Performance Analysis of Reciprocating Refrigerant Compressor

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Abstract: ***In this work, a performance analysis of the compression cycle of a reciprocating compressor for domestic refrigeration was carried out. A numerical simulation model of a refrigerant reciprocating compressor has been developed in MATLAB Simulink R2009a. An advantage of numerical simulation model of a reciprocating compressor is for the design, development, improvement and optimization of the elements constituting the compressor circuit. A specific numerical simulation model of the whole vapour compression refrigeration system has been developed coupling different simulation models for each element (i.e., evaporator, compressor ,condenser and expansion valve), more detailed mathematical models has been incorporated in the simulation model of the compressor and adaptability to different configurations without changing the program. Also, to match with the actual working condition of a reciprocating compressor, different losses such as pressure drop, leakage losses etc. is also incorporated in the simulation model of the compressor. Result presented show the influence of different aspects (speed, l/d ratio, geometry, working conditions) on the basis of meaningful non-dimensional parameters, which describe the compressor behavior (volumetric efficiency, coefficient of performance). The refrigerants taken for the analysis in this paper is R134a, but any refrigerants would be considered for the analysis. The idea of this paper is to give a better understanding of the performance behavior under different aspects to improve the design of this equipment.***

Keywords: Reciprocating Compressor; Numerical Simulation Model; Calculation; Parameter; Performance Analysis; Geometry

1. Introduction

A vapour compression refrigeration system comprised of four components: compressor, evaporator, condenser and expansion device. Out of which, the most power consumption device is a compressor. Hermetic reciprocating compressors are widely used in refrigerators, residential air conditioners etc. These compressors have the advantage of small volume, low levels of noise and energy consumption absence of leakage to the ambient especially at low back pressure and also less effect on global warming. So, the need of the detailed performance analysis of the reciprocating compressor is there. Based on this analysis, the effect of various performance parameters and geometrical parameters would be found out. This will ease our understanding about the behavior of compressor when altering the speed of the compressor (N), Evaporator Temperature (Te), Condenser Temperature (Tc), L/D Ratio of the compressor etc. Rigola et, al.[1] develop a numerical simulation model for the parametric studies on hermetic reciprocating compressors and based on the result of the simulation model, different performance curves has been plotted, showing the influence of different aspects which is also carried out by experimental verification and validation. There were authors [2]-[11], who evaluate thermodynamically the behavior of hermetic reciprocating compressors at periodical conditions, by means of global energy and mass balances including spring-mass valve dynamic models. Models by Perez-Segarra et al. [12], [13] and Rigola et al.[14], which are based on full integration of the one dimensional and transient governing equation in their simulation model. In this paper, a specific numerical simulation model of the whole vapour compression refrigeration system has been developed coupling different simulation models for each element (i.e., evaporator, compressor, condenser and expansion device). With the help of different references mentioned in this paper and by considering different losses a better simulation model has been developed. And a closed analysis has been done by drawing the graphs between different parameters.

Nomenclature

m Mass flow rate (Kg/s) Tc Condenser Temperature (^{0}C)
TeEvaporator Temperature (^{0}C) Pe Evaporator Pressure (bar)
η_V Volumetric efficiency Pc Condenser Pressure (bar)
C Clearance factor n Isentropic Index
V_pSwept volume (m³/s)
L Length of strokeSubscripts
DBore diameter1 at the inlet of the compressor
Q Refrigerating effect (KW)2 at the outlet of the compressor
H Enthalpy of suction vapour 3 at the outlet of the condenser
(saturated type) (KJ/Kg)
4 at the inlet of the evaporator
$h_f(h_4)$Enthalpy of liquid refrigerant at
condenser temperature (saturated type)
(KJ/Kg)
Wc Compressor work (KW)
N Speed of the compressor (r.p.m)
v Specific volume (m³/Kg)

