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Design and Analysis of Automotive Air-Conditioning System

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Abstract--- *Automotive air conditioning systems are traditionally designed by choosing the right components from the upholstery of available component sizes and specifications, thereby expecting such chosen components would meet the heat load estimations in the automobile. This is almost a complete practical approach of choosing components, form a cycle and testing them to ensure desired results. If often happens, that a certain design parameter, say size of a component has to be modified to arrive at optimized results. This would again require new prototypes and testing which is both time consuming and expensive. In this project, a simulation methodology is formed and executed which provides a quicker estimation of air conditioning performance, thereby avoiding unnecessary loss of time and money. The work is based on assigning a suitable functional relationship and performance model of each components, thereby forming a one-dimensional analysis cycle. The modelling consists in first describing the characteristics of the various components of the system and the study of their characteristics. Major components like compressors, heat exchangers and expansion values are discussed in detail. Suitable functional relations are used in the mathematical description of these components. The component sub-models are integrated to form a holistic or a cycle simulation model. The output of the cycle is attempted to be plot on a time axis to deduce the cooling pattern of the automobile.*

Index Terms--- *Automobile, Calculation, Clear-sky, Fenestration, Glazing, Heat Load Estimation, Solar Angle, Solar Radiation, Windshield Glass.*

I. INTRODUCTION

The main components of an automotive air conditioning system are the same as those in other air conditioning systems, namely, the compressor, condenser, expansion valve and the evaporator. Except for this similarity in functionality, automotive air conditioning systems differ from conventional systems in many ways. Comfort conditions are to be provided in automobiles in all weather conditions. The automobile is directly exposed to different kinds of weather; cold, mild, damp, rainy and hot. In addition, provision is to be made for heating (if necessary), and defogging and de-icing of glazing. The control of dust, smoke and odours are additional important factors. By far, the most critical conditions arise in very hot weather and in slow running when the air flow over the condenser and the compressor speed can become insufficient to provide the necessary cooling. Solar load and fenestration are the main contributions to cooling load. Older automobiles with worn-out or defective door gasket seals can contribute to significant infiltration load, although this load is not large for new automobiles.

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When the vehicle is parked in hot weather with the doors closed, the hot soak temperature can reach 65-70°C.

The cooling capacities range from a minimum of 1 TR for small automobiles to about 4 TR for larger vehicles like vans (1 TR = 1 ton of refrigeration = 3.52 kW). Under-capacity results in discomfort while over-capacity leads either to frequent cycling of the system or necessitates heating of air in a heater with radiator hot water (engine cooling water). The latter choice leads of course to some thermal bucking and wastage of energy. The usual inside design temperature is about 10□ C below the ambient when the vehicle moves at a speed of typically 50 kmph. As the vehicle speeds up, the cooling capacity becomes better and vice versa. At reasonable vehicle speeds, the compressor consumes about 1.5 kW of engine power for every 1 TR capacity. This is almost double compared to the power consumption by the conventional vapour compression systems. The extra fuel consumption of the vehicle due to the provision of air conditioning depends upon urban, suburban and motorway driving. In a 27°C and 60% relative humidity weather, the extra fuel consumption for air conditioning can be as much as 20 per cent. However, year-round values for urban driving range from 5 to 10% because during some seasons fresh air can directly be used without the need for cooling and dehumidification.

II. LITERATURE REVIEW

Literature review reveals that a limited amount of work has been done for the evaluation of performance of an automotive air-conditioning as it is different from the conventional air-conditioning system. Moreover, whatever is the work done, is carried out in the research and development laboratories of automotive companies and hence remains proprietary. The components used in the automotive air-conditioning system are the swash-plate compressor and flat tube heat exchangers (rather than hermetic compressor and round tube heat exchangers), which are new fields of interest. Hence most of the papers available on these topics are published in recent years.

