IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)
e-ISSN: 2278-1684,p-ISSN: 2320-334X, Volume 13, Issue 3 Ver. IV (May- Jun. 2016), PP 109-117
www.iosrjournals.org

Effect of Suction-Line Heat Exchangers on the Performance of Alternative Refrigerants to R-22

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Abstract: *The paper presents an Excel-based computer model for assessing the effect of suction-line heat exchangers (SLHXs) on the performance of vapour-compression refrigerant systems. The paper verifies the model by comparing it's results with previous analyses that used the more established REFPROP and CoolPack software. The model is then used to compare the effect of SLHXs on the performance of six refrigerants as suitable alternatives for refrigerant R-22 in air-conditioning applications. Four of these alternatives are halocarbon refrigerants (R-134a, R-152a, R-407C, R-410A) and two are hydrocarbon refrigerants (R-290 and R-600a). The comparison was made for an evaporator temperature of 0^{o}C, condenser temperature of 55^{o}C, and subcooling degrees from 2 to 10^{o}C. The results show that three refrigerants are suitable alternatives to R-22, which are R-152a, R-407C, and R-600a.*
Keywords: *Refrigerant R-22**, a**lternative refrigerants, suction-line heat exchanger, computer simulation*

I. Introduction

Chlorofluorocarbon refrigerants (CFCs) and hydro-chlorofluorocarbon refrigerants (HCFCs) offered significant advantages compared to natural fluids, such as ammonia and hydrocarbons (HCs), in terms of safety and economy. However, extensive use of these refrigerants resulted in two serous environmental problems, which are the ozone-layer depletion and global-warming. Because of their high ozone-layer depletion potential (ODP), CFCs have already been phased out and, because of their high global-warming potential (GWP), HCFCs which have will be phased out in the near future. HCFC refrigerant R-22, which currently dominates air-conditioning applications, has a relatively high GWP. Therefore, it is targeted to be phased out by 2030 in developing countries and by 2040 in developing countries. Worldwide research efforts are currently being made so as to find suitable alternatives to R-22. Since a large number of hydrofluorocarbon refrigerants (HFCs) and natural hydrocarbon refrigerants are currently being investigated, computer models can play an important role in the selection process by minimising the cost and time needed for experimental investigations.

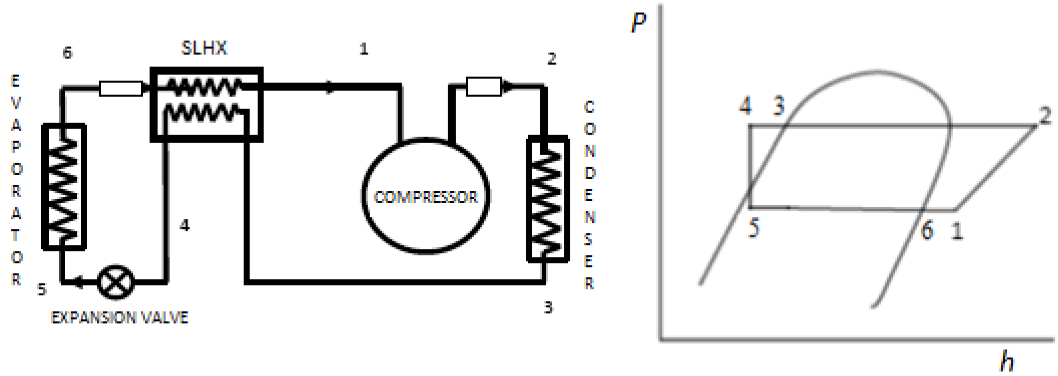
Energy-efficiency of the refrigerant, as measured by its coefficient of performance (COP), is an important consideration in the selection of future alternative refrigerants. In this respect, some alternative refrigerants might require some modification or optimisation of the systems in order to reach a performance which is comparable to that of refrigerant R-22 for air-conditioning applications. For example, depending on the physical characteristics of the particular refrigerant, adding a suction-line heat exchanger (SLHX) may improve the performance of certain refrigerants and, therefore, can be economically justified. Computer analyses of refrigeration cycles can be particularly useful for investigating the effect of adding SLHXs on the performance of candidate alternatives. Apart from energy-efficiency, there are other practical considerations for selecting the suitable refrigerant such as its compatibility with low-cost compressor lubricants.

