Performance Analysis of Condenser Subcooling Effects on Vapor Compression Refrigeration System

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Abstract
In this paper, a theoretical study of the condenser subcooling effects on the performance of vapor compression refrigeration system is presented. Generally a condenser subcooling temperature varies with environmental conditions. An alternative system architecture, which utilize a Secondary Condenser Coil, is used to obtain a desired subcooling temperature. An analysis methodology helps to determine the Coefficient of Performance of the refrigeration cycle with subcooling for various refrigerants. The performance of R134a with high GWP is compared to different low GWP hydrocarbon refrigerant mixtures with various proportions. Simulation result shows that, the increase in subcooling temperature maximizes the COP of the system, refrigerating effect and minimizes the power consumption, isentropic compression work. On the other hand, the refrigerant blends of R600/R1270/R290 (20/70/10 by wt.%) and R600/R1270/R290 (20/60/20 by wt.%) are gives the desirable results in all stages of readings and also they are found to be most suitable alternative refrigerants for existing R134a in VCR system.

Keywords: Vapor Compression system, Subcooling, secondary condenser coil, COP, Alternative refrigerants.

1. Introduction
Generally a liquid refrigerant at saturated condition entered into the expansion device, which helps to make the refrigerating effect in vapor compression refrigeration system. Whenever the liquid refrigerant entered as subcooled state can increase the refrigerating effect as well as the coefficient of performance (COP) of the system. The subcooling effects on the performance of vapor compression system has been studied theoretically and experimentally by several authors [9-13]. One of the familiar refrigerant from hydro fluoro carbon (HFC) group is R134a having zero ozone depletion potential (ODP) and high global warming potential (GWP) (1430 for 100 years), would be used in many industrial and domestic applications. Due to its high GWP, the production and usage of existing R134a will be terminated in near future. In order to control the environmental effects, the hydrocarbon (HC) refrigerants having low GWP are used as an alternatives for HFC group refrigerants in recent years. Many authors had done the experiments on alternative refrigerant.

Hammadet al. [1] investigated the performance of a domestic refrigerator using four ratios of HC refrigerant blends in order to find an alternative for CFC-12. Hydrocarbon mixture with 50% propane, 38.3% butane and 11.7% isobutene was found to be the best alternative refrigerator among the four refrigerant mixtures investigated. Wongwiseset al. [2] presented an experimental study on the application of a mixture of propane, butane, and isobutene to replace HFC134a in a domestic refrigerator. The results showed that a 60%/40% propane/butane mixture was the most appropriate alternative refrigerant. Wongwiseset al. [3] performed an experimental study on the application of hydrocarbon mixtures to replace HFC134a in automotive air conditioners. The hydrocarbons chosen for the investigation are propane (R290), butane (R600), and isobutene (R600a). The measured data are obtained from an automotive air conditioning test facility utilizing HFC134a as the refrigerant. Their results shows that the mixture of 50% propane, 40% butane and 10% isobutene produce the better performance among all other mixtures. Han et al. [4] studied a new hydrocarbon refrigerant mixture instead of R407C for vapor-compression refrigeration systems experimentally. As a result of the experimental and theoretical analysis, their new ternary non-azeotropic mixture of R32/R125/R161, whose ODP and GWP are zero and lower than R407C respectively. The new mixture of R32/R125/R161 showed better refrigerating capacity and coefficient of performance (COP) than R407C. Park et al. [5] studied a performance analysis of pure hydrocarbons and seven mixtures composed of propylene, propane, HFC152a, and dimethyl ether as an alternative to HCFC22 in residential air-conditioners and heat pumps. Their experimental results showed that the coefficient of performance (COP) of these mixtures are up to 5.7%
higher than that of HCFC22. K. Maniet et al. [6] performed experiments using a vapor-compression refrigeration system with the new R290/R600a refrigerant mixture as a substitute refrigerant for CFC12 and HFC134a. According to the results of their experiments, the refrigerant R290/R600a had a refrigerating capacity 19.9% to 50.1% higher than that of R12 and 28.6% to 87.2% than that of R134a. The R290/R600a blend's coefficient of performance (COP) is improved by 3.9–25.1% compared to that of R12 at lower evaporating temperatures and by 11.8–17.6% at higher evaporating temperatures. Syed. M et al. [7] investigated an experimental analysis of a vapor compression refrigeration system with dedicated mechanical sub-cooling. He used a sub cooling loop for obtain a required sub cooling range. R22 is employed as the refrigerant in the main cycle whereas R12 is flowing in the dedicated subcooling cycle. The experimental outcomes indicate that the load carrying capacity of the evaporator increased by approximately 0.5 kW when R22 was subcooled in the main cycle by 5–8°C. The experimental work proves that dedicated subcooling can be used for increasing cooling capacity and efficiency. Justin et al. [8] investigated a theoretical and experimental analysis of optimal subcooling in vapor compression systems via extremum seeking control. He used an alternative system architecture, which utilizes a receiver and an additional electronic expansion valve, is used to provide independent control of condenser subcooling. Simulation and experimental results show there exists an optimal sub cooling which maximizes the system efficiency. Experimental results demonstrate a 9% increase in efficiency using the alternative architecture system and extremum seeking control.

