# THERMODYNAMIC COMPARISON BETWEEN HCFC22 AND PROPANE ON A LABORATORY-SCALE VAPOR-COMPRESSION SYSTEM

**Raquel da Cunha Ribeiro da Silva, raquelcrs@fem.unicamp.br**
**Eduardo Nogueira Pavan, eduardo.nog.pavan@gmail.com**
**Araí Augusta Bernárdez Pécora, arai@fem.unicamp.br**
State University of Campinas, School of Mechanical Engineering, R. Mendeleyev, 200 – Zip Code 13083-860, Campinas, SP, Brazil

***Abstract.*** *Natural fluids such as hydrocarbons are a good choice to substitute synthetic refrigeration fluids which are harmfull to the environment. These refrigerants may present similar thermodynamics properties and do not affect the ozone layer, as they do not have chlorine atoms. The objective of this research was to verify the thermodynamical performance of HC290 as a substitute to HCFC22 in a refrigeration system. Experimental tests took place on a laboratory-scale vapor-compression system consisting of a compressor, two equal compact heat exchangers and a capillary tube as expansion device. Pressures and temperatures of the refrigeration cycle were monitored. Result analysys were made using the software EES (Engineering Equation Solver). The developed EES program has calculated operational parameters as compression work, refrigerating capacity and coefficient of performance. The program also allowed to draw pressure-enthalpy diagrams of the cycles. Results showed that it is possible to perform a retrofit in refrigeration systems using HC290 as alternative to HCFC22.*
*Keywords: Hydrocarbon HC290, vapor-compression refrigeration cycle, alternative refrigerant to HCFC22*

## 1. INTRODUCTION

Commercial chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) are recognized for their severe harmful effects on the environment when they are released to the atmosphere. Specific concerns about their usage in air conditioning and refrigeration equipment are related to their potential contribution to global warming and their depletion effect on the stratospheric ozone layer due to chlorine chemical effect. As a result, environmental groups and the Montreal Protocol called for halting CFCs and HCFCs production. The alternative refrigerants to them should present the required thermodynamic and physical properties, comparable cost, low toxicity and low flammability (Hammad and Alsaad, 1998).

The ozone depleting potential (ODP) and global warming potential (GWP) have become one of the more important criteria in analyzing new alternatives to CFCs and HCFCs refrigerants in vapor-compression refrigeration. Both, CFCs and HCFCs, have high ODP and GWP index, which shows that they have a high contribution to ozone layer depletion and global warming. The general consensus regarding ozone layer depletion is that free chlorine radicals remove ozone from the atmosphere, and later, chlorine atoms continue to convert more ozone to oxygen. The presence of chlorine in the stratosphere is the result of the migration from chlorine containing chemicals (Akash and Said, 2003).

Hydrofluorcarbons (HFCs) refrigerants have been studied as working fluids in refrigeration and air conditioning systems and also used in many industrial and domestic applications in the last decade as they do not have chlorine atoms in their composition, resulting in ODP equal to zero (Dalkilic and Wongwises, 2010). HFCs refrigerants present others suitable specifications such as non-flammability, chemical stability and similar vapor pressure to the refrigerant CFCs and HCFCs at one specified temperature. Meanwhile, HFCs have a significant global warming potential.

The problems of the depletion of ozone layer and increase in global warming motivated scientists to investigate more environmental friendly refrigerants than HFCs. Much attention has been paid to the so-called "natural fluids", which are claimed to be more environmental friendly than the synthetic ones. Among these natural alternatives are the hydrocarbons (HCs) as propane, isobutene, n-butane and hydrocarbon mixtures which offer the advantages of low cost and high availability. HCs present zero ozone depletion potential and do not cause the greenhouse warming effect (Lorentzen, 1995). Garland and Hadfield (2005) commented that HFCs have been replaced by HCs refrigerants in a lot of vapor-compression refrigeration systems due to these environmental advantages.

