Q_{RAD} can then be expressed as a function of coolant temperature T_W, coolant quantity $\dot{m}_W$, cooling air flow rate $\dot{m}_A$, and air temperature T_A as follows:

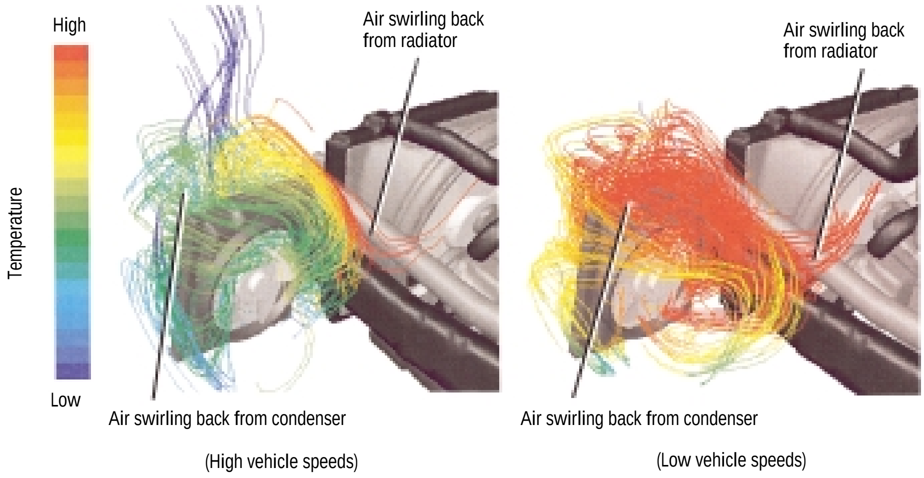

Fig. 14 Airflow and thermal fields around condenser

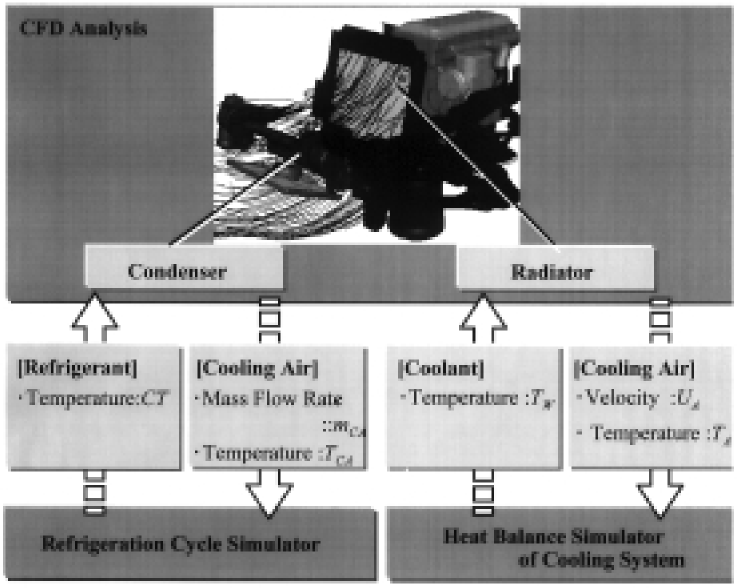

Fig. 15 General flow of cooling performance calculation

$$Q_{RAD} = f_{RAD}\,(\dot{m}_A,\ \dot{m}_W,\ T_A,\ T_W) \tag{5}$$

Among the abovementioned parameters, T_A and $\dot{m}_A$ can be obtained using the first quadrant. The $\dot{m}_W$ parameter can be obtained based on the water pump characteristic from the engine speed. The radiator exchanged heat quantity Q_{RAD} has already been obtained using the third quadrant. This means that T_W in a state of thermal equilibrium can be calculated through an inverse operation, provided the heat exchange performance of radiator f_{RAD} is given (fourth quadrant). The temperature thus obtained represents the thermal equilibrium point of the engine cooling system.

The heat balance calculation of the engine cooling system involves calculating the result of interaction between the abovementioned element factors using a one-dimensional model. However, a CFD analysis is used for calculating the heat transport of the cooling air flow through the engine compartment (handled in the first quadrant) as it significantly affects the cooling performance of the radiator and condenser and involves predicting the flow and thermal fields, which requires highly accurate calculations. The one-dimensional code for the calculation of the coolant system and the CFD analysis code for the calculation of the cooling air system are thus combined and used for predicting the coolant temperature through repeated calculation.

