Journal of Engineering and Development, Vol. 18, No.5, September 2014, ISSN 1813- 7822

Exergy Analysis and Thermodynamic Model for Reciprocating and Scroll Compressors Used in an Air Conditioning Packaged Unit

Prof. Dr. Sabah Tarik Ahmad *Prof. Dr. Wahid S. Mohammad*
Lecturer. Dr. Louay Abd-Alazez Mahdi

Machine and Equipment department., University of Technology

Abstract :

An exergical thermodynamic model was used to analyze and optimize reciprocating and scroll compressors of an air condition packaged unit. The experimental work was carried out using 3.0 TR packaged unit manufactured by Carrier Company. The experimental and the analysis show that the exergy dissipative due to friction losses in bearings, suction and discharge valves and the transformation of power are larger than the heat losses. The exergy efficiency for the compressor was varying between (60-68) %. The scroll compressor was found to be better than the reciprocating compressor at an environmental temperature of 35℃ due to its low total losses which is 10% less than the reciprocating compressor ones . The heat losses percentages were 3% for scroll and 2% for the reciprocating of the total power input. The thermodynamic model has shown to be reliable in dealing with a change in the environmental temperatures and such system components and size.

Keywords: Exergy analysis, thermodynamic model, reciprocating compressor, scroll compressor.

تحليل للطاقة المتاحة مع نموذج ثرموديناميكي للضواغط الترددية و الاوربيتالية في وحدة تكييف مجمعه

ا.د. صباح طارق احمد ا.د. وحيد شاتي محمد م.د. لؤي عبد العزيز مهدي
قسم هندسة المكائن والمعدات / الجامعة التكنولوجية

الخلاصة :

تم استخدام موديل ديناميكي حراري باستخدام مبدأ الطاقة المتاحه (Exergy) لتحليل وتحديد أداء نوعين من الضواغط الأول ترددي والأخر اوربيتالي ،مثبتين في جهاز تكييف سعة 3طن تبريد مصنع من قبل شركة Carrier . بينت النتائج التحليلية والمختبرية ان خسائر الاحتكاك اكبر من الخسائر الحرارية. وان الكفاءة الحرارية المتقدمة تتراوح بين 60-68 %. وجد ان الضاغط الاوربتالي أفضل من الضاغط الترددي عند درجة حرارة جو قدرها 35 م بسبب قلة خسائره بمقدار 10 % مقارنة بالضاغط الترددي. تبين ان الخسائر الحرارية للضاغط الاوربتالي تمثل 3 % من الطاقة

الكهربائية الكلية المجهزة و 2 % للضاغط الترددي. ووجد ان التحليل الحراري المتقدم كان نافعا ومرنا في التعامل مع تغيير درجة حرارة الجو ومع مكونات وحدة التكييف وحجمها.

Nomenclatures

Symbol	Definition	Unit
A	Area	m^2
A_r	Heat transfer area on the refrigerant side	m^2
A_s	Surface area	m^2
D	Diameter	m
D_i	Inside diameter of pipe	m
D_o	Outside tube diameter	m
ED	Exergy destruction	W
Ex	Exergy	W
Gr	Grashof Number ($Gr=g\ \beta\ \Delta T\ L_c^3/v^2$)	---
h_n	Enthalpy at state n (n=1,2,3 …)	kJ/kg
H	heat transfer coefficient	$W/m^2.°C$
M	Mass flow rate	kg/s
N	Number	---
Nu	Nusselt number ($Nu= h\ L_c\ /\ k_a$)	---
P	Pressure	N/m^2
P	Power	W
P_r	Prandtl number ($Pr=\mu\ c_p/k_a$)	---
Q	Heat transfer rate	W
Ra	Rayleigh number (Ra=Gr Pr)	---
Re	Reynolds number($Re= u\ D_i\ /v$)	---
S	Entropy	kJ/kg.K
T	Temperature	°C/K
W	Work	W

Greek characters

β	extend coefficient for air in natural convection	K^{-1}
ε_{scomp}	Compressor surface emissivity	---
η	Efficiency	---
ζ	exergy dissipate	---

Subscripts

symbol	**Definition**
a	Air
act	Actual
amb	ambient
comp	Compressor
dis	Discharge
e	Exit
Ex	Exergy
F	Fluid
g	Gas
i	In
l	Liquid state
o	Dead state temperature 35°C
r	Refrigerant
S	Surface
ST	Short tube
suc	Suction
shellr	Shell reciprocating
shells	Shell scroll
TEV	Thermal expansion valve
II	Second
rev	Reversible

Introduction:

Recently the exergy approach has been used to improve the performance of small systems such as refrigerators ,window type air conditioners and domestic deep freezers [1,2] . However the packaged air conditioning units , air handling units and central air conditioning systems which represent a medium to large sizes air conditioning systems required performance optimization too to reduce their energy bill . Hence and based on years of experience in the design and manufacture of the reciprocating and scroll compressors for such systems , Bristol compressors engineering recommended that the scroll compressor is a better choice for air conditioning application at or above 3.5 TR , while reciprocating compressor is preferred for 1.5 up to 3 TR in meeting the new seasonal electric efficiency ratio(SEER 13) requirement .Therefore, the two types of compressors have been subject to several researching studies in the recent years for performance optimization :

Kim and Bullard (2002) [1] developed a simple physical model for small hermetic reciprocating, rotary and scroll compressors based on thermodynamic principles and large data sets from the compressor calorimeter and experimental tests. Pressure losses along the refrigerant path were neglected and the compression process assumed isentropic. A linear relationship between the discharge and shell temperature extracted from the large data sets and applied to the model for calculating the discharge temperature. The accuracy of the model for calculating the mass flow rate and power consumption used are within ±3.0%.

Chen et al. (2002) [3] investigated a compressor's performance under different operating conditions specially the compression process of a scroll compressor. They were combined the conservation equations with models for the refrigerant flow in the suction and discharge processes, radial and flank leakage, and heat transfer between the gas and scroll wraps and solved simultaneously using the a nonlinear equations solver. The lumped capacitance method was used to study the energy balance equations. The results indicated that the comprehensive scroll compressor model was capable of predicting real compressor behavior and useful to the design and optimizing scroll compressors.