A review of the literature shows that most theoretical analyses of the vapour-compression refrigeration (VCR) cycle did not take into consideration the effects on the refrigerants' performance that result from adding a suction-line heat exchanger [1-8]. Although some studies considered the effects of subcooling and superheating, they prescribed the degrees of both subcooling and superheating rather than determining these from energy balance over the suction-line heat exchanger [9-14]. Some of the computer studies that addressed the issue of suction-line heat exchangers also fixed both the degree of subcooling and the degree of superheating [15,16]. Such studies did not take into consideration the differences in thermal properties of the refrigerants. Sunardi et al [17], who took these differences into consideration, considered only one possible alternative to R-22 in their study, which is R-290.

The present paper presents a computer model for the VCR cycle with a suction-line heat exchanger that takes into consideration the difference in thermal properties of the various alternative refrigerants. The computer model, that adopts Microsoft Excel as a computational platform, determines the refrigerants properties by using the Thermax add-in [18]. The paper first verifies the model by comparing it's results with published results of previous analyses that used the more established REFPROP and CoolPack software. The model is then used to analyse the performance of six alternative refrigerants to R-22, which are R-134a, R-152a, R-407C, R-410A, R-290, and R-606a, at fixed evaporating and condensing temperatures but various degrees of subcooling

II. Analytical Model For The VCR System With A Suction-Line Heat-Exchanger

Figure 1.a shows a VCR system that consists of five basic components: an evaporator, compressor, condenser, throttling valve and a suction-line heat exchanger (SLHX). After the refrigerant absorbs heat in the evaporator to vaporise, it is passed through the SLHX where it is superheated by cooling the hot liquid refrigerant leaving the condenser. In the compressor, the refrigerant is compressed to the condenser pressure. The refrigerant leaving the compressor enters the condenser where it is completely condensed by rejecting heat to the surrounding environment. The high-pressure liquid refrigerant leaving the condenser passes through the SLHX before being throttled in the expansion valve which reduces its pressure and temperature to those of the evaporator. Subcooling the refrigerant after the condenser increases the refrigeration effect while superheating it after the evaporator protects the compressor by removing liquid refrigerant droplets. Figure 1.b shows the P-h diagram for the cycle.



(a) (b)

Figure 1. Schematic and *P-h* diagrams of the VCR cycle with a suction-line heat exchanger (Adapted from Venkataiah and Rao [15])

The present analytical model neglects pressure losses in the evaporator, condenser, and SLHX, but takes into consideration the adiabatic efficiency of the compressor. It is assumed that the refrigerant leaves the evaporator as dry saturated vapour at the evaporator pressure (point 6) to enter the SLHX, where it is superheated to T_1 before it enters the compressor. Enthalpy of the refrigerant after the adiabatic compression process (h_2) is obtained from:

$$h_2 = h_1 + (h_{2s} - h_1)/\eta_c \tag{1}$$

Where h_{2s} is the enthalpy of the refrigerant after an ideal isentropic compression process and η_c is the isentropic efficiency of the compressor. In order to account for the effect of pressure ratios of the different refrigerants on the compressor efficiency, the following relationship is adopted from Sunardi et al [17]:

$$\eta_c = 0.874 - 0.0135\frac{P_2}{P_1} \tag{2}$$

It is assumed that the amount of heat rejected in the condenser (Q_H) is such that the refrigerant leaves the condenser as saturated liquid at the condenser pressure (point 3) before entering the SLHX. Conservation of mass and energy between two streams passing through the heat-exchanger leads to:

$$h_1 = h_6 + (h_3 - h_4) \tag{3}$$

The liquid refrigerant leaving the SLHX passes through the adiabatic throttling valve and then enters the evaporator as a saturated liquid-vapour mixture at state 5. The refrigerant absorbs the amount of heat it needs to vaporise (Q_L) in the evaporator. It is a common practice now that refrigeration systems are designed such that some degrees of subcooling and superheating are achieved in the condenser and evaporator, respectively. Depending on whether there is a SLHX or not, the refrigerating effect (Q_L) can be determined from:

$$Q_L = \dot{m}(h_1 - h_4) \quad \text{without SLHX} \tag{4.a}$$
$$= \dot{m}(h_1 - h_5) \quad \text{with SLHX} \tag{4.b}$$

Similarly, the condenser duty (Q_H) is given by:

$$Q_H = \dot{m}(h_2 - h_3) \qquad \text{without SLHX} \qquad (5.a)$$
$$= \dot{m}(h_2 - h_4) \qquad \text{with SLHX} \qquad (5.b)$$

The coefficient of performance (COP) of the refrigeration system is defined as:

$$\text{COP} = \frac{\text{Rate of heat removal from the evaporater}}{\text{Rate of work-input to the compressor}} = \frac{Q_L}{W_c} \qquad (6)$$

Where W_c is the compressor work input, which is given by:

$$W_c = \dot{m}(h_2 - h_1) \qquad (7)$$

While subcooling the refrigerant after the condenser increases the refrigeration effect (Q_L), superheating it after the evaporator increases the compressor work (W_c).

