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Evolving an optimal composition of HFC407C/HC290/HC600a mixture as an alternative to HCFC22 in window air conditioners

D.B. Jabaraj, A. Narendran, D. Mohan Lal *, S. Renganarayanan

Department of Mechanical Engineering, Anna University, Sardar Patel Road, Chennai 600 025, India

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Abstract

Countries that have ratified Montreal Protocol have to phase out HCFC22 in the near future due to its ozone depleting potential (ODP) and hence new eco-friendly refrigerants are being evolved as substitutes. At Present HFC407C is one of the promising drop-in substitutes for HCFC22. But it is immiscible with mineral oil and hence polyol ester (POE) oil is recommended. Since POE oil is highly hygroscopic in nature it is not user friendly. However such oil immiscibility issue of HFC134a has been overcome [M. Janssen, F. Engels, The use of HFC134a with mineral oil in hermetic cooling equipment, Report 95403, No. 07, presented in the 19th International Congress of Refrigeration, The Hague, 1995] by the addition of HC blend to it, which also resulted in performance improvements. In the present work an attempt has been made to study the possibility of using HFC407C/HC290/HC600a refrigerant mixture as a substitute for HCFC22 in a window air conditioner and to evolve an optimal composition for the mixture. Experiments were carried out in a room calorimeter setup fitted with 1050 W capacity window air-conditioner. Condenser inlet air temperatures were held constant at 30, 35, 40 and 45 °C, while evaporator inlet air temperatures were varied over a range viz. 21, 23, 25, 27 and 29 °C during the experimentation. The HC percentage was also varied from 10 to 25% in steps of 5%. The new refrigerant mixtures demand longer condenser length to decrease the high discharge pressure matching with HCFC22 systems and hence the length has been increased while testing the mixtures. This also resulted in better heat transfer in condenser. The performance analysis revealed that the new refrigerant mixture performed better than that of HCFC22. It has in fact been found that the new mixture can improve the actual COP by 8 to 11% and hence it can reduce the energy consumption by 5 to 10.5%. The overall performance has proved that the new refrigerant mixture could be an excellent substitute for HCFC22.

Keywords: HCFC22 Phase-out; HFC407C with mineral oil; Room calorimeter; COP improvement for window air conditioner; HFC407C/HC blend mixture

1. Introduction

At present for air conditioning applications HCFC22 is the most widely employed refrigerant. The published literature revealed that the only drop-in substitute for HCFC22 is HFC407C, because it offers a close match to HCFC22 in existing window air-conditioner with respect to energy efficiency and compressor discharge temperature [2]. HFC407C is a zeotropic refrigerant mixture of HFC32/HFC125/HFC134a (23/25/52% by weight). However, with HFC407C, Polyol Ester (POE) oil must be used instead of mineral oil. This POE oil is highly hygroscopic leading to several service issues. It is also expensive and it causes irritation if it comes in contact with our skin. If HFC407C could be made to work with mineral oil the above service issues could be alleviated. It is possible to mix suitable HC Refrigerants with HFCs to solve the miscibility issues with mineral oil [1,3–6]. Preliminary investigations proved that addition of HC blend with HFC407C could solve the immiscibility issue with mineral oil and also improve the system performance [7,8]. The only drawback of HC is its flammability, but a reduction in flammability can be achieved by mixing HCs and HFCs [5].

In the present work an attempt has been made to study the possibility of using HFC407C/HC blend refrigerant mixture as a substitute for HCFC22 in a window air conditioners and to evolve an optimal composition for the mixture.

* Corresponding author. Tel.: +91 44 22203261; fax: +91 44 22203255.
E-mail address: mohanlal@annauniv.edu (D.M. Lal).

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Nomenclature

A	area	m^2
BIS	Bureau of Indian Standards	
COP	coefficient of performance	
$CIAT$	condenser inlet air temperature	
$EIAT$	evaporator inlet air temperature	
$HCFC$	hydrochlorofluorocarbons	
He	height	m
HFC	hydrofluorocarbons	
L	length	m
P	pressure	bar
Q	refrigeration capacity	W
T	temperature	K
U	overall heat transfer coefficient	$W\,m^{-2}\,K^{-1}$
W	power	W

Subscripts

ac	actual
ciat	condenser inlet air temperature
c	compressor
d	discharge
eiat	evaporator inlet air temperature
ei	evaporator inlet
eo	evaporator outlet
he	heater load
hl	heat infiltration
m	mean
s	suction

