A Review Investigation Of One Ton Vapour Compression Refrigeration R-12 Refrigerate with Alternative ECO Friendly Refrigerants R410. R-413a And R-423a

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Abstract

The Montreal Protocol has sealed the use of Halogenated Hydrocarbons; Keeping in view they affect the environment in the form of Ozone layer depletion and Global warming. Thus one of the major thrust areas is to identify substitute of Halogenated Hydrocarbons, especially CFCs. The substitute refrigerant should be eco-friendly, chemically stable and compatible with existing refrigeration systems with transport and thermal properties similar to or better than CFCs. Since the refrigeration system use CFC- 12 as refrigerant universally, the search for a suitable alternative for CFC-12 is inevitable.HFC-134a is currently the leading alternative to CFC-12. Other promising substitutes are R-423as a binary mixture of R-227ea & R- 413a ternary mixture of R-134a, R-218 andR-600a. In this thesis, performance evaluation of eco-friendly alternate refrigerant R-134a, R-413a & R-423a for replacing CFC12 has been done and a suitable alternative refrigerant for retrofitting has been identified.

Introduction

the search for a suitable alternative for CFC-12 is inevitable.HFC-134a is currently the leading alternative to CFC-12. Other promising substitutes are R-423a a binary mixture of R-227ea & R-413a ternary mixture of R-134a, R-218 and R-600a. The present work aims to analysis suitability of two newly discovered mixtures R-423a & R-413a for replacement of CFC-12 in existing refrigeration units. For this purpose it is necessary to evaluate the performance of refrigeration system. This requires thermodynamic and transport properties of the mixture, most of which have already computed. A cycle analysis has been carried out to predict the performance of system under various operation conditions.

The aim of this project is to propose a suitable design of a vapor Compression Refrigeration System using eco-friendly refrigerants such as R-423a and R-413a. The design changes of the components of refrigerating system will be based on the similarities in the desirable properties of the refrigerants with that of R-12.

1.1 Classification of refrigerants

Most of the existing refrigerants belong to halocarbon group which is classified into three categories:-

1. Chlorinated hydrocarbons
2. Fluorinated hydrocarbons
3. Bromines

In the present usages halogenated hydrocarbons are divided into three groups:-

1. CFCs (Chloro Flouro Carbon compounds)
2. HCFCs (Hydro Chloro Flouro Carbons)

3. HFCs (Hydro Flouro Carbons)

1.2 Global warming & CFCS

Problem of global warming is due to greenhouse effect. The green house effect refers to the trapping of infra-red radiation of sun by the atmosphere and subsequent warming of earth. Although the greenhouse effect is primarily due to CO_2 and the concentration of CFCs are very low compared to CO_2. CFCs absorb strongly the I-R region particularly in the wavelength between 7-13 microns where the atmosphere is largely transparent. The absorption is due to the C-Cl and C-F bonds present are the CFCs. If the greenhouse gases keep on emitting at present rate, the average temperature of earth will increase which will cause many problems like submerging of coastal areas and also lead to ecological unbalance.

Table 1.1 : Ozone layer depletion factors and control measures

Chemical	Uses	Ozone Depletion Factor	Control Under 1987 Montreal Protocol	New Controls with Amendments London, June 1990
CFC -11	A.C./solvent	1.0	50% cut on 1989 level by 1999	Complete Phase out by 2000 AD
CFC-12	Room Refrigeration & A.C.	1.0		
CFC -22	Mobile A.C. A.C. / Cold	0.05		
CFC -113	Room solvent	0.80	Freeze on 1992 level Production	
Halon 1211	Fire extinguisher	3.00		
Halon 1301	Fire Extinguisher	10.00		Total Phase out by 2000
Carbon Tetrachloride	Solvent	1.06	No control	85% cut by 1995, phase out by 2000
Methyl chloroform	Solvent	0.10	No Control	70% cut by 2000 phase out 2005
HCFCs	Replacement Refrigerant	0.02 0.10	No Control	phase out by 2040 suggested

1.3 Thermodynamic assessment of alternative refrigeration to R-12 for refrigeration system

The main substitutes for CFC -12 are :-

1. HFC - 134 a
2. R-423 a,
3. R- 413a
4. HFC-152a
5. A near azeotropic ternary blend of HCFC-22, HCFC -124 and HFC-152 a
6. Hydrocarbons etc.