III. Validation Of The Computer Model

The computer model developed for the present study to compare the performance of various future substitute refrigerants to that of R-22 uses Microsoft Excel as a computational platform. Since Excel does not have built-in functions for determining the thermodynamic properties of the various refrigerants considered in the study, these are determined by using the Thermax add-in [18]. Thermax, which supports the 32 refrigerants listed in ASHRAE [19], determines the properties of saturated and subcooled refrigerants by interpolating the tabulated data at a given pressure or temperature. Enthalpies and entropies of superheated refrigerants are determined by using the formulae described by El-Awad [20] that apply ideal-gas relationships to calculate these properties from their values as saturated vapours at the corresponding pressure or temperature. Specific volumes of superheated refrigerants are calculated from the Soave-Redlich-Kwong equation of state.

To validate the Excel-Thermax model, its estimations were compared with the results obtained by Pavani et al [14] who analysed the performance of the simple ideal VCR cycle shown in Figure 2. In this cycle, the refrigerant absorbs heat in the evaporator to become a dry vapour (point 1). It is then compressed by the compressor to the condenser pressure (point 2). Since the compression process is both adiabatic and reversible, it is an isentropic process. The refrigerant leaves the condenser as saturated liquid (point 3). Ideally, both vaporisation and condensation are reversible processes that occur at constant pressures.

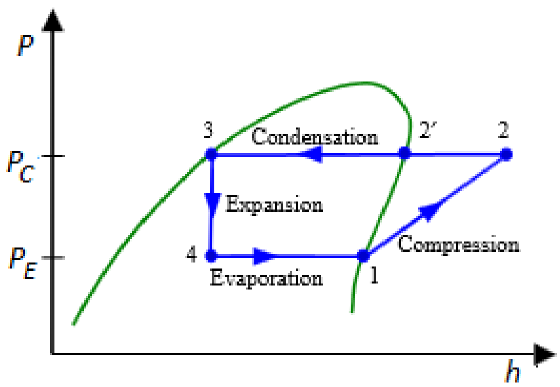


Figure 2. Ideal vapour-compression refrigeration cycle on a *P-h* diagram (adapted from Bolaji et al [1])

Pavani et al [14] analysed the performance of the simple ideal VCR cycle with R-404A, R-507A, and R-410A as alternative refrigerants to R-22 at various evaporator and condenser pressures. They used REFPROP [21] to determine the thermodynamic properties of the refrigerants and provided numerical data from their analyses for the performance of R-404A with a constant condenser pressure of 1.182 MPa and evaporator pressures in the range 164 – 364 kPa. Table 1 compares estimates of the present model with those reported by Pavani et al [14] for the evaporator temperature and the cycle's COP. The figures in the table show that estimates of the present model for the evaporator temperature deviated by less than 0.3^{o}C while values of the COP deviated by less than 0.6%.

The compressor discharge temperature (T_2) is frequently obtained by using the following approximate constant-specific heat method:

$$T_2 = T_1 \times \left(\frac{P_2}{P_1}\right)^{(k-1)/k} \tag{8}$$

Where k is the ratio of specific heats: $k = c_p/c_v$. Thermax function that determines the refrigerant's temperature at the compressor discharge is based on the following relationship [20]:

$$T_2 = T_C + \frac{(h_2 - h_g)}{Cp_g} \tag{9}$$

Where T_C is the refrigerant's saturation temperature at the condenser pressure and h_g and C_{pg}, respectively, are the corresponding values of the enthalpy and specific heat of the refrigerant as saturated vapour. Table 2 compares estimates of the discharge temperature (T_2) as calculated by Equation (8) and Equation (9) with the values given by Pavani et al [14] for R-404A. In applying Equation (8), values of the ratio of specific heats $k = c_p/c_v$ were taken from ASHRAE [19] as saturated vapour at T_1. While the temperatures calculated by Equation (8) deviates by nearly 14^{o}C, those calculated by Equation (9) deviated by less than 0.5^{o}C.