HC290 (propane) is a common HC refrigerant that could be considered as a mixture constituent with HFC407C due to its higher latent heat, volatility and miscibility with mineral oil. However, the discharge pressure of HFC134a with HC290 was higher than that of HCFC22 as well as HC290 [9] and hence if HC290 is added to HFC407C it would result in still higher discharge pressure only. It was also observed that HC600a (isobutane) could also be considered as an additive with HFC134a while the vapor pressure did not shoot up [1]. But, the boiling point of HC600a was much higher when compared to that of the HFC407C and there would be a greater composition shift in the heat exchanger [10], which might lead to oil return problems in the evaporator. Hence to utilize the above advantages of HC290 and HC600a, a HC blend consisting of 45.2% of HC290 and 54.8% of HC600a was considered to be mixed with HFC407C. Further since HC blend is a commercially available mixture, for all practical reasons it was preferred. From the literature it was found that addition of 9% of HC blend with HFC134a could solve the miscibility problem and also improves the performance of the system [11–13]. Based on the above observations, in the present work, experiments were conducted for the mixtures containing 10, 15, 20 and 25% HC blend (by weight) in HFC407C. These mixtures are further referred in this paper as M10, M15, M20 and M25, respectively. Preliminary analysis using REFPROP software indicated that with the increase in the mass percentage of HC blend, the suction and discharge pressure shoots up. The maximum permissible pressure in the compressor as per manufacturer catalog was limited to 27 bar. In order to limit the pressure within these levels the condenser length had to be altered. Therefore in this study, the condenser surface area was increased by 19% for the mixtures to control the increase in discharge pressure. This can prevent compressor failure and realize better heat transfer too in the condenser [2,14]. This also resulted in reduced pressure ratio for the mixtures as compared to HCFC22 leading to higher COP. The COP was the parameter to be optimized. The charge quantity, the capillary length and diameter, the mixture composition and the condenser length were the variables. However the range of variations was limited by the system operating conditions.

2. Experimental setup

Fig. 1 shows the schematic diagram of the experimental setup used for this performance study. The experimental setup consists of a room calorimeter, a window air conditioner of 1050 W capacity, instruments and accessories fitted to facilitate performance study as detailed in Sections 2.1–2.3.

2.1. Room calorimeter

The outer dimensions of the room calorimeter are $2300 \times 2300 \times 2800\ mm^3$. The walls of the room were insulated with glass wool of thickness 200 mm in order to maintain the heat infiltration to be less than 5% of the air conditioner capacity [BIS: 1391–1992]. A 2000 W heating capacity air heater was placed inside the room calorimeter as the source for cooling load. The heater was connected through a variac and wattmeter ($\pm 0.5\%$ accuracy) to the power supply, to facilitate variation and measurement of heat load. In order to have a uniform temperature throughout the calorimeter room, a fan (40 W) was used to circulate the air inside the calorimeter.

2.2. Window air-conditioner

To periodically check the oil level in the compressor an oil level indicator was attached suitably to the compressor as shown in Fig. 1. To optimize the capillary, 8 capillaries of different diameter and length viz. 1.1176 mm diameter: 1.25, 1.5, 1.75, 2, 2.25 m long and 1.27 mm diameter: 1.5, 1.75, 2 m long were fixed to a header. Suitable ball valves were used to select the required capillary to be included in the circuit. A thermally insulated duct was used to control the temperature of air passing over the condenser to simulate various ambient conditions without obstructing the flow of air [15]. In order to ensure oil miscibility with the mixture and in turn proper oil return an oil level indicator was fixed to the compressor as shown in Fig. 1.

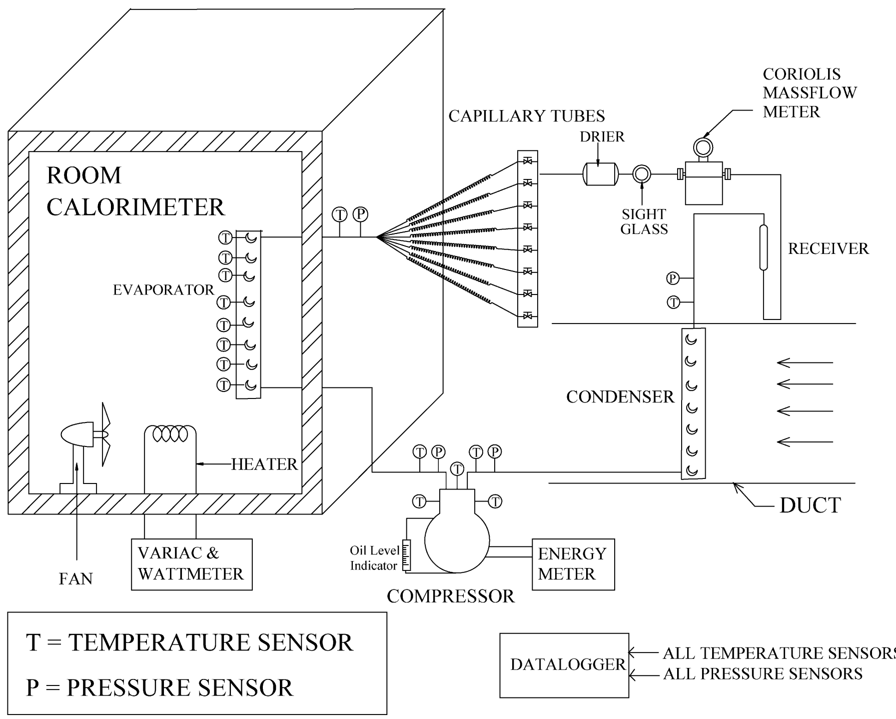

Fig. 1. Schematic diagram of the experimental setup.

