Purdue University
Purdue e-Pubs

International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

1998

Computational Comparison of R22 and R407C Air Conditioners with Rotary Vane Compressors

I. B. Vaisman
Mobile Climate Control Inc.

COMPUTATIONAL COMPARISON OF R22 AND R407C AIR CONDITIONERS WITH ROTARY VANE COMPRESSORS

IGOR B. VAISMAN
Mobile Climate Control Inc., Research & Development Manager
80 Kincort Street, North York, Ontario M6M 5G1, Canada

Computational comparison of performance of an air conditioner with a rotary vane compressor operating on R22 and R407C is produced taking into account actual system configurations. Exergy approach in evaluation of air conditioner thermodynamic efficiency is applied. Calculated results are presented. The paper shows that R407C is compatible with R22 in terms of air conditioner performance. However, optimal design for R22 system is not necessarily optimal for R407C and this should be taken into account while developing new air conditioners.

INTRODUCTION

When R22 is phased out it would be desirable that existing air conditioners, charged with replacement refrigerant, produce the same capacity and consume the same power. R407C is considered the best replacement for R22. The major concern and drawback of R407C is a glide of 5.4K.

Comparison of performance of air conditioners, heat pumps and refrigeration systems are made in references /1/ - /6/.

In general terms /1/ a comparison of performance of different refrigerants in the vapour compression cycle can be carried out on the basis of the internal temperatures of the refrigerant or on the basis of the external temperatures of the heat transfer fluids which acts as a heat source and a heat sink. In the reference /1/ the first approach is applied. An assumption is made that the heat exchangers have equivalent thermal transmittance values (i.e. the product of heat transfer area multiplied by the overall coefficient of heat transfer). Based on this assumption it is shown a substantial equivalence of the two refrigerants from point of view of energy efficiency for most current application of refrigeration.

Reference /2/-/5/ compares the performance of R22 and R407C at certain condensing and boiling temperatures. Reference /2/ defines that performance of R407C is the same as R22 in air conditioning applications, but the R407C cooling capacity and COP are lower for refrigeration applications. Reference /3/ states that R407C shows a lower cooling capacity and COP than R22 does. References /4/ and /5/ conclude that difference in COP and cooling capacity is fairly small.

In real life, air conditioner performance depends on actual system components - condensers, evaporators, compressors, condenser fans, evaporator blowers, suction pipe, etc. Different refrigerants show different heat transfer ratios and pressure drops in condensers and evaporators. The pressure drops increase discharge and decrease suction pressures. Moreover, R407C is a non-azeotropic refrigerant blend and this adds influence of glides of condensing and boiling isotherms. As a result operating discharge, suction, condensing and boiling pressures and temperatures that are built up in the system are different for different refrigerants. Therefore, the system with its real components, and not thermodynamic cycles only at some equivalent parameters should be compared. This approach, which is the second in the reference /1/, takes into consideration influence of both, thermodynamic and transport refrigerant properties.

In reference /6/ the performance evaluation was carried out in a psychometric calorimeter test facilities located at national Research Council Canada using Canadian Standards Association (CSA) and Air Conditioning and Refrigeration Institute (ARI) rating conditions. Performance of a residential central heat pump with reciprocating compressor operating on R22 and R407C was investigated. Cooling capacity and energy efficiency ratio of R407C appeared slightly lower than those of R22 for the same system arrangement.

The aim of the paper is to present a computational comparison of an air conditioner equipped with actual condenser and evaporator units and rotary vane compressor operating on R22 and R407C.

PRINCIPLES OF COMPUTATIONAL COMPARISON

Air conditioner system includes compressor, condenser unit, expansion valve and evaporator unit. The condenser unit includes condenser coil(s) and fan(s). The evaporator unit includes evaporator coil(s) and blower(s). The air conditioner shows performance based on compressor capabilities, coils' design, suction line design, discharge line design, liquid line design and performance of fans and blowers. The computational procedure defines performance and all operating parameters of the air conditioner, based on performance data of actual compressor, design parameters of evaporator and condenser coils, shape and length of suction, discharge and liquid pipes, air flow rates of the fans and blowers. Thermodynamic cycle is defined "inside" the air conditioner configuration with actual pressure drops in condenser, evaporator and suction line.

Performance of the rotary vane compressor operating on R407C has been defined from R22 performance data taking into account the fact that for both refrigerants volumetric and isoentropic efficiencies are the same at the same discharge pressure and pressure ratio.