Modeling and experimental evaluation of an automotive air conditioning system with a variable capacity compressor did by Jabardo et al. [19] gives an overall picture of automotive air-conditioning system. Lee et al. [21] have done similar kind of work, which gives the performance analysis of automotive air-conditioning for different parallel arrangement of heat exchanger, rather than cross flow type. Brown et al. [20] compares the performance of an automotive air-conditioning with two different refrigerants i.e. CO_2 and R-134a. The designs of a wobble plate compressor and thermostatic expansion valve have been illustrated well in Jabardo et al. [19] and Brown et al. [20]. Formulas for calculation of refrigerant mass flow rate through the compressor along with the volumetric efficiency of the compressor are taken from the Jabardo et al. [19] and the expression for isentropic efficiency of the compressor is taken from the Brown et al. [20], which is required for the determination of refrigerant properties at compressor's outlet. Constraints for using a thermostatic expansion valve (TXV) at lower condensing temperature have been explained by Yu et al. [22]. Conde et al. [23] have presented a mathematical simulation model for TXV, which explains the simulation, functions and applications of a TXV in broad manner. Brown et al. [20] have given a simple modeling of TXV, whose parameter for calculation of mass flow rate is given as the function of evaporating temperature and superheat temperature. Another approach for simulation of TXV is made referring James et al. [27], which has given TXV parameter as a function of TXV needle displacement.

Webb et al. [11] first showed the possibility of replacement of fin-and-tube heat exchangers by flat tube heat exchangers by showing their advantages. The advantages of flat tube heat exchangers have been further studied by Webb et al. [1]. Wu et al. [2] showed thermal and hydraulic analysis of brazed aluminium heat exchangers, which has been very useful while designing the same. Different kinds of possible geometries in a flat tube heat exchanger and their respective friction and heat transfer correlations have been presented by Chang et al. [5] in their different papers. Effects of wet surface and inclination angles of tube and fins have been studied by Kim et al. [3].

Performance and effects of different refrigerants is given by James et al. [13] and further study has been done computationally by Joudi et al. [4]. Relations for estimating thermo-physical properties of the fluids in vapour state are taken from Chung et al. [8]. This work is based on kinetic theory of gases. Other thermal properties are available in the paper of Cleland [9] and in the manual of Solvay Fluor Company [10].

III. METHODOLOGY OF RESEARCH

Each and every component in the VCRS cycle along with heat load estimation are capsuled as separate mathematical models and then combined to form the system as a whole. The models which are considered are

1) *Head Load Estimation Model*
2) *Compressor Model*
3) Heat Exchanger Model
4) *Expansion Valve Model*
5) *Simulation Cycle Model.*

Modeling of these first four parts (models) of the system, corresponding to their actual behavior is to be done and the integration of four parts to simulate the actual refrigeration cycle running at its steady state condition is described in the following section.

The algorithm or computer code which is produced for the simulation model demonstrates the simulation of complete cycle, which runs for basic operating conditions, initially provided i.e. compressor speed, condenser inlet air velocity and temperature and condenser pressure. Evaporator pressure and superheat at the evaporator outlet are initially guessed, on convergence of these values for given basic operating conditions and input parameters, cycle provides the overall performance parameters of the system. To study the system performance for different operating conditions, algorithm or code was run required number of times with different operating conditions and system performance parameters are produced.

IV. MODELING

The five different models are explained below

Heat Load Estimation Model

The automotive surfaces are made of several layers of sheet metal and plastic trim parts. This sandwich of layers contributes to the overall transmission load and has to be deduced using the expression

$$\frac{1}{UA_i} = \frac{1}{A_i}\left(\frac{1}{h_{ai}} + \frac{L_{i1}}{k_{i1}} + \frac{L_{i2}}{k_{i2}} + \frac{L_{i3}}{k_{i3}} + \frac{L_{i4}}{k_{i4}} + \frac{1}{h_{ao}}\right)$$

where U is the overall heat transfer coefficient, A is the area, h is the convective heat transfer coefficient, k is the thermal conductivity, L is the thickness of layers. Transmission load is given by

$$Q_{tr} = UA\Delta T$$

Solar load is given by

$$Q_{sg} = AI_t\left[\tau + \frac{N\alpha}{\left(1 + {h_{ao}}/{h_{ai}}\right)}\right]$$

where, It is the total irradiance in W/m^2, τ is the transmittance, N is the unsteady state factor i.e. equal to one for steady state, α is the absorptance, h_{ao} and h_{ai} are the convective heat transfer coefficient for exterior and interior surfaces respectively.

Other loads like infiltration load and internal loads are also added to the total heat transfer to arrive at the complete heat load estimation model.