2.3. Instrumentation

To monitor the mass flow of the refrigerant in the system, a mass flow meter with ±0.25% accuracy was installed next to the condenser as shown in Fig. 1. To measure the compressor power a wattmeter with ±0.5% accuracy was used. The energy consumption per day was also measured with an energy meter with ±0.5% accuracy. Pressure transducers with ±0.25% accuracy and film type PT100 RTD temperature sensors with ±0.1°C accuracy were fixed appropriately to measure the respective parameters across each component. Since the mixture is zeotropic in nature, to measure the temperature distribution along the evaporator coil eight temperature sensors were fixed suitably. Computerized data acquisition system (Agilent 34970 A & polling frequency 60 channels/second) was used to record the entire temperatures (T) and pressures (P). Five temperature sensors were fixed at various state points inside the room calorimeter to ensure that the variations in the temperature inside is not exceeding 1 °C [15] at steady state conditions before making observations.

3. Experimental procedure

The entire test was conducted according to BIS: 1391–1992. In this study refrigerant side performance of the air conditioner was measured. Before starting the experiment, heat infiltration test was carried out. For a temperature differential between evaporator inlet air temperature and the atmosphere ranging from 15 to 0 °C. It was found that for the maximum temperature differential of 15 °C, the heat leak was 47.1 W, which was less than 5% of the air conditioner capacity.

During experimentation, utilizing the duct arrangement as shown in Fig. 1, the condenser inlet air temperature was varied from 30 to 45 °C in steps of 5 °C, whereas evaporator inlet air temperature was varied from 21 to 29 °C in steps of 2 °C for each condensing temperature. The evaporator inlet air temperature and condenser inlet air temperature were the main variables in the test matrix. The refrigerating capacity of the system for a particular evaporator inlet air temperature and condenser inlet air temperature was obtained by reading the room heater load, which was controlled by a variac to maintain steadily the required evaporator inlet air temperature. At each test condition the respective heat infiltration was added to the heater load to get the actual refrigeration capacity.

To have a realistic comparison of the performance of the proposed mixtures, the experiment was carried out initially with the conventional refrigerant HCFC22. The capillary tube diameter, length and the refrigerant charge were optimized as the refrigerant flow volume had changed due to alterations made to fix instruments, receiver etc. During capillary tube optimization, the system was initially charged with 750 g of HCFC22 (as per the manufacturer's catalog). An evaporator inlet air temperature of 27 °C and a condenser inlet air temperature of 35 °C were maintained during testing. In the test the COP of the system was maximum with 1.1176 mm diameter capillary tube at 1.75 m length and it was selected. Subsequently for the selected capillary tube the charge quantity of HCFC22 was optimized for maximum COP by varying charge from 600 to 1100 gm in steps of 50 gm. The optimal charge of HCFC22 was found to be 950 gm. After that, the performance study of the system was carried out for various sets of condenser and evaporator inlet air temperatures for the optimal capillary and optimal charge.

Table 1
Comparison of measured mass flow rate with calculated mass flow rate

Condenser inlet air temperature	Mass flow rate [g s^{-1}] for 27 °C evaporator inlet air temperature									
	Measured					Calculated				
	HCFC22	M10	M15	M20	M25	HCFC22	M10	M15	M20	M25
30 °C	6.460	6.606	6.693	6.875	6.790	6.570	6.620	6.730	6.880	6.840
35 °C	6.642	6.800	6.858	7.028	6.922	6.600	6.670	6.810	6.970	6.900
40 °C	6.858	6.955	7.023	7.188	7.114	6.730	6.870	7.030	7.180	7.070
45 °C	6.989	7.150	7.267	7.426	7.324	6.810	6.980	7.180	7.340	7.240

A pull down test to find the cooling rate of the system for the refrigerants was also conducted during the experimentation. Initially, both the evaporator and the condenser inlet air temperatures were maintained at 35 °C. After attaining equilibrium, the system was made to run and the power consumption for every 10 seconds was manually noted while the temperatures were recorded using the data acquisition system. The readings were taken until the system reached its cut-off for an evaporator inlet air temperature of 27 °C.