3.2 CFD analysis of airflow and thermal fields around condenser

In actual vehicles, part of the high-temperature air which has passed through heat exchangers such as the condenser and radiator often swirls back to the front surface of the heat exchangers; for this reason, the air temperature distribution at the inlet of each heat exchanger must be precisely predicted. The CFD analysis code, therefore, is designed to use as inputs the refrigerant temperature (condensation temperature) or coolant temperature inside the heat exchangers so that the temperatures of the air at the front of each heat exchanger, including that of swirling-back air, can be obtained. **Fig. 14** is an example of the calculated effects of the air swirling back to the front of the condenser. This example shows that the temperature at the front of the condenser of an air conditioning system varies with the vehicle speed due to a change in the swirling back air even if the specifications of the system are the same.

4. Cooling performance prediction scheme

Fig. 15 shows an outline of the air conditioning performance prediction scheme formed by combining the refrigeration cycle simulator and the code for calculating the engine cooling system heat balance. With this scheme, the air temperatures in front of the condenser and radiator are first calculated for each temperature of the refrigerant and coolant in these heat exchangers.

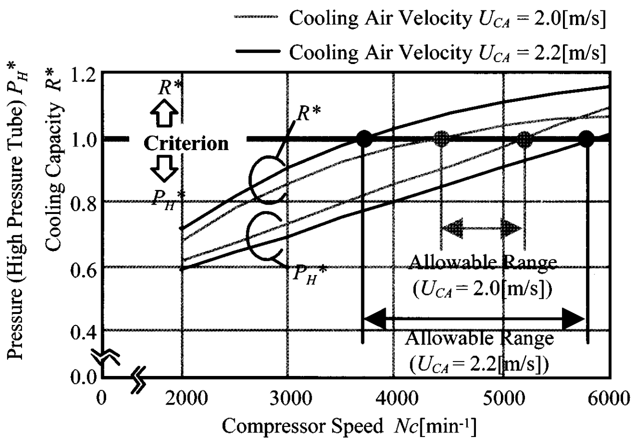

Fig. 16 Cooling capacity and refrigerant pressures

Next, the temperatures of the refrigerant and coolant are calculated back using the air temperatures in front of the condenser and radiator thus obtained and also the outcomes of the refrigeration cycle simulator and heat balance calculation of the engine cooling system. This process is carried out repeatedly until the solution of the coolant temperature converges and the air conditioning performance is finally predicted.

5. Application of cooling performance prediction scheme

This section describes, in comparison with the experimental measurements, the results of the cooling performance prediction obtained by applying the abovementioned scheme to the analysis model of a vehicle under development.

5.1 Cooling capacity R and refrigerant pressure P_H

In general, the higher the compressor speed Nc and condenser cooling air velocity U_{CA}, the higher the cooling capacity of the air conditioning system. On the other hand, the pressure of the high-pressure refrigerant P_H decreases when U_{CA} increases although it increases as Nc becomes higher. In the graph of **Fig. 16**, the ratio between R and P_H (represented by R^* and P_H^*, respectively) when the criterion is assumed to be 1.0 is plotted on the vertical axis. It is desirable that R should account for a larger part (or P_H for a smaller part) of the target cooling capacity, so $R^* \geqq 1.0$ and $P_H^* \leqq 1.0$ hold true. The graph shows that the allowable compressor speeds that satisfy this condition largely depend on U_{CA} values obtained through CFD analysis in order to satisfy all the relevant conditions.

5.2 Specifications of study subject

Fig. 17 shows the analysis model of a front under mounted condenser on a light-duty truck used for the study. With a front under mounted condenser, U_{CA} is generally insufficient. As a measure to increase the U_{CA} level, the addition of wind deflectors A and B are considered since relocating the condenser is not feasible due to truck restrictions such as minimum ground clearance and approach angle.