Input of the calculation procedure is:

- outdoor and indoor conditions
- specification of compressor, which presents polynoms of volumetric and isoentropic efficiency on discharge and suction temperature
- specification of refrigerant that runs an equation of state and all related thermodynamic calculations
- specification of condenser coil design, which is associated with the number of rows, fins' pitch, finned length, fin's surface and shape, surface inside pipes, etc.
- specification of evaporator coil design, which is associated with the number of rows, fins' pitch, finned length, fin's surface and shape, surface inside pipes, etc.
- flow rates produced by condenser fans
- flow rates produced by evaporator blowers
- suction line length and shape that defines pressure drop in suction pipe
- discharge line length and shape that defines pressure drop in suction pipe
- liquid line length and shape that defines pressure drop in suction pipe

Calculated output includes:

- pressure temperature, density, enthalpy and entropy in each thermodynamic state of the air conditioner
- evaporator capacity, refrigerant and air flow pressure drops in the evaporator, logarithmic temperature difference and thermal transmittance of the evaporator, indoor air outlet temperature and relative humidity,
- condenser capacity, refrigerant and air flow pressure drops in the condenser, logarithmic temperature difference and thermal transmittance of the condenser, outdoor air outlet temperature
- compressor volumetric and isoentropic efficiency, mass flow rate, isoentropic work, power consumption
- COP and exergy efficiency of the air conditioner
- exergy efficiencies of the air conditioner components

Exergy efficiency of an air conditioner is defined as a relation of exergy capacity (output exergy) to exergy spent for air conditioner operation:

$$\eta_e = \frac{E_q}{E_q + \Delta E_{loss}}, \qquad (1)$$

where E_q - is exergy capacity, ΔE_{loss} - is exergy losses in the air conditioner.

By analogue with system exergy efficiency, exergy efficiency of the *i*-th system component is:

$$\eta_{ei} = \frac{E_q}{E_q + \Delta E^i_{loss}}, \tag{2}$$

where ΔE^i_{loss} - is exergy losses in the ***i***-th system component.

System exergy efficiency is connected with the component exergy efficiencies as follows:

$$\frac{1}{\eta_e} - 1 = \sum_{i=1}^{n}\left(\frac{1}{\eta_{ei}} - 1\right), \tag{3}$$

where ***n*** – is the number of the components in the air conditioner.

The equation (3) shows that increase of the lowest exergy efficiencies gives the strongest effect on the system exergy efficiency.

The air conditioner to be simulated consists of a rotary vane compressor with a displacement rate of 80.4 *cub. m* at 3000 *rpm*.

The condenser consists of 2 equal condenser coils. The condenser coil is of 1" x 0.866" staggered, 2 cycle sine wave fin made of aluminium, 10 fin per inch, 3 rows deep, 36 rows high and 48" of finned length. The number of circuits in the condenser coil is 18. Liquid subcooling at condenser outlet is of $9\,^0R$. Air flow rate provided by the condenser fans is 10000scfm per coil.

The evaporator consists of 2 equal evaporator coils. The evaporator coil is of 1" x 0.866" staggered, 2 cycle sine wave fin made of aluminium, 10 fin per inch, 4 rows deep, 36 rows high and 36" of finned length. The number of circuits in the condenser coil is 24. An expansion valve generates superheating of $9\,^0R$ at the evaporator coil outlet. Air flow rate provided by the evaporator blower is 4000csfm per coil.

Condenser and evaporator coils are arranged in order to approximate air to refrigerant flow as a cross-counter flow configuration.

Suction pipe is designed for 5psi of refrigerant pressure drop and additional refrigerant heating by $12.6\,^0R$. Refrigerant vapour pressure drop in discharge line after the compressor and in liquid line after the condenser are negligible.

Calculated results per one condenser coil and one evaporator coil for the air conditioner operating on R22 and R407C at outdoor temperature of $95\,^0F$, indoor temperature of $80.6\,^0F$ and relative humidity of 47% are presented in Table 1. Such calculations are also made for the operation at outdoor temperature of $104\,^0F$, indoor temperature of $95\,^0F$ and relative humidity of 30.74%. Results are presented in Table 2.

ANALYSIS OF CALCULATED RESULTS

Analysis of the data in Table 1 shows the following.

Cooling capacity of the air conditioner operating on R407C is higher than on R22 in a relation equal to 137.554/132.491 = 1.038. It is in spite that specific evaporator capacity of R407C is lower then the capacity of R22. The reason is in increased refrigerant mass flow rate.

The mass flow rate is proportional to a product of refrigerant vapour density and volumetric efficiency. Since volumetric efficiency is about the same for both refrigerants, the higher value of R407C refrigerant vapour density at the compressor suction defines the increase of mass flow rate.