The per day energy consumption was studied for evaporator inlet air temperature of 27 °C and different condenser inlet air temperatures set at 30, 35, 40 and 45 °C. In order to maintain the evaporator inlet air temperature the thermostat cut-in and cut-off temperatures were set at 28 and 26 °C, respectively. The energy consumption for 24 hours was noted in the energy meter for all condenser inlet air temperatures.

To conduct the experiment using mixtures, they were prepared separately in four different cylinders, which were initially cleaned and flushed thoroughly. For each mixture the equivalent charge quantity for 950 gm of HCFC22 was obtained along with % composition of HFC407C and HC blend considering the specific volume ratios at suction condition. Each mixture component was weighed individually in an electronic balance with an accuracy of ±0.1 gm and filled in the respective cylinders with the help of suitable manifold. While doing experiments with mixtures, after realizing the high condensing pressure the tube length was increased by 19% from that of HCFC22 so that the discharge pressure was maintained within 27 bar. Of the four mixtures, M10 was initially selected for the performance analysis. The capillary tube optimization for the mixture was carried out as mentioned earlier. Taking the equivalent charge of 816 gm, it was found that the COP of the system was found to be highest for a capillary tube diameter of 1.1176 mm and length 1.5 m. Having found the optimal capillary tube; the next step was to arrive at an optimal charge for the mixture. The COP of the system was studied as the mixture was charged from 500 gm and raised in steps of 50 gm up to 1000 gm. The COP of the system was found to be highest for the optimal charge of 800 gm for the mixture, which was closer to its equivalent charge of 816 gm. Hence the system performance study as carried out with HCFC22 was repeated with the equivalent charge of the mixture. Having observed a superior system performance for the mixture at its equivalent charge, the performance tests for the other three mixtures M15, M20 and M25 were also carried out with their equivalent charges of 774, 736 and 702 gm, respectively. At all the test conditions the mass flow measured through the meter and that which was calculated from refrigerating capacity was compared to confirm the observations. The calculated mass flow rate refers to the mass flow rate obtained by dividing the refrigerating capacity (kJ s^{-1}) by the refrigerating effect (kJ kg^{-1}) at the evaporator pressure and temperature. A typical table of such data is shown in Table 1. It has to be mentioned that no oil was changed during the entire test.

4. Results and discussion

In this section, performance parameters are compared between the mixtures M10, M15, M20, M25 and HCFC22.

Fig. 2 shows the variation of the refrigeration capacity with condenser inlet air temperature. Among the mixtures M20 is characterized with maximum refrigeration capacity. It is observed that the improvement in refrigeration capacity of M20 mixture is 9.54 to 12.76% higher than HCFC22 at the various condenser inlet air temperatures. This increase in refrigeration capacity can be attributed to the higher latent heat of evaporation. Even though M25 has higher latent heat, its mass flow rate is lower than that of M20 as shown in Table 1. From the table it is to be noted that the deviation of the measured mass flow rate from the calculated mass flow rate is 0.07 to 2.56% for various condenser inlet air temperatures. Thus it could be considered that the experimental observations are quite consistent with calculated results. For M20 the mass flow rate is found to be 4.64 to 9.04% higher than that of HCFC22 at various condenser inlet air temperatures. This increased mass flow rate could be attributed to the higher volumetric efficiency that might result due to small pressure ratio.

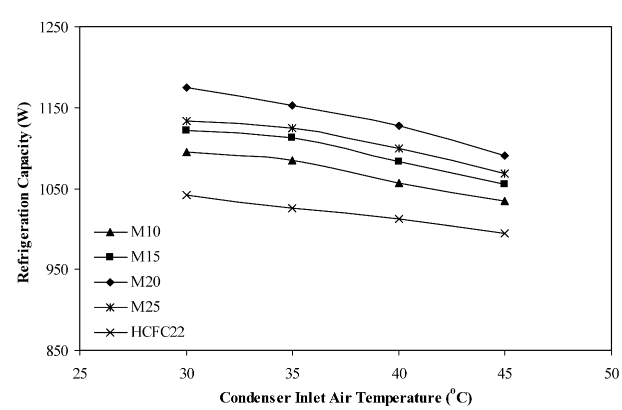

Fig. 2. Variations of refrigeration capacity with condenser inlet air temperature at 27 °C evaporator inlet air temperature.

