EXPERIMENTAL INVESTIGATIONS WITH ECO-FRIENDLY REFRIGERANTS USING DESIGN OF EXPERIMENTS TECHNIQUE
Mathematical Modeling and Experimental Validation

by

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In this paper mathematical models were developed using design of experiments technique for the performance prediction of refrigeration system parameters such as refrigerating capacity, power consumption, and coefficient of performance. The models developed were checked for their adequacy using F-test. The performances of vapor compression refrigeration system with different refrigerants R12, R134a, and R290/R600a were compared. The R290/R600a mixture showed 10.7-23.6% higher coefficient of performance than that with R12 and R134a and it was found that the hydrocarbon mixture with 68% propane and 32% iso-butane could be used as a substitute for R12 and R134a.

Keywords: *design of experiments, mathematical model, hydrocarbonmixture, R134a, global warming potential, refrigeration, COP, power consumption*

Introduction

The chlorofluorocarbons (CFC) and hydrofluorocarbons (HFC) both used as working fluids in refrigeration and heat pump applications need to be replaced due to their influence in the depletion of the stratospheric ozone layers and climate change. CFC are already banned in most countries in 1996 because of their high ozone depletion potential (ODP) and global warming potential (GWP). R134a has been developed as an alternative refrigerant to R12 [1]. The refrigerant R134a was found to cause climate change because of its high global warming potential. The Montreal Protocol and the Kyoto Protocol have been subscribed to call for complete phasing out of CFC and HFC. The use of natural fluids as refrigerants has attracted renewed interest during the last decade. Hydrocarbon (HC) fluids are generally considered as environmentally benign. They have zero ODP and negligible GWP. There are many studies and publications on HC as substitutes for R12. HC refrigerants have several positive characteristics such as zero ODP, very low GWP, non-toxicity, high miscibility with mineral oil, good compatibility with the materials usually employed in refrigerating systems. The main disadvantage of using HC as refrigerant is their flammability [2, 3]. If safety measures are taken to prevent refrigerant leakage from the system then a flammable refrigerant could be as safe as other refrigerants.

The thermodynamic properties of R12, R134a, propane, n-butane and iso-butane are given in tab. 1. From the table it is evident that the thermodynamic properties of pure HC

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Table 1.Thermodynamic property of refrigerants

Refrigerants	Molecular weight [gmol^{-1}]	Boiling point [°C]	Freezing point [°C]	Critical temperature [°C]	Critical pressure [MPa]	Latent heat [kJkg^{-1}]
R12	120.9	–29.8	–158	112	4.14	166.2
R134a	102	–26.1	–104	101.1	4.06	217
R290	44.1	–42.1	–188	96.7	4.25	421.4
R600	58.12	–0.51	–139	152	3.79	386
R600a	58.12	–11.7	–160	134.7	3.64	364.4

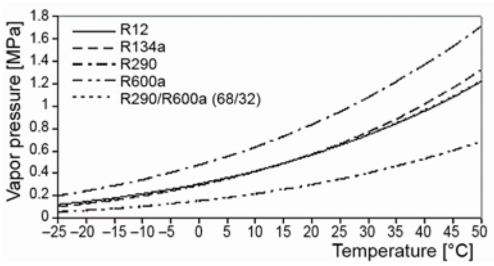

Figure 1. Vapor pressure curves of refrigerants

do not match with that of R12 and R134a. The variation of saturated vapor pressure with temperature of R12, R134a, propane, iso-butane, and a propane/iso-butane mixture is depicted in fig.1. From the figure it is clear that the R290/R600a (68/32 by wt.%) mixture could be used as a potential retrofit refrigerant instead of R12 and R134a as the vapor pressure curve of the mixture is very close to that of R12 and R134a. The thermodynamic properties of the refrigerants were taken from the NIST REFPROP database [4].

The R290/R600a is a zeotropic refrigerant mixture.A zeotropic refrigerant [5, 6] needs a temperature range to condense or to evaporate at a given pressure, with the dew point temperature always higher than the corresponding bubble point temperature. This leads to a shift in the composition of the phase changing mixture. The temperature glide of R290/R600a mixture is 7.6 °C at 101 kPa. The refrigerant 290/R600a mixture was charged in the liquid state to assure proper mixture**.** The specific volume of HC mixture is more than that of CFC and HFC [7]. Therefore the amount of HC refrigerant charged was approximately one-third of CFC refrigerants [8].