Literature Review:

Miguel Padilla investigates Exergy analysis of R413A as replacement of R12 in a domestic refrigeration system (9). She deals with an exergy analysis of the impact of direct replacement (retrofit) of R12 with the azeotropic mixture R413A on the performance of a domestic vapour-compression refrigeration system originally designed to work with R12. Parameters and factors affecting the performance of both refrigerants are evaluated using an exergy analysis. In the literature, no experimental data for exergy efficiency are reported, so far, for R413A. Twelve tests (six for each refrigerant), are carried out in a controlled environment during the selected cooling process from evaporator outlet temperature from 15^0C to 10^0 C. The evaporator and condenser air-flows are modified to simulate different evaporator cooling loads and condensers ventilation loads. The overall energy and exergy performance of the system working with R413A is consistently better than that of R12.

Ciro Aprea et al. investigate through an experiment global environmental impact of the R22 retrofits with R422D (11**).** In recent years a new refrigerant, R422D, has been introduced as substitute of R22 for refrigeration systems. This new fluid is an easy-to-use, non-ozone-depleting HFC refrigerant and, differently from its predecessor (R407C), it is compatible with mineral oil. However, R422D has a very high GWP, and it tends to worsen the efficiency of retrofitted R22 systems. Consequently, even if R422D respects the limits of Montreal Protocol, its global environmental impact could be high. In this paper, we report an experimental analysis in terms of TEWI aimed to identify the global environmental impact of R22 systems retrofitted with R422D. For this purpose, we considered a direct expansion refrigerator for commercial applications and we investigated energy consumption with the temperature of the cold reservoir set to 5, 0, 5, 10 ^{0}C. The experimental investigation confirmed that the system, when retrofitted with R422D, leads to an increase of TEWI. Therefore an optimization analysis aimed to eco-friendly scenarios was performed.

Ciro Aprea et al. investigate an experimental analysis for a vapor compression refrigeration plant for a walk-in cooler for change in energy performance as a result of a R422D retrofit (12). In this paper, the energy performance of a walk-in cooler working with R22 and its substitute R422D are experimentally studied. The experimental investigation was carried out considering three different operating conditions; in particular, the AHRI standard has been used as reference for operating conditions. All tests were run at steady state conditions and keeping the external air temperature at 35^0C. The experimental analysis allowed the determination of cooling capacity, the electrical power absorbed, the COP and other variables characterizing the working of the plant. The results demonstrated that the cooling capacity for R422D was lower than for R22, while the electrical power absorbed with R422D was higher than that with R22. As consequence, the COP of R422D was lower than that of R22 furthermore; technical proposals are introduced with the aim of improving the overall performances of those plants, which could be retrofitted with R422D.

A.N. Leiper et al. investigate energy conservation in ice slurry applications (13). Significant improvements in coefficient of performance (COP) can be made when the evaporating temperature of a vapour compression cycle is raised. Producing the ice slurries used in cooling applications and ice pigging by crushing blocks of ice made from pure water and mixing the particles with a solution of freezing point depressant (FPD) later enables higher evaporator temperatures than if the solution itself is frozen in a scraped surface ice maker. Predictions of the possible improvements to COP can be made in Cool Pack, a refrigeration cycle simulation program, over a range of evaporator temperature series of experiments designed to verify these predictions were carried out where the operating conditions of a scraped surface ice maker were altered by retrofitting compressor speed control to a scraped surface ice maker. Following verification of the Cool Pack results, further experiments were conducted that evaluated the energy required for two stage combinations of large ice blocks into fine particles. Two theoretical slurry production systems were then compared: mixing crushed ice with an FPD solution and a scraped surface ice maker with FPD solution feedstock. Although energy conservation was shown with combination the proposed method introduces a number of challenges that require careful consideration.

R. Llopis et al. investigate HCFC-22 replacement with drop-in and retrofit HFC refrigerants in a two-stage refrigeration plant for low temperature (14). The world community has committed to eliminate the HCFC-22 refrigerant to a series of deadlines according to the agreements taken during the 19th Montreal Protocol meeting in September 2007. This phase-out, which is already in progress in European Countries, has been accelerated in Article 5 countries. Refrigerant manufactures offer different drop-in refrigerants to replace R22 in existing equipment by non-ozone depleting substances in order to be able to make full use of the remaining life of the plants or different retrofit refrigerants, the use of which implies modifications to the existing systems. This work aims to contribute to the understanding of the implications of the process of R22 substitution, either with drop-in or retrofitting processes, by presenting a theoretical and experimental analysis of the performance of R22, of two drop-in fluids (R422A, R417B) and a retrofit refrigerant (R404A), in a two-stage vapour compression plant over a wide range of evaporating temperatures for a fixed condensing temperature of 40^0C. In this communication the main energy parameters, such as cooling capacity and COP are analyzed and discussed.