Ooi (2003) [4] presented an analytical study on heat transfer and temperature distribution for a hermetic reciprocating refrigeration compressor using the lumped thermal conductance approach. The lumped thermal conductance method was applied to all components of the compressor to form simultaneous equations, the convection heat transfer effects of the fluid and solid surface boundaries, and the simplification made in distributing the various components of the compressor into discrete parts. The results obtained had good agreement with test measurement.

Perez-Searra et al. (2005) [5] analyzed different thermodynamic efficiencies usually used to characterize hermetic compressors. Attention was focused on the volumetric efficiency, the isentropic efficiency, and the combined mechanical–electrical efficiency. The volumetric efficiency split into partial efficiencies related to pressure drop and heat transfer effects, supercharging effects, super discharging effects, leakages, etc. The isentropic efficiency was detached using two different points of view: The work associated to the individual sub-processes (compression, discharge, expansion, suction), and the work associated to the under pressures, overpressures, and between the inlet and outlet mean compressor pressures. Finally, the combined mechanical–electrical efficiency related to the heat transfer losses gains, and to the exergy transfers and exergy destroyed. They argued that the criteria developed was useful tools for comparison purposes, to characterize compressors, and to assist designers during the optimization process.

Rovarisand Deschamps (2006) [6] used the large eddy simulation(LES)to predict the performance of the hermetic reciprocating compressor utilized in vapor compression refrigeration system combine with grid model . The mathematical model depends on the simulation methodology which combines differential and integral formulations for the governing equation and using k-ε model with (LES) which implies a transient three-dimensional simulation. The methodology still requires validation with reference to experimental data.

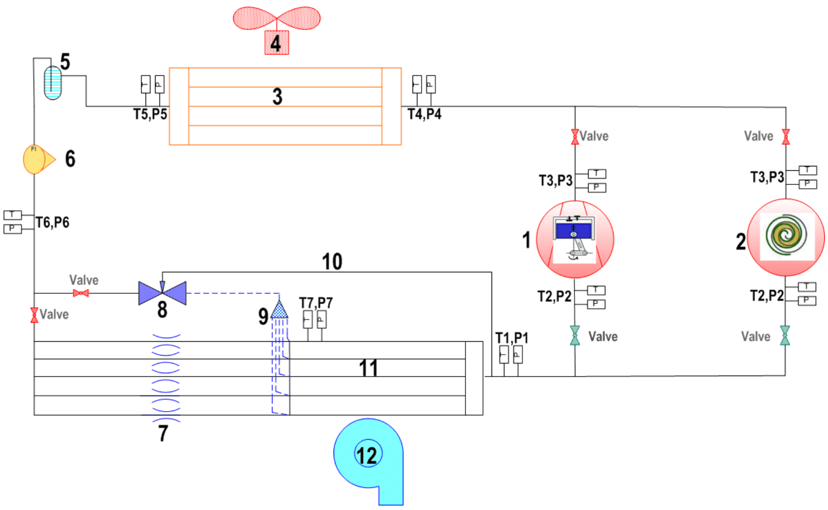
Duprez et al. (2007) [7] presented a thermo dynamical realistic models of two types of compressors (reciprocating and scroll). These models calculated the mass flow rate of refrigerant and the power consumption from the knowledge of operating conditions and parameters. These parameters were found in the technical datasheets of compressors. This study was limited to compressors with a maximum electrical power of 10 kW and for the following special operating conditions: i.e. [Evaporating temperatures ranging from -20 to 15 ℃ and condensing temperatures ranging from 15 to 60 ℃]. The average discrepancies on mass flow rate and power for reciprocating compressors were found to be 1.10 and 1.69% and for scroll compressors, were 2.42 and 1.04%respectively.

Navarroa et al. (2007) [8] presented a model for hermetic reciprocating compressors. The model was able to predict compressor and volumetric efficiency in terms of a certain number of parameters representing the main sources of losses inside the compressor. The model provided users with helpful information about the way in which the compressor was designed and working. The model can predict compressor performance at most points with a maximum deviation of 3%.

The above review shows that there are few articles in the open literature that use the exergy approach to model the thermodynamic behavior of air conditioning systems. The object of the current research is to carry out a theoretical analyses using the exergy approach in order to evaluate the thermodynamic performance of compressors (i.e. reciprocating and scroll compressors) in a packaged air conditioning unit. An experimental work is also carried out to analysis the performance of the compressors via the replacement of the reciprocating compressor in the packaged air conditioning unit by a scroll type. The experimental tests covered the changes in the ambient air temperature and its effect on the performance of the unit.

Experimental apparatus and Measuring devices:

The apparatus is a(3TR) packaged air conditioning unit manufactured by Carrier company equipped with a reciprocating compressor [model H23A423DBEA] manufactured by Bristol company. A Scroll compressor [model HRM045U4LP6]manufactured by Danfoss company was added to the unit and installed beside the reciprocating compressor as shown in **Figures (1&2)**.**Table (1)**includes the measuring devices that were used in the experimental tests.



1	Reciprocating compressor	7	Short tube restrictor
2	Scroll compressor	8	Thermal Expansion Valve
3	Air cooled condenser	9	Distributor
4	Condenser fan	10	External Equalizer
5	Receiver	11	Evaporator
6	Rota meter	12	Evaporator fan

Fig .(1) Vapor compression refrigeration system with the locations of temperature and pressure measurement .