Literature review

Many studies have been concentrated on the research of substitutes for R12. Some of the investigations are Richardson and Butterworth [9], Jung *et al.* [10], Kuijpers *et al.* [11], Hammad and Alsaad [12], Jung *et al.* [13], Akash and Said [14], and Fatouh and Kafafy [15]. Recently Dalkilic and Wongwises [16] studied the performance of refrigerants HFC134a, HFC152a, HFC32, HC290, HC1270, HC600, and HC600a for different ratios in a traditional vapor compression refrigeration system and compared with CFC12, CFC22, and HFC134a. Their results showed that refrigerant blend of HC290/HC600a (40/60 by wt.% instead of CFC12 found to be replacement refrigerant among other alternatives. Sarkar and Bhattacharyya [17] studied the performance of heat pump using blends of R744/R600 and R744/R600a as working fluids. Mohanraj *et al.* [18] experimented with R290/R600a in the ratio of 45.2:54.8 by weight as an alternative to R134a in a domestic refrigerator. The results reported that the above mixture could be the best long term alternative to phase out R134a. Ari and Aryadi [19] discussed the possibility to use HC based refrigerants in air conditioning and refrigeration sectors. In the present investigation the proposed R290/R600a mixture is a HC blend composed of propane 68% and iso-butane 32% on mass basis and performed better than the other propane/iso-butane mixtures so far studied by various investigators.

Design of experiments

Design of experiments (DOE) is a standard statistical technique used to identify factors and levels that have the most and least impact on system performance [20]. The statistical analysis of the results allows the determination of the significance of the results and to obtain a mathematical equation that relates the variables and the results. The DOE technique is used in many fields such as welding, grinding, machining, *etc.*In this work, a new environmentally friendly alternative refrigerant was proposed and comparison of its performance with R12, R134a was carried out using design of experiment technique to prove its potential as a promising substitute. An attempt has been made to develop a mathematical model to predict and compare the performance of the refrigeration system parameters. Design of experiments technique reduces the number of experiments to be conducted.

Experimental set-up and procedure

An experimental set-up of a vapor compression refrigeration system was built to investigate the performance of R12, R134a, and R290/R600a mixture. A schematic diagram of the experimental set-up is shown in fig. 2, which consists of two loops; a main loop and a secondary loop. The main loop is composed of compressor, condenser, a filter-drier, refrigerant flow meter, sight glass, expansion valve, and evaporator. The compressor is an open, reciprocating type. The compressor speed could be changed by a variable diameter belt pulley of the electrical motor. The condenser and evaporator are made of copper double tubes. In the double tube condenser, the refrigerant flows through the inner tube while the cooling water flows through the annular space between the inner and outer tubes. In the double tube evaporator, the brine solution (calcium chloride/water solution) flows through the inner tube and the refrigerant flows through the annular space between the inner and outer tube. For minimizing the heat loss, the outer tube is well insulated. Two sight glasses are incorporated into the system, one in the liquid line at the condenser outlet and another in the vapor line at the evaporator outlet in order to give a visual indication of the refrigerant circulation. The secondary loops are composed of a pump, a flow meter and an electrically heated unit within the insu-