Purkayastha and Bansal (1998) studied the performance of a vapor-compression refrigeration cycle using a hydrocarbon mixture composed by HC290 (98.95%), HCI70 (1.007%) and HC600a (0.0397%). The coefficient of performance (COP) obtained was compared with both obtained for operations using pure HC290 (propane or $C_3H_8$) and HCFC22 (chlorodifluoromethane or $CHClF_2$) as refrigerant. They observed that both hydrocarbon refrigerants presented higher COPs than HCFC22 (18% higher for HC mixture and 12% higher for pure HC290) and smaller condenser capacity (10% lower for HC mixture and 15% lower for HC290). They observed that the refrigerant mass flow rates in the cycles using both HC290 and LPG mixture were about 44% lower than using HCFC22.

HCs have many advantages over HFCs concerning energy efficiency; critical temperature, solubility, transport and heat transfer properties although, their flammability is an important disadvantage. Kabul *et al.* (2008), Bayrakc and Ozgur (2009), Devotta et. al (2005) , Lee and Su (2001) recommended that the usage of alternative refrigerants as retrofit requires some changes in the refrigeration systems and system components. Lorentzen (1995) recommended HC290 as alternative to HCFC22 due to its excellent thermodynamic and transport properties.

Bayakc and Ozgur (2009) made a COP comparison for cycles using four different pure HCs (R290, R600, R600a and R1270), HCFC22 and HFC134a. Their results showed that the system operating with R1270 presented the best coefficient of performance. HCFC22, HC600a, HFC134a showed almost the same COP. But comparing HC290 and HCFC22 the coefficient of performance were lower using HC290 than HCFC22 (5.7% less) in vapor compression heat pumps.

Owing to the listed reasons, researchers are still working in order to find alternative refrigerants to HCFCs. Energetic analysis (first law of thermodynamics) involving each main component of the refrigeration cycle allows the comparison between performance parameters of systems operating with different fluids as refrigerant. However, as indication of possibilities for thermodynamic improvements, energy analysis is inadequate and exergy analysis is needed (Bayrakc and Ozgur, 2009).

The goal of this research was to compare the thermodynamic performance of a lab-scale compression refrigeration cycle using two different refrigerants: HCFC22 and HC290 in order to verify the possibility of the retrofit operation. The analysis was made by comparing some cycle's performance parameters at steady state operation as mass flow rate of refrigerant; electrical power input; rate of heat transfer released at condenser, refrigeration effect; refrigeration capacity and coefficient of performance.