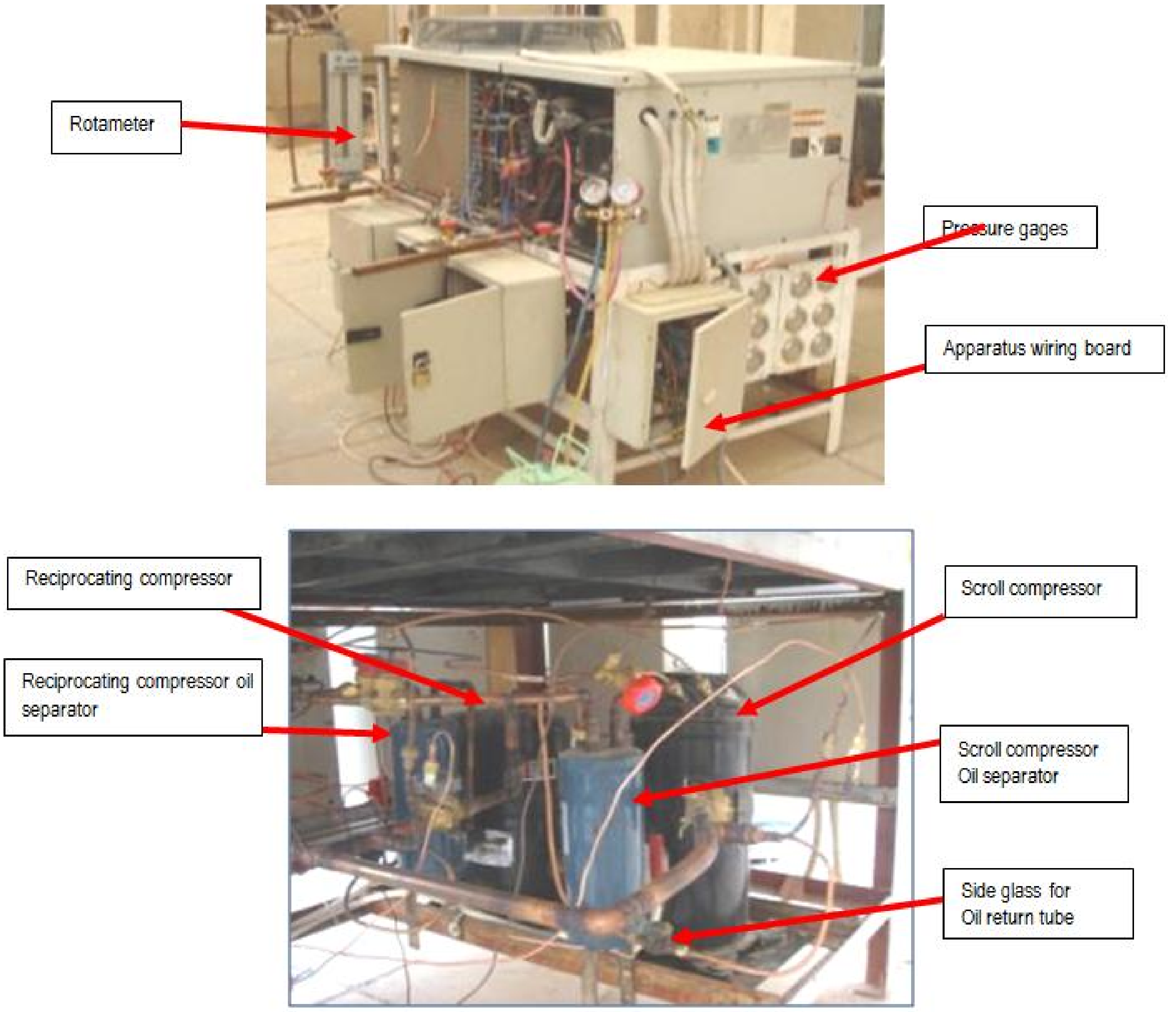


Fig .(2) Packaged unit, compressors and oil separators installation.

Table .(1) Measuring devices used in the VCRS and its modifications

	Device type	Manufacturing Nationality	Range	unit	Error
1	Pressure transducer is KELLER AG fürDruckmesstechnik, SERIES 21 R / 21 SR / 21 MR with plug	Switzerland	0-10 0-40	bar	± 1.0 -2% max.
	UDL100-4 interface				
	USB line				
2	Pressure gages AIRMENDER,	USA	-30 – 250 0 - 500	psi	± 1.5 psi

Compressor modeling and Exergy analysis:

There are three methods for modeling the thermal performance of the compressors; these methods are related to the appearing time :(1) Loss-efficiency based model [2,4,7], (2) Manufacturer's data model [1,7] and (3) Computational fluid dynamic (CFD) model [3,6].

The first method is simple but it is an old method while the CFD method is a recent method but is complicated and required high experience in mashing the geometry of the compressor[6]. The second method which called manufacturer's data depends on the manufacturing data that measure according to Standard Performance Rating of Positive Displacement Refrigerant Compressors and Compressor Units ANSI/AHRI Standard 540-2004 and Rating test point A [9]. The using of the standards provides a valuable basis for device evaluation. It appeared at the last 20 years.

The data depends on the condenser and evaporator saturated temperature. Capacity, power, and refrigerant mass flow rate, which can be considered as three dimension map representing the behavior of the compressor .Each of the parameters, capacity, power, and refrigerant mass flow rate has a correlation with ten constant and third order equation for the condenser and evaporator temperatures . The first step in this method is to find the total power input, and mass flow rate .Then the second step is to find the approximate properties of the refrigerant outlet, which lead to the calculation of the power required to compress the refrigerant only, and by omitting it from the total power input to find the total losses (friction and heat). This method is reliable and accurate, and gives the details of the total losses [9] .

The second method will be used to analyze the compressors due to the availability for machining the types of compressors manufacturing data. The question is whether one can find a compressor formula that covers several types of compressor working at the same conditions of ANSI/AHRI Standard 540-2004, rating test point A [9] ?

The reciprocating and scroll compressors have a capacity around 10.551 kW (3 TR) and working at same the conditions of ANSI/AHRI Standard 540-2004, rating test point A [9].

The reciprocating compressors that were used to find the mathematical formula are shown in **Table(2).**

Table .(2) Reciprocating compressors from several companies

Company	**BRISTOL**	**COPELAND**	**TECUMSEH**	**MANEUROP**
model	H23A423DBE	CRKQ-0325-TDF	TFH5542E	MT40JH4
Capacity kW	10.300	9.62	9.951	10.476
Displacement	13.1	12.8	12.6	11.8

Using Matlab software R2010a, and surface fitting method, polynomials were found for the cooling capacity (Q_e), power (P), and refrigerant mass flow rate (m_r).These polynomials are presented in **Table (3)** based on the data obtained from the fourth compressor. The maximum deviation for capacity was found to be (-9.7 to10.6) %, for power (-13to 20) %, and for mass flow rate (-20 to 20) %.