Table 1

No.	Parameter	R22	R407C
1	Suction Temperature, ^0F Suction Pressure, psig Density at Compressor Suction, kg/cub. m	67.25 71.99 23.57	75.58 80.55 25.45
2	Discharge Temperature, ^0F Isoentropic Discharge Temperature, ^0F Discharge Pressure, psig	181.75 177.24 257.58	184.94 180.70 317.51
3	Saturated Temperature at Condenser Outlet, ^0F Temperature at Condenser Outlet, ^0F Pressure at Condenser Outlet, ^0F	119.06 110.06 256.52	124.00 115.00 316.28
4	Saturated Temperature at Evaporator Inlet, ^0F Pressure at Evaporator Inlet, ^0F Vapour Quality, %	46.79 78.79 22.072	47.76 88.50 28.910
5	Temperature at Evaporator Outlet, ^0F Pressure at Evaporator Outlet, psig	54.65 76.99	62.98 85.55
6	Mass Flow Rate, lb/h	1947.6	2092.59
7	Pressure Ratio Specific Isoentropic Work, *kJ/kg* Isoentropic Compressor Work, *hp* Compressor Work, *hp* Compressor Volumetric Efficiency, % Compressor Isoentropic Efficiency, %	3.141 30.745 10.114 16.414 93.25 61.62	3.488 33.289 11.766 17.289 92.77 68.05
8	Specific Condenser Capacity, *kJ/kg* Condenser Capacity, *kBTUH* Thermal Transmittance, $kW/^0K$ Logarithmic Mean Temperature Difference, ^0F Pressure Drop in Condenser, psig	194.956 161.364 5328.5 15.98 1.06	192.826 171.128 3558.0 25.37 1.23
9	Specific Evaporator Capacity, *kJ/kg* Evaporator Capacity, *kBTUH* Thermal Transmittance, $kW/^0K$ Logarithmic Mean Temperature Difference, ^0F Pressure Drop in Evaporator, psig	158.246 132.491 3433.9 20.36 1.80	152.910 137.554 4448.2 16.32 2.95
10	Cycle COP System COP	5.147 3.171	4.593 3.126
11	Air Conditioner Exergy Efficiency, %	8.45	8.33
12	Compressor Exergy Efficiency, % Condenser Exergy Efficiency, % Expansion Valve Exergy Efficiency, % Evaporator Exergy Efficiency, % Suction Line Exergy Efficiency, %	18.05 30.67 52.35 27.17 69.44	20.70 25.18 29.94 40.82 70.92

Condenser capacity of R407C is higher than the capacity of R22, although specific condenser capacity is about the same for both refrigerants. This is also because of the increased mass flow rate.

System COP of the R407C air conditioner appears lower than the COP of the R22 – 3.126/3.176 = 0.984. Relation of R407C cycle COP to R22 cycle COP is equal to 4.593/5.147 = 0.892.

Table 2

No.	Parameter	R22	R407C
1	Suction Temperature, ^0F Suction Pressure, psig Density at Compressor Suction, kg/cub. m	74.36 83.59 26.63	82.16 92.66 28.64
2	Discharge Temperature, ^0F Isoentropic Discharge Temperature, ^0F Discharge Pressure, psig	194.72 189.70 302.74	196.78 192.8 369.00
3	Saturated Temperature at Condenser Outlet, ^0F Temperature at Condenser Outlet, ^0F Pressure at Condenser Outlet, ^0F	131.27 122.27 301.83	135.64 126.64 367.93
4	Saturated Temperature at Evaporator Inlet, ^0F Pressure at Evaporator Inlet, ^0F Vapour Quality, %	53.88 90.53 24.67	54.85 100.86 32.58
5	Temperature at Evaporator Outlet, ^0F Pressure at Evaporator Outlet, psig	61.76 88.59	69.56 97.66
6	Mass Flow Rate, lb/h	2182.08	2360.8
7	Pressure Ratio Specific Isoentropic Work, *kJ/kg* Isoentropic Compressor Work, *hp* Compressor Work, *hp* Compressor Volumetric Efficiency, % Compressor Isoentropic Efficiency, %	3.23 31.517 11.616 21.936 92.47 52.95	3.574 33.882 13.510 20.129 93.02 67.12
8	Specific Condenser Capacity, *kJ/kg* Condenser Capacity, *kBTUH* Thermal Transmittance, $kW/^0K$ Logarithmic Mean Temperature Difference, ^0F Pressure Drop in Condenser, psig	187.938 173.890 4912.8 18.67 0.91	183.441 183.593 3574.4 27.099 1.07
9	Specific Evaporator Capacity, *kJ/kg* Evaporator Capacity, *kBTUH* Thermal Transmittance, $kW/^0K$ Logarithmic Mean Temperature Difference, ^0F Pressure Drop in Evaporator, psig	150.356 141.041 3380.7 22.01 5.32	142.832 144.958 4275.6 17.89 3.2
10	Cycle COP System COP	4.771 2.526	4.216 2.829
11	Air Conditioner Exergy Efficiency, %	4.10	4.59
12	Compressor Exergy Efficiency, % Condenser Exergy Efficiency, % Expansion Valve Exergy Efficiency, % Evaporator Exergy Efficiency, % Suction Line Exergy Efficiency, %	8.01 19.49 34.96 16.08 58.8	12.25 16.00 18.66 23.04 59.52