G. potkeret et al. [9] presented a theoretical study about the effect of condenser subcooling on the performance of vapor-compression systems. It is shown that, as condenser subcooling increases, the COP reaches a maximum as a result of a trade-off between increasing refrigerating effect and specific compression work. Theoretical results show that the refrigerants with large latent heat of vaporization tend to benefit less from condenser subcooling. For an air conditioning system, results indicate that the R1234yf (+8.4%) would benefit the most from condenser subcooling in comparison to R410A (+7.0%), R134a (+5.9%) and R717 (+2.7%) due to its smaller latent heat of vaporization. On the other hand, the value of COP maximizing subcooling does not seem to be a strong function of thermodynamic properties. D. Azzouziet et al. [10] studied a wire-on-tube condenser subcooling effect on the performance of vapor compression refrigeration system. He used an analysis methodology, which makes it possible to determine the COP of the refrigeration cycle with subcooling for R12, R134a and R600a. His theoretical results show that, in the subcooling temperature interval from 0°C to 14°C, the condenser additive surface is lower for R600a refrigerant compared to R134a. Moreover, the increase in subcooling temperature plays a significant role in the rise of refrigeration cycle efficiency. The purpose of this paper is to study the condenser subcooling effects on the performance of vapor compression system using secondary condenser coil having small diameter and minimum number of rows than primary condenser coil. A sample condenser coil arrangement is shown in fig. 1. In the next part consists of the performance comparison of R143a with three hydrocarbon refrigerant mixtures named as propane (R290), propylene (R1270), butane (R600) at five different proportions are taken for analysis.

2. Analysis methodology

A conventional ideal vapor compression system with and without subcooling (basic) cycle is shown in fig.2. In order to find any parameters of the vapor compression system, we must need the thermodynamic properties of the refrigerants. The thermo-physical properties of each refrigerant are determined by REFPROP 7.1 computer software. The physical and environmental properties of selected refrigerants are given in table.1.

2.1 Theoretical analysis

Generally, a simple vapor compression system consists of four basic components. There are compressor, condenser, expansion device and evaporator. A secondary condenser coil would be connected to the primary condenser coil, which helps to improve the subcooling process. Let consider a theoretical vapor compression cycle with subcooling and super heating after compression. The vapor compression system with above mentioned cycle is shown in Fig.2 and the cycle consist of the following processes:

1. Process 1-2 & 1-2a: Isentropic Compression process,
2. Process 2-3 & 2a-3a: Constant pressure heat rejection,
3. Process 3-4 & 3’-4’: Isenthalpic process,
4. Process 4-1 & 4’-1: Constant pressure heat absorption.

The condenser subcooling temperature is defined as follows:

$$T_{sub} = T_3 - T_3'$$

Where $T_3$ is the saturation temperature at condensation pressure, while $T_3'$ is the temperature existing the condenser.