Table .(3) Reciprocating compressor polynomials coefficients

Reciprocating compressors			
Parameter	Capacity Qe	Power P	Refrigerant Mass flow rate mr
Unit	kW	kW	Kg/s
C0	11.86	1.997	0.1615
C1	0.5171	-0.06204	0.002256
C2	0.05544	-0.006679	-0.005822
C3	0.007499	-0.002776	-3.848e-005
C4	-0.00198	0.003505	2.01e-006
C5	-0.004317	0.0008199	0.0001031
C6	3.605e-005	-2.476e-005	-5.693e-007
C7	-4.492e-005	5.119e-005	1.348e-006
C8	-2.223e-005	-2.377e-005	-4.38e-008
C9	3.25e-005	-5.862e-006	-6.421e-007
SSE	0.06035	0.05268	1.006e-005
R-square	0.9999	0.9977	0.9994
Max. Deviation from original data	-9.7%and +10.6%	-13% and +20%	±20%

Table (4) shows the deviations for each compressor type. The Bristol compressor which was used in the packaged unit has deviations within the acceptable range.

Table (4) The deviations of reciprocating compressor

	Bristol		**Copeland**		**Tecumseh**		**MANEURO P**
	H23A423DBE		CRKQ-0325-TDF		TFH5542E		MT40JH4
Capacity%	-9.7	+5.4	-7.6	+2.2	-6	+6.8	-7.6
Power%	-13.5	+8.46	-18.66	------	-----	+20.7	--------
mr%	-19.8	+20	-14.46	------	-9.38	+14	-13.8

The scroll compressors types that used to find the mathematical formula are shown in **Table(5).**

Table .(5) Scroll compressors data taken from several companies

Company	**Bristol**	**Copeland**	**Danfoss**	**Carlyle**
Model	H21R453DBE	ZR45K3-TFD	HRM045U4	XCH542H
Capacity kW	10.800	10.551	10.939	11.049
Displacement	10.7	10.62	10.7	10.5

The polynomials representing the cooling capacity (Q_e), power (P), and refrigerant mass flow rate (m_r) are presented **Table (6)** . The maximum deviation for power was (±6) % and for mass flow rate (±5) %.

Table .(6) Scroll compressor polynomials coefficients

Coefficients	Capacity Qe	Power P	Refrigerant Mass flow rate
	kW	kW	kg/s
C0	12.81	0.7775	0.06394
C1	0.4927	0.006223	0.002096
C2	-0.06378	0.03015	-0.0004054
C3	0.007552	-8.542e-005	3.321e-005
C4	-0.003655	-0.0007456	-6.281e-006
C5	-0.0001584	0.0001197	8.738e-006
C6	-9.137e-006	8.101e-007	-3.501e-008
C7	-7.677e-005	-2.57e-006	-3.175e-007
C8	1.208e-005	1.004e-005	7.499e-008
C9	-3.57e-006	4.414e-006	-8.948e-008
SSE	0.07105	0.02097	3.179e-006
R-square	0.9998	0.9992	0.9997
Max. Deviation	∓6%	∓6%	∓5%

Table (7) shows the deviation for each compressor which is in an acceptable range.

Table .(7) The deviations of scroll compressor polynomials

Deviation from original data				
	Bristol	**Copeland**	**Danfoss**	**Carlyle**
	H21R453DBE	ZR45K3-TFD	HRM045U4	XCH542HA
Capacity%	-0.67 +1.48	-1.26 +7.97	-3 +0.84	-3.19 +0.7
Power%	-5.1 +3.18	-6.7 +2.92	-3.28 +7.52	-3.44 +7.85
mr%	--------	-1.27 +3.37	-2.63 +0.7	-3 +0.55

The Danfoss compressor used in the second modification has also acceptable deviation. The polynomial equation is given by:

f(x,y) =c1*x+c2*y+c3*x^2+c4*x*y+c5*y^2+c6*x^3+c7*y*x^2+c8*x*y^2+c9*y^3

Where: f(x,y) represent Qe (kW) ,Power (kW) or refrigerant mass flow rate m_r (kg/s) ,x represent evaporator temperature (°C) and , y represent condenser temperature (°C).

Basic Equations : The power input to the compressor is the summation of the work and the power required to overcome the friction and the heat losses as shown in Figure (3,4), which can be formulated in the follows equations:

$$P = W + Q_{lossestott} \qquad \text{.................(1)}$$

$$W = \dot{m}_r * (h_3 - h_2) \qquad \text{.................(2)}$$

$$Q_{lossestot} = Q_{lossesconv\&ad} + Q_{lossesfrication} \qquad \text{.................(3)}$$

$$Q_{lossesconv} = h_{acomp} * A_{compsuface} * (T_{shellcomp} - T_{amb}) \qquad \text{.................(4)}$$

$$T_{shellcomp} = f(T_{dis}, T_{suc}, T_{amb}) \qquad \text{.................(5)}$$

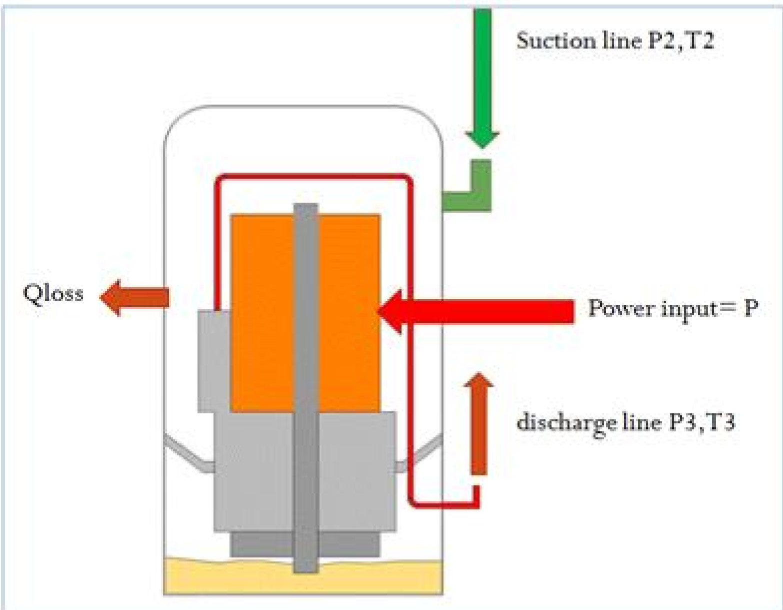


Fig .(3) Scheme of the thermodynamic parameters for reciprocating compressor.