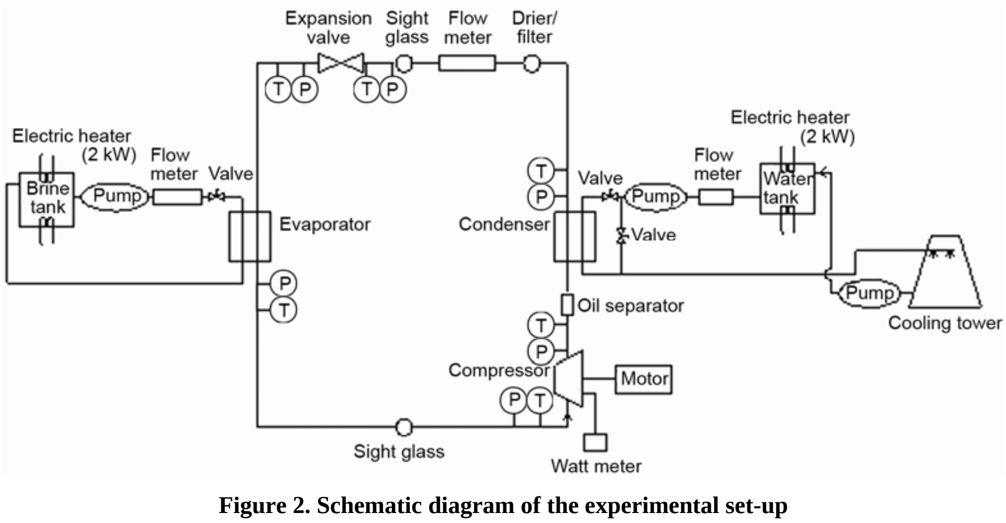

Figure 2. Schematic diagram of the experimental set-up

lated tank. One tank is filled with cooling water and circulated through the condenser tubes while the other tank is filled with brine solution and circulated through the evaporator tubes. The hot water coming out of the condenser tube is supplied to a cooling tower to be cooled. This cooled water is pumped to the cooling water tank *via* a separate pump.

Rotameters were used to measure the flow rates of the cooling water and brine solution with an accuracy of ±0.05 lpm. The refrigerant flow meter was used to measure the refrigerant flow rate with an accuracy of ±0.0125kg/min. RTD type thermocouples were used to measure the temperatures with an accuracy of ±0.1 °C and pressures were measured using calibrated pressure gauges with an accuracy of ±0.00689MPa.

The temperatures and pressures (T and P) of the refrigerant and secondary fluid temperatures were measured at various locations in the experimental set-up as shown in fig. 2. The compressor power consumption was measured using a wattmeter. The compressor power consumption was measured using a wattmeter with an accuracy of ±0.01kWh. An expansion device was used to regulate the mass flow rate of refrigerant and to set pressure difference. The refrigerant was charged after the system had been evacuated. Drop-in experiments were carried out without any modifications of the experimental set-up. The experiment was started with R12 to set up the base reference for further comparisons with the other two refrigerants. The desired evaporating and condensing temperatures were obtained by adjusting all other parameters in the system such as cooling water flow rate and its temperature, refrigerant flow rate, brine solution flow rate and its temperature. The readings were taken after the system had reached steady state conditions and all observed values were recorded.

Development of mathematical models

The independent controllable parameters such as evaporating temperature (T_c), condensing temperature (T_e) and compressor speed (N) were identified to carry out the experimental work. The upper limit of a parameter was coded as +1.682 and the lower limit as –1.682. The coded values for the intermediate values which were taken as levels are calculated from the relationship:

$$X_i = 1.682 \frac{2X - (X_{\max} + X_{\min})}{X_{\max} + X_{\min}} \tag{1}$$

where X_i is the required coded value of a variable X. X is any value of the variable from $X_{\min}$ to $X_{\max}$. $X_{\min}$ and $X_{\max}$ are the lower and upper limits of the variable X, respectively.

The decided levels of the selected parameters with their units and notations are given in tab. 2. The design matrix was developed consisting of three factors and five levels central composite rotatable design [21]. It composed of a full replication of $2^3 = 8$ factorial design plus 6 center points and 6 star points. The selected design matrix is shown in tab. 3. The experiments were conducted according to the design matrix at random to avoid systematic errors creeping into the system. The data obtained from these experiments were used to develop the mathematical models and analyze the performance of the refrigerants R12, R134a and R290/R600a mixture.

The response function representing any of the refrigeration system output can be expressed as:

$$Y = f(T_e, T_c, N) \tag{2}$$

The second order polynomial (regression) equation used to represent the response function for three factors is given by:

$$Y = b_0 + b_1T_e + b_2T_c + b_3N + b_{11}T_e^2 + b_{22}T_c^2 + b_{33}N^2 + \\ + b_{12}T_eT_c + b_{13}T_eN + b_{23}T_cN \qquad (3)$$

Table 2. Control parameters and its levels

Parameters	Unit	Notation	Parameter levels				
			–1.682	–1	0	1	1.682
Evaporatingtemperature	°C	T_e	–18	–14	–8	–2	2
Condensingtemperature	°C	T_c	32	35	40	45	48
Compressorspeed	rpm	N	552	675	855	1035	1157

Table 3. Design matrix and calculated responses of refrigeration system parameters