PROBLEM FORMULATION AND PROPERTY CORRELATION

In this chapter the aim of the present work is briefly describes the plan to complete the work successfully.

3.1 Problem formulation

The objectives of the present work are as follows.

1. To study various alternatives to CFC-12 for refrigeration system. This involves the study of various aspects including the thermodynamic as well as operating and maintenance aspects.
2. To develop properties and property correlation for R-423a & R-413a, which are not available in literature.
3. To evaluate the performance of 1 ton refrigeration system using HFC-134a, R-423a & R-413a at various conditions and to compare them with CFC -12.
4. To carry out the thermodynamic design of refrigeration system and identify the design changes required, for each component of the system.
5. To study the feasibility of the design modification of various components and modify some of the components, if necessary.
6. To identify the design changes with respect to R-12 and to suggest modification (if any) in each system component.

This chapter deals with the cycle analysis of the refrigeration system using R-423a (binary) and R- 413a (ternary) mixture. Performance of system using CFC-12 and R-134a also studied for comparison. The first portion of this chapter is devoted to property correlation and second portion is devoted to identify the design changes required.

3.2 Property correlation

Analysis of vapor compression refrigeration system can be carried out using thermodynamic properties of the working fluid. Prediction of the performance of compression refrigeration system at design and off design conditions needs repeated calculations.

This necessitates the development of correlation for various properties such as vapor pressure, specific volume, saturated vapor and saturated vapor enthalpy as a function of temperature and pressure.

For thermodynamic properties correlation have already been developed. The reference state for liquid phase enthalpy and entropy is taken at temperature of 0^oC.

Href= 200kJ/kg, Sref. = 1.0 kJ/kg

For design of vapor compression refrigeration system in addition to thermodynamic properties, transport properties of the working fluid are also required. These have been developed in the present work, taking data mainly from TPRC handbooks. The data is available for R-423a and R -413 a separately.

All the correlations have been developed using Regression method.

3.2.a Correlation for vapour pressure : Vapour pressure correlation for R-12, R-134a, R-413a & R-423a binary blend and ternary blend as a function of temperature are given below. The form is as follows. -

$$P_v=A+B*T+C*T^2+D*T^3+E*T^4+F+T^5$$

The constants A,B,C,D,E,F are given in table 3.1

The RMS error for binary mixture in PL correlation is 0.081% and in PV correlation is 0.053%.Here unit of pressure is kPa and temperature is in Kelvin.

3.2.b Correlation for saturated vapour specific volume

Correlation for liquid and vapor enthalpy for R-12, R-134a, R-413a & R-423a binary blend and ternary blend as a function of temperature are given below. The form is as follows.

$$V_g=(A+B*T)/(1+C*T+D*T^2)$$

The constants A, B, C, and D are given in table 3.2

Table 3.1 : Coefficients for vapor pressure correlation
(Formula: P_v =A +B.T +C.T^2 +D.T^3 +E.T^4+F.T^5)

Refrigerant	**A**	**B**	**C**	**D**	**E**	**F**
R-12	308.6	10.144	0.125753	0.0006640294	9.964117e-007	5.09385 e-010
R-134a	293.4	10.642	0.146852	0.0008955292	2.092317e-006	1.553163 e-009
R-423 a	262.4	9.5324	0.131655	0.0008186726	2.149774e-006	2.292466 e-009
R-413a	319.0	11.489	0.157428	0.0009389806	2.093660e-006	3.654824 e-009

Table 3.2: Coefficients for vapor specific Volume correlation
(Formula: V_g = (A+B*T)/ (1+C*T+D*T^2)

Refrigerant	**A**	**B**	**C**	**D**
R-12	0.055386671	-0.0003223	0.0255523	0.0001983527
R-134a	0.069321776	-0.0004813	0.0280071	0.0002395776
R-423 a	0.062864672	-0.0004436	0.0280426	0.0002377446
R-413a	0.062040924	-0.0004482	0.0276292	0.0002285484

METHDOLOGY

The Refrigeration system

The refrigeration system of 1-ton cooling capacity has been chosen for the study. The system consists of the four usual components i.e. the compressor, condenser, evaporator and capillary tube (throttling device). The design conditions for the refrigeration unit in Indian conditions are as follows.

S.No		Case-1	Case-2
1.	Condensing Temperature	45^0C, 50^0C and 55^0C	450C,500C and 550C

2	Evaporator Temperature	-10^0C,-5^0C, 0^0C, 5^0C, 10^0C	-10^0C, -5^0C, 0^0C,5^0C,10^0C
3.	Superheating	0^oC	10^0C
4	Sub-cooling	0^oC	5^0C

Analysis of the Refrigeration system

The aim of the performance analysis is to compare the refrigerant R-134a, R-423a and R-413a with R-12. The simple vapor compression refrigeration cycle considering no losses is shown in figure 3.1. T-s diagrams are shown in figure 3.1.a and 3 .1.b for case - 1 and Case 2 respectively.