Compressor Model

Mass flow rate inside the refrigeration system is one of important factors for the optimal performance of the cycle. It depends mainly upon the function of compressor and thermostatic expansion valve. The mass flow rate through the compressor is decided by the speed and volumetric efficiency of the compressor for the given suction and delivery pressures. The volumetric efficiency depends upon several factors like speed, pressure ratio, suction gas heating, flow inertial in the suction line and the like. The mass flow-rate of the refrigerant through the compressor is obtained by following correlation:

$$M_r = \rho_1 \eta_v V_{dis} \frac{RPM}{60}$$

where ρ_1 is the density at the inlet, η_v is the volumetric efficiency, V_{dis} is the total piston displacement and RPM is the rotation per minute of the compressor. A suitable relationship for η_v is arrived based empirical curve fitting of the datas available on compressor under consideration.

Heat Exchanger Model

In designing any refrigeration system, heat exchangers design is the most important and most challenging design because this is the part where heat transfer takes place between the refrigerant and the air. It should also be noted that the heat exchangers are very compact in nature with very small hydraulic diameter where chances of pressure drop is very high.

Moreover, in case of evaporator a wet film is formed over the outer surface of the tube which affects the overall heat transfer rate. It needs high accuracy design for the optimal performance of the cycle. The flow in a heat exchanger consists of

1) Two-phase region
2) Superheated region
3) Liquid deficient region
4) Entry and Exit losses

Also the approach of heat transfer is taken in two ways

1) Appraoch (A) ε–NTU Method
2) Approach (B) Energy Balance Method

TXV Model

The thermostatic expansion valve is modeled as an orifice through which the liquid is throttled from high condensing pressure to low evaporating pressure.

As a result, enthalpy remains constant during the process. It is taken equal to the enthalpy at outlet of the condenser.

The pressure drop during the process is decided by the mass flow of refrigerant and geometry of the orifice. The flow rate through it can be calculated according to the following equation:

$$M_r = C_v A_o \sqrt{2\rho_1 \Delta P}$$

where, ΔP is the pressure variation across the valve orifice. The flow coefficient, Cv depends upon the degree of opening of the valve.

The maximum value of Cv is reached when the valve is fully open. Ao could be considered as a minimum flow area across the orifice, which, in general, does not coincide with the orifice cross sectional area. Instead, Ao is the so called “vena contracta” area. Both Cv and Ao are loosely defined and difficult to evaluate separately.

Simulation of Cycle

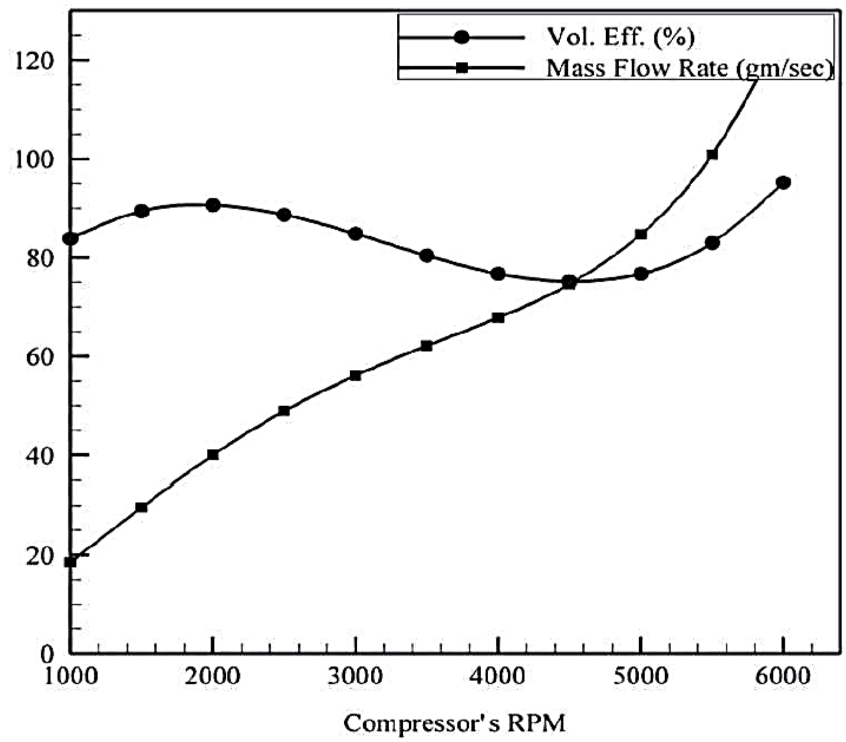
The simulation of complete cycle runs for basic operating conditions, initially provided i.e. compressor speed, condenser inlet air velocity and temperature and condenser pressure. Evaporator pressure and superheat at the evaporator outlet are initially guessed, on convergence of these values for given basic operating conditions and input parameters, cycle provides the overall performance parameters of the system.

