COMPUTER-AIDED SIMULATION OF AUTOMOTIVE AIR CONDITIONING SYSTEM

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Abstract

Traditionally, automotive air conditioning systems have been developed by selecting the appropriate components to meet the heat load estimates of a vehicle. The practical approach of component selection and testing to get desired output is expensive and time-consuming. Hence, computer-aided simulation plays a vital role to achieve optimal results without loss of money and time. For this purpose, this research work develops and implements a simulation methodology that accurately estimates the performance of the air conditioning system. An integrated model comprising of sub-models for heat exchangers, compressors, and expansion valves has been developed. The simulation model involves assigning each component a functional and a performance model that forms an analysis cycle.

Keywords: Simulation, automotive, air conditioning, HVAC, computer-aided, heat load
1. Introduction

Almost all automobile air conditioning systems use the principle of vapor compression. There are few differences between automobile and traditional air conditioning systems. Apart from that, the compressor, condenser, expansion valve, and evaporator are all same in automotive and conventional air conditioning systems. Fig. 1 shows the cooling system for comfort air conditioning of an automobile [1-2].

The compressor, condenser, expansion valve and evaporator component form the conventional VCRS cycle with a receiver/Dryer unit. Thermal comfort for an individual is said to be in a state if the person does not prefer warmer or cooler surroundings, which depends on many variables related to the human body and the controlled environment. For any air-conditioning system, the following is a basic design condition: Summer: 24-26°C with 60% relative humidity (RH), and winter: 20-22°C with 30% RH [3]. Different types of loading conditions must be considered when calculating the heat capacity of the air-conditioned volume.

An automobile is subjected to a wide range of weather conditions, including cold, mild, humid, wet, and hot. But it must provide a comfortable ride in all types of weather [4]. The temperature might reach 65-70°C, when an automobile is parked in hot weather with the doors shut [5].

![Fig.1. Cooling system for comfort air conditioning of an automobile [2]](image)

Under-capacity causes discomfort, while over-capacity causes the system to cycle frequently, resulting in some thermal bucking and energy waste [6]. When a
vehicle is traveling at a speed of 50 mph, the inside temperature is normally 10°C cooler than the ambient temperature. The cooling capacity improves as the vehicle speeds up, and vice versa.

The compressors in small to medium capacity air conditioning systems are open, semi-hermetic, or fully hermetic with an electric motor drive [7]. Whereas, the compressor of an automotive climate control system needs to be coupled through the engine shaft and it should be able to disconnect from the shaft as per requirement [8]. Because of this, as well as space limits, wobble-plate or swash-plate compressors are the best choices for automotive air conditioning systems. These are reciprocating compressors that have a specific application in the automotive industry. An oblique plate is mounted to a spinning shaft in a wobble-plate compressor. The wobble plate's periphery is surrounded by bearing shoes with ball bearings [9].

The condenser, heater, evaporator heat exchangers are part of the Heating, Ventilation, and Air Conditioning system. Evaporators and condensers were continuous heat exchangers with continuous fins and tubes [10]. However, due to space constraints, evaporators, condensers, heaters, and even radiators are increasingly using small brazed aluminum heat exchangers.

In this paper, an integrated model comprising of sub-models for heat exchangers, compressors, and expansion valves has been developed for the computer-aided simulation of automotive air conditioning systems. The results reveal that the computer simulation helps in predicting the performance of the system.

2. Methodology

2.1 Heat exchanger model

The heat exchanger model is explained with a flowchart in Fig 2. It depicts how to approach the concept in computer programming and exhibits the various fluid zones.
Assume dQ and dAi have converged for initial calculations. The humidity ratio and temperature of the air leaving the heat exchangers can be computed using mass and energy conservation at the inlet and outlet points of the airflow through the heat exchangers.

Heat exchangers made of brazed aluminum with high-performance louvered fins soldered to the tubes and flat multichannel tubes with tiny hydraulic diameters were employed [11]. Both condenser and evaporator mathematical formulations are similar, with the overall heat transfer rate calculation being the only difference. In the heat exchangers, the total heat transfer rate and pressure drop are determined incrementally [12]. The values received from each increment are combined together.
to generate the needed final values.

In a superheated area, Petukhov's equation for turbulent flow is used to calculate the coefficient of heat transfer and friction factor. The heat balance in a condenser and evaporator can be utilized to estimate the incremental heat transfer rate after computing the heat transfer coefficients of refrigerant and moist air [13]. A simple technique is used for the condenser, while the evaporator uses the concept of a cooling and dehumidification coil with a Lewis number of one.

The mass and energy balance in the control volume, which includes inlet air, heat exchanger, and output air, is used to compute the overall heat transfer rate through the heat exchangers. Further, it leads to the computation of the parameters of moist air leaving the heat exchanger.

