CHAPTER 4

PERFORMANCE STUDIES ON VAPOUR COMPRESSION REFRIGERATION SYSTEM

4.1 INTRODUCTION

Mathematical modelling is one of the most widely accepted practices to study the performance of vapour compression refrigeration system with environmental friendly refrigerants. This chapter deals with the theoretical modelling and prediction of performance of the vapour compression refrigeration system with CFC12, HFC134a and the proposed alternative refrigerant mixture.

4.2 MATHEMATICAL MODELLING

The theoretical model used to compare the performance of the refrigerants is based on a one stage simple vapour compression refrigeration cycle consisting of compressor, condenser, expansion valve and evaporator. The schematic diagram of the vapour compression refrigeration cycle is shown in Figure 4.1. The Pressure-Enthalpy (P-H) and Temperature-Entropy (T-S) diagrams for pure and zeotropic refrigerant mixture is shown in Figure 4.2.

In this theoretical vapour compression cycle, the refrigerant enters the compressor at state 1 at low pressure, low temperature and saturated vapour state. From state 1 to 2, the refrigerant is compressed by the
compressor and is discharged at state 2 as a high pressure, high temperature and superheated vapour refrigerant. At the superheated state 2, it enters the condenser and releases heat to the environment. The superheated refrigerant vapour is cooled to saturation temperature (state $2'$). During condensation the refrigerant temperature decreases for mixtures (gliding temperature effect) while that remains constant for pure refrigerants if pressure drop is not considered. The refrigerant leaves the condenser at state 3 at high pressure and saturated liquid state. From state 3, the refrigerant enters the expansion valve where its pressure is reduced in a throttling process from high pressure (condenser pressure) to low pressure (evaporator pressure). After this it enters the evaporator (state 4) where it absorbs heat from the conditioned space and it leaves the evaporator at low pressure, low temperature and saturated vapour state. During evaporation the refrigerant temperature increases for mixture (gliding temperature effect) while that remains constant for pure refrigerants if pressure drop is not considered. In the theoretical cycle, it is also assumed that there is no superheating in the suction line, no subcooling in the liquid line and no pressure drop throughout the cycle.

![Schematic Diagram of Vapour Compression Refrigeration Cycle](image)

**Figure 4.1** Schematic Diagram of Vapour Compression Refrigeration Cycle
It is also assumed that steady state and uniform flow conditions exist throughout the elements of this simple vapour refrigeration cycle and changes in kinetic, potential energies and heat loss from the compressor are neglected. Therefore, specific compressor work $w_{\text{comp}}$ can be written as

$$w_{\text{comp}} = h_2 - h_1$$  \hspace{1cm} (4.1)
where $h_1$ and $h_2$ are the enthalpies of refrigerant at the compressor inlet and exit, respectively.

During the throttling process in the expansion valve, it is assumed that there is no heat transfer to the environment, which results in

$$h_3 = h_4$$  \hspace{1cm} (4.2)

The specific refrigerating effect of the cycle calculated from the rate of enthalpy change in the evaporator

$$q_{\text{ref}} = (h_1 - h_4)$$  \hspace{1cm} (4.3)

where $q_{\text{ref}}$ is the specific refrigerating effect of the refrigeration cycle. The COP of the theoretical refrigeration cycle is then calculated by

$$\text{COP}_{\text{th}} = \frac{q_{\text{ref}}}{W_{\text{comp}}}$$  \hspace{1cm} (4.4)

The parameters such as pressure ratio, specific refrigerating effect, specific compressor work and coefficient of performance are studied and compared for the refrigerants R12, R134a, R290/R600a (68/32), R290/R600 (79/21), LPG and LPG/R134a (60/40).

