Computer Simulation of an Automotive Air-Conditioning in a Transient Mode

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Key words: Automotive air-conditioning system, Transient modelling, Mass balance, Energy balance, Thermal performance, Computer simulation

Abstract

The cool-down performance after soaking is very important in an automotive air-conditioning system and is considered as a key design variable. Therefore, transient characteristics of each system component are essential to the preliminary design as well as steady-state performance. The objective of this study is to develop a computer simulation model and estimate theoretically the transient performance of an automotive air-conditioning system. To do that, the mathematical modelling of each component, such as compressor, condenser, receiver/ drier, expansion valve, and evaporator, is presented first of all. The basic balance equations about mass and energy are used in modelling. For detailed calculation, condenser and evaporator are divided into many sub-sections. Each sub-section is an elemental volume for modelling. In models of expansion valve and compressor, dynamic behaviors are not considered in this analysis, but the quasisteady state ones are just considered, such as the relation between mass flow rate and pressure drop in expansion device, polytropic process in compressor, etc. Also it is assumed that there are no heat loss and no pressure drop in discharge, liquid, and suction lines. The developed simulation model is validated by comparing with the laboratory test data of an automotive air-conditioning system. The overall time-tracing properties of each component agreed well with those of test data in this case.

Nomenclature

A : heat transfer area [m^2]
c : specific heat [J/kgK]
d : piston diameter [m]
E : system energy [J]
h : heat transfer coefficient [W/m^{2}K]
i : enthalpy [J]
k : thermal conductivity [W/mK]
l : length [m]
M : mass [kg]
$\dot{m}$: mass flow rate [kg/sec]
n : polytropic exponent

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no : number of cylinder
P : pressure [Pa]
Q : heat transfer rate [W]
rpm : revolution speed [rpm]
s : stroke [m]
T : temperature [℃]
t : thickness [m]
U : internal energy [J]
V : volume [m^3]
v : specific volume [m^3/kg]
W : power [W]
w : humidity ratio [kgw/kga]

Greek symbols

η : efficiency
ρ : density [kg/m^3]
τ : time [sec]
ϕ : surface effectiveness

Subscripts

a : air
c : compressor
f : fin
i : inlet
o : outlet
r : refrigerant
t : tube
w : condensed water

1. Introduction

In recent years, the period to develop a new car is getting shorter. This means the design of an air-conditioning system makes no exception. Therefore, the preliminary and basic design of an air-conditioning system is very important. Direct hardware experiments take long time, and slow response to the system change. Hence, as a design tool to verify hardware modifications of the system, it is desirable to develop an analytical computer simulation program, based on the correlation of experimental data.

From 1970s, theoretical studies about automotive air-conditioning have been conducted systematically in the advanced countries. Conklu developed a computer simulation model using a digital computer[1] and Davis et al., performed a computer simulation including heat load into passenger space.[2] Cherng and Wu, summarized the overall design concept about automotive air-conditioning system[3] and Yamada et al., presented an automotive air-conditioning system using HFC-134a.[4] Won et al., also performed a computer simulation of an automotive air-conditioning system with a simple model.[5] Khamsi, et al., developed an analytical program, which combined systematically heat load calculation program and air-conditioning system program.[6,7] Mathur modelled behaviors of refrigerant in evaporator and condenser, and then compared theoretical results with empirical ones. He presented that the ratio of superheated/saturated/subcooled heat transfer rate in condenser was 16%/72%/12% and that of saturated/superheated heat transfer rate in evaporator was 60%/23%.[8,9] However, in spite that transient characteristics after soaking are very important in the design, most presented model treated just steady-state characteristics or focused on dynamic behaviors of the compressor.[10,11,12]

The objective of this study is to develop a computer simulation model and estimate theoretically the transient thermal performances of an automotive air-conditioning system. To do that, the mathematical modelling of each component, such as compressor, condenser, receiver/drier, expansion valve, and evaporator, is presented, first of all. Then the developed simulation model is validated by comparing with the laboratory test data.