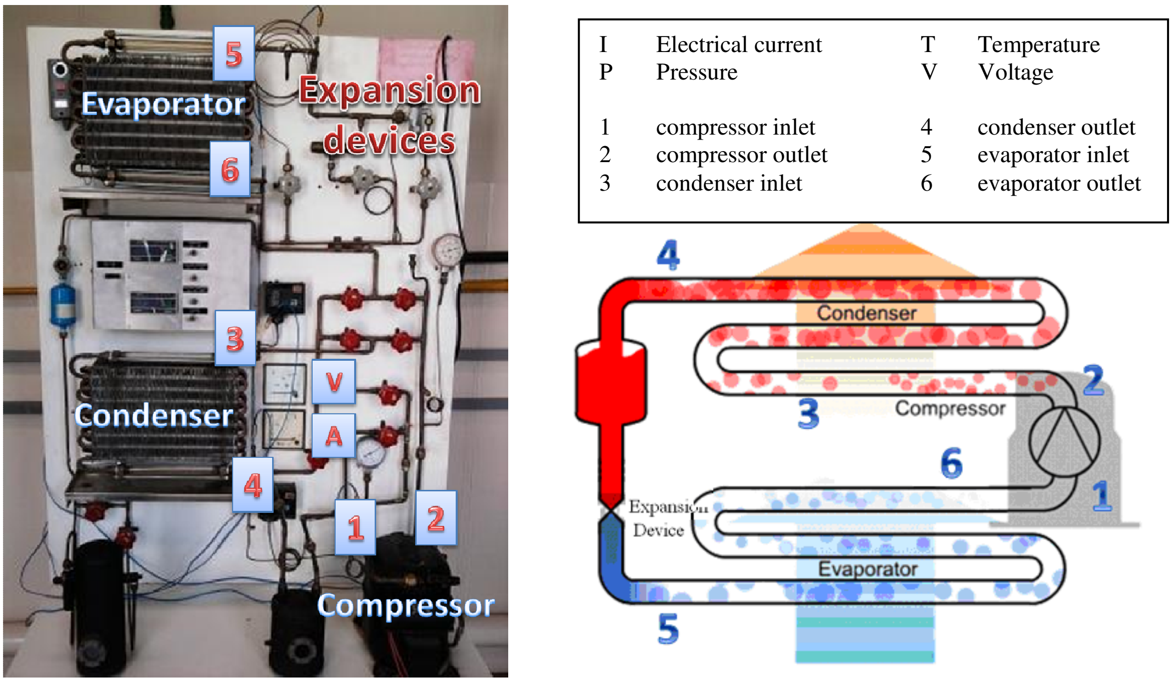
## 2. EXPERIMENTAL SETUP AND PROCEDURE

Experimental tests took place in the Thermal Processes and Environmental Engineering Laboratory based in the School of Mechanical Engineering at University of Campinas.

The main components of the system are two compact heat exchangers (evaporator and condenser), a hermetic compressor (Copeland model ARE51C4E-CAA, displacement rate, CD, equal to 1.13E-3 $m^3/s$) and three types of expansion devices (capillary tube, thermostatic valve and pressostatic valve). Tests were conducted using the capillary tube (1.27E-3 m diameter and 1.42 m length) as expansion device. The experimental set up is shown in Fig. 1(a) as the schema of the vapor-refrigeration cycle in Fig. 1 (b). Points 1 to 6 show the locations where temperature (T) and pressure (P) measurements were made.

The refrigerant load ($m_R$) was made after the system had been evacuated with the help of a vacuum pump. The system was flushed with nitrogen to check leakage and to eliminate impurities and moisture inside the system, which may affect the accuracy of the experimental results. To determine the necessary mass of refrigerant into the system, the weight of the refrigerant cylinder was measured before and after loading process.

Experiments started using HCFC22 into the system to create a reference for further comparison with HC290 as alternative refrigerant. All the experimental measurements were at steady state conditions which took about 30 minutes. The steady state was achieved when temperature measurements at each point (1 to 6) were almost constant (a maximum variation of 1°C was adopted). To produce a reliable set of data, each experiment was made three times, each one in a different day.




(a) (b)

Figure 1. Laboratory scale vapor-compression system

The temperature of the refrigerant on inlet/outlet of each component was measured using cooper-constantan thermocouples (T type) with an uncertainty of ± 0.6 °C.

Pressure measurements were made using pressure-gauge manometers with an uncertainty ± 1.8 kPa. The electric power input in the compressor was measured using an amperage meter and voltage meter with uncertainties of ± 0.26 A and ± 5V respectively. The displacement rate of the compressor (CD) was obtained from manufacturer catalog as 1.12E-3m$^3$/s.

Temperature and pressure measurements were necessary to find out the enthalpy in and out of each component of the system in order to investigate the refrigeration cycle performance.

An EES (Engineering Equation Solver) program was developed to perform the system energetic analysis for each tested condition.

Cycle performance was obtained through mass and energy balances in a control volume enclosing each main component of the refrigeration system, using the following hypothesis:

- Steady state operation;
- Kinetic and potential energy changes are neglected in the analyses of the components;
- No heat transfer to or from the compressor;
- Saturated liquid at the condenser outlet;
- Pressure drop between points 2 and 3 and between points 1 and 6 are neglected;
- Isenthalpic expansion.