The various processes in the cycle are as follows (fig. 3.1.b)

1-2 Compression process in compressor
2-3 Condensation of vapor in condenser
3-4 Throttling (expansion) in the expansion device
4-1 Evaporation in the evaporator
1'-1 Superheating before compression
2-2' Superheating after compression

3-3' Sub-cooling during condensation

Performance characteristics for vapour compression refrigeration system:

1. Volumetric efficiency of compressor: The ratio of clearance volume v_0 to the swept volume v_p is called the clearance factor(c).

$$c= v_0 /v_p$$

This factor has been taken as 0.03(3%of v_p).

The expression of volume efficiency is given by-

$$\eta =1+c-c*(v_{suction}/v_{discharge})$$

On decreasing suction pressure, or increasing pressure ratio, the suction volume v_1 and hence the volumetric efficiency η_v decreases. At constant suction pressure, an increase in discharge pressure will cause a reduction in volumetric efficiency due to higher compression ratio.

2. Degree of superheat at exit to compressor, dsup2 (^{0}c): Superheating after compression increases the refrigerating effect and the amount of work done in the compressor. Since the increase in refrigerating effect is less as compared to the increase in work done, therefore, the net effect of superheating is to have low COP. But Superheating after compression up to some degree is necessary in order to ensure complete vaporization of refrigerant before entering to the condenser, as the condenser can't handle two phase liquid.

$$\text{Degree of superheat } \Delta T(k)=T_{sup}-T_{sat}$$

3. Refrigerating effect, R (kJ/kg): An ideal refrigerant should have higher refrigerating effect per kg. The refrigerating capacity decreases with decrease in evaporator temperature or pressure. Increase in condenser pressure results in decrease of compressor refrigerating capacity. The high latent heat of vaporization results in high refrigerating effect per kg refrigerant which reduces mass of refrigerant to be circulated per ton of refrigeration.

$$\text{Refrigerating effect R (kJ/kg)}=h_1-h_3$$

4. Work of compressor, w_c(k.j/kg): For a good refrigerant, work of compressor per kg should be as low as possible. Increase in discharge pressure or decrease in suction pressure will result in increase in specific work, w_c. With increase in suction volume, work of compressor increases.

Work of compressor W_c(kj/kg)= h_2-h_1

5. COP of Refrigeration cycle: It should be as high as possible. Decrease in suction pressure and increase in discharge pressure decreases the COP of the refrigerating system.

COP of Refrigeration cycle = Refrigerating effect R (kJ/kg)/ Work of compressor w_c(kj/kg)

6. Mass flow rate of refrigerant, m (kg/min): The mass flow of refrigerant in a reciprocating compressor is given by:

$$m=\left(v_p/\upsilon_1\right)*\eta_v$$

As the suction pressure decreases or discharge pressure increases the mass flow rate for unit ton refrigeration will increase.

7. Power of compressor P_c(kW): The power of compressor in a reciprocating compressor is given by work done per second.

Power of compressor P_c(kW) =Work of compressor W_c(kj/kg)* Mass flow rate of refrigerant, m (kg/min)

8. Heat rejected Q_r (kW): It is the amount of heat rejected in condenser by refrigerant.

Heat rejected in Q_r (kW): h_2-h_3

9. Dimensions of Compressor L & D (cm) : The bore and stroke of cylinder are the required dimensions of compressor.

10. Piston Displacement Va(cm^3/s): The volume of cylinder is known as piston displacement volume.

RESULT AND DISCUSSION

Performance Analysis of Refrigeration system: The performance analysis has been carried out by manual calculations for evaporator temperatures -10^0c,-5^0c, 0^0c, 5^0c,and 10^0c, and condenser temperatures 45^0c, 50^0c and 55^0c. The following parameters have been calculated by considering case1 & case2 separately, as discussed below.

Case 1: Without sub-cooling & superheating.

Case 2: With sub-cooling of 5^0c & superheating of 10^0c.