2. Mathematical modelling

General mass and energy balance equations about the system shown in Fig. 1 are given by

$$\dot{m}_i = \frac{dM}{d\tau} + \dot{m}_o \quad (1)$$

$$Q + \dot{m}_i\left(i_i + \frac{v_i^2}{2} + z_i\right) = \frac{dE}{d\tau} + \dot{m}_o\left(i_o + \frac{v_o^2}{2} + z_o\right) + W \quad (2)$$

Here, m is mass flow rate, M and E are mass and energy of the system, i is enthalpy, and Q and W are heat transfer rate and power, respectively. These are basically used to the transient modelling of each component of an automotive air-conditioning system.

2.1 Heat exchanger modelling

The evaporator and condenser are important heat exchangers in the air-conditioning system. Hence, it is extremely crucial to model the heat exchanger. In order to predict accurately the time varying properties, the heat exchanger is divided into many sub-subsections more than thirty. A sub-section is an elemental control volume for modelling. Fig. 2 shows the schematic diagram of a heat exchanger and the bold line is a typical boundary of an elemental control volume. The control volume is assumed to consist of three parts, air, tube with fin, and coolant. Applying the 1st law of thermodynamics to the control volume, we can obtain three transient energy balance equations in case of evaporator are as follows (K.E. and P.E. neglected):

Air side

$$\dot{m}_a \cdot i_{ai} = \frac{dE_a}{d\tau} + \dot{m}_a \cdot i_{ao} + Q_{air \to tube} \quad (3)$$

Tube side

$$Q_{air \to tube} = \frac{dE_t}{d\tau} + Q_{tube \to ref} \quad (4)$$

Refrigerant side

$$Q_{tube \to ref} + \dot{m}_r \cdot i_{ri} = \frac{dE_r}{d\tau} + \dot{m}_r \cdot i_{ro} \quad (5)$$

For condenser, similar equations are obtained.

2.1.1 Evaporator

The refrigerant enters the evaporator as a mixture of liquid and vapor and leaves generally as a superheated vapor. If the surface temperature of the tube is less than dew point temperature, thin moisture film is formed on the tube and fin surfaces. Therefore, it is necessary to classify the dry surface and the wet surface in air side and tube side.

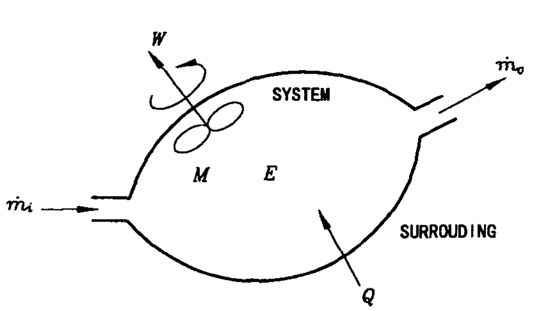


Fig. 1 A transient system.

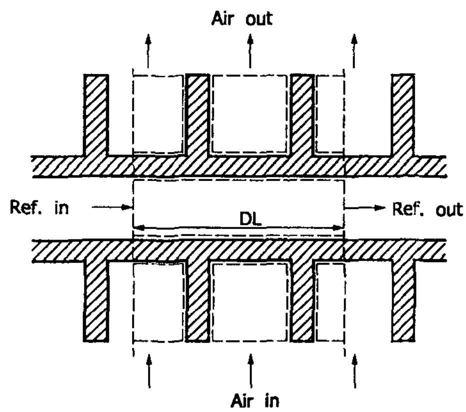


Fig. 2 Schematic of a control volume.

Balance equation at refrigerant side:

$$\frac{dU_r}{d\tau} = \dot{m}_r(i_{ri} - i_{ro}) + h_r A_r (T_w - T_r) \quad (6)$$

Here, U_r is the internal energy of refrigerant and $\dot{m}_r$ is the mass flow rate of refrigerant. Also h represents the heat transfer coefficient and A is the heat transfer area. Subscripts r and w represent refrigerant and tube wall, respectively.