Energetic analyses were made from P and T measurements concerning the following performance parameters:

- Mass flow rate of refrigerant ($\dot{m}_R$), through a compressor is proportional to the rate of displacement of the compressor (CD) and to the clearance volumetric efficiency ($\eta_{vc}$), as shown in Eq. (1),

$$\dot{m}_R = \frac{CD * n_{vc} / 100}{v_1} \tag{1}$$

The clearance volumetric efficiency was obtained from Eq. (2) assuming a clearance volume ratio (C) equal to 4.5% (Stocker and Jones, 1982),

$$n_{vc} = 100 - C\left(\frac{v_1}{v_2} - 1\right) \tag{2}$$

where $v_1$ and $v_2$ are the specific volume of vapor entering and leaving the compressor, respectively.

- Rate of power input per unit mass of refrigerant ( $w$ ) obtained from Eq. (3),

$$w = h_2 - h_1 \tag{3}$$

where the specific enthalpy $h_1$ and $h_2$ were obtained from $P_1$,$T_1$ and $P_2$,$T_2$ measurements, respectively.

- Actual electric power input to the compressor ($\dot{W}$) calculated from electrical current (I) and voltage (V) measurements, Eq. (4).

$$\dot{W} = IV \tag{4}$$

- Rate of heat transfer from the refrigerant at the condenser ( $\dot{Q}_{out}$ ) calculated by Eq. (5),

$$\dot{Q}_{out} = \dot{m}_R (h_4 - h_3) \tag{5}$$

where $h_3$ was obtained from $T_3$ and $P_2$ measurements (as $P_3 \cong P_2$) and $h_4$ was obtained from $T_4$ and $P_4$ measurements.

- Refrigeration effect ($q_R$) given by Eq. (6),

$$q_R = h_6 - h_5 \tag{6}$$

where $h_5=h_4$ (isenthalpic expansion) and $h_6$ was obtained from $T_6$ and $P_1$ measurements (as $P_6 \cong P_1$).

- Refrigeration capacity ($\dot{Q}_R$) calculated by Eq. (7)

$$\dot{Q}_R = \dot{m}_R q_R \tag{7}$$

- Coefficient of performance (COP) obtained from Eq. (8) as the performance of refrigeration cycles can be described as the ratio of the energy received by the system ($\dot{Q}_R$) to the net work transfer of energy into the system ($\dot{m}_R w$).

$$COP = \frac{q_R}{w} \tag{8}$$

A computer program was developed using the software EES (Engineering Equation Solver) to obtain refrigerant properties, performance parameters and the uncertainties of the results.

## 3. RESULTS

Tables 1 and 2 show measured values for each experimental test using HCFC22 and HC290, respectively.

Table 1. Measured values for tests using HCFC22.

| Test | $m_R$ (kg) | $T_1$ (°C) | $T_2$ (°C) | $T_3$ (°C) | $T_4$ (°C) | $T_5$ (°C) | $T_6$ (°C) | $P_1$ (kPa) | $P_2$ (kPa) | $P_4$ (kPa) | V (v) | I (A) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 0.380 | 30.0 | 116.0 | 80.0 | 30.0 | -22.0 | 30.0 | 159.9 | 1170.0 | 1136.0 | 120 | 6.3 |
| 2 | 0.380 | 29.0 | 102.0 | 67.0 | 27.0 | -23.0 | 26.0 | 128.9 | 1135.5 | 1101.1 | 110 | 5.5 |
| 3 | 0.380 | 33.0 | 118.0 | 78.0 | 31.0 | -22.0 | 30.0 | 135.8 | 1218.3 | 1204.5 | 120 | 6.5 |

Table 2. Measured values for tests using HC290.