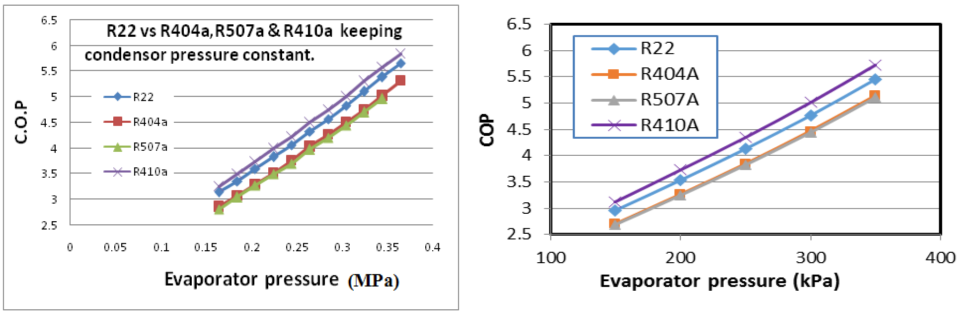
Table 1. Comparison of the evaporator temperature and cycle COP obtained by the present model with their corresponding values reported by Pavani et al [14]

P_E	Evaporator temperature (T_E)			COP		
	Ref [14]	Model	Deviation %	Ref [14]	Model	Deviation %
164	-35.95	-35.65	-0.294	2.859	2.861	0.054641
184	-33.24	-32.946	-0.288	3.071	3.083	0.37705
204	-30.75	-30.462	-0.284	3.297	3.307	0.308904
224	-28.44	-28.156	-0.28	3.524	3.537	0.37329
244	-26.29	-26.01	-0.284	3.763	3.771	0.206469
264	-24.27	-23.986	-0.276	4.031	4.012	-0.46616
284	-22.36	-22.084	-0.276	4.269	4.260	-0.21195
304	-20.55	-20.274	-0.27	4.512	4.515	0.076875
324	-18.83	-18.56	-0.266	4.756	4.778	0.460368
344	-17.19	-16.924	-0.262	5.024	5.053	0.572834
364	-15.62	-15.358	-0.294	5.319	5.338	0.360987

Table 2. Comparison of the values obtained by the present model for the compressor discharge temperature with the values reported by Pavani et al [14]

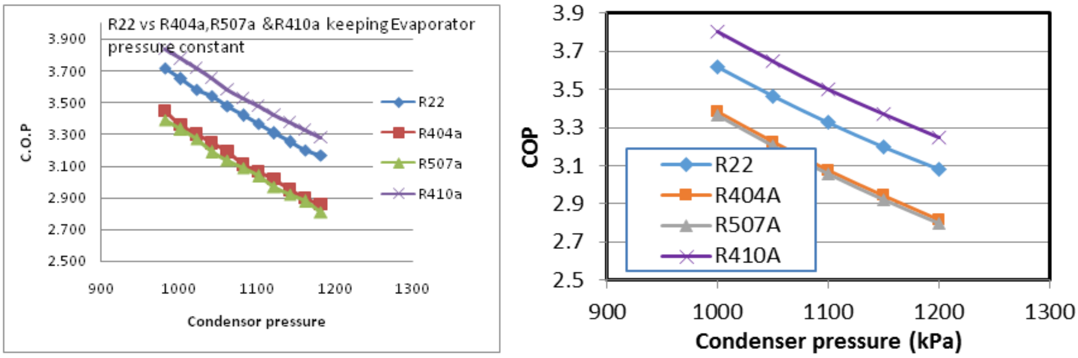
P_E	Ref [14] ^{o}C	Thermax function		Equation (8)	
		^{o}C	Deviation ^{o}C	^{o}C	Deviation ^{o}C
164	31.005	31.363	-0.358	45.342	13.979
184	30.489	30.724	-0.235	44.703	13.979
204	29.974	30.177	-0.203	43.919	13.742
224	29.459	29.687	-0.228	43.539	13.852
244	28.946	29.264	-0.318	43.190	13.926
264	28.433	28.871	-0.438	42.813	13.942
284	28.177	28.524	-0.347	42.261	13.737
304	27.921	28.207	-0.286	41.996	13.789
324	27.666	27.931	-0.265	41.726	13.795
344	27.411	27.660	-0.249	41.458	13.798
364	27.861	27.414	0.447	41.189	13.775

Figure 3 and Figure 4 compare the cycle COP as determined by the present model with those obtained by Pavani et al [14] for R-22 and the three alternative refrigerants, R-404A, R-507A, and R-410A, at various condenser and evaporator pressures. The figures show that the model estimated the COP variation for each refrigerant, as well as the relative COP magnitudes for the four refrigerants, with good accuracy. The results show that R-410A has higher COP than R-22 as well as the other two alternative refrigerants.