Balance equations at tube side:
For dry surface,

$$c_{pw} m_w \frac{dT_w}{d\tau} = h_a \phi A_a (T_a - T_w) - h_r A_r (T_w - T_r) \quad (7)$$

For wet surface,

$$c_{pw} m_w \frac{dT_w}{d\tau} = \frac{h_a \phi A_a}{c_{pa}} (i_a - i_w) - h_r A_r (T_w - T_r) \quad (8)$$

Here, ϕ is the surface effectiveness of tube and defined by

$$\phi = 1 - \frac{A_f}{A_a}(1 - \eta_f)$$

$$\eta_f = \frac{\tanh Z}{Z}$$

$$Z = \left(\frac{2h_a}{k_f t}\right)^{0.5} l_f$$

Subscript a means air side, k_f is the thermal conductivity of fin, and t and l_f are thickness and length of fin, respectively.

Balance equations at air side:
For dry surface,

$$c_{va} m_a \frac{dT_a}{d\tau} = \dot{m}_a c_{pa} (T_{ai} - T_{ao}) - h_a \phi A_a (T_a - T_w) \quad (9)$$

For wet surface,

$$\frac{dU_a}{d\tau} = \dot{m}_a [(i_{ai} - i_{ao}) - (w_i - w_o) i_{fo}] - \frac{h_a \phi A_a}{c_{pa}} (i_a - i_w) \quad (10)$$

Here, w is the humidity ratio and subscript f means condensed water.

To estimate the heat transfer rates correctly, heat transfer coefficients of air and refrigerant sides are very important. The air side heat transfer coefficient for the evaporator can be sought by means of reference.(13) The refrigerant side for two-phase can be obtained by Kandlikar equation(14) and for single phase by Dittus-Boelter equation.(15) The pressure drop for two-phase is estimated by Pierre correlation(16) and for single phase by Darcy equation.(15)

2.1.2 Condenser

The refrigerant enters the condenser as a superheated vapor and leaves generally as a subcooled liquid. Hence, it is necessary to classify the three regions. Of course, balance equation itself is identical in three regions.

Balance equation at refrigerant side:

$$\frac{dU_r}{d\tau} = \dot{m}_r(i_{ri} - i_{ro}) - h_r A_r (T_r - T_w) \quad (11)$$

Balance equation at tube side:

$$c_{pw} m_w \frac{dT_w}{d\tau} = h_r A_r (T_r - T_w) - h_a \phi A_a (T_w - T_a) \quad (12)$$

Balance equation at air side:

$$c_{va} m_a \frac{dT_a}{d\tau} = \dot{m}_a c_{pa} (T_{ai} - T_{ao}) + h_a \phi A_a (T_w - T_a) \quad (13)$$

The refrigerant side heat transfer coefficient can be estimated by Cavallini-Zechin correlation[14] and the pressure drop by Lockhart-Martinelli correlation[17] for two phase. For single phase, the same equations in relation to evaporator can be used. The air side heat transfer coefficient can be estimated by a suitable correlation sought from reference.[13]

2.2 Compressor modelling

The compressor consists of pulley, cylinder and piston, suction and discharge valves, oil, refrigerant, miscellaneous body, etc., and temperatures of each element are different in fact. Important parameters to be determined in the compressor are such as compressor power and the refrigerant state at compressor exit. Hence, as shown in Fig. 3, the other elements except the cylinder volume are assumed to be a body with identical density and temperature. Therefore, the balance equation in compressor is expressed by

$$W + \dot{m}_r \cdot i_i = \frac{dE_c}{d\tau} + \dot{m}_r \cdot i_o + Q_{loss} \tag{14}$$

Here, $E_c = c \cdot M_c \cdot T_c$, c=specific heat of compressor body, $Q_{loss} = h_a \cdot A_c \cdot (T_c - T_a)$, A_c= surface area of compressor body.

Assuming that the compression process is polytropic, the $P-v$ relation is

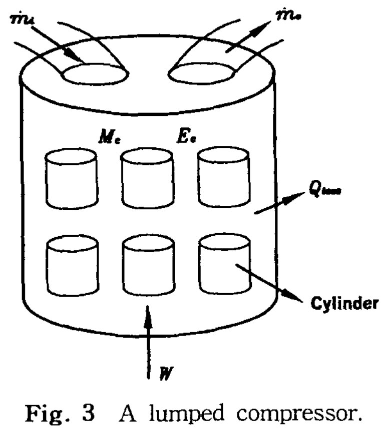


Fig. 3 A lumped compressor.