2. Mathematical Model

The mathematical model of a vapour compression refrigeration system will include the preparation of mathematical model of each elements (i.e., evaporator, compressor, condenser and expansion valve), which consists of various thermodynamic relations. For the analysis following assumptions are made:

1. Bore diameter of cylinder (D) = 64 mm
2. Stroke length of cylinder (L) = 56 mm
3. Clearance factor (C) = 0.04
4. Pressure drop at suction = 0.2 bar
5. Pressure drop at discharge = 0.4 bar
6. Evaporator temperature (Te) = -10^{0}C, -8^{0}C, -6^{0}C, -4^{0}C and -2^{0}C
7. Condenser temperature (Tc) = 30^{0}C
8. Isentropic index (n) = 1.118 (for R-134a)
9. Speed of the compressor (N) = 1080 r.p.m, 1180 r.p.m, 1280 r.p.m, 1380 r.p.m and 1480 r.p.m
10. Length of stroke to Bore diameter ratio (L/D ratio) = 0.875, 0.9206, 0.9677, 1.016 and 1.067

2.1 Thermodynamic Relations Used

2.1.1Volumetric Efficiency

It is denoted by η_v

$\eta_v = 1 + C - C(\frac{Pc}{Pe})^{1/n}$....equation (1)

2.1.2 Mass Flow Rate

It is denoted by m.

$m = (\eta_V \times V_p) \div V_1$, Kg/s .... equation (2)

Where,

V_p=Swept Volume,m³/s

V_p=((π/4)*D²*L*N)/60 .... equation(3)

2.1.3 Refrigeration effect

It is denoted by Q.

$Q = m \times (h_1 - h_f)$.... equation (4)

2.1.4 Compressor work

It is denoted by W_c

$W_c = m \times (h_2 - h_1)$.... equation (5)

2.1.5 Coefficient of performance

It is denoted by $C.O.P$

$C.O.P = Q \div W_c$ .... equation (6)

2.1.6 Losses Considered

Following are the losses which had considered in the simulation model of compressor

a. Pressure Drop

For the flow of any fluid, the pressure must drop in the direction of flow. Both suction and discharge valves will open only when there is a pressure drop across them. As a result of throttling or pressure drop on the suction side the pressure inside the cylinder at the end of the suction stroke is P_s while the pressure at the suction flange is P_1. The pressure in the cylinder rises to the suction flange pressureP_1. The pressure in the cylinder rises to the suction flange pressure P_1 only after the piston has travelled a certain distance inward during which the volume of the fluid has decreased from $(V_p + V_c)$to V_1.

Considering the pressure drop at suction is 0.2 and Pressure drop at discharge is 0.4

Therefore, the volumetric efficiency,

$\eta_v = (1 + C)(\frac{P_s}{P_1})^{1/n} - C(\frac{P_d}{P_1})^{1/n}$.... Equation (7)

b. Leakage loss

The effect of leakage past piston rings and under the suction valve elements is normally accounted for by allowing 1.5 per cent leakage per unit of the compression ratio r, which is equal to P_2/P_1.

Old worn out compressors tend to have more leakage and hence they lose their cooling capacity.

2.1.7 Overall Volumetric Efficiency

$\eta_v = (1 + C)\left(\frac{P_s}{P_1}\right)^{1/n} - C\left(\frac{P_d}{P_1}\right)^{1/n} - 0.015r$...Equation (8)

3. Analysis of the Results

Based on the above simulation model, various graphs has been plotted by varying different parameters, the graphs which are plotted are based on the following parameters concerned -:

1. By varying the speed of the compressor at different fixed evaporator temperature (i.e., -10^{0}C,-8^{0}C,-6^{0}C,-4^{0}C and -2^{0}C)
2. By varying the evaporator temperature at different fixed speed of the compressor
(i.e., 1080 r.p.m, 1180r.p.m,
1280 r.p.m, 1380r.p.m and 1480 r.p.m)
3. By varying the L/D ratio (i.e., Length of stroke to Bore diameter ratio) at different fixed evaporator temperature

All numerical experiments have been performed assuming a single stage vapour compression low pressure cycle at the following conditions: both outlet compressor temperature and inlet condenser temperature of 30^{0}C and evaporation temperature of -10^{0}C, -8^{0}C, -6^{0}C, -4^{0}C and -2^{0}C.