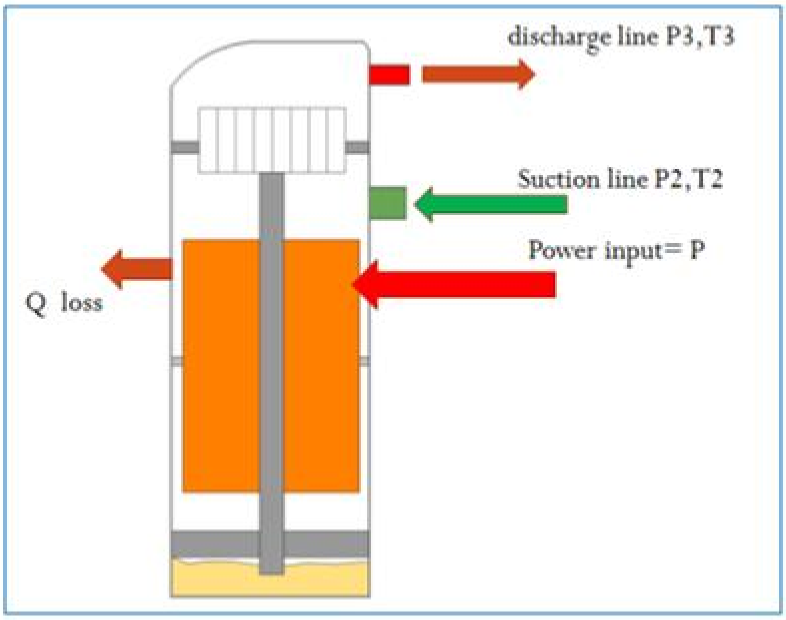


Fig .(4) Scheme of the thermodynamic parameters for Scroll compressor.

$T_{\mathbf{shellcomp}}$ in equation (4) can be found from the test data of the average surface temperature of the compressor (at the upper, middle, and lower locations), discharge temperature, suction temperature, and ambient temperature. The discharge temperature alone may be used in some application where the compressor placed at stationary place(no moving air around the compressor). In this case the natural convection and radiation being the most effected thermodynamic forces on heat transfer losses with the environment.

For **reciprocating** compressor the correlation found from the test data [9] is:

$$T_{shellr} = 8.42667 - 0.9333 * T_{dis} + 031111 * T_{suc} + 1.51778 * T_{amb} \qquad \ldots\ldots\ldots\ldots\ldots\ldots (6)$$

For the ***scroll*** compressor the correlation is:

$$T_{shells} = 5.37765 - 0.20743 * T_{dis} + 1.11097 * T_{suc} + 0.77631 * T_{amb} \qquad \ldots\ldots\ldots\ldots\ldots\ldots (7)$$

To find the heat transfer coefficient, the relations of vertical cylinder, heated surface facing up, heated surface facing down will be used as in **Table (8) :**

$$Nu = c * Ra^{m} \qquad \ldots\ldots\ldots\ldots\ldots\ldots (8)$$

$$Nu = \frac{h_{acomp} * L_c}{k_a} \qquad \ldots\ldots\ldots\ldots\ldots\ldots (9)$$

$$Ra = \frac{g * b * (T_{ashellcomp} - T_{amb}) * r^2 * Lc^3}{m^2} * Pr \qquad \text{.................. (10)}$$

Where $Lc = \frac{A}{Permiter}$, $b = \frac{1}{T_{film}}$, and $T_{film} = \frac{T_{shellcomp} + T_{amb}}{2}$

Air properties recalculated at T_{film} , the values of the constants c and m in equation (8) are given in **Table (8)** .

Table (8) The correlation used for natural convection [10 , 11 , 12]

Geometry	Correlation	Limitations
Vertical cylinders and planes	$Nu = 0.59 * Ra^{1/4}$	$10^4 < Ra < 10^9$
	$Nu = 0.1 * Ra^{1/3}$	$10^4 < Ra < 10^9$
Heated surface facing upward	$Nu = 0.54 * Ra^{1/4}$	$10^4 < Ra < 10^9$
	$Nu = 0.15 * Ra^{1/3}$	$10^9 < Ra < 10^{11}$
Heated surface facing	$Nu = 0.58 * Ra^{1/5}$	$10^5 < Ra < 10^{11}$

To calculate the radiation heat transfer from the compressor shell which is very important with natural convection due to the large effect of the total heat transfer losses:

$$Q_{lossesrad} = F_{1-2} * s_B * A_{scomp} * e_{scomp} * (T^4_{shellcomp} - T^4_{amb}) \qquad \text{.................. (11)}$$

Where: F_{1-2}: Shape factor, equal to 1
σ_B: Stefan-Boltzmann constant, $5.669*10^{-8}$(W/m^2.K)
A_{scomp}: Compressor surface area (m^2)
ε_{scomp}: Compressor surface emissivity=0.96(the compressor black paint is treated as a black body surface)

T: Shell and ambient temperature (K)

Energy balance:

$$\dot{m}_r * \sum_{in} h_2 - Power = \dot{m}_r * \sum_{out} h_3 + Q_{lossestot}$$

Rearranging gives :

$$Q_{lossestot} = \dot{m}_r * (h_2 - h_3) - Power \text{; Where } W = \dot{m}_r * (h_3 - h_2) \qquad \text{.................. (12)}$$

The kinetic and potential energies are neglected due to their small magnitude compared with the other terms.

Exergy balance:

Exergy destruction can be given by :

$$ED_{comp} = (1 - \frac{T_{amb}}{T_{shellcomp}})Q_{lossestot} - Power + \sum_{in} Ex - \sum_{out} Ex \qquad \text{..................} (13)$$

Where :

$$\sum_{in} Ex = \dot{m}_r * ex_2 = \dot{m}_r * \{(h_2 - h_o) - T_o * (s_2 - s_o)\}$$

$$\sum_{out} Ex = \dot{m}_r * ex_3 = \dot{m}_r * \{(h_3 - h_o) - T_o * (s_3 - s_o)\}$$

Compressor dissipative:

$$z_{comp} = \frac{ED_{comp}}{Power} \qquad \text{..................} (14)$$

Exergy efficiency:

$$h_P = h_{ex.comp} = 1 - z_{comp} \qquad \text{..................} (15)$$

Results and discussion:

Figure (5) represents the variation of compressors exergy efficiencies for the reciprocating and scroll types via the ambient temperature. The exergy efficiency decreases with the increase in the ambient temperature due to the decrease in the refrigerant mass flow rate which decreased the volumetric efficiency. The exergy efficiency for the scroll compressor is higher than the reciprocating compressor. One of the reasons is the refrigerant mass flow behavior, for the reciprocating, the refrigerant mass flow is a pulsing flow, while the scroll compressor has a continues flow. This fact is confirmed in **Figure (6)** where the scroll compressor exergy efficiency is better than that of the reciprocating compressor.