$$P_i \cdot v_i^n = P_o \cdot v_o^n = \text{constant} \tag{15}$$

Hence, the compression power is given by

$$W = \dot{m}_r \int_i^o v dP = \dot{m}_r \frac{n}{n-1} (P_o v_o - P_i v_i) \tag{16}$$

Also the polytropic efficiency P and the volumetric efficiency V are defined by

$$\eta_P = \frac{W}{\dot{m}_r (i_o - i_i)} \tag{17}$$

$$\eta_V = C_1 \left[1 - C_2 \left\{ \left(\frac{P_o}{P_i} \right)^{1/n} - 1 \right\} \right] \tag{18}$$

where C_1 and C_2 are empirical constants. The refrigerant mass flow rate can be derived by volumetric efficiency, theoretical displacement per a cylinder, V_d, and compressor revolution speed, rpm.

$$\begin{aligned} \dot{m}_r &= \eta_V \rho_i V_d \frac{rpm}{60} \\ V_d &= \frac{\pi d^2}{4} s \cdot no \end{aligned} \tag{19}$$

Here, d is bore, s is stroke, and no is the number of cylinder in the compressor.

2.3 Expansion valve modelling

The throttling process takes place in the expansion valve. This process is isenthalpic at any time.

$$i_i = i_o \tag{20}$$

In general, the relation between refrigerant mass flow rate and pressure drop is given by

$$\dot{m}_r = C \sqrt{\rho_i \cdot \Delta P} \cdot f(L_{valve}) \tag{21}$$

where C is a correction factor and f is a function representing valve action. Those are determined from empirical data.

2.4 Receiver/drier modelling

The receiver/drier is assumed to be a body with uniform density and temperature except flowing refrigerant. So the balance equation is given by

$$\dot{m}_r \cdot i_i = \frac{dE_{r/d}}{d\tau} + \dot{m}_r \cdot i_o + Q_{loss} \qquad (22)$$

where $E_{r/d} = c \cdot M_{r/d} \cdot T_{r/d}$, c=specific heat of body, $Q_{loss} = h_a \cdot A_{r/d} \cdot (T_{r/d} - T_a)$, $A_{r/d}$=surface area of receiver/drier

2.5 Discharge, liquid, and suction lines modelling

To simplify the analysis, it is assumed that there are no heat loss and no pressure drop in discharge, liquid, and suction lines.

3. Solution to equations

The differential equations developed in the previous modelling represent a coupled set of equations, which must be solved. By integrating each equation in a control volume from time τ to $\tau + \Delta\tau$ with fully implicit scheme, we can obtain a set of discretized equations. The resulting matrix in evaporator or condenser is solved using Gauss-Jordan method and the equation in other component is solved using a repetition method.

Fig. 4 show the flow chart of the simulation program developed. The input data are such as specifications of the system, initial conditions of refrigerant and air, and various empirical constants and functions. The program starts with a compressor outlet pressure assumed. At the expansion valve outlet, the mass balance is checked, and if not satisfied, the loop is repeated from the compressor to the expansion

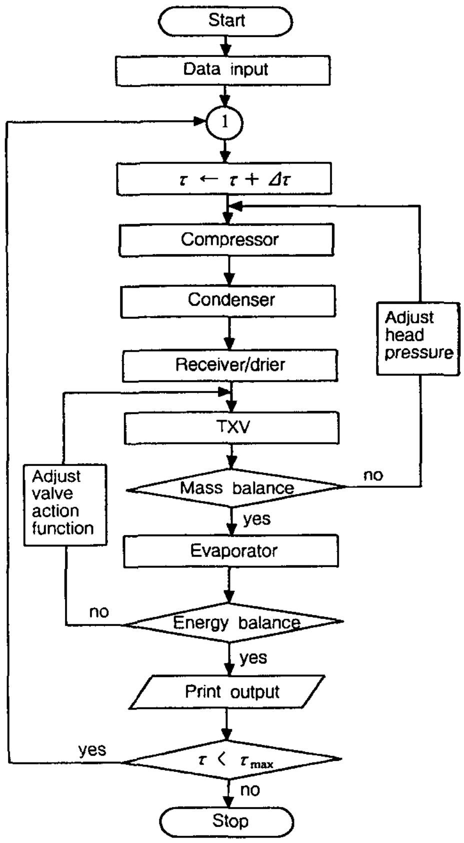


Fig. 4 Flow chart of the simulation program.

valve. In the program, various properties are calculated, which are thermodynamic properties like temperature and pressure in interested points, heat transfer rates, refrigerant mass flow rate, compression power, etc.