3.1 By varying the speed of the compressor at different fixed evaporator temperature (Compressor Speed influence)

The influence of the compressor speed for different evaporator temperature has been numerically studied. Figure 4 to figure 8 shows the C.O.P, mass flow rate (kg/s), refrigerating effect (KW) and compressor work (KW) evolution depending on compressor speed for different evaporator temperature.

The comparative results shows that the C.O.P will remain constant for a particular temperature and it will increase with the increase in evaporator temperature. This will increase around 23.48% from evaporator temperature of -10^{0}C to -8^{0}C.

From equation (6),

C.O.P = Q ÷ Wc

From equation (4) and equation (5),

Q = m × (h_1-h_f)

Wc = m × (h_2- h_1)

Therefore, the above equation can be written as,

C.O.P = (h_1-h_f) ÷ (h_2- h_1)

Therefore, the m i.e., mass flow rate will be cancelled out from both numerator and denominator and thus, the resultant C.O.P will only be the ratio of the change of enthalpy in evaporator and compressor. So, it can be seen that there will be no effect of mass flow rate (m) on the C.O.P of the system and therefore, there will be no effect of change of speed of compressor on the C.O.P of the system. Therefore, the C.O.P of the system will remain constant for the particular temperature. But, as the evaporator temperature increases the refrigeration effect $(h_1\text{-}h_f)$ increases marginally and the compressor work $(h_2\text{-}\ h_1)$ reduces sharply. As a result the C.O.P of the system increases rapidly as the evaporator temperature increases as shown in figure 4.

The comparative results between evaporator temperatures from -10^{0}C to -8^{0}C presents an increase of mass flow rate around 11% at low compressor speed and is also around 11% at high compressor speed. This is due to the fact that the mass flow rate (m) is directly proportional to swept volume (V_p) and the swept volume is directly proportional to the compressor speed (N), so as the compressor speed increases, the mass flow rate also increases as shown in figure 5. Also by increasing the evaporator temperature, the volumetric efficiency (η_V) increases and specific volume at the inlet of the compressor (v_1) decreases and the mass flow rate is directly proportional to the volumetric efficiency and is inversely proportional to the specific volume at the inlet of the compressor. Therefore, the resultant mass flow rate (m) increases by increasing the particular evaporator temperature.

The comparative result between evaporator temperatures from -10□C to -8□C present an increase of refrigerating effect around 12% at low compressor speed and is also around 12% at high compressor speed. This is due to the fact that the mass flow rate of refrigerant is directly proportional to the compressor speed, so as the compressor speed increases the mass flow rate of refrigerant also increases which will increase the refrigerating effect, so the refrigerating effect increases with the compressor speed. Also, the change in enthalpy of evaporator increases marginally as the evaporator temperature is increased. Therefore, the refrigerating effect increases by increasing the particular evaporator temperature as shown in figure 6.

The comparative results between evaporator temperature from -10^{0}C to -8^{0}Cpresent an increase of compressor work around 5% at low compressor speed and is also around 5% at high compressor speed. This is due to the fact that the compressor work is the product of mass flow rate of refrigerant and change in enthalpy in compressor. And the mass flow rate increases with the increase in compressor speed and thus, the compressor work is also increases with the compressor speed. Also, as the evaporator temperature increases, the compressor work is also increases up to the certain evaporator temperature at which the peak point for the compressor work lies and it start decreasing to the right of that peak point. Therefore, initially the compressor work increases with the increase in evaporator temperature as shown in figure 7.