(a) (b)

Figure 3. Variation of COP with evaporator pressure: (a) Pavani et al [14], (b) present model



(a) (b)

Figure 4. Variation of COP with condenser pressure: (a) Pavani et al [14], (b) present model

Venkataiah and Rao [15] analysed an air-conditioning system with 1.5 ton (5.276 kW) refrigeration capacity using refrigerant R-22. The evaporator temperature was 7.2°C and condenser temperature was 40.5°C. The cycle had 6°C of subcooling and 8°C of superheating, and the compressor adiabatic efficiency was 85.0%. The same cycle was analysed by the present model and the results are compared in Table 3 with those obtained by Venkataiah and Rao [15] who used CoolPack [22] in their analysis. The table compares the values obtained for the rate of heat rejection ($\dot{Q}_H$), mass flow rate of refrigerant ($\dot{m}$), compressor power ($\dot{W}_c$), compressor discharge temperature (T_2), quality of the refrigerant entering the evaporator (x_5), and the cycle's COP. The figures show close agreements between the values obtained by the present model and those obtained by Venkataiah and Rao [15] for all the parameters except for the compressor discharge temperature which deviates from their value by about 2.5°C.

Table 3. Comparison with the analysis by Venkataiah and Rao [15] for R-22

	Ref [15]	Present model	Deviation
$\dot{Q}_H$ [kW]	6.127	6.123889	-0.051 (%)
$\dot{m}$ [kg/s]	0.03081	0.030788	-0.070 (%)
$\dot{W}_c$ [kW]	0.851	0.8479	-0.366 (%)
T_2	68.0	65.4994	-2.5 °C
x_5	0.17	0.1703	0.172 (%)
COP	6.2	6.2225	0.363 (%)

IV. Performance of R-22 Alternatives with Suction-Line Heat Exchangers

A suction-line heat exchanger increases both the refrigeration effect and the compressor work and, therefore, its net effect on the COP depends on the characteristics of the particular refrigerant being used. For the present analyses of the effect of SLHXs on the performance of R-22 alternatives, six candidate refrigerants were selected, which are R-134a, R-152a, R-407C, R-410A, R-290, and R-600a. The six refrigerants have been identified by previous analyses as good alternatives to R-22 in air-conditioning systems [5,8,10,13-17]. The air-conditioning system considered for the analysis has a cooling capacity of 1.5 ton (5.276 kW). The analyses were done for an evaporator temperature of 0°C and condenser temperature of 55°C. The degree of subcooling was varied from 2°C to 10°C The degree of superheating for each refrigerant, which depends on the refrigerant's enthalpies at the liquid and vapour phases, was determined from the energy balance equation (Equation 3). For each refrigerant, the study determined the mass flow rate, volume flow rate at the compressor inlet, compressor power and discharge temperature, rate of heat rejection in the condenser, and COP of the system.

Figure 5 shows the mass flow rates for the six alternative refrigerants compared to that of R-22. The figure indicates that the mass flow rates of all refrigerants decrease as the degree of subcooling increases. The two HC refrigerants, R-290 and R-600a, require considerably lower flow rates (≈40%) compared to that of R-22 followed by R-152a (≈35%), while refrigerant R-134a requires a slightly higher mass flow rate (≈10%). The closest alternative refrigerants to R-22 in this respect are R-407C and R-410A, which require about 5% more flow rate over the entire range of subcooling degrees. Figure 6, which shows the volume flow rates at the compressor inlet (state 1 in Figure 1) indicates that it is almost constant for all refrigerants over the full range of subcooling degrees. This is due to the fact that increasing the degree of subcooling also increases the degree of superheating at the compressor inlet and, therefore, the specific volume. Since the mass flow rate decreases with the degree of subcooling, the volume flow rate ($\dot{V} = \dot{m}v$) remains almost constant. Note that R-134a and R-152a have nearly identical volume flow rates, which are about 70% higher than that of R-22 over the entire range of subcooling degrees. While the volume rate of R-290 is only about 20% higher than that of refrigerant R-22, that of R-600a is twice that of R-22, which is the highest volume flow rate among the six alternative refrigerants. Refrigerants R-407C and R-410A are also the closest alternatives to R-22 in this respect