### 4.3 STUDIES ON THE PERFORMANCE PARAMETERS

The system performance parameters such as pressure ratio, specific refrigerating effect, specific compressor work, specific compressor displacement and coefficient of performance are calculated with evaporating temperature varying from -20°C to 15°C and condensing temperature of 35°C using the following mathematical models.
4.3.1 Pressure Ratio

Figure 4.3 shows the variation of Pressure Ratio (PR) with evaporating temperature for a condensing temperature (T_co) of 35°C. The pressure ratio of R134a is higher than that of R12. The PR values of hydrocarbon (HC) mixtures are lower than that of R12 and R134a. LPG/R134a mixture has the lowest PR values. At the rating conditions, the descending order of pressure ratios for the refrigerants is R134a, R12, R290/R600a, R290/R600, LPG and LPG/R134a.

![Diagram showing variation of Pressure Ratio with Evaporating Temperature](image)

Figure 4.3 Variation of PR with T_ev for T_co = 35°C
4.3.2 Specific Refrigerating Effect

Figure 4.4 shows the variation of Specific Refrigerating Effect (SRE) with evaporating temperature for $T_{co} = 35^\circ$C. From the figure it is observed that the HC mixtures have higher SRE than that of R12 and R134a. This is due to the higher enthalpy values of the saturated vapour of HC mixtures with higher evaporating temperatures. R134a has 25% higher SRE than that of R12. The refrigerant mixtures R290/R600a, R290/R600, LPG and LPG/R134a has 158%, 172%, 201%, 135% and 105%, 117%, 140%, 88% higher SRE than that of R12 and R134a respectively.

![Graph showing variation of SRE with evaporating temperature](image)

**Figure 4.4 Variation of SRE with $T_{ev}$ for $T_{co} = 35^\circ$C**

4.3.3 Specific Compressor Work

Figure 4.5 shows the Specific Compressor Work (SCW) with evaporating temperature for $T_{co} = 35^\circ$C. The SCW of all the refrigerants increases with decreasing evaporator temperature. This behaviour is due to
the constant entropy lines in the superheated region on P-H diagram. All the alternative HC refrigerant mixtures require higher SCW than that of R12. The refrigerant R12 requires the lowest SCW. R134a requires 32% higher SCW than that of R12. The descending order of refrigerants for SCW is R290/R600, R290/R600a, LPG, LPG/R134a, R134a, and R12.

Figure 4.5  Variation of SCW with $T_{ev}$ for $T_{co} = 35^\circ C$

4.3.4 Specific Compressor Displacement

The variation of Specific Compressor Displacement (SCD) with evaporating temperature, $T_{ev}$, is shown in Figure 4.6. The SCD of R134a and LPG is slightly higher than R12. The SCD value of R290/R600a, R290/R600 and LPG/R134a mixture is lower than that of R12, R134a and LPG.
4.3.5 Coefficient of Performance

Figure 4.7 shows the Coefficient of Performance (COP) for R12, R134a and alternative refrigerants for various evaporating temperatures for $T_{co} = 35^\circ C$. The COP of HC mixtures is higher than that of R12 and R134a. R134a has COP values very close to that with R12. LPG/R134a mixture has the highest COP. The R290/R600a and R290/R600 has 7-17% and 13-26% higher COP than that of R12 and R134a respectively for the range of temperatures considered in this study. The descending order of refrigerants for COP is LPG/R134a, LPG, R290/R600 and R290/R600a, R12 and R134a.
4.4 CONCLUSION

The behaviour of the selected refrigerant mixtures is analyzed theoretically and the following conclusions are made:

1. The specific refrigerating effect of the selected alternative refrigerant mixtures R290/R600a and R290/R600, LPG, LPG/R134a are higher than that of R12 and R134a.

2. The specific compressor work of all the selected alternative refrigerants is higher than that of R12 and R134a. The SCW of all the refrigerants increases with the decreasing evaporator temperature.

3. The COP of the HC mixtures is higher than that of R12 and R134a.
4. The pressure ratios of the HC mixtures are lower than that of R12 and R134a.

5. The refrigerant mixtures R290/R600a, R290/R600, LPG and LPG/R134a have been identified as promising alternatives to CFC12 and HFC134a in a vapour compression refrigeration system.