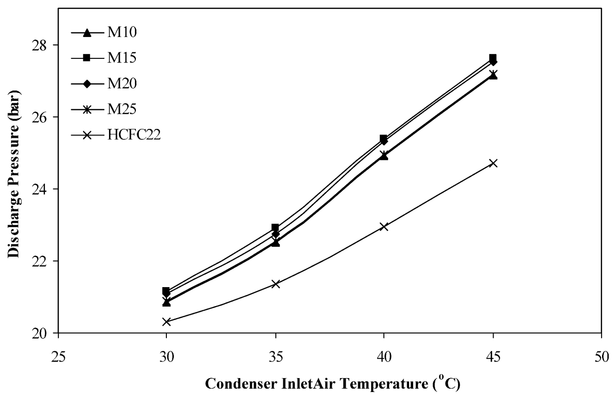

Fig. 3. Variations of discharge pressure with condenser inlet air temperature at 27 °C evaporator inlet air temperature.

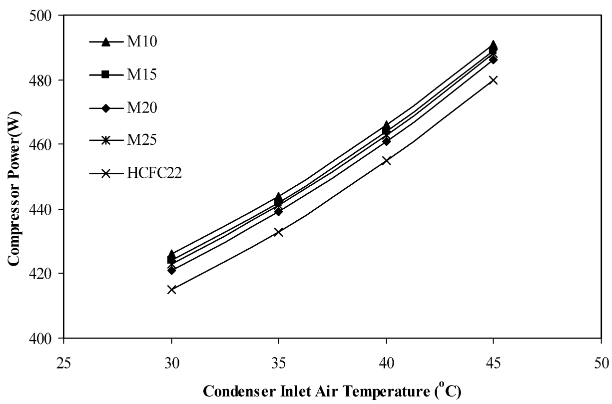

Fig. 4. Variations of compressor power with condenser inlet air temperature at 27 °C evaporator inlet air temperature.

From Fig. 3 it could be noted that the discharge pressure of HCFC22 is found to be lowest among the refrigerants. For M20 the discharge pressure is found to be 3.73 to 11.46% higher than that of HCFC22 for different condenser inlet air temperatures. Even though the discharge pressure of HCFC22 is lower, it was observed that the pressure ratio of HCFC22 was the highest and it could be attributed to the lower suction pressure of HCFC22 as compared to that of mixtures. Typically pressure ratio for M20 was the lowest and was found to be 3.56 to 4.97% lower than that of HCFC22 for various condenser inlet air temperatures.

Fig. 4 shows the variation of compressor power with condenser inlet air temperature for the refrigerants. It could be noticed that the compressor power was lowest for HCFC22. This can be due to the lower mass flow rate than other mixtures. However for the mixtures, with the increase in HC blend, the compressor power was found to be decreasing and approaching that of HCFC22. It is observed that among the mixtures M20 is found to be having the lowest power consumption which is 1.25 to 1.45% higher than that of HCFC22 for various condenser inlet air temperatures. This can be attributed to its lowest pressure ratio than other refrigerants.

The actual COP at various condenser inlet air temperatures (30 to 45 °C) for 27 °C evaporator inlet air temperature is presented in Fig. 5. It could be noted that even though with mix-

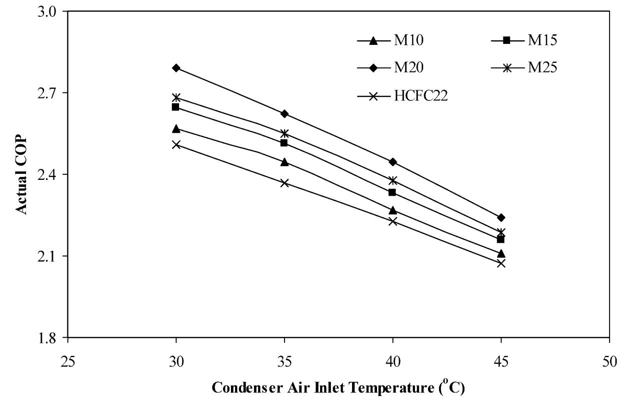

Fig. 5. Variations of actual cop with condenser inlet air temperature at 27 °C evaporator inlet air temperature.

tures the power consumed by the compressor is higher than that of HCFC22, the COP is also higher because of the higher mass flow rates and possibly better heat transfer characteristics. It is to be noted that, among all the mixtures M20 has the maximum COP, which is 8.19 to 11.15% higher than that of HCFC22 at various condenser inlet air temperatures. Even though M25 has lesser density, due to its higher-pressure ratio, the compressor work is higher while the mass flow rate is lesser than that of M20. These lead to a lower COP for M25.

The actual COP of the different cases at various condenser inlet air temperatures (30 to 45 °C) for various evaporator inlet air temperatures (21 to 29 °C) is shown in Table 2. It is to be noted that for all refrigerants the improvement in actual COP is found to be increasing with the increase in evaporator inlet air temperature and decreasing with increase in condenser inlet air temperatures. It is to be noted that, among all the mixtures the improvement in COP of M20 over HCFC22 was the maximum at all evaporator inlet air temperatures and is found to be 10 to 13.49% for 35 °C condenser inlet air temperature. Thus M20 is found to be better in all the room temperature conditions. At all test conditions, uncertainty analyses were carried out and the uncertainty in COP was less than 2.3%.