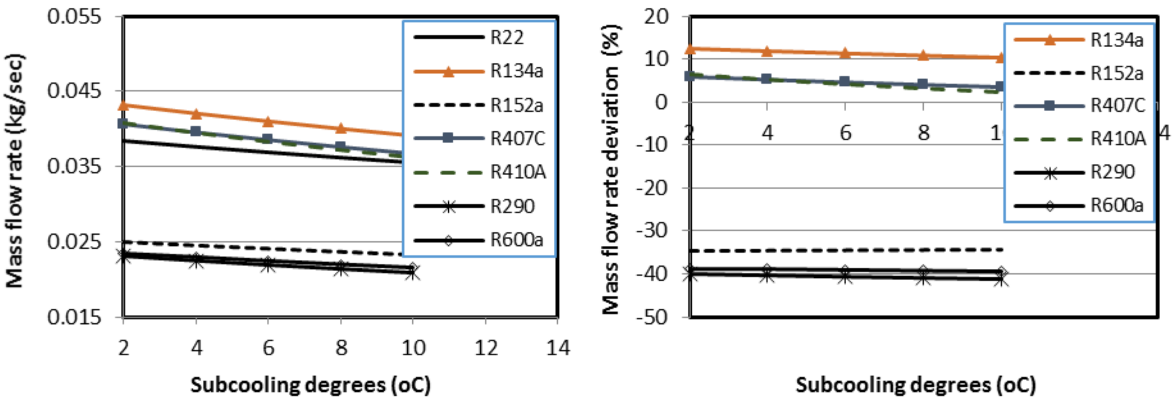


Figure 5. Variation of refrigerants' mass flow rates with subcooling degree

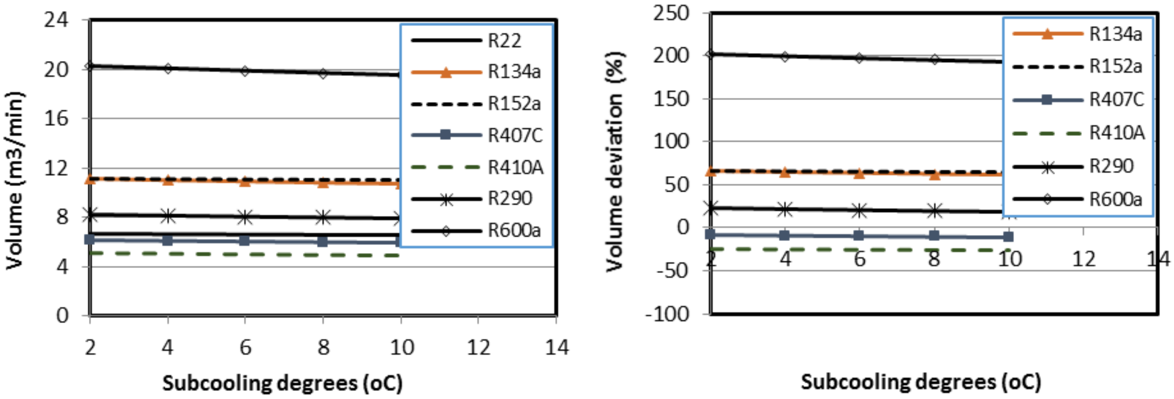


Figure 6. Variation of refrigerants' volume flow rates at the compressor inlet with subcooling degree

As shown in Figure 7, the compressor power slightly decreases as the degree of subcooling increases for all seven refrigerants. Although increasing the degree of subcooling also increases the degree of superheating and, therefore, the specific volume flow rate at the compressor inlet, it must be recalled from Figures 5 and 6 that the mass flow rate actually decreases with subcooling degree while the volume flow rate remains almost fixed. This explains why the compressor work decreases slightly with degree of superheating as shown in Figure 7. While R-410A required about 10% higher compressor power compared to R-22, R-152a required 4% less power. Refrigerants R-407C, R-134a, R-600a, and R-290 showed similar power requirements to that of R-22, but the refrigerant with the nearest compressor power to that of R-22 is R-407C.

Figure 8 shows the variation of heat rejection rate with the degree of subcooling for R-22 and its alternative refrigerants. It should be noted that the trend of this parameter is almost identical to that of the compressor power shown in Figure 7. The rates of heat rejection by five alternative refrigerants are within 1% of that given by R-22; which are R-134a, R-152a, R-407C, R-290, and R-600a. While R-410A leads to higher rate of heat rejection compared to R-22, refrigerants R-407C and R-600a give lower rates of heat rejection. The alternative refrigerant with the nearest rates of heat rejection to R-22 is R-407C.