To study the system performance for different operating conditions, code was run required number of times with different operating conditions and system performance parameters are produced.

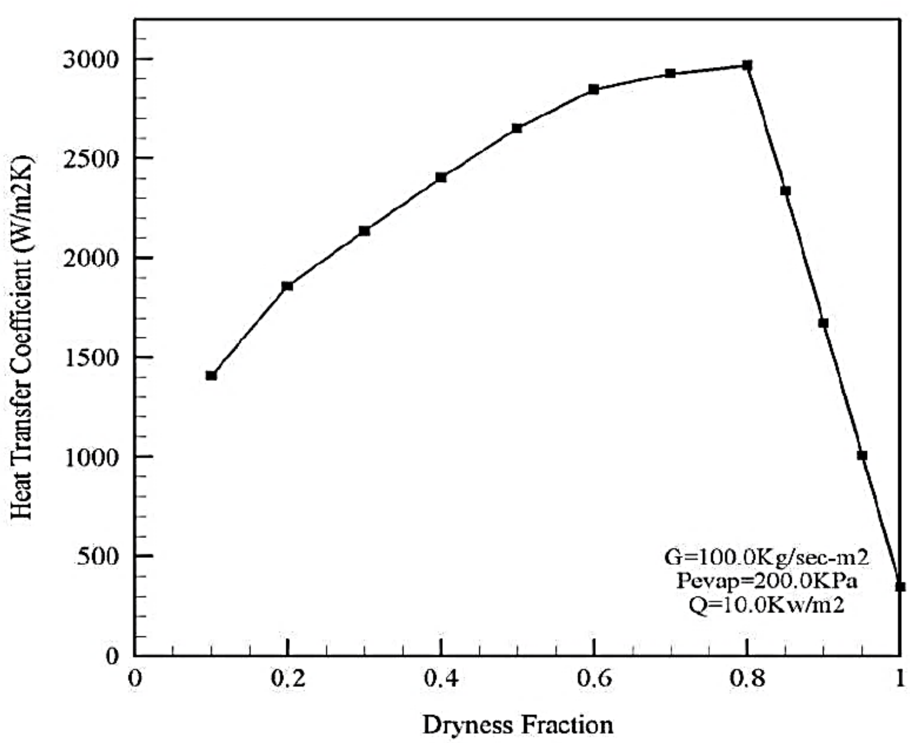
V. RESULTS & DISCUSSION

Based on the computer code models the following results are obtained

a) Compressor Model - Mass flow rate and volumetric efficiency Vs Compressor Speed:



b) Heat Exchanger Model - Heat transfer coefficient of refrigerant inside the channel Vs its Dryness Fraction for Two Phase region and Liquid Deficient region