Sl. no	Design matrix			Responses of refrigerant								
				R12			R134a			R290/R600a		
	T_e	T_c	N	*RC*[1]	*PC*[2]	*COP*	*RC*	*PC*	*COP*	*RC*	*PC*	*COP*
1	–1	–1	–1	0.53	0.34	1.57	0.56	0.39	1.43	0.64	0.4	1.58
2	1	–1	–1	0.74	0.52	1.43	0.67	0.52	1.29	1.15	0.62	1.86
3	–1	1	–1	0.42	0.41	1.04	0.45	0.42	1.07	0.42	0.41	1.03
4	1	1	–1	0.64	0.53	1.19	0.56	0.51	1.1	0.82	0.63	1.3
5	–1	–1	1	1.24	0.51	2.41	1.11	0.51	2.17	1.44	0.48	3.03
6	1	–1	1	1.44	0.67	2.14	1.46	0.7	2.08	1.86	0.79	2.36
7	–1	1	1	1.03	0.53	1.94	0.95	0.52	1.82	1.03	0.52	1.97
8	1	1	1	1.24	0.71	1.74	1.28	0.7	1.82	1.65	0.81	2.03
9	–1.682	0	0	0.72	0.36	1.96	0.58	0.4	1.45	0.72	0.38	1.88
10	1.682	0	0	1.03	0.67	1.54	0.89	0.66	1.36	1.43	0.74	1.94
11	0	–1.682	0	1.03	0.53	1.95	0.87	0.56	1.57	1.27	0.57	2.25
12	0	1.682	0	0.8	0.55	1.44	0.75	0.56	1.33	0.44	0.47	0.94
13	0	0	–1.682	0.35	0.4	0.88	0.28	0.39	0.72	0.53	0.48	1.1
14	0	0	1.682	1.53	0.69	2.22	1.53	0.66	2.34	2.12	0.79	2.68
15	0	0	0	0.87	0.53	1.65	0.85	0.52	1.64	1.11	0.6	1.85
16	0	0	0	0.87	0.53	1.64	0.85	0.52	1.63	1.11	0.6	1.85
17	0	0	0	0.87	0.53	1.65	0.85	0.52	1.62	1.11	0.6	1.84
18	0	0	0	0.87	0.53	1.66	0.85	0.52	1.63	1.11	0.6	1.85
19	0	0	0	0.87	0.54	1.63	0.85	0.52	1.62	1.11	0.61	1.84
20	0	0	0	0.87	0.53	1.64	0.85	0.52	1.64	1.11	0.6	1.85

1 – refrigerating capacity [kW]; 2 – power consumption [kW]

The values of the coefficients of the polynomial were calculated by regression [21] with the help of the eqs. (4)-(7):

$$b_0 = 0.1663\Sigma(Y) - 0.0568\Sigma\Sigma\,(X_{ii}Y) \qquad (4)$$

$$b_i = 0.0732\Sigma(X_iY) \tag{5}$$

$$b_{ii} = 0.0625\Sigma(X_{ii}Y)+0.00689\Sigma\Sigma(X_{ii}Y)-0.0568\Sigma(Y) \tag{6}$$

$$b_{ij} = 0.1250\Sigma(X_{ij}Y) \tag{7}$$

A SYSTAT software package was used to calculate the values of these coefficients for direct responses. The mathematical models with parameters in coded form, developed by the above analysis are represented in eqs. (8)-(16) for refrigerant R12, R134a, and R290/R600a:

– for refrigerant R12

$$RC = 0.875 + 0.1N - 0.075\,T_c + 0.337\,T_e - 0.001N^2 + 0.013\,T_c^2 + 0.023\,T_e^2 - 0.001T_eN + 0.001T_cN - 0.025T_cT_e \tag{8}$$

$$PC = 0.532 + 0.086N + 0.013T_c + 0.082T_e - 0.006N^2 + 0.002T_c^2 + 0.004T_e^2 - 0.004T_cN + 0.004\,T_eN - 0.003\,T_cT_e \tag{9}$$

$$COP = 1.645 - 0.086N - 0.183T_c + 0.385T_e + 0.04N^2 + 0.03T_e^2 - 0.022T_c^2 + 0.044T_cN - 0.059T_eN - 0.014\,T_cT_e \tag{10}$$

– for refrigerant R134a

$$RC = 0.846 + 0.104N - 0.056T_c + 0.342T_e - 0.025N^2 + 0.034T_e^2 + 0.001\,T_c^2 + 0.056T_eN - 0.003T_cN - 0.015T_cT_e \tag{11}$$

$$PC = 0.521 + 0.074N + 0.003T_c + 0.077T_e + 0.013T_c^2 + 0.002N^2 + 0.001T_e^2 - 0.006T_cN + 0.018T_eN - 0.001T_cT_e \tag{12}$$

$$COP = 1.597 - 0.114T_c + 0.418T_e - 0.025N - 0.044N^2 - 0.032T_c^2 - 0.006T_e^2 + 0.032T_cN + 0.003T_eN - 0.007T_cT_e \tag{13}$$