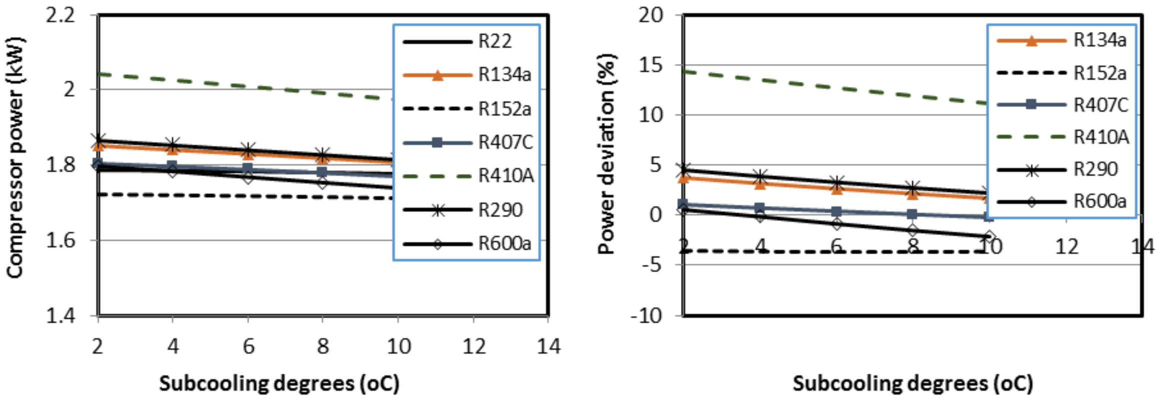


Figure 7. Variation of compressor power with subcooling degree

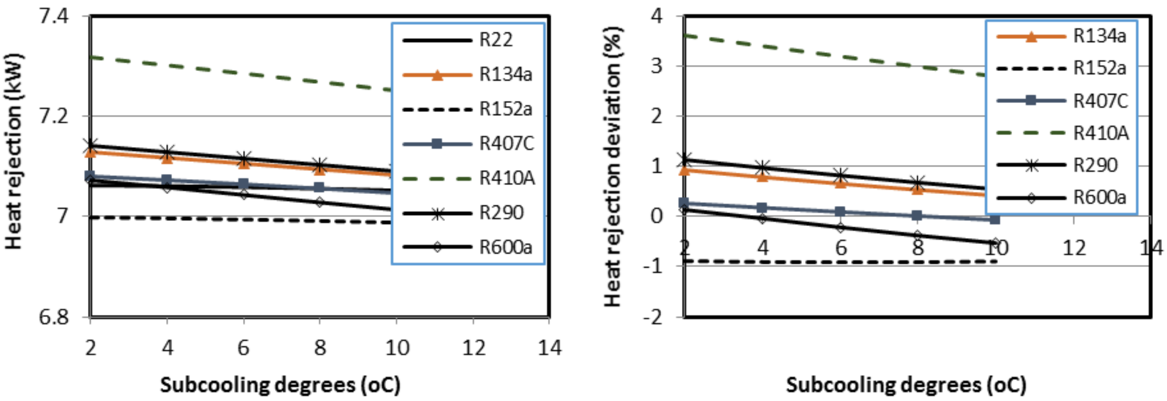


Figure 8. Variation of rate of heat rejection with subcooling degree

Variation of the compressor discharge temperature with the degree of subcooling is shown in Figure 9. The figure shows that subcooling increases the compressor discharge temperature for R-22 and all the six alternative refrigerants. However, all alternative refrigerants produced lower discharge temperatures compared to R-22. The figure also indicates that subcooling leads to discharge temperatures higher than 90^{o}C when it exceeds 4^{o}C for R-22 and when it exceeds 6^{o}C for R-152a. The compressor discharge temperatures for refrigerants R-134a, R-407C, R-410A, R-290, and R-600a remain lower than 90^{o}C for all degrees of subcooling in the range considered, with R-600a giving the lowest discharge temperature. The present results agree well with those of Sunardi et al [17] for R-22 and R-290. Figure 10 shows that the COPs for R-22 and all its alternative refrigerants increase with the increase in degree of subcooling, but R-152a gives the highest COP, which is about 4% higher than that of R-22 over the entire range of subcooling. Both R-134a and R-290 produce slightly lower COPs compared to R-22, but the lowest COP is that of R-10A, which is about 10% lower than that of R-22. While both R-600a and R-407C produce comparable COPs to that of R-22, the nearest alternative to R-22 in this respect is R-407C.

Table 4 summarises the findings of the present study on the effects of adding suction-line heat exchangers on the performance of alternative refrigerants to R-22. For each of the six performance parameters shown in Figures 5 to 10, if the effect is favourable, e.g. a lower mass flow rate or a higher COP, the relevant cell is given a positive (+) sign and if the change is unfavourable, e.g. a larger volume flow rate or a higher discharge temperature, the cell is given a negative (-) sign. Since all alternative refrigerants produced less compressor discharge temperatures than that of R-22, all the cells for this parameter have "+" signs. The last column shows the net result for each alternative refrigerant as the number of "+"or "–" signs that remain after equal numbers of "+" signs and "-" signs cancel out. From the table, adding a SLHX with both R-407C and R-600a leads to favourable effect in four cycle parameters compared to R-22. The third refrigerant for which a SLHX produces favourable effects is R-152a, which has two "+" signs. For the remaining three refrigerants, which are R-134a, R-410A, and R-290, the net effect of adding SLHXs is unfavourable.