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The refrigeration cycle efficiency can be measured in terms of coefficient of performance (COP), which is expressed as follows:

\[
\text{COP} = \frac{\text{Refrigerating Effect}}{\text{work done by the compressor}}
\]

COP of cooling = \( \frac{h_1 - h_4}{h_2 - h_1} \) (2)

COP of heating = \( \frac{h_2 - h_4}{h_2 - h_1} \) (3)

For subcooling cycle, the coefficient of performance is expressed as follows:

\[
\text{COP}_{\text{sub}} = \frac{Q_c}{W_c} (4)
\]

Where;

\[
Q_c (\text{Refrigerating Effect}) = h_1 - h_4 \text{ kJ/kg} \] (5)

\[
W_c (\text{Compressor Work}) = h_2 - h_1 \text{ kJ/kg} \] (6)

The expression of COPsub becomes:

\[
\text{COP}_{\text{sub}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{(h_1 - h_4) + (h_3 - h_4)}{(h_2 - h_1) + (h_2 - h_1)} = \text{COP}_b + \frac{(h_3 - h_4)}{h_2 - h_1} = \text{COP}_b + \frac{\Delta H_{\text{sub}}}{\Delta H_w} \] (7)

Where; coefficient of performance of the basic cycle is follows:

\[
\text{COP}_b = \frac{h_1 - h_4}{h_2 - h_1} \] (8)

Enthalpy difference due to subcooling is expressed as follows:

\[
\Delta H_{\text{sub}} = h_4 - h_4' \] (9)

Enthalpy difference due to compressor work expressed as follows:

\[
\Delta H_w = h_2 - h_2' \] (10)

Power consumption is defined as follows:

\[
P = \frac{m (h_2 - h_1)}{60} \text{ kW} \] (11)

Where; \( m \) denotes the mass flow rate of the refrigerant, which can be defined as follows:

\[
m = \frac{\text{Refrigerator Capacity}}{\text{Refrigerating Effect}} \text{ kg/min} \] (12)

2.2 Methodology

The various thermo-physical properties of the chosen refrigerants were calculated and tabulated. The refrigerant mixtures were taken at five different proportions and their corresponding properties also calculated. Then a simulation program was written based on the mathematical formulae of vapor compression refrigeration system and making the use of refrigerant property tables. Finally, the simulation software named as vapor compression simulator (VACSI) was developed using Visual Basic.net (VB.net) for simulating the system performance. An algorithm of the simulation is shown in fig.3. Various graphs are plotted against the subcooling temperature using the results obtained from the simulation modeling. Consider an ideal vapor compression system (cycle with subcooling and super heating after compression) operates at following conditions:

- Average Condenser Temperature = 40°C
- Average Evaporator Temperature = -5°C
- Refrigerator Capacity = 1 TON = 3.5 kW
- Subcooling Temperature = 0°C to 10°C
- Super heating Temperature = 10°C to 2°C
- Volumetric Efficiency = 0.9

As assumption, the pressure losses through the condenser and evaporator pipe lines are neglected.

3. Tables and Figures

3.1 Tables and Figures

Physical properties of the refrigerants like molecular weight, boiling point, critical temperature and critical pressure and the environmental properties (GWP and ODP) are listed in Table.1. Sample design of primary and secondary condenser coils, T-h diagram of ideal vapor compression system and flow chart of the simulation model are shown in Fig.1, 2 and 3 respectively.

| Table 1: Physical and environmental properties of R134a, R290, R600 and R1270 |
|-----------------|--------|--------|--------|--------|
| Refrigerant     | R134a | R290  | R600   | R1270  |
| Molecular weight (g/mol) | 102.032 | 44.095 | 58.122 | 42.08  |
| Boiling point (°C)     | -26    | -42    | -0.56  | -47.5  |
| Critical temperature (°C) | 101    | 97     | 152    | 93     |
Critical pressure (kPa) | 4059.28 | 4247.09 | 3796.0 | 4664.6
---|---|---|---|---
GWP (100 yr) | 1300 | 3 | 3 | 2
ODP | 0 | 0 | 0 | 0

Fig. 2 T-h diagram for ideal vapor compression system with and without subcooling.

4. Result and Discussion

The analysis of the effect of condenser subcooling temperature on the physical parameters of vapor compression system such as refrigeration effect \(Q_e\), isentropic compression work \(W\), power consumption \(P\), coefficient of performance of heating \((\text{COP}_h)\), coefficient of performance of cooling \((\text{COP}_c)\), mass flow rate of the refrigerant \(m\) are investigated theoretically for pure and blend refrigerants chosen for analysis, and these parameters are plotted against the degree of subcooling under the constant condenser and evaporator temperatures as shown in Figs. 4 – 9.
Fig. 4 shows the variation of refrigerating effect with respect to change in subcooling temperature for the refrigerant R134a and the alternative refrigerant mixtures considered for this analysis, which is R600/R1270/R290 in five different proportions. As the subcooling temperature increases the refrigerating effect also increases linearly for all kind of refrigerants taken for analysis. When compared to R134a, all other mixtures are having maximum amount of refrigerating effect. Among those mixtures, three proportions namely R600/R1270/R290 (40/40/20 by wt.%), R600/R1270/R290 (40/50/10 by wt.%) and R600/R1270/R290 (50/30/20 by wt.%) gives better refrigerating effect with small deviation from other two refrigerant mixtures in the proportion of 20/70/10 by wt.% and 20/60/20 by wt.%.