Table 1: Numerical results of compressor speed influence at Te = -10^{0}C

N (r.p.m)	*Q (KW)*	*Wc (KW)*	*m (Kg/s)*	*C.O.P*	η_V*(%)*
1080	3.614	0.6045	0.02392	5.978	73.9
1180	3.949	0.6605	0.02613	5.978	73.9
1280	4.283	0.7165	0.02835	5.978	73.9
1380	4.618	0.7725	0.03056	5.978	73.9
1480	4.953	0.8285	0.03278	5.978	73.9

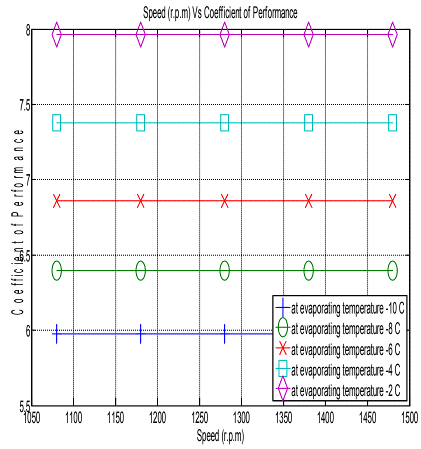

Figure 1: Speed (r.p.m) Vs Coefficient of Performance

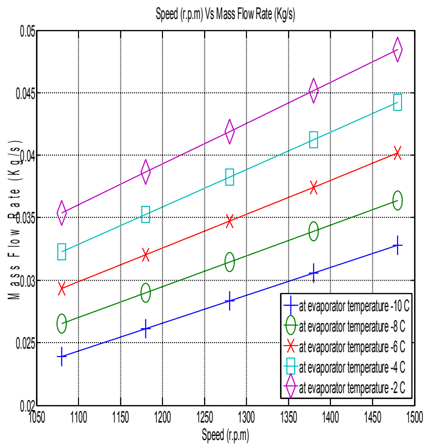

Figure 2: Speed (r.p.m) Vs Mass Flow Rate (Kg/s)

3.2 By varying the evaporator temperature at different fixed compressor speed (Evaporator Temperature influence) -:

The influence of evaporator temperature for different compressor speed has been numerically studied. Figure 9 to figure 13 shows C.O.P, refrigerating effect, volumetric efficiency and compressor work evolution depending on evaporator temperature for different compressor speed. The comparative result shows that the C.O.P of the system increases with the evaporator temperature. This is due to the fact that as the evaporator temperature increases the refrigeration effect increases marginally and the work of compression reduces sharply. As a result the C.O.P of the system increases rapidly as the evaporator temperature increases as shown in figure 9. Also, there will be no effect of compressor speed on the C.O.P of the system; therefore, the C.O.P will be constant for the particular speed as shown in figure 9.

The comparative result between compressor speed from 1080 r.p.m to 1180 r.p.m present an increase of refrigeration effect around 10% at low evaporator temperature and is also around 10% at high evaporator temperature. This is due to the fact that the refrigeration effect is the product of mass flow rate and change in enthalpyin evaporator and the mass flow rate is directly proportional to the compressor speed, so the increase in the compressor speed will also increase the refrigeration effect. Also, for a particular compressor speed the refrigeration effect increases with the evaporator temperature. This is due to the fact that the mass flow rate of refrigerant is directly proportional to the volumetric efficiency and inversely proportional to the specific volume of refrigerant entering the compressor. And with the increasing temperature, the volumetric efficiency increases but the specific volume of refrigerant entering the compressor decreases, which will increase the resultant mass flow rate of refrigerant. Also, the change in enthalpy in evaporator increases with the evaporator temperature, which will increase the product of mass flow rate of refrigerant and change in enthalpy in refrigerant. Therefore, the refrigeration effect increases with the evaporator temperature as shown in figure 10.

The comparative result shows that the volumetric efficiency increases with the evaporator temperature. This is due to the fact that as the evaporator temperature increases, the evaporator pressure also increases, which will decreases the pressure ratio (Pc/Pe), which will increases the volumetric efficiency. Therefore, the volumetric efficiency increases with the evaporator temperature. Also, there will be no effect of compressor speed on the volumetric efficiency as shown in figure 11.