– for refrigerant R290/R600a

$$RC = 1.112 + 0.231N - 0.187T_c + 0.411T_e - 0.081T_c^2 + 0.084T_e^2 - 0.005N^2 + 0.014T_eN + 0.011T_cN - 0.010T_cT_e \tag{14}$$

$$PC = 0.603 + 0.120N + 0.012T_c + 0.077T_e - 0.018N^2 + 0.010T_e^2 - 0.008T_c^2 - 0.002T_cN + 0.020T_eN + 0.006T_cT_e \tag{15}$$

$$COP = 1.842 - 0.370T + 0.459T_e + 0.003N - 0.098T_c^2 + 0.051N^2 + 0.044T_e^2 + 0.089T_cN - 0.146T_eN - 0.035T_cT_e \tag{16}$$

Results and discussion

From the developed mathematical models, the direct effects of the vapor compression refrigeration system parameters on refrigerating capacity, power consumption, and coefficient of performance were obtained and presented in figs.3 to11.

Direct effect of evaporating temperature (with T_c = 40 °C and N = 855 rpm) on refrigerating capacity, power consumption, and coefficient of performance

Figure 3 shows the variation of the *RC* for various evaporating temperatures. It is observed that as T_e increases, the refrigerating capacity of the refrigerants is increased. The *RC* of R290/R600a is 21.3%-26% higher for temperatures above –14 °C and 44% higher at

–18 °C than that with R12.The increasing percentage of propane in the HC mixture reduces the specific volume of the refrigerant vapor results in increase of the refrigerant circulated per unit of time and the *RC*. The *RC* of R134a showed a very close match with R12 for all the operating conditions. This is due to the similar thermophysical properties of R134a to R12 [22]. R290/R600a mixture showed a higher cooling rate than that with R12 and R134a at higher evaporating temperatures.

The relationship between the power consumption and the evaporating temperature of R12, R134a, and R290/R600a mixture is shown in fig. 4. It is found that when the T_e increases, the power consumed by the compressor increases. This is due to the increased mass flow rate of refrigerants at higher evaporator temperatures. The power consumed by the system with R290/R600a mixture is higher by 10.5%-19.3% and 14.2%-21.5% than that with R12 and R134a, respectively. The refrigerant R134a consumed 2.8%-4.3% lesser power than that with R12 for all the operating conditions.

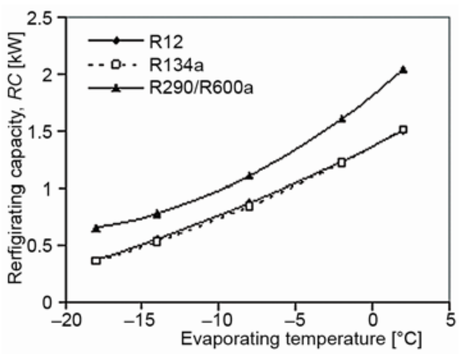

Figure 3. Direct effect of evaporating temperature on *RC*

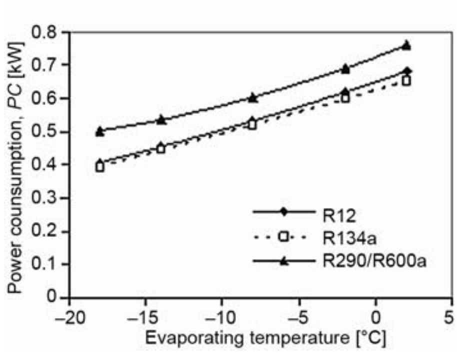

Figure 4. Direct effect of evaporating temperature on *PC*

Figure 5 shows the coefficient of performance for R12, R134a and R290/R600a mixture for various evaporating temperatures. It is observed that the *COP* of R290/R600a mixture is 10.7%-23.6% higher than that of R12 due to the increased refrigerating capacity. The *COP* of R134a closely matches with that of R12.