From Fig. 6 it could be noted that the compressor discharge temperatures for mixtures are less than that of HCFC22 and it was the lowest for M20. This can be attributed to lower compression ratio of the mixtures than that of HCFC22. In the case of M20 the reduction was 12.07 to 14.09% as compared to HCFC22. Thus, it can be concluded that all the refrigerant mixtures would be workable alternatives from the viewpoint of system reliability and stability of the refrigerant, however M20 seems to have an edge over the other mixtures because of its higher COP.

The temperature distribution along the length of the evaporator coil at 27 °C evaporator inlet air temperature and 35 °C condenser inlet air temperature is plotted in Fig. 7. It is to be noted that the temperature difference across the evaporator coil for the mixtures is more due to their zeotropic nature compared to HCFC22. Since HCFC22 is a single component no variation is observed up to 70% of the coil length. In the case of M20 the temperature variation along the length of the coil is about 10 °C up to 70% of the coil length, which is quite comparable with

Table 2
Performance parameters at various evaporator and condenser inlet air temperatures

Refrigerant	EIAT [°C]	Performance parameters for different CIAT [°C]											
		30 °C			35 °C			40 °C			45 °C		
		Q	W_c	COP	Q	W_c	COP	Q	W_c	COP	Q	W_c	COP
HCFC22	21	700	415	1.686	685	420	1.631	665	443	1.501	645	470	1.372
M10	21	755	438	1.725	740	434	1.705	710	456	1.557	690	481	1.435
M15	21	768	438	1.754	750	432	1.736	724	454	1.595	700	479	1.461
M20	21	806	431	1.868	792	427	1.855	768	449	1.710	745	474	1.572
M25	21	780	434	1.798	768	430	1.786	745	452	1.648	720	477	1.509
HCFC22	23	820	428	1.916	805	425	1.894	780	448	1.741	762	474	1.608
M10	23	860	444	1.938	840	438	1.918	812	460	1.765	794	485	1.637
M15	23	872	443	1.969	858	436	1.968	830	458	1.812	805	483	1.667
M20	23	920	436	2.110	906	432	2.097	880	454	1.938	853	479	1.781
M25	23	898	445	2.020	882	434	2.032	853	456	1.871	830	481	1.726
HCFC22	25	924	411	2.248	908	430	2.112	899	452	1.989	870	477	1.824
M10	25	965	424	2.276	950	442	2.149	926	464	1.996	910	489	1.861
M15	25	990	422	2.346	977	440	2.220	950	462	2.056	928	487	1.906
M20	25	1039	418	2.486	1021	436	2.342	994	458	2.170	968	483	2.004
M25	25	1007	420	2.398	987	438	2.253	960	460	2.087	940	485	1.938
HCFC22	27	1042	415	2.511	1025	433	2.367	1013	455	2.226	995	480	2.073
M10	27	1095	426	2.570	1085	444	2.444	1056	466	2.266	1035	491	2.108
M15	27	1122	424	2.646	1112	442	2.516	1083	464	2.334	1055	489	2.157
M20	27	1175	421	2.791	1152	439	2.624	1128	461	2.447	1090	486	2.243
M25	27	1134	423	2.681	1125	441	2.551	1100	463	2.376	1068	488	2.189
HCFC22	29	1143	418	2.734	1134	436	2.601	1108	458	2.419	1090	483	2.257
M10	29	1200	428	2.804	1188	446	2.664	1155	468	2.468	1135	493	2.302
M15	29	1226	426	2.878	1214	444	2.734	1190	466	2.554	1167	491	2.377
M20	29	1277	424	3.012	1265	442	2.862	1240	464	2.672	1212	489	2.479
M25	29	1245	425	2.929	1232	443	2.781	1203	465	2.587	1186	490	2.420

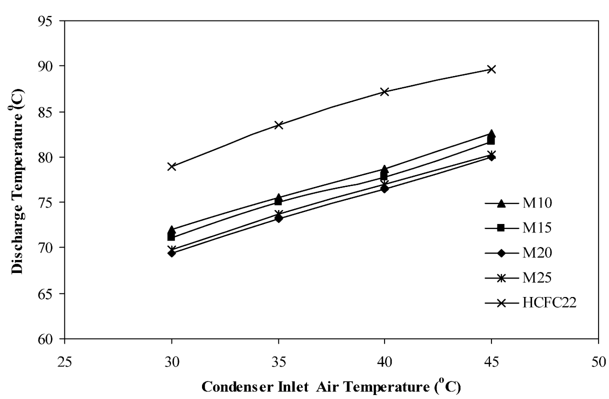

Fig. 6. Variations of discharge temperature with condenser inlet air temperature at 27 °C evaporator inlet air temperature.