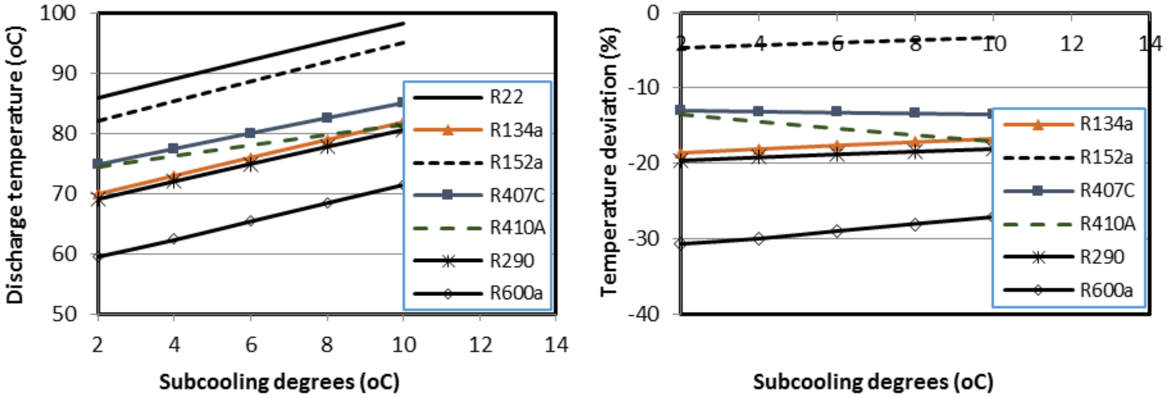


Figure 9. Variation of compressor discharge temperature with subcooling degree

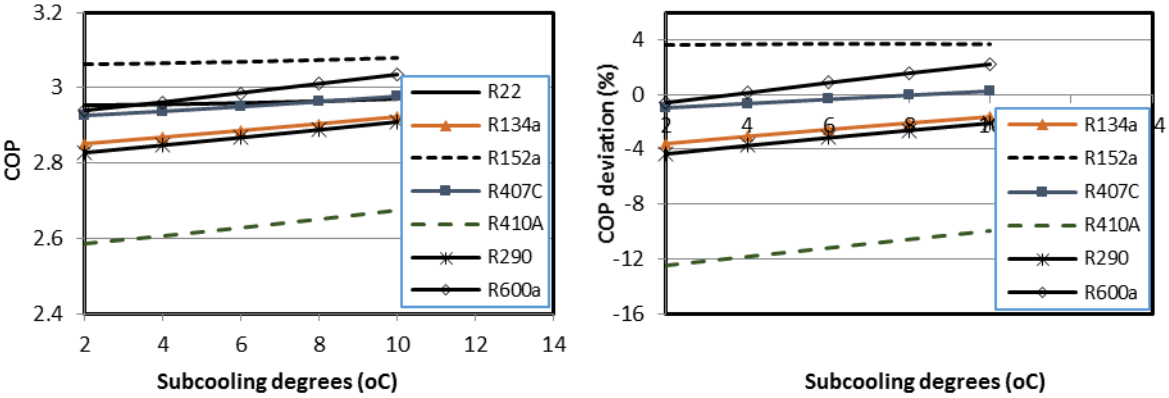


Figure 10. Variation of refrigerants' COP with subcooling degree

Table 4. Merits of alternative refrigerants compared to R-22

	Mass flow rate	Volume flow rate	Power	Q_{out}	T_2	COP	Net effect
R-134a	-	-	-	-	+	+	2-
R-152a	+	-	+	+	+	-	2+
R-407C	-	+	+	+	+	+	4+
R-410A	-	+	-	-	+	-	2-
R-290	+	-	-	-	+	-	2-
R-600a	+	-	+	+	+	+	4+

V. Conclusion

Based on the results of the present analysis, three refrigerants emerge as suitable alternatives to R-22, which are R-152a, R-407C, and R-600a. These three refrigerants should be scrutinised further on the basis of other practical factors such as the flammability of the refrigerant and its compatibility with low-cost compressor lubricants. For the other three refrigerants, R-134a, R-410A and R-290, the study shows that adding a SLHX will not improve the refrigerant's performance and, therefore, cannot be justified. The Excel-based computer model presented in this paper can also be used to analyse the effects of other factors related to VCR systems and alternative refrigerants. For example, the model can be used to analyse the performance of cascade and multi-stage refrigeration systems. The Thermax add-in used by the model for determining fluid properties supports all 32 refrigerants listed in ASHRAE [19]

Acknowledgements

The author is indebted to the Ministry of Higher Education (Oman) and to the College of Applied Sciences at Sohar for their support without which this work would not have been possible.

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