Cycle COP relates specific evaporator capacity to specific isoentrropic compressor work. It takes into account pressure drops, which are sources of energy dissipation. That is why the specific condenser capacity, specific evaporator capacity and specific compressor look as unbalanced. System COP includes actual compressor, which is the main source of irreversibility losses. Since the exergy losses of the R22 compressor are higher then the losses of R407C, then the system COP of the air conditioner operating on R407C looks much better in comparison with system COP of R22 air conditioner than cycle COP does.

Exergy efficiencies of both compressors are lower than any from all other exergy efficiencies since the compressors are the main source of irreversibility and that is why any improvement of compressor efficiencies strongly influences on air conditioner efficiency. The R407C compressor shows exergy efficiency higher than the R22 one due to its better isoentropic efficiency. R22 condenser exergy efficiency is higher than the efficiency of R407C. Exergy efficiency of the R22 expansion valve is much higher than exergy efficiency of the R407C expansion valve. It is because vapour quality and pressure ratio of the R22 air conditioner is lower. Exergy efficiency of the evaporator operating on R407C is higher than exergy efficiency of the evaporator operating on R22. The reason is in the boiling pressure glide of R407C, which is helpfully straightened by pressure drop in the evaporator.

It is known from the entropy analysis that the lower temperature level is the lower values of temperature heads should appear. Since current air conditioner configuration is optimised for use of R407C at the stated indoor and outdoor conditions, logarithmic mean temperature difference in its condenser is higher than that in its evaporator. Logarithmic mean temperature difference of R22 evaporator is higher than that in the condenser. It means that the current configuration is not optimised for use of R22 at the stated indoor and outdoor conditions.

Table 2 shows that increase of operating temperature level decrease performance of R407C air conditioner in relation to R22 application. Cooling capacity of an air conditioner operating on R407C is higher than on R22 – 144.958/141.041 = 1.028. System COP of the R407C air conditioner appears lower 2.526/2.829 = 0.893.

CONCLUSIONS

A computational comparison of R22 and R407C air conditioners with rotary vane compressors has been produced.

Cooling capacity of the air conditioner operating on R407C at outdoor temperature of 95 ^0F, indoor temperature of 80.6 ^0F and relative humidity of 47% is higher than on R22 by factor of 1.038. System COP of the R407C air conditioner appears lower by factor 0.984.

Cooling capacity of an air conditioner operating on R407C at outdoor temperature of 104 ^0F, indoor temperature of 95 ^0F and relative humidity of 30.74% is higher than R22 by factor 1.028. System COP of the R407C air conditioner appears lower by factor 0.893.

R407C is compatible with R22 in terms of air conditioner performance. However, optimal design for R22 system is not necessarily optimal for R407C and this should be taken into account while developing new air conditioners.

REFERENCES

1. S. Bobbo, R. Camporese, G. Cortella, E. Fornasieri, 1995. Theoretical Evaluation of the Performance of Zeotropic Mixtures in Refrigerating Cycles, 19-th International Congress of Refrigeration, The Hague, Aug. 20-25.
2. N. J. Hewitt, J.T. McMulan and B. Mongey, 1995. The Replacement of R22 in Refrigeration Systems, 19-th International Congress of Refrigeration, The Hague, Aug. 20-25.
3. P. Weiss, M. Barreau, S. Macaudiere, 1995. Field Tests using HCFC-22 replacements, 19-th International Congress of Refrigeration, The Hague, Aug. 20-25
4. K. Berglof, J. Morley, 1996. Practical Experience in Use of R407C in Smal Chillers and Heat Pumps in Sweden. International Refrigeration Conference, Purdue University, West Lafayette, July 23-26.
5. P. Lindqvist, L. Herbe, 1996. Laboratory Evaluation and field Tests of Replacements for R22 and R502. International Refrigeration Conference, Purdue University, West Lafayette, July 23-26.
6. J.W. Linton, W.K. Snelson, P.F. Hearty and A.R. Triebe, F.T. Murphy, R.E. Low and B.E. Gilbert, 1996. Comparison of R407C and R410A with R22 in a 10.5 kW (3.0 TR) Residential Central Heat Pump. International Refrigeration Conference, Purdue University, West Lafayette, July 23-26.