Case 1 : (No Superheat & Subcool)
Table 5.1: Properties at Te =-10^0C
Table 5.1.a: For T_c =45^0 c

S.N.	Parameters	Unit	R-12	R-134a	R-413a	R-423a
For unit kilogram of refrigerant						
1	Volumetric efficiency	η_v	0.89045	0.86131	0.85981	0.853164
2	Degree of superheat	ΔT (K)	9.183216	6.672174	4.85086	3.25245
3	Refrigeration Effect	R(kJ/kg)	103.48149	128.7666	116..262	96.076328
4	Work of compressor	W_c(kJ/gk)	28.413576	36.63578	35.1824	29.377087
5	C.O.P. of cycle	COP	3.6419735	3.514777	3.304457	3.2704510
For unit ton of refrigeration						
6	Mass flow rate	m(kg/min)	2.0293483	1.630857	1.80625	2.1857921
7	Power of compressor	P_c(kW)	0.96101	0.99579	1.05913	1.0701887
8	Heat rejected in KW	(Q_r)(kW)	4.461017	4.4957956.1	4.058	3.924
		D (cm)	4.577000	2546	5.930	6.569000

9	Dimensions of Compressor	L(cm)	6.334	7.3227	7.11246	7.31265
10	Piston Displacement	Va(cm^3/s)	2593.6273	2708.837	2682.48	2398.2240

Table 5.1: b For T_c=50^{0}C

S.N.	**Parameters**	**Unit**	**R-12**	**R-134a**	**R-413a**	**R-423a**
For unit kilogram of refrigerant						
1	Volumetric efficiency	η_v	0.8523	0.80486	0.8051	0.7955
2	Degree of superheat	ΔT (K)	23.880	16.5444	5.21337	3.97865
3	Refrigeration Effect	R(kJ/kg)	104.48	121.039	108.4067	89.0063
4	Work of compressor	W_c(kJ/gk)	30.6511	39.3226	37.7037	31.595
5	C.O.P. of cycle	COP	3.215	3.078	2.5752	2.817
For unit ton of refrigeration						
6	Mass flow rate	m(kg/min)	2.137	1.7349	1.9371	2.3593
7	Power of compressor	P_c(kW)	1.088	1.1370	1.217	1.2424
8	Heat rejected in KW	(Q_r)(kW)	4.588	5.268	4.058	3.924
		D (cm)	3.844	3.810	4.7172	4.296
9	Dimensions of Compressor	L(cm)	6.123	6.3227	6.3246	6.81265
10	Piston Displacement	Va(cm^3/s)	2881.779	2881.77	2876.88	3560.21

Table 5.1.c: For T_c=55^{0}C

S.N.	**Parameters**	**Unit**	**R-12**	**R-134a**	**R-413a**	**R-423a**
For unit kilogram of refrigerant						
1.	Volumetric Efficiency	η_v	0.9107	0.888	0.8861	0.883
2.	Degree of superheat	Δ T(k)	18.419	14.836	13.0439	12.6754
3.	Refrigeration Effect	R(kJ/kg)	117.144	147.916	135.900	114.38
4.	Work of compressor	W_c(kJ/kg)	26.639	34.309	33.0442	27.410
5.	C.O.P. of Cycle	COP	4.397	4.311	4.11516	4.173
For unit ton of refrigeration						
6.	Mass flow rate	m(kJ/kg)	1.7959	1.419	1.5452	1.8358
7.	Power of compressor	P_C(kW)	0.7959	0.8118	0.8505	0.8387
8.	Heat rejected in KW	Q_r(kW)	4.2959	4.3118	0.8505	0.8387
9.	Dimensions of Compressor	D(cm)	5.45	5.51	5.4702	5.82
		L(cm)	6.54	6.612	6.564	6.984
10.	Piston Displacement	Va(cm^3/s)	2013.82	2033.094	1979.455	2386.08

CLUSION AND FUTURE SCOPE

Conclusion

Following conclusion may be drawn from the comparison above.

1. R-423a and R-413a are almost drop in substitutes. No major hardware changes are required in the existing unit.
2. The optimum change of pressure R-423a and R-413a is approximately same as R-12
3. Theoretically, almost same power is required for running the refrigeration unit.
4. Evaporator temperatures are slightly higher in case of modification refrigerator.
5. Same type of minerals oils can be used but the viscosity required is higher
6. Same type of construction can be used.

Making an overall comparison, both refrigerants R-423a and R-413a are attractive alternatives to CFC-12. Research results are favorable, with no major drawback.

Future Scope

As we have seen from the theoretical analysis of the performance of the refrigerant of R-423a and R-413a, they are almost drop in substitutes. No major hardware changes are required in the existing unit from the theoretical analysis point of view. So after successfully testing in laboratory these refrigerant can be suitable retrofit for the vapour compression refrigeration system. For further research on this field exergy analysis of R-413a and R-423a refrigerant can be analyze. For further research mixture of other refrigerants can be used for analysis and software can be use for the analysis.

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