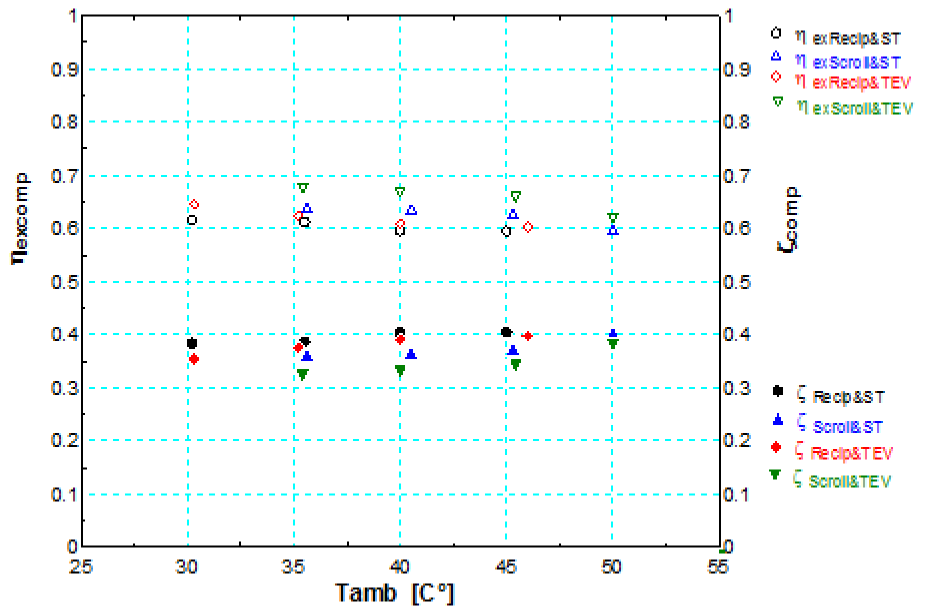


Fig .(5) Variation of exergy efficiencies of reciprocating and scroll compressor with ambient temperature.

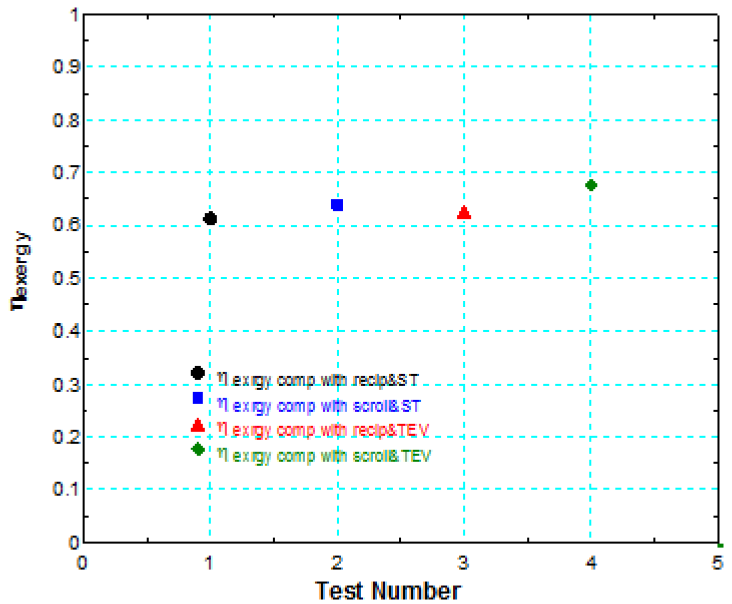


Fig .(6) Exergy efficiencies of reciprocating and scroll compressor with 35 °Cambient temperature.

Figures (7 and **8)** represent the actual power consumption, the power increases as the ambient temperature increases due to the rising in the condenser pressure. These figures also contain the actual work, total losses, friction losses, and heat losses for the reciprocating and the scroll compressors. The actual work took 72% of the total power input and the total losses are 28% for the reciprocating compressor. The actual work took 82% of the total power input and the total losses18% for the scroll compressor. The friction losses took 26% of the total power input and the heat losses covered 2% for the reciprocating while the friction losses took 15% of the total power input and 3% heat losses for the scroll compressor. The main reason for the higher percentage of the friction losses in the reciprocating compressor is the number of the moving parts comparing with scroll compressor; also these losses include losses dissipated due to the power transformation.

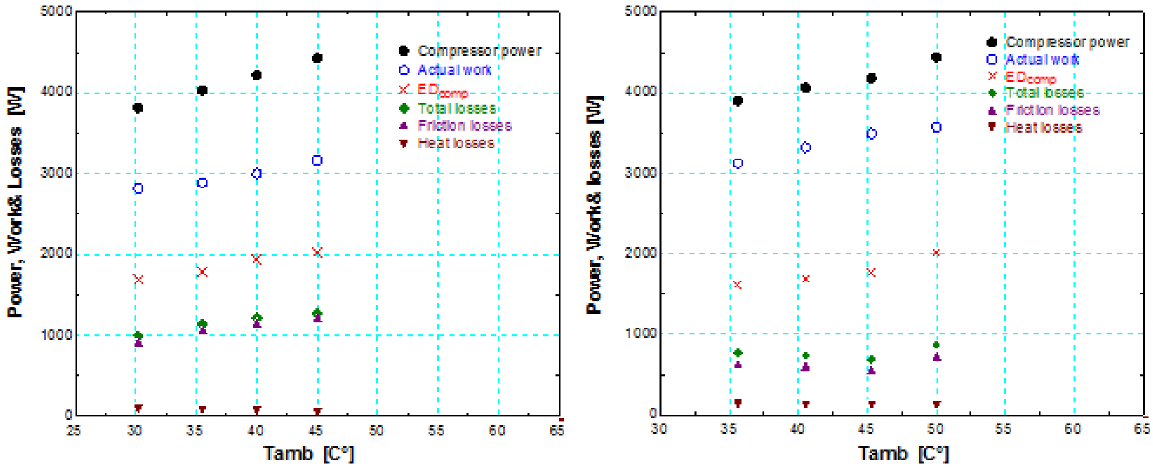


Fig .(7) Power, Work, and losses via ambient temperature for a) reciprocating and b) scroll compressor.