Fig. 5 shows the variation of isentropic compression work with respect to change in subcooling temperature at constant condenser and evaporator temperature of 40°C and -5°C respectively. The isentropic compression work decreases with increasing the subcooling temperature. As we considering the refrigerant the work done could not varies linearly. The refrigerant R134a gives small amount of work done performance as compared to all other refrigerant mixtures. Among those mixtures, the proportion 20/70/10 by wt.% consumes the small isentropic compression work.

Fig. 6 shows that the power consumption of the compressor decreases with increasing subcooling temperature at a constant condenser and evaporator temperature of 40°C and -5°C respectively. When the analysis was done using the mixture of HC refrigerants R600, R1270 and R290 in the proportions of 20/70/10 by wt.% and 20/60/20 by wt.%, the results from the analysis shown that they consumed less power when compared to R134a. Other three proportions namely R600/R1270/R290 (40/40/20), R600/R1270/R290 (40/50/10) and R600/R1270/R290 (50/30/20) consumed more power consumption than R134a.

The variation of coefficient of performance of heating with subcooling temperature is illustrates in Fig. 7. From this figure it can be seen that the coefficient of performance of heating increases as the subcooling temperature increases at a constant condenser and evaporator temperature of 40°C and -5°C respectively. Out of the five different mixing proportions of R600, R1270 and R290, two...
proportions namely R600/R1270/R290 (20/70/10) and R600/R1270/R290 (20/60/20) have higher values of coefficient of performance of heating when compared to the refrigerant R134a. The remaining three mixing proportions of R600/R1270/R290 are found to have lower values of coefficient of performance of heating when compared to the pure refrigerant R134a.

The variation of coefficient of performance of cooling or the COP of the system with subcooling temperature is illustrated in Fig. 8. From this figure it can be seen that the coefficient of performance of cooling increases as the subcooling temperature increases at a constant condenser and evaporator temperature of 40°C and -5°C respectively. Out of the five different mixing proportions of R600, R1270 and R290, two proportions namely R600/R1270/R290 (20/70/10) and R600/R1270/R290 (20/60/20) have higher values of coefficient of performance when compared to the refrigerant R134a. The remaining three mixing proportions of R600/R1270/R290 are found to have lower values of coefficient of performance when compared to the pure refrigerant R134a.

5. Conclusion

Theoretical analysis of condenser subcooling effects on ideal VCR system have been studied. An ideal vapor compression refrigeration system and its mathematical model was considered. Then the mathematical model was...
used to write a simulation program for the performance analysis of alternative refrigerant mixtures and the pure refrigerant R134a. The major findings contained in this work are as follows:

- The refrigeration effect (RE), coefficient of performance of heating (COP_h) and coefficient of performance of the system (COP_C) increases with increasing subcooling temperature (up to 10°C) for a constant condenser and evaporator temperature.

- The isentropic compressor work (W), power consumption (P) and the mass flow rate of the refrigerant (m) decreases with increasing the subcooling temperature (up to 10°C) for a constant condenser and evaporator temperature.

- Considering the comparison of coefficients of performance of heating, coefficients of performance of system and power consumption of the tested different proportions of HC refrigerants mixtures and also the main environmental impacts of ozone layer depletion and global warming potential, refrigerant blends of R600/R1270/R290 (20/70/10 by wt.%) and R600/R1270/R290 (20/60/20 by wt.%) are found to be most suitable alternatives among the considered refrigerants mixing proportions tested for R134a.

- The remaining three HC refrigerant blends of R600/R1270/R290 (40/40/20 by wt.%), R600/R1270/R290 (40/50/10 by wt.%) and R600/R1270/R290 (50/30/20 by wt.%) are found to be inferior when tested for R134a as a possible alternative.

References