Direct effect of condensing temperature (with T_e = –8 °C and N = 855 rpm) on refrigerating capacity, power consumption, and coefficient of performance

As T_c increases, the *RC* of the refrigerants decreases as shown in fig.6. It can be seen from the figure that when T_c increases, the *RC*decreased because of the reduced flow rate of the refrigerant. The R290/R600a mixture shows a higher *RC* at T_c = 35 °C and gets reduced at higher condensing temperatures. The *RC* of the R290/R600a mixture is higher for the condensing temperatures below 45 °C and lower at the condensing temperature of 48 °C than that with R12. For all the operating conditions the *RC* of R134a is 4% lower than that with R12.

Figure 7 shows the direct effect of condensing temperature on power consumption. The power consumed by the compressor increases as $T_c$increases. R290/R600a mixture consumed 8%-11.7% more power than R12 and R134a at all the operating conditions. This is due to its higher heat of compression. The power consumed by the system with R134a is very close to that of R12.

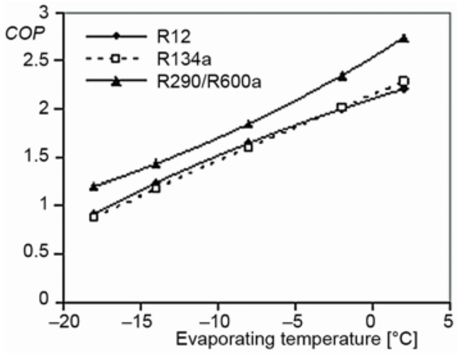

Figure 5. Direct effect of evaporating temperature on *COP*

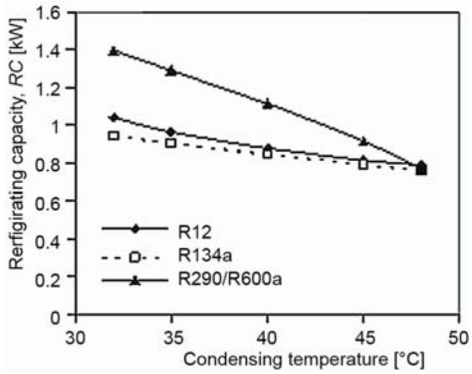

Figure 6. Direct effect of condensing temperature on *RC*

The variation in *COP* as a function of condensing temperature with T_e = –8 °C and N = 855 rpm is shown in fig. 8. It is observed that the *COP* of the refrigerants decreases as the condensing temperature increases. The R290/R600a mixture has higher *COP* at lower T_c and lower *COP* at higher T_c than that with R12 and R134a. The *COP* of R134a is less than that of R12 at lower value of T_c and matches with R12 at the higher condensing temperatures.

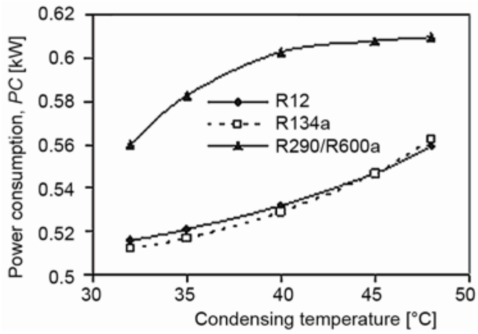

Figure 7. Direct effect of condensing temperature on *PC*

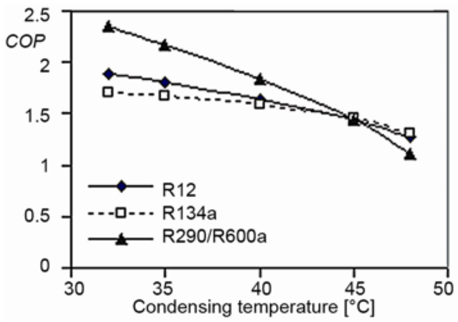

Figure 8. Direct effect of condensing temperature on *COP*

Direct effect of compressor speed (with T_e= –8 °C and T_c= 40 °C) on refrigerating capacity, power consumption, and coefficient of performance

Figure 9 shows the direct effect of N on *RC*. From the figure it is observed that the *RC* increases as the N increases. This is due to the increased refrigerant flow rate at higher N. The *RC* of R290/ /R600a mixture is close to R12 at the lower compressor speed of 552 rpm and increases at a higher rate for the higher N. The *RC* of R134a is slightly lower than that of R12 for all the operating conditions.

Figure 10 shows the variation of power consumed by the compressor at different speeds for R12, R134a, and R290/R600a. As the compressor speed increases, the mass flow rate of the refrigerant increases causing compressor work to increase. R290/R600a mixture consumed 5.7% less power at the lower N of 552 rpm and 12.6% more power at the higher speeds than that of R12. R134a consumed 7.9% more power at the lower compressor speed of 552 rpm and 2.5% lesser power at higher speeds than that of R12.