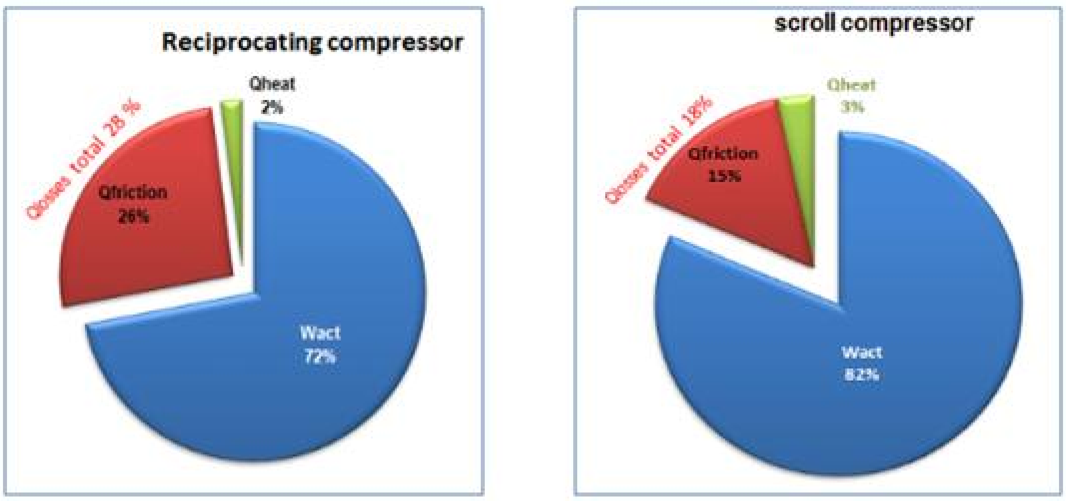


Fig .(8) The percentage of the actual work and total losses for compressors.

The heat losses for the scroll compressor is higher than that of the reciprocating, this because the discharge of the hot gas in the scroll type is part of the compressor shell and also the stator winding of the motor is in direct contact with the inside wall of the shell, while the reciprocating type has no touch between stator winding and the inside wall and the hot gas line is separated from the shell. This is shown in **Figure (9)** where the shell temperature of the scroll compressor is higher than that of the reciprocating compressor.

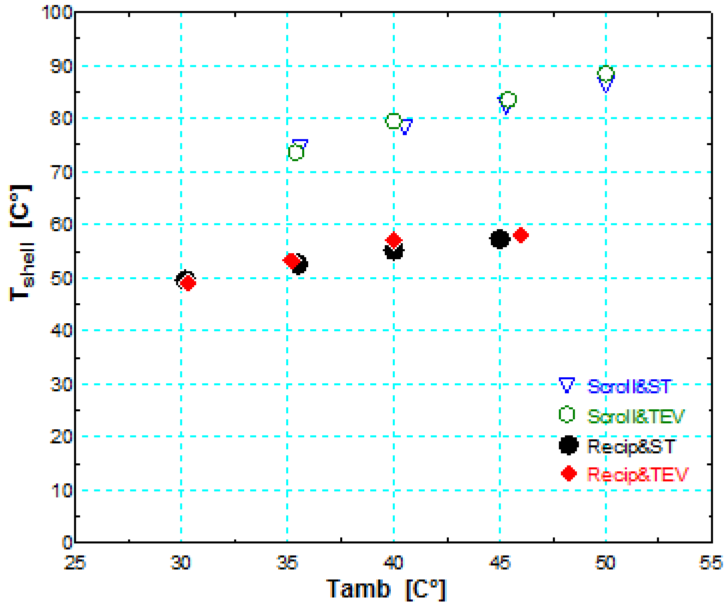


Fig .(9) The variation of shell temperature of the scroll and reciprocating compressor with the ambient temperature.

The operation points of the system deviated from the set point of the ARI 540 standard (T_e=7.2°C, T_c=54.4°C) as shown in **Figure (10).**

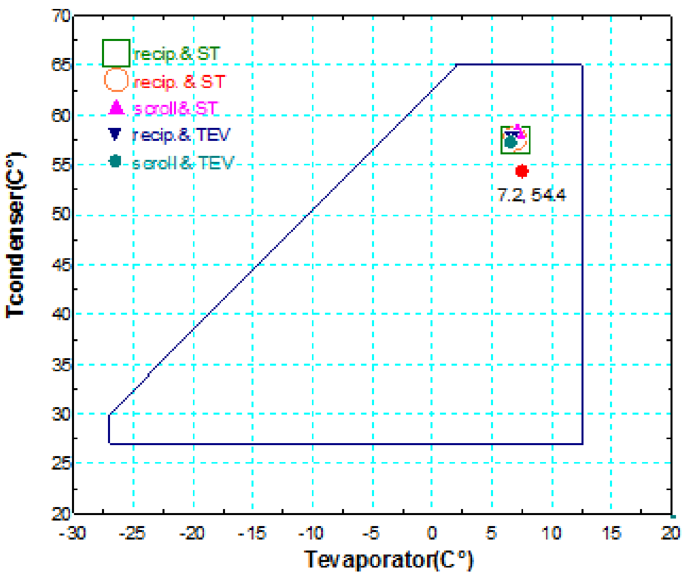
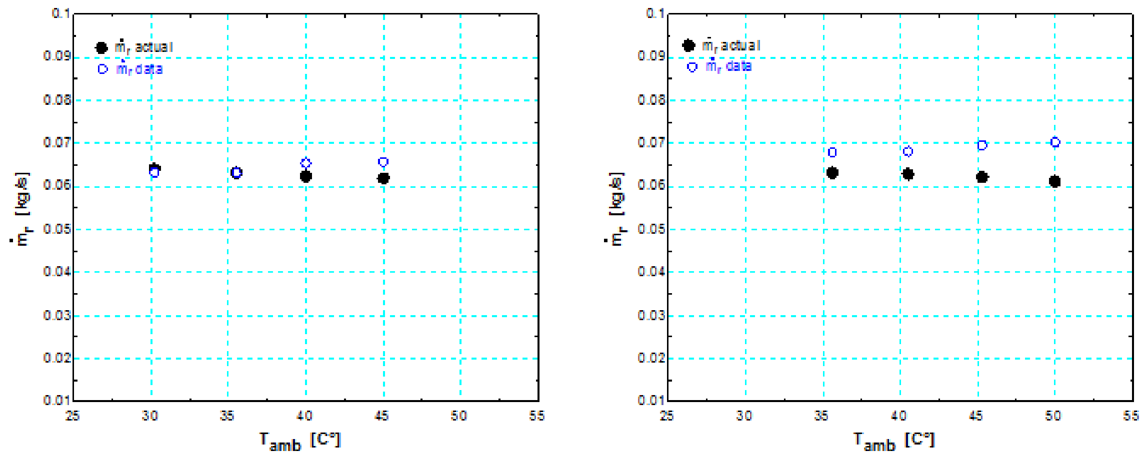


Fig .(10) The Envelop of the operation points for reciprocating and scroll compressor at 35°C ambient temperature.