The comparative result between compressor speed from 1080 r.p.m to 1180 r.p.m present an increase of compressor work around 10% at low evaporator temperature and is also around 10% at high evaporating temperature. This is due to the fact that the mass flow rate of refrigerant increases with the compressor speed and the compressor work is the product of mass flow rate of refrigerant and change in enthalpy in compressor. Therefore, the compressor increases with the increase in compressor speed. Also, the compressor work increases with the increase in evaporating temperature as shown in figure 12.

Table 2: Numerical results of evaporator temperature (Te) influence at N = 1080 r.p.m

Te (⁰C)	*Q (KW)*	*Wc (KW)*	*m (Kg/s)*	*C.O.P*	*η_V (%)*
-10⁰C	3.614	0.6045	0.0239	5.978	74
-8⁰C	4.042	0.6322	0.0265	6.394	76.2
-6⁰C	4.500	0.6562	0.0293	6.858	78.2
-4⁰C	4.989	0.6763	0.0323	7.377	80
-2⁰C	5.512	0.6922	0.0354	7.962	81.7

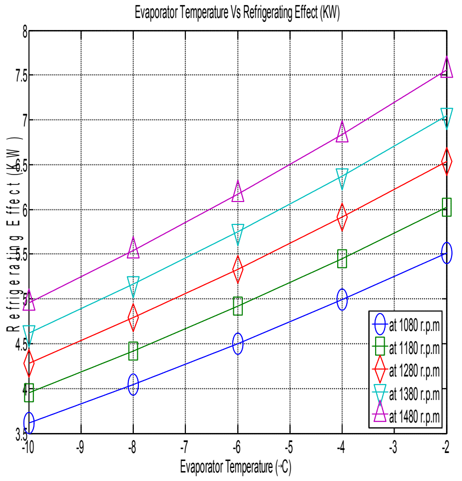

Figure 3: Evaporator Temperature (^{0}C) Vs Refrigerating Effect (KW)

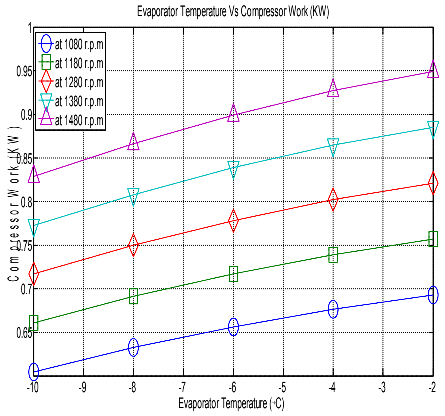

Figure 4: Evaporator Temperature (^{0}C)Vs Compressor Work (KW)

3.3 By varying the L/D ratio (Length of stroke to Bore diameter ratio) at different fixed evaporator temperature (Stroke to bore ratio influence)

The influence of L/D ratio for different evaporator temperature has been numerically studied. Figure 14 to figure 17 show compressor work, mass flow rate, C.O.P and refrigerating effect evolution depending on evaporator temperature for different compressor speed. A whole group of numerical simulation results has been obtained modifying the stroke to bore ratio, although maintaining the compressor chamber volume and getting the resultant numerical results at different evaporator temperature. Results show that compressor work has a maximum value and then begins to decrease with the increase in L/D ratio. The compressor work maximum value depends on the evaporation temperature. The maximum compressor work is around 1.0164 of stroke to bore ratio at -2^{0}C of evaporation temperature and around 1.0164 of stroke to bore ratio at -4^{0}C. The compressor work increases by increasing the particular evaporator temperature as shown in figure 14.