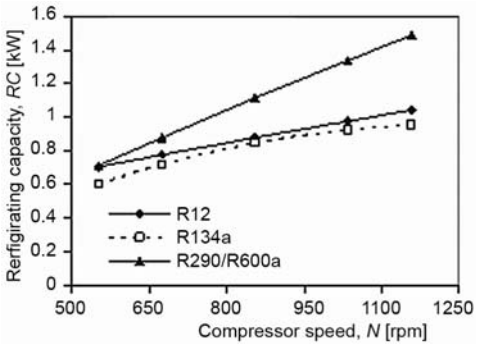

Figure 9. Direct effect of compressor speed on *RC*

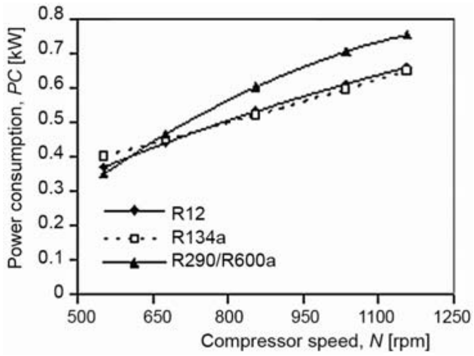

Figure 10. Direct effect of compressor speed on *PC*

Figure 11 shows the variation of *COP* for different *N*. The *COP* of R290/ /R600a mixture varies from 1.99 to 1.84 for all the operating conditions and is higher than that with R12 and R134a. The *COP* of R134a is lesser than that with R12 and decreases as the *N* increases. The *COP* of R134a closely matches with R12 at the compressor speed of 855 rpm.

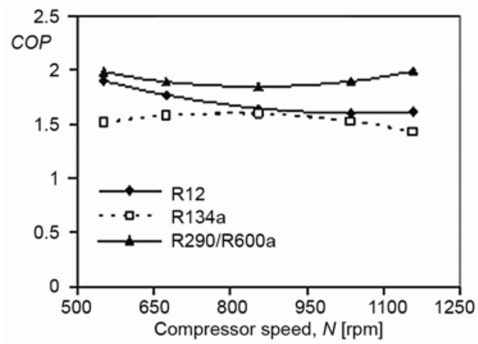

Figure 11. Direct effect of compressor speed on *COP*

Checking the adequacy of the models

The adequacies of the models were tested using the analysis of variance (ANOVA) technique. According to this technique, if the calculated value of *F*-ratio of the model exceeds the standard tabulated value of the *F*-ratio for a desired level of confidence (say 95%), then the model will be considered adequate within the confidence limit. Table 4 shows that all the models were adequate. The square multiple *R* va-

Table 4. Analysis of variance for testing adequacy of models for R12, R134a, and R290/R600a

Refrigerants	System responses	Sum of squares		Degrees of freedom		Mean square		*F*-ratio	Remarks
		regression	residual	regression	residual	regression	residual		
R12	*RC*	1.778	0.002	9	10	0.198	0.000	845.03	adequate
	PC	0.195	0.002	9	10	0.022	0.000	125.65	adequate
	COP	2.671	0.034	9	10	0.297	0.003	86.67	adequate
R134a	*RC*	1.843	0.026	9	10	0.205	0.003	78.89	adequate
	PC	0.161	0.000	9	10	0.018	0.000	687.38	adequate
	COP	2.625	0.128	9	10	0.292	0.013	22.75	adequate
R290/ /R600a	*RC*	3.739	0.094	9	10	0.415	0.009	44.39	adequate
	PC	0.292	0.005	9	10	0.032	0.000	69.18	adequate
	COP	5.208	0.202	9	10	0.579	0.020	28.62	adequate

F-ratio = Mean sum of squares for regression/Mean sum of squares for error; *F*-ratio (9, 10, 0.05) = 3.02

lues, adjusted squared multiple *R* values, and standard error of estimate of the models are presented in tab. 5.