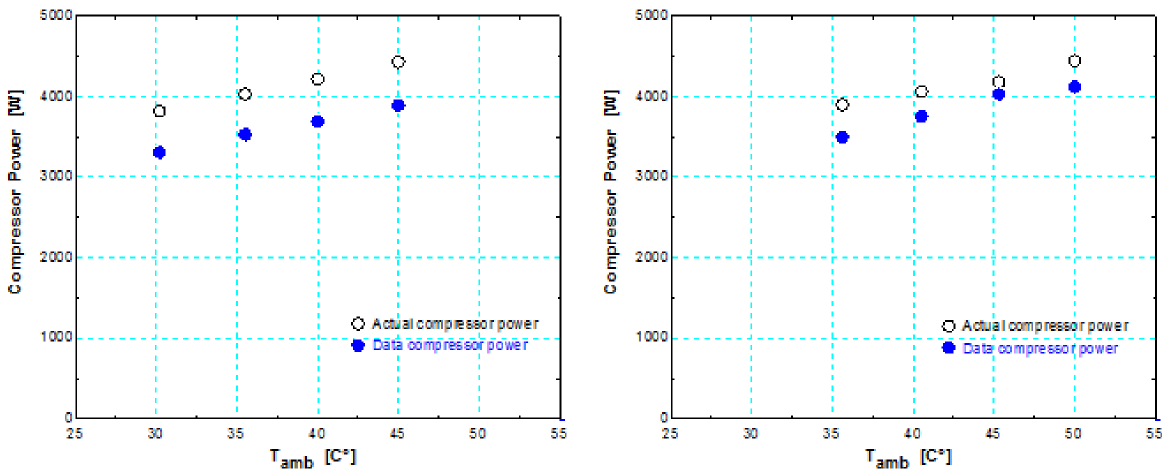
This deviation is appearing to be normal due to the difficulty in keeping the operating conditions at the set point. This led to a decrease in the cooling capacity, an increase in the power consumption, and the refrigerant mass flow rate is affected as shown in **Figure (11)** for reciprocating and scroll compressor.



a) b)

Fig .(11) Variation of refrigerant mass flow rates with ambient temperature for a) reciprocating compressors and b) scroll compressors.

The effect of the power consumption is shown in **Figure (12)** for reciprocating and scroll compressor. The refrigerant mass flow calculated from the data equation is higher by 8% than that of the experimental for the reciprocating type and 10% for scroll type, and the power consumption is lower by 12% than that of the experimental for reciprocating and 7% for the scroll. These results high light the areas of large losses and this will help other researches to modify their design in order to improve future systems.



a) b)

Figure (12) Compressor power along with ambient temperature for a) reciprocating compressors b) scroll compressors.

Conclusions:

A thermodynamic model for steady state vapor compression refrigeration compressors working with R-22 as a working fluid has been developed using the exergy approach with EES software supported by Mat lab program. The following conclusions have been drawn from the application of the model:

1- The compressor (scroll and reciprocating) has a large exergy dissipative (18%-28%) of the total input power. However the scroll type is better than the reciprocating according to the total losses
2- The friction losses are always higher than the heat losses (26% for friction losses and 2% for the heat losses)
3- The heat losses for the scroll compressor are higher than that for the reciprocating compressor.

References:

1. **Kim, M.H., Bullard, C.W. (2002)[Thermal performance analysis of small hermetic refrigeration and air-conditioning compressors. JASE International journal,vol.45, No.4,pp857-864]**
2. **Winandy, E., O, C.S., Lebrun, L. (2002)[Experimental analysis and simplified modeling of a hermetic scroll refrigeration compressor. Applied Thermal Engineering 22, pp.:107-120]**
3. **Chen, Y., Halm, N.P., Groll, E.A., Braun, J.E. (2002)[Mathematical modeling of scroll compressors-part 1: compression process ,International journal of refrigeration 25, 2002, pp 731-750. Mathematical modeling of scroll compressors-part 2: overall scroll compressor, International journal of refrigeration 25, pp751-764.]**
4. **Ooi, K.T. (2003). [Heat transfer study of a hermetic refrigeration compressor. Applied Thermal Engineering 23, pp.: 1931-1945.]**
5. **Perez-Searra, C.D., Rigola, J., Soria, M., Oliva, A. (2005) [Detailed thermodynamic characterization of hermetic reciprocating compressors. International journal of refrigeration 28, pp:579-593]**
6. **Rovaris, J.B., Deschamps, C.J. (2006)[Large eddy simulation applied to reciprocating compressors , Journal of the Brazilian soc. Of mech. Sci&Eng, April-June, vol xxviii, No.2,pp;208-215]**
7. **Duprez, M.E., Dumont, E., Frere, M. (2007) [Modeling of reciprocating and scroll compressors ,International journal of refrigeration 30, pp.; 873-886]**

8. **Navarroa, E., Granryd, E., Urchueguia, J.F., Corberan, J.M. (2007) [A phenomenological model for analyzing reciprocating compressors ,International journal of refrigeration 30, pp:1254-1265]**
9. **ANSI/AHRI Standard 540-2004.**
10. **Holman J. P. (2011) [Heat transfer , Tenth Edition , Mac Raw Hill]**
11. **Bejan, A.(1993) ,[Heat transfer] By John Wiley & Sons, Inc.**
12. **Bejan, A.(1996) ,[Entropy Generation Minimization] By CRC's Mechanical Engineering Series**