Results shows that the mass flow rate of refrigerant is maintained almost constant with a small increase when stroke to bore ratio increases. Also, the mass flow rate of refrigerant increases with the increase in particular evaporator temperature as shown in figure 15. Results shows that the C.O.P will be constant for a particular compressor temperature but it will increases with the increase in L/D ratio. Also, the C.O.P is more at high evaporator temperature as compare to the C.O.P at low evaporator temperature as shown in figure 16.

Results shows that the refrigerating effect has a maximum value and then begins to decrease with the increase in L/D ratio as shown in Figure 17. Differences on compressor work between stroke to bore ratios of 0.8750 and 1.0164 are approximately 1% for evaporator temperatures of -10, -8, -6, -4 and -2^{0}C. Differences on refrigerating effect for the same L/D ratios are approximately 1% for the same evaporation temperatures values.

Table 3: Numerical results of stroke-to-bore ratio influence at Te = -10^{0}C

L/D	*Q (KW)*	*Wc (KW)*	*m (Kg/s)*	*C.O.P*	η_V *(%)*
0.875	5.1002	0.8532	0.0338	5.9781	76.2
0.921	5.1186	0.8562	0.0339	5.9781	76.2
0.968	5.1283	0.8579	0.0339	5.9781	76.2
1.016	5.1297	0.8581	0.0339	5.9781	76.2
1.067	5.1230	0.8570	0.0339	5.9781	76.2

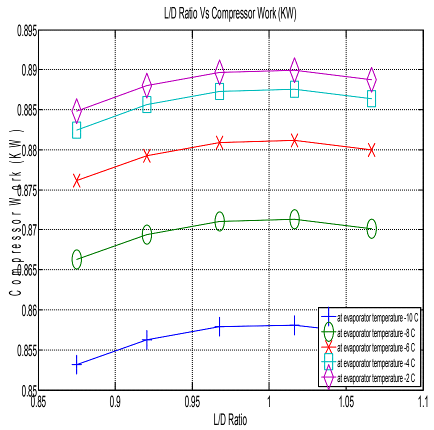

Figure 5: L/D Ratio Vs Compressor Work (KW)

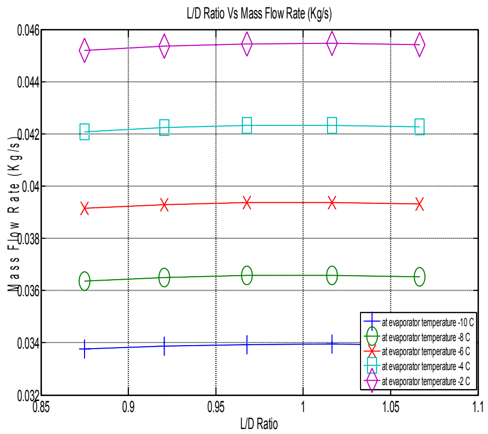

Figure 6: L/D Ratio Vs Mass Flow Rate (Kg/s)

4. Conclusions

An advanced numerical simulation model of refrigerant reciprocating compressor has been used in a wide range of geometries and working conditions. A parametric study of reciprocating compressors behavior has been presented based on the numerical simulation mentioned above. The simulation model was able to evaluate the mass flow rate (m), refrigeration effect (Q), compressor work (Wc), volumetric efficiency (η_V) and coefficient of performance (C.O.P) of the whole refrigeration system based on the specific input parameters and varying some input parameters. Results presented of C.O.P show, there is no effect of compressor speed on it but it changes with the evaporator temperature showing its dependency on evaporator temperature. The result presented of refrigeration effect and compressor work shows that both depend on the compressor speed, evaporator temperature and L/D ratio. The result presented of volumetric efficiency shows that it will only depend on evaporator temperature and does not depends on compressor speed and L/D ratio. The result presented of mass flow rate of refrigerant shows that it will depends on compressor speed, evaporator temperature and L/D ratio. The simulation model can be easily adapted to different compressor geometry and different operative fluids; therefore, it can be useful tool for the analysis, the design and the development of the reciprocating compressors. Any new parameters can be introduced without affecting the whole model. The design of highly efficient reciprocating compressors increasingly requires the adaption of simulation methodologies to reduce the costs associated with the design of new products. The present study also made evident the impact of different parameters which need to be optimizing so as to increase the performance and to reduce the various losses incurred. The simulation model of the reciprocating compressor offers flexibility in handling complex compressor circuitry and quick adaptability to new arrangements by addition or removal of the required elements.