Table 5. Comparison of squared multiple *R*, adjusted squared multiple *R*, and standard error of estimate for the developed models

Refrigerants	Response	Squared multiple *R*	Adjusted squared multiple *R*	Standard error of estimate
R12	*RC*	0.999	0.998	0.015
	PC	0.991	0.983	0.013
	COP	0.987	0.976	0.059
R134a	*RC*	0.986	0.974	0.051
	PC	0.998	0.997	0.005
	COP	0.953	0.912	0.113
R290/R600a	*RC*	0.976	0.954	0.097
	PC	0.984	0.970	0.022
	COP	0.963	0.929	0.142

Validation of models

Conformity tests were conducted using the same experimental set-up to validate the model developed and to determine the accuracy of the models. Table 6 shows the results of conformity test for refrigerant R12, R134a and R290/R600a. From the conformity test it was found that the developed models were able to predict the system responses with reasonable accuracy.

Figures 12-14 depict the scatter diagram of the developed mathematical model, in which the measured and predicted values of *RC, PC,* and *COP* are scattered on both sides as well as closer to 45° line indicating the adequacy of the developed model.

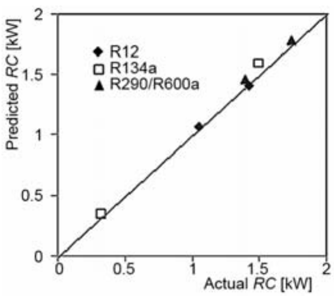

Figure 12. Scatter diagram for refrigerating capacity (*RC*) model

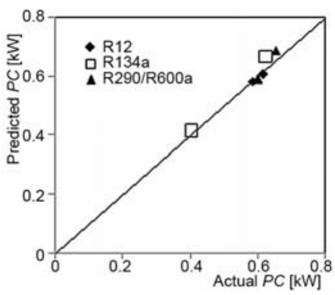

Figure 13. Scatter diagram for power consumption (*PC*) model

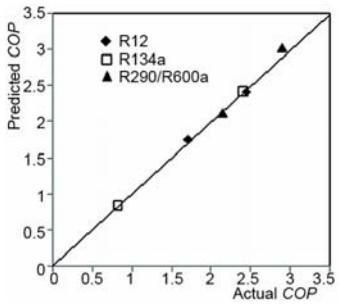

Figure 14. Scatter diagram for coefficient of performance (*COP*) model

Conclusions

The following conclusions are made from the research.

- A five level factorial experimentation technique was employed for developing mathematical models and the performance of R12, R134a, and R290/R600a (68/32 by wt.%) mixture was compared.

- The mathematical models can be used to predict the performance of this particular vapor compression refrigeration system for the range of parameters used in the investigation.
- The refrigerating capacity of R290/R600a (68/32 by wt.%) was 21.3-26% higher at T_e above –14 °C and 44% at T_e = –18 °C than R12 and R134a, respectively.
- The energy consumed by the R290/R600a (68/32 by wt.%) mixture was higher by 10.5--19.3% than R12 and 14.2-21.5% than R134a for the range of evaporating temperatures.
- The *COP* of R290/R600a (68/32 by wt.%) mixture was 10.7-23.6% higher than R12 and the *COP* of R134a was close to that of R12 for the range of evaporating temperatures.
- The investigated hydrocarbon mixture R290/R600a (68/32 by wt.%) could be used as a possible alternative refrigerant for CFC12 and HFC134a.

Table 6. Results of conformity test for refrigerants R12, R134a, and R290/R600a

Refrigerants	System variables			Actual values			Predicted values			Error, [%]		
	T_e	T_c	*N*	*RC*	*PC*	*COP*	*RC*	*PC*	*COP*	*RC*	*PC*	*COP*
R12	–8	35	1035	1.051	0.616	1.706	1.061	0.605	1.753	–0.94	1.82	–2.68
R12	2	40	675	1.428	0.586	2.437	1.407	0.582	2.416	1.49	0.69	0.87
R134a	2	35	855	1.502	0.624	2.408	1.599	0.665	2.405	–6.06	–6.16	0.12
R134a	–18	45	855	0.33	0.406	0.813	0.337	0.412	0.818	–2.08	–1.46	–0.61
R290/R600a	–8	35	1035	1.398	0.653	2.141	1.455	0.687	2.118	–3.92	–4.95	1.09
R290/R600a	2	40	675	1.742	0.602	2.894	1.781	0.589	3.024	–2.19	2.21	–4.29

% Error = [(Actual value – Predicted value)/Predicted value] × 100

Acronyms

ANOVA – analysis of variance
GWP – global warming potential
N – compressor speed
ODP – ozone depletion potential

Subscripts

c – condensing/condenser
e – evaporating/evaporator

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