4. Results and discussion

The specifications of the automotive air-conditioning system used in this study are shown in Table 1. Externally equalized thermostatic expansion valve is used. Fig. 5 shows a test bench equipped with the system using R134a as working fluid and the test bench was used to obtain some empirical data for comparison. In the test bench, air and refrigerant temperatures were measured at 42 points by T-type thermocouples. The uncertainty of thermocouple

Table 1 Specifications of the automotive air-conditioning system

	Type	Specifications
Compressor	Swash plate type	Cylinder, diameter : 29.01 mm, stroke : 25.73 mm No. of cylinder : 10
Evaporator	Laminated type single tank louvered fin	Size : 236.5×235×78 cm, 4 pass
Condenser	Parallel flow type luovered fin	Size : 679×363×18 mm, 17-10-8-6 tube pass

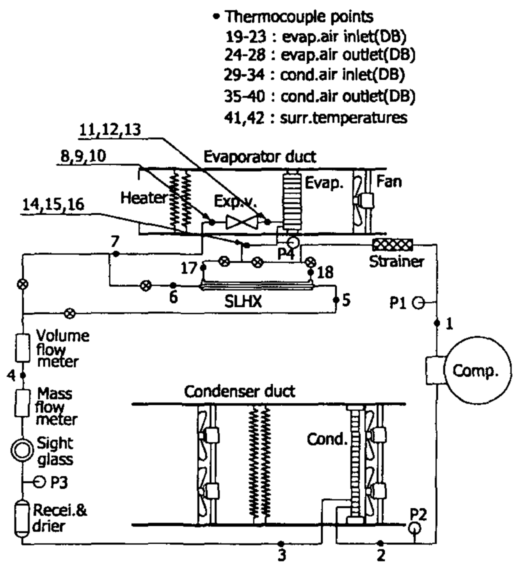


Fig. 5 Schematic diagram of the experimental apparatus and measured points.

readings was estimated to be within ±0.1℃. The refrigerant mass flowrate was measured with mass flowmeter manufactured by Smith Meter Inc. (Model S25), whose error was estimated to be within ±0.12%. The air flowrate was measured with orifice meter installed evaporator duct inlet and pitot tube in condenser duct. Two pressure transducers with an accuracy of ±0.1% were installed at the inlet and outlet of compressor. The temperatures and pressures were collected by a data acquisition system, which was connected to a computer.

The initial condition of refrigerant in the evaporator is assumed to be a saturated liquid state at 35℃ and that in the condenser to be a saturated vapor state at 35℃. Air flow rates in evaporator and condenser are 400 m^3/hr and 3500 m^3/hr, respectively, and kept constant. Also air temperature and relative humidity at evaporator inlet are 35℃ and 50% and the same at condenser inlet. The compressor speed is constant at 1600 rpm. In the experiment, all the above conditions are maintained. The compressor speed levels up to 1600 rpm as soon as possible and is kept constant.

Fig. 6 shows the time traces of pressure and temperature of refrigerant and air in the evap-