| Test | $m_R$ (kg) | $T_1$ (°C) | $T_2$ (°C) | $T_3$ (°C) | $T_4$ (°C) | $T_5$ (°C) | $T_6$ (°C) | $P_1$ (kPa) | $P_2$ (kPa) | $P_4$ (kPa) | V (V) | I (A) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **1** | 0.200 | 34.0 | 105.0 | 88.0 | 41.0 | -13.0 | 30.0 | 232.0 | 1445.0 | 1442.0 | 130 | 6.5 |
| **2** | 0.200 | 32.0 | 107.0 | 89.0 | 39.0 | -12.0 | 29.0 | 231.6 | 1444.0 | 1443.2 | 130 | 6.4 |
| **3** | 0.200 | 33.0 | 105.0 | 91.0 | 41.0 | -13.0 | 30.0 | 232.3 | 1445.8 | 1441.3 | 130 | 6.5 |

It was observed that the pressure at the compressor inlet and outlet ($P_1$ and $P_2$) were bigger in the tests using HC290, even though the refrigeration mass load is lower. So, evaporating and condensing temperatures are higher in the refrigeration cycles using this alternative refrigerant.

Tables 3 shows the average results, regarding tests 1 to 3 shown in Tabs. 1 and 2, for the performance parameters obtained from Eqs. (1) to (8).

Table 3. Performance parameters for the studied for HCFC22 and HC290 cycle

| Fluid | $v_1$ (m³/kg) | $v_2$ (m³/kg) | $\eta_{vc}$ (%) | $\dot{m}_R$ x10³ (kg/s) | $w$ (kJ/kg) | $q_R$ (kJ/kg) | $\dot{W}$ (kW) | $\dot{Q}_{out}$ (kW) | $\dot{Q}_R$ (kW) | COP |
|---|---|---|---|---|---|---|---|---|---|---|
| **HCFC22** | 0.2049 | 0.02920 | 72.9 | 3.76 | 48.9 | 215.1 | 0.713 | 0.876 | 0.785 | 3.98±0.03 |
| **HC290** | 0.2402 | 0.04358 | 79.7 | 3.74 | 113.9 | 324.8 | 0.845 | 1.481 | 1.181 | 2.85±0.04 |

We can verify that HC290 presents higher specific volumes at compressor inlet and outlet ($v_1$ and $v_2$) resulting in a higher clearance volumetric efficiency. Also, the refrigerating effect ($q_R$) is higher using HC290 than HCFC22 which is a desired result regarding the performance of refrigeration cycles. Meanwhile, the compression work using HC290 was more than twice the compressor work obtained using HCFC22 resulting in a lower coefficient of performance. The average COP value was 28% lower in cycles using HC290 which agrees with Purkayastha and Bansal (1998) and

Devotta *et al.* (2005). However, our results disagree with Bayakc and Ozgur (2009) results which were obtained in a heat pump system.

Figures 2 and 3 show points 1 to 6, obtained from the experimental data, in the P-h diagram for HCFC22 and HC290 respectively.

Pressure at point 5 (evaporator inlet) was obtained from $T_5$ as we have a saturated condition after the expansion device. These results show that the retrofit effect in refrigerating cycles is a function of the refrigeration capacity of the system.