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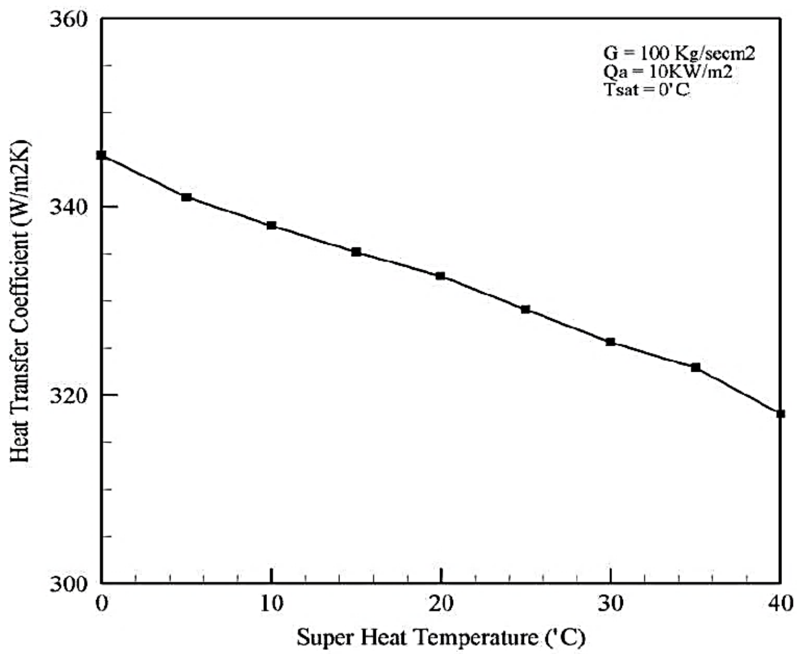
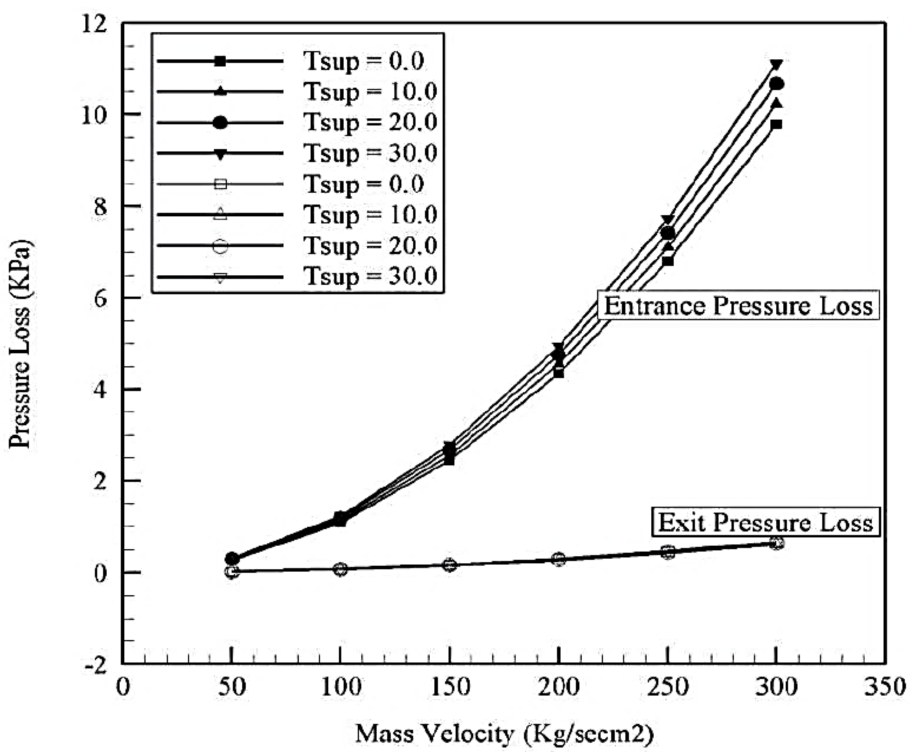
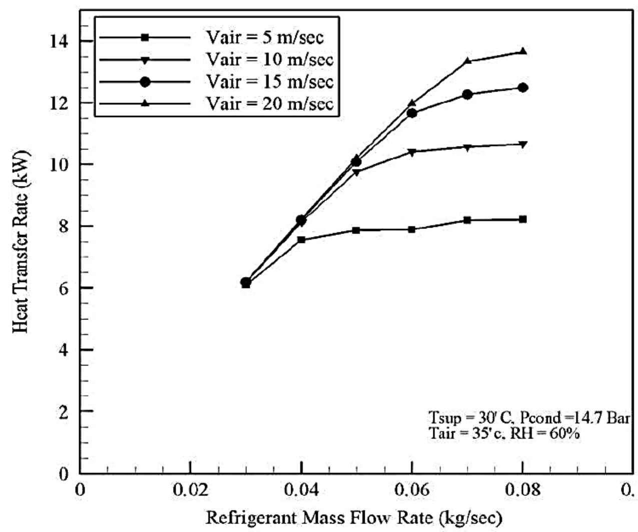
c) Heat Exchanger Model – Heat transfer coefficient of refrigerant inside the channel Vs its mass velocity for superheated region



d) Heat Exchanger Model – Entry and Exit losses



e) Condenser heat transfer rate Vs refrigerant mass flow rate for different air inlet velocity



VI. CONCLUSION

From the results presented in the previous section, which includes individual characteristics of each component and performance analysis of the complete system for different operating conditions, the following conclusions can be made:

Since the refrigerant is falling into two-phase region at the exit of condenser for extreme conditions of operation, i.e. at high air temperature and at very low vehicle speed, it seems that condenser size and geometry should be checked. Either condenser size should be increased for single pass flow or a two pass flow heat exchanger can be used with a little smaller cross-section area of air flow.

The evaporator and thermostatic valve combination should be checked again for their flow rate matching.

Although a few results have been obtained for demonstration and discussion, the computer code is capable of producing results for any desired input data and operating conditions..

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