2.2 Thermostatic Expansion Valve

The thermostatic expansion valve, which is modelled as an orifice, throttles the liquid from high condensing pressure to low evaporation pressure. [14]. This process results to maintain enthalpy throughout, which is similar to the enthalpy at the outlet of the condenser. When the valve is fully open, the maximum value of $C_v$ is reached. $A_o$ is assumed as the minimum flow area across the orifice, which does not always correspond to the cross-sectional area of the orifice. The evaporator outlet's superheat causes the valve to open. Therefore, the mass flow rate is given by,

$$M_r = C(P_b - P_e - P_{sp})\sqrt{P_1(P_c - P_e)} \quad \text{--------------------------- (1)}$$

where,

- $P_b$ - bulb pressure
- $P_e$ - evaporator pressure
- $P_c$ - condenser pressure

The constant in the equation is determined from the valve shape and the spring pressure, $P_{sp}$. The bulb pressure is determined from the superheat at the evaporator outlet, and the spring pressure is determined by the initial setting of the valve to fix the lowest superheat at the evaporator outlet [15]. The force balance applied can be demonstrated in Fig. 3.
Using the minimum superheat of 5°C and 3°C respectively for front and back TXV and the respective mass flow rate for operating superheat of 10°C obtained from the data available, the values of the constant C and $p_{sp}$ are estimated.

### 2.3 Cycle simulation

For fundamental operating conditions and input parameters, the cycle provides overall system performance characteristics based on the convergence of evaporator pressure and superheat at the evaporator outlet. [16-17]. The P-H diagram of the VCRS cycle is shown in Fig.4. To investigate the system performance under various operating situations, the code was run a certain number of times under various operating conditions, and system performance parameters were generated [18-20]. The flowchart of the cycle simulation is shown in Fig.5.

![Fig. 4 P-H diagram of VCRS cycle](image)
Fig. 5 Flowchart of cycle simulation
3. Results and Discussion

A simple algorithm is generated to derive the various characteristics of the cycle simulation. The model is applied and input for specific components is fed to obtain different graphs which give the characteristics or behavior of the entire system, which is shown in Fig. 6.

Fig. 6 Characteristics of the system

Fig. 6 (a) shows a considerable decrease in heat load for the lower velocity of the vehicle but low influence at its higher velocity. The outside air temperature influence is considerably higher than the vehicle speed. As can be seen, even a temperature rise of 5°C increases heat load by nearly 1 kW.

The core geometry for a typical compressor taken into consideration are:

- Number of cylinders = 5
- Piston action = double
- Cylinder bore = 28.5 mm
- Cylinder stroke = 21.7 mm
- Displacement volume = 138.3 cc

From Fig. 6 (b) and (c), the dependence of compressor mass flow rate on its speed and pressure ratio can be observed. Both parameters are affected equally, compressor speed tries to increase it, whereas pressure ratio tries to decrease with an increase in their values.

The heat exchanger characteristics are presented in Fig. 7 for different basic operating conditions using the core geometric data provided by the company and extracting some typical values from the provided data which are essential for the calculation part. Since there are three heat exchangers used in the cycle namely, condenser, front evaporator and back evaporator, three different data sets are produced and basic operating conditions are chosen accordingly.

![Fig.7 Heat exchanger characteristics](image-url)
When it enters the superheat region, the heat transfer coefficient value continues to decrease as the superheat temperature rises. When the heat transfer coefficient is calculated, the mass-velocity of the refrigerant inside the channel increases. Pressure gradient, i.e. pressure loss per unit length inside the channel is given in Fig. 8 to compare the values for the two-phase region and superheated region.

![Fig. 8. Entrance & exit pressure loss of super-heated vapour Vs mass velocity](image)

The more pressure loss observed in the two-phase region is because of the liquid layer present in the channel wall. If the entrance and exit pressure losses are compared for the two-phase and superheat region, a higher range is observed for the first case. The heat transfer coefficient of air is calculated with variation in its velocity and temperature as shown in Fig. 9.

![Fig. 9. Heat transfer coefficient of air at different air velocity and temperature](image)

Since most of the air properties do not vary much in the temperature range of 0 to 50°C, and are also not much sensitive to its humidity, the heat transfer coefficient gives only a slight decrement with an increase in temperature as well as relative humidity. The heat transfer rate and superheat temperature for different air inlet temperatures is shown in Fig. 10.
Fig. 10. Heat transfer rate and superheat temperature for different air inlet temperature

Air velocity is the governing property that decides the heat transfer coefficient of moist air. In comparison of graphs for heat transfer of air and refrigerant, it is observed that the range of values for air is almost one-tenth of those of refrigerant. Therefore the air-side heat transfer resistance is the controlling resistance.

The results show that for a constant condenser pressure and constant superheat of 10°C at the evaporator outlet for a constant condenser inlet air temperature of 35°C, the cycle's performance improves constantly as the air velocity, i.e. vehicle speed, increases. For air velocity less than 5 m/s, the refrigerant enters a two-phase area, indicating that an external fan should be provided to be employed after a specified value of air velocity for the relevant air temperature.

4. Conclusion

For the computer-aided simulation of automotive air conditioning systems, an integrated model comprising of sub-models for heat exchangers, compressors, and expansion valves has been developed. The component sub-models are integrated to form a holistic or a cycle simulation model. For single-pass flow, the condenser size should be increased, or a two-pass flow heat exchanger with a slightly lower cross-section area of airflow can be
employed. It is concluded that the computer simulation program is capable of predicting the performance of the system ahead of time.

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