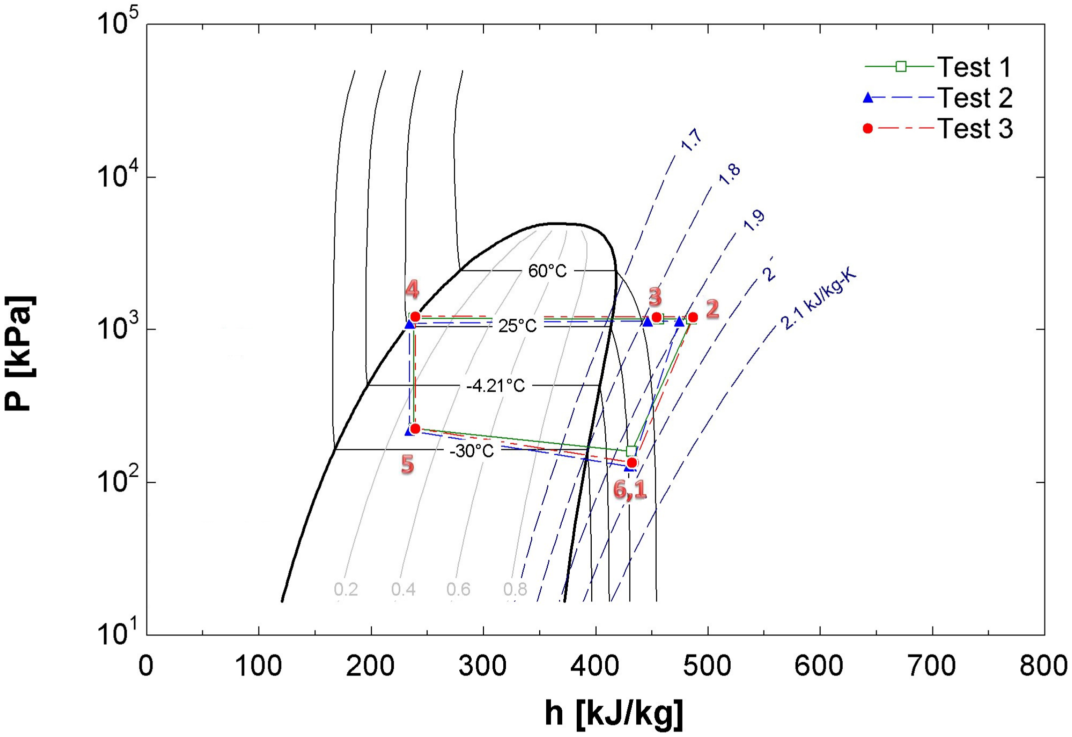



Figure 2. Pressure-enthalpy cycle for HCFC22

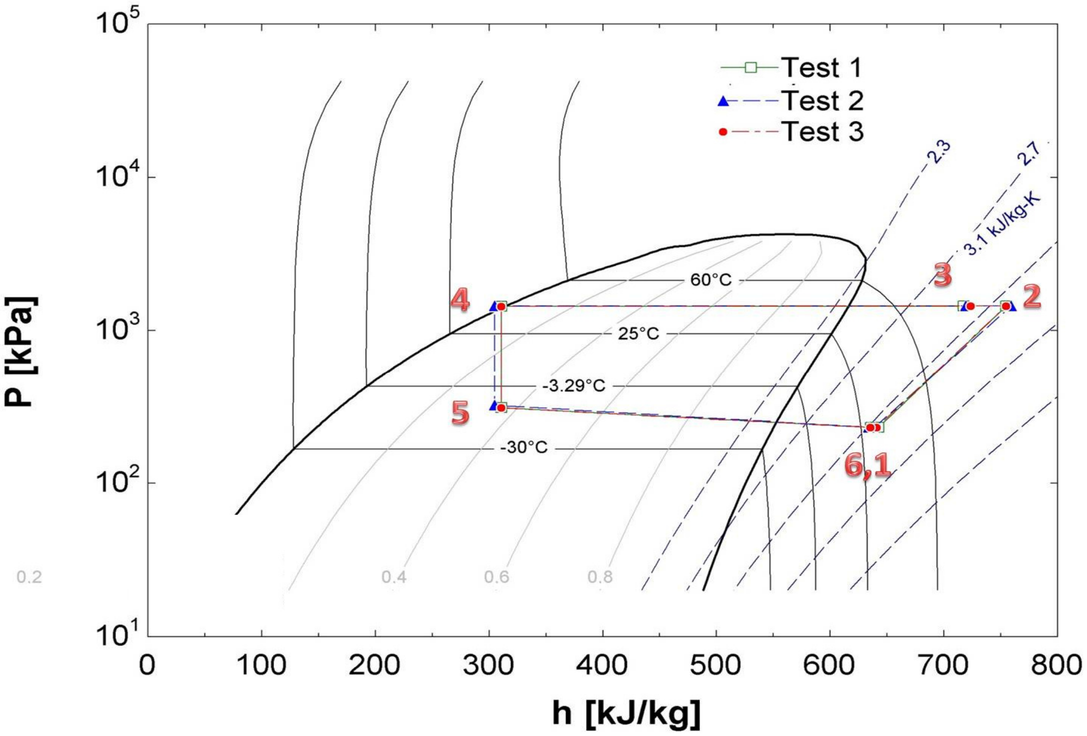



Figure 3. Pressure-enthalpy cycle for HC290

Comparing Figs. 2 and 3 it can be noticed that the low evaporating temperature in cycles using HCFC22 causes the cooling of the compressor wall which provokes a reduction on the entropy of the refrigerant at the compressor outlet.

## 4. CONCLUSION

Based on the experimental investigation of the performance of HC290 as a retrofit substitute to HCFC22, the following conclusions were made:

COP values for refrigeration cycles using R290, although satisfactory, were 28 % lower than the obtained using HCFC22, in spite of the hydrocarbon had presented a higher refrigeration capacity. More power was required by the compression system using HC290 as refrigerant, which caused the reduction on the coefficient of performance. These results could be explained by the different operational pressure and thermodynamics properties regarding HC290 and HCFC22.

Results showed that it is possible to perform a retrofit on small refrigeration systems using propane to replace HCFC22 but the pressure levels into the system operating with HC290 will be higher.

## 5. REFERENCES

Akash, B. A. and Said, S. A., 2003. "Assessment of LPG as a possible alternative to R-12 in domestic refrigerators". Energy conversion and Management , Vol. 44, No. 3, pp. 381-388.

Bayrakc, H. C. and Ozgur, A. E., 2009. "Energy and exergy analysis of vapor compression refrigeration system using pure hydrocarbon refrigerants". International Journal of Energy Research,Vol. 33, No.12, pp. 1070-1075.