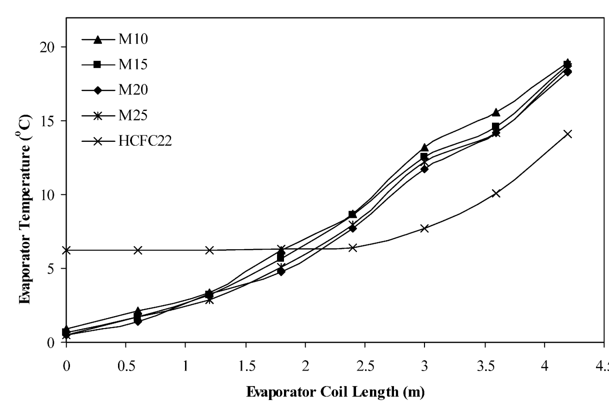

Fig. 7. Variations of evaporator coil temperature across the length at 35 °C condenser inlet air temperature and 27 °C evaporator inlet air temperature.

that of HFC407C that has a temperature glide of 6 °C. Among the four mixtures M20 and M25 show almost the same temperature distribution throughout the length of the coil. The glide realized is within acceptable limits and it is also beneficial in respect of heat exchange in such cross flow configurations [16].

The power consumption of the compressor with time during the pull down is shown in Fig. 8. As the performance of M20 was superior to that of all the mixtures, only M20 was chosen to compare the power consumption with that of HCFC22. It shows that the compressor power of M20 mixture is 2.34 to 10.45% higher than HCFC22 during the pull down. But for the same thermostat setting the cut-off took place in 40.83 minutes for M20 and in 60.50 minutes for HCFC22. Thus the total energy

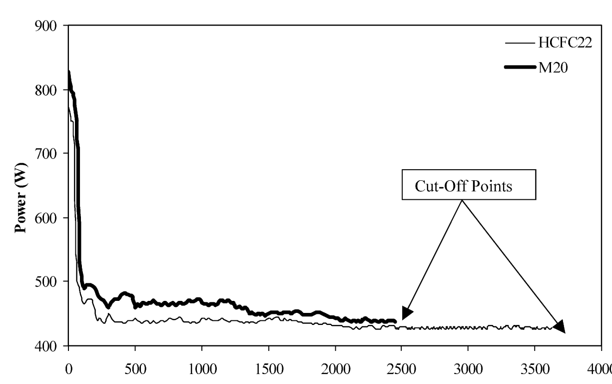

Fig. 8. Variations of compressor power with time during pull-down.

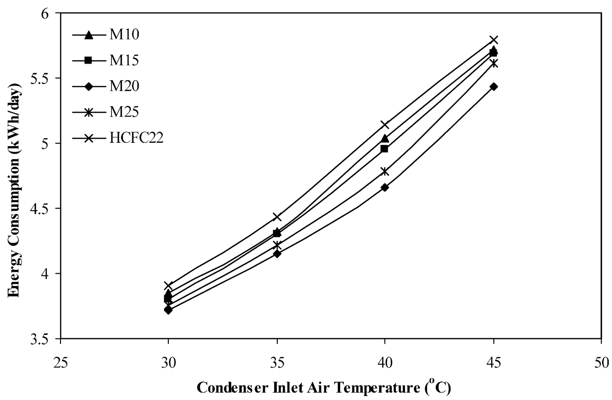

Fig. 9. Variations of energy consumption per day with condenser inlet air temperature at 27 °C evaporator inlet air temperature.

consumed by M20 during pull down is 0.31 kWh as against 0.36 kWh consumed by HCFC22. This shows that M20 mixture is more energy efficient than HCFC22.

The per day energy consumption data obtained for all the refrigerants at 27 °C evaporator inlet air temperature for various condenser inlet air temperatures are plotted in Fig. 9. From that it is found that the energy consumption of the mixtures is less than that of HCFC22. For the mixtures as HC blend percentage increases the energy consumption decreases but beyond 20% the energy consumption again increases. Even though the compressor power is higher than that of HCFC22 the reduced running time due to higher refrigerating capacity has resulted in lower energy consumption for mixtures. Among the mixtures M20 is characterized with lowest energy consumption. Typically it was 5.08 to 10.45% less than that of HCFC22 for various condenser inlet air temperatures.

The oil level was continuously noted on the oil level indicator during the operation of the system for all the refrigerants. Compared to the initial level, a 1 mm drop in oil level was observed. This could be due to oil lost with the refrigerant during change over of mixtures. Thus from the above observation it is proved that the miscibility issue of HFC407C refrigerant with mineral oil can be over come with the addition of HC blend. The improvement in performance is an additional credential for the new mixture to be considered as a substitute for HCFC22.