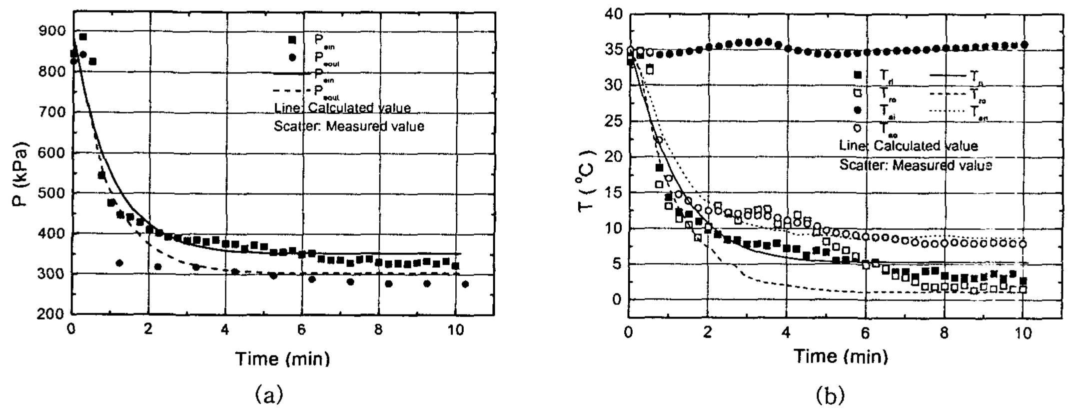
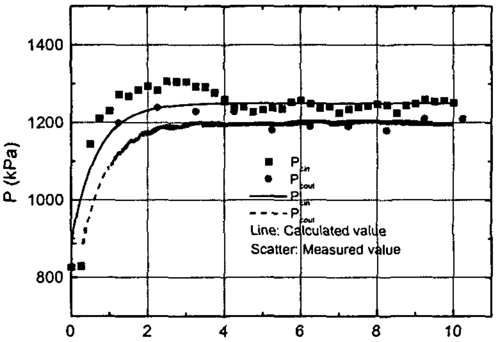
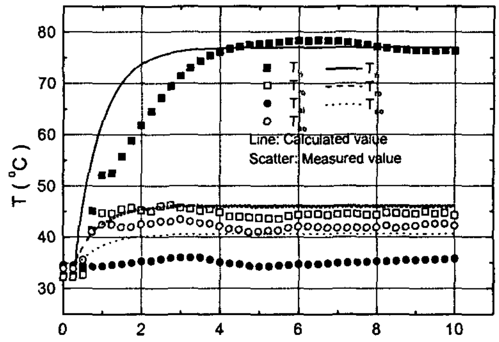


Fig. 6 Variations of pressures and temperatures according to time in the evaporator.



(a)



(b)

Fig. 7 Variations of pressures and temperatures according to time in the condenser.

orator, compared to empirical data. In the figure, the symbols represent empirical data and the lines represent analytical results. Considering experimental error, the analytical results coincide fairly well to experimental results. However, we can see the gradients of experimental data are steeper than those of analytical data at the initial stage up to 2 minutes and the time-tracing temperatures of refrigerant at evaporator exit are much different between analytical and experimental data. Since the compressor speed goes up to 1600 rpm within 10 seconds after start in the experiment, the suction force of compressor is thought to be very large at the initial stage. Therefore, the change of empirical temperature at evaporator exit is shown in three steps such as rapid drop (0~2 min), going-up (2~5 min), and slow drop (after 5 min). This fact shows this simulation model does not catch up with the phenomena, We believe the difference results from negligence of dynamic behaviors of the system.

Fig. 7 shows the time traces in the condenser. The analytical results coincide fairly well to experimental results as the same as in the evaporator. In general, the gradients of experimental data except condenser inlet refrigerant temperature are steeper than those of analytical data at the initial stage up to 2 minutes. Especially, the temperature difference shows quite large difference between analytical and empirical data at condenser inlet. This represents the compressor model in this study should be corrected a little bit in any way.

5. Conclusions

In this study, a transient simulation model about an automotive air-conditioning system was presented and the calculated results were compared with the bench test ones. Of course, the comparison was not performed over wide range, but in the time-trace point of view about temperature and pressure, we could see fairly coinciding tendency.

The conclusion is as follows: First, the time traces of pressure and temperature of refrigerant and air in the evaporator, coincide fairly well to experimental results. The changing rates of theoretical time-traces are a little bit steeper than those of experimental ones. Second, the time traces of pressure and temperature of refrigerant and air in the condenser, coincide fairly well to experimental results. But the changing rates of theoretical time traces except condenser inlet temperature are a little bit slower than those of experimental ones, unlike in the evaporator. We believe that the small differences of changing rates are due to negligence of dynamic behaviors of the compressor.

Through the comparison with wide range of test data, the simulation model and program will be revised and enhanced.

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