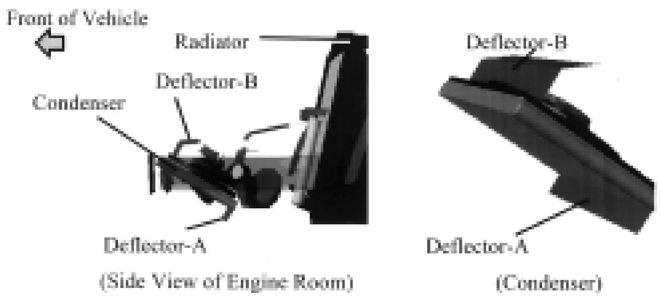

Fig. 17 CFD analysis model

5.3 Calculation results and their evaluation

A comparison of path lines of airflow corresponding to trucks with and without the wind deflectors in **Fig. 18** shows that the deflector A guides air appropriately to increase airflow passing under the bumper (part ①) and the deflector B successfully controls the swirling back of the air which has passed the condenser (part ②). The distribution of cooling air velocity U_{CA} at the front of the condenser and that of temperature T_{CA} both reflect the airflow representation by path lines (parts ③ and ④), showing an approximately 25 % improvement in the average cooling air velocity and an approximately 4 % improvement in the air temperature in front of the condenser. **Fig. 19** shows improvements achieved in the airflow and thermal fields in terms of R and P_H. The errors in the calculated results from the experimental results are approximately 5 % for R and approximately 3 % for P_H, both of which are acceptable practical levels for the accuracy expected in a predictive calculation.

6. Summary

(1) In order to make it possible to concurrently study the performance of an air conditioning system and appropriateness of the arrangement of its components during the development of a vehicle's air conditioning system, a mathematical scheme has been constructed by combining the refrigeration cycle simulator and the engine compartment thermal flow analysis code (CFD analysis code).
(2) The refrigeration cycle simulator formulates simplified models of air conditioning system characteristics and carries out calculations through rearrangement using systematically obtained bench-test performance data.
(3) With the CFD analysis, a computational code is constructed which allows complicated airflow and thermal fields around the condenser to be calculated considering the heat balance in the engine cooling system.
(4) As a result of applying the mathematical scheme to performance prediction in the actual development of a vehicle, it has been confirmed that the scheme can predict the cooling performance of an air conditioning system with acceptable accuracy for practical purposes.