Dalkilic, A. S. and Wongwises, S., 2010. "A performance comparison of vapour-compression refrigeration system using various alternative refrigerants". International Communication in Heat and Mass Transfer, Vol. 37, No. 9, pp. 1340-1349.

Devotta, S., Padalkar S.A.,Sane N.K. 2005. "Performance assessment of HC290 as a drop-in substitute to HCFC22 in a window air conditioner". International Journal de Refrigeration, Vol. 28, No. 9, pp. 594-604.

Garland, N. P. and Hadfield, M., 2005. "Environmental implications of hydrocarbon refrigerants applied to the hermetic compressor". Materials and Design, Vol. 26 , No. 7, pp. 578-586 .

Hammad, M. A. and Alsaad, M. A., 1998. "The Use of Hydrocarbon Mixtures as Refrigerants in Domestic Refrigeratores". Applied Thermal Engineering, Vol. 19, No. 11, pp. 1181-1189.

Kabul, A., Kizilkan, O., Yak, A. K., 2008. "Performance and exergetic analysis of vapor compression refrigeration system with an internal heat exchanger using a hydrocarbon, isobutane (R600a)". International Journal of Energy Research, Vol. 32, No. 10, pp. 824-836.

Lee Y.S and Su C. C., 2001. "Experimental studies of isobutene (R600a) as the refrigerant in domestic refrigeration system". Applied Thermal Engineering. Vol.22. No.1, pp.507-519

Lorentzen, G., 1995. "The use of natural refrigerants: a complete solution to the CFC/HFC predicament". International Journal of Refrigeration, Vol. 18, No. 3, pp.190-197.

Purkayastha, B. and Bansal, P.K., 1998. "An experimental study on HC290 and a commercial liquefied petroleum gas (LGP) mix as suitable replacements for HCFC22". International Journal of Refrigeration, Vol. 21, No. 1, pp. 3-17.

Stoecker, W.F. and Jones, J.W., 1982. *Refrigeration and air conditioning*, 2nd Ed., McGraw Hill Inc.

## 6. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.