From the above in general it is inferred that as the proportion of HC blend increases the COP of the mixtures increases. However, even though M25 has higher proportions of HC blend its higher compressor power due to higher-pressure ratio lead to a lower COP than M20. Thus M20 is the best choice.

5. Uncertainty analysis

The uncertainty in actual COP of the system was calculated from the equations listed below:

$$COP_{\mathrm{ac}} = \frac{Q_{\mathrm{he}} + Q_{\mathrm{hl}}}{W_c}$$

$$Q_{\mathrm{hl}} = UA(T_{\mathrm{ciat}} - T_{\mathrm{eiat}})$$

$$Q_{\mathrm{he}} = f(Q_{\mathrm{he},m}, T_{\mathrm{ciat}}, T_{\mathrm{eiat}}, T_{\mathrm{ei}}, T_{\mathrm{eo}}, P_{\mathrm{ei}}, P_{\mathrm{eo}})$$

$$W_c = f(W_{c,m}, T_s, T_d, T_{\mathrm{ciat}}, P_s, P_d)$$

The uncertainty is expressed as [17]

$$\delta COP_{\mathrm{ac}} = \left[\left(\frac{\delta Q_{\mathrm{he}}}{Q_{\mathrm{he}}}\right)^2 + \left(\frac{\delta Q_{\mathrm{hl}}}{Q_{\mathrm{hl}}}\right)^2 + \left(\frac{\delta W_c}{W_c}\right)^2\right]^{\frac{1}{2}}$$

$$\delta Q_{\mathrm{hl}} = \left[\left(\frac{\delta L}{L}\right)^2 + \left(\frac{\delta He}{He}\right)^2 + \left(\frac{\delta T_{\mathrm{ciat}}}{T_{\mathrm{ciat}}}\right)^2 + \left(\frac{\delta T_{\mathrm{eiat}}}{T_{\mathrm{eiat}}}\right)^2\right]^{\frac{1}{2}}$$

$$\delta Q_{\mathrm{he}} = \left[\left(\frac{\delta Q_{\mathrm{hl},m}}{Q_{\mathrm{hl},m}}\right)^2 + \left(\frac{\delta T_{\mathrm{eiat}}}{T_{\mathrm{eiat}}}\right)^2 + \left(\frac{\delta T_{\mathrm{ciat}}}{T_{\mathrm{ciat}}}\right)^2 + \left(\frac{\delta T_{\mathrm{ei}}}{T_{\mathrm{ei}}}\right)^2 + \left(\frac{\delta T_{\mathrm{eo}}}{T_{\mathrm{eo}}}\right)^2 + \left(\frac{\delta P_{\mathrm{ei}}}{P_{\mathrm{ei}}}\right)^2 + \left(\frac{\delta P_{\mathrm{eo}}}{P_{\mathrm{eo}}}\right)^2\right]^{\frac{1}{2}}$$

$$\delta W_c = \left[\left(\frac{\delta W_{c,m}}{W_{c,m}}\right)^2 + \left(\frac{\delta T_{\mathrm{ciat}}}{T_{\mathrm{ciat}}}\right)^2 + \left(\frac{\delta T_s}{T_s}\right)^2 + \left(\frac{\delta T_d}{T_d}\right)^2 + \left(\frac{\delta P_s}{P_s}\right)^2 + \left(\frac{\delta P_d}{P_d}\right)^2\right]^{\frac{1}{2}}$$

6. Conclusion

The behavior of HCFC22 and HFC407C with various proportions of HC blend (10 to 25%) with mineral oil as compressor lubricant has been experimentally analyzed with a range of test conditions in a window air conditioner. From the discussion it is found that the actual COP of M20 is 8.19 to 11.15% higher than that of HCFC22 at various condenser inlet air temperatures. The power consumption of M20 during pull down was 2.34 to 10.45% higher than that of HCFC22. However the pull down time was reduced by 32.51% resulting in lower energy consumption. This mixture demanded lengthening of condenser by 19% in order to maintain discharge pressure with in acceptable limits.

During the continuous operations of the system no significant deviation from the initial oil level in the indicator was observed and hence the oil miscibility of M20 with mineral oil is ascertained. The fact that POE oil can be dispensed with by using M20 in the place of HFC407C is a significant finding in this work. Among the mixtures considered M20 would be the best choice for HCFC22 window air-conditioners without changing the mineral oil. However the price of obtaining solubility with mineral oil is likely to be flammability. This may not be a high risk to the consumer because of the small charge and sealed system but to the manufacturer who has to handle bulk quantities in the factory, it is of importance.

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