References

[1] J. Rigola, C.D. Perez-Segarra, A. Oliva, Parametric studies on hermetic reciprocating compressors, 2004. P.253-266.

[2] W. Soedel. Introduction to computer simulation displacement type compressors. Purdue University Short Courses, IN, USA; 1972.

[3] P. Pandeya, W. Soedel, A generalized approach toward compressor performance analysis, Compressor Technology Conference, Purdue University 1978; 135–143.

[4] R. Prakash, R. Singh, Mathematical modelling and simulation of refrigerating compressors, Proceedings of the International Compressor Engineering Conference, Purdue University, IN, USA, 1974. p. 274–285.

[5] P.J. Singh, A digital reciprocating compressor simulation program including suction and discharge piping, Proceedings of the International Compressor Engineering Conference, Purdue University, IN, USA, 1984. p. 129–138.

[6] F. Fagotti, M.L. Todescat, R.T.S. Ferreira, A.T. Prata, Heat transfer modelling in a reciprocating compressor, Proceedings of the International Compressor Engineering Conference, Purdue University, West Lafayette, IN, USA, 1994. p. 605–610.

[7] A. Cavallini, L. Doretti, G.A. Longo, L. Rosseto, B. Bella, A. Zanneiro, Thermal analysis of a hermetic reciprocating compressor, Proceedings of the International Compressor Engineering Conference, Purdue University, IN, USA, 1996. p. 535–540.

[8] P. Stouffs, M. Tazerout, P. Wauters, Thermodynamic analysis of reciprocating compressors, Int J Thermal Sci 40 (2001) 52–66.
[9] K.T. Ooi, Heat transfer study of a hermetic refrigeration compressor, Appl Thermal Engng 23 (2003) 1931–1945.
[10] B. Raja, S. Joseph Sekhar, D. Mohan Lal, A. Kalanidhi, A numerical model for thermal mapping in a hermetically sealed reciprocating refrigerant compressor, Int J Refrig 26 (2003) 652–658.
[11] C.A. Longo, A. Casparella, Unsteady state analysis of the compression cycle of a hermetic reciprocating compressor, Int J Refrig 26 (2003) 681–689.
[12] C.D. Perez-Segarra, J. Rigola, A. Oliva, Modelling and numerical simulation of the thermal and fluid dynamic behaviour of hermetic reciprocating compressors – Part 1:theoretical basis. Int.j. Heating, Ventilating, Air-Condition. Refrig. Res. 9, (2003) 215-236.
[13] C.D. Perez-Segarra, J. Rigola, A. Oliva, Modelling and numerical simulation of the thermal and fluid dynamic behaviour of hermetic reciprocating compressors – Part 2:experimental investigation. Int.j. Heating, Ventilating, Air-Condition. Refrig. Res. 9, (2003) 237-249.
[14] J. Rigola, G. Raush, C.D. Perez-Segarra, A. Oliva, Detailed experimental validation of the thermal and fluid dynamic behaviour of hermetic reciprocating compressors, Int. J. Heating, Ventilating, Air-Condition, Refrig. Res. 10, (2004) 291-306.
[15] T. Morel, R. Keribar, Comprehensive model of a reciprocating compressor applicable to component design issues, Proceedings of the International Compressor Engineering Conference, Purdue University, IN, USA, 1988. P. 375-384.
[16] S. Porkhial, B. Khastoo, M.R. ModarresRazavi, Transient characteristics of reciprocating compressors in household refrigerators, Appl Thermal Engng 22 (2002) 1391-1402.