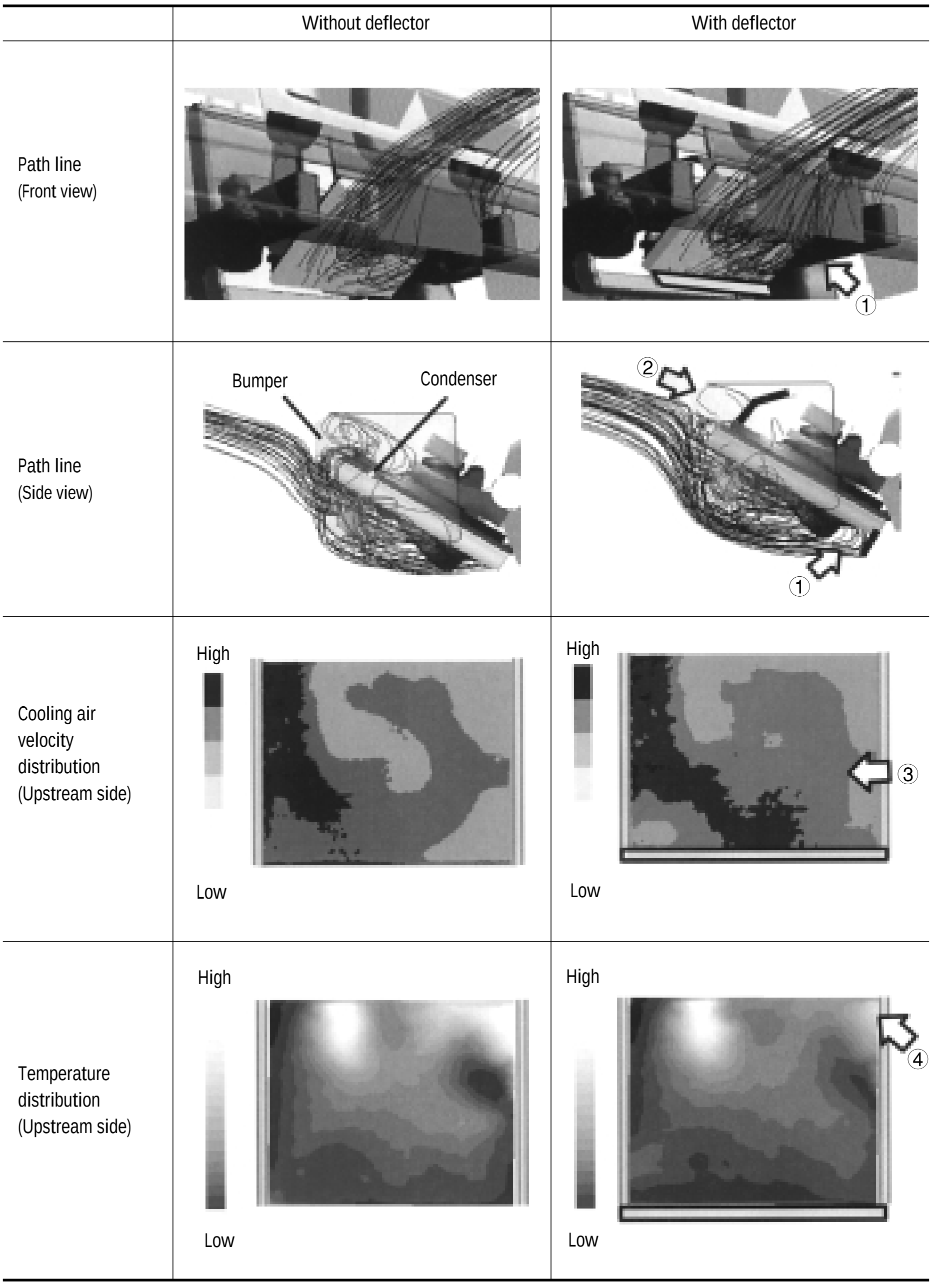

Fig. 18 Airflow and thermal fields

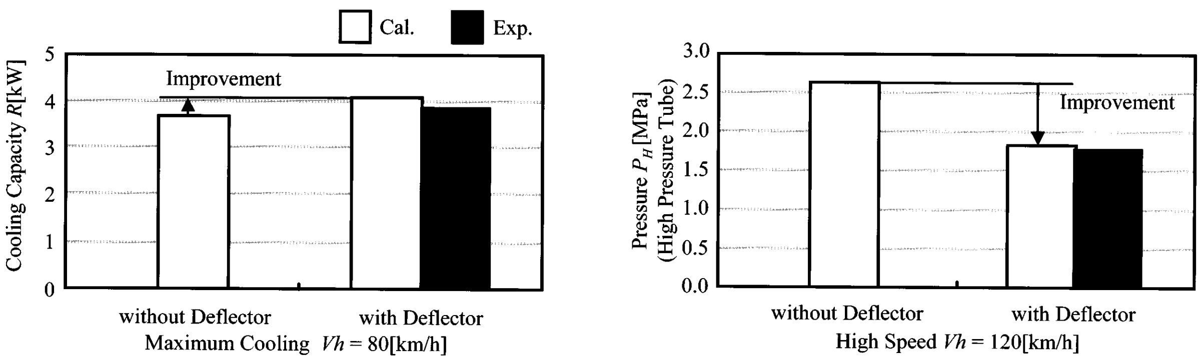

Fig. 19 Comparison between Prediction and Experiment

References

(1) Huang D. C., Oker E., Yang S. L., and Arici: A Dynamic Computer-Aided Engineering Model for Automobile Climate Control System Simulation and Application Part 1: A/C Component Simulations and Integration, SAE Paper 1990-01-1195, 1990

(2) Eisenhour R. S., Kawakami K., and Tsunada N.: HVAC System Analysis Method for Testing, SAE Paper 960684, 1996

(3) Kitada, Asano and Kamihara: Development of Car Air Conditioner Basic Performance Simulator, Proceedings of JSAE Convention, 20005538, 2000

(4) Takeuchi T., Kakishita N., and Kohri I: The Prediction of Refrigeration Cycle Performance with Front End Air Flow CFD Analysis of an Automotive Air Conditioner, SAE Paper 2002-01-0512, 2002

(5) Takeuchi, T. and Kohri I.: Development of Prediction Method of Engine Cooling Performance, JSAE Proceedings, 20005312, 2000

(6) Kohri I., Takeuchi T., and Matsunuma Y.: Computational Design of Commercial Vehicle for Reconciling Aerodynamics and Engine Cooling Performance, FISITA, F2000H248, 2000

(7) Takeuchi T., Kakishita N., and Kohri I.: Development of Method for Predicting Performance of Air Conditioner on a Truck, Proceedings of JSAE Convention, 20016068, 2001

(8) Sakai: Analysis of Thermal Equilibrium Point in Air Conditioning Refrigerator, JSME Paper No. 246, Vol. 33, 1967

(9) Kenichi F.: “Car Air Conditioners”, Sankaido

(10) Marthur G. D.: Performance of Serpentine Heat Exchanger, SAE Paper 980057, 1998

(11) Lee K. H. and Won J.-P.: Thermal Design Study of a High Performance Evaporator for the Automotive Air Conditioner, SAE Paper 1999-01-1191, 1990


Toshio TAKEUCHI


Naoya KAKISHITA


Itsuhei KOHRI