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Orkan Kurtulus
orkan@purdue.edu

Burak Olgun

Emre Oguz

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Orkan KURTULUS*, Burak OLGUN, Emre OGUZ

[1]Purdue University, School of Mechanical Engineering,
West Lafayette, IN, USA
orkan@purdue.edu

[2]Zeta Information Technologies,
Istanbul, TURKEY
burak@zetabt.com

[3]Research and Technology Development Center, Arcelik A.S.,
Istanbul, TURKEY
emre.oguz@arcelik.com

* Corresponding Author

ABSTRACT

The vapor compression cycle using refrigerant CFC (chlorofluorocarbon) and HCFC (hydrochlorofluorocarbon) is widely used in refrigeration, air-conditioning and water heating industries. Utilization of these gases are prohibited by the Montreal Protocol. The critical temperature of carbon dioxide (CO_2) is much lower than that of the other refrigerants, namely 31.06°C. The gas cooling process occurs at a constant pressure but at variable temperatures takes the place of the condensation process that occurs at both constant pressure and temperature. Domestic type refrigerators use smaller compressor capacities between 100 W to 300 W. Data on CO_2 (R744) refrigeration cooling cycle applications and calorimetric measurements are not available in the market and literature. Utilizing the experimental setup, our research group has developed a model for mass flow rate and compressor power of commercially available R744 compressors at different pressures and temperatures. In this article, experimental results and mathematical model calculations will be presented and discussed.

Keywords: R744, Transcritical, Refrigeration,

1. INTRODUCTION

The vapor compression cycle which is widely used in refrigeration, air-conditioning and water heating industries requires a refrigerant that will absorb heat while evaporating at a low temperature and transfer this thermal energy to a different medium by condensing at a higher temperature. After it was established that chlorofluorocarbon (CFC) and HCFC gases that have been used for a long time in such systems, damaged the ozone layer, the utilization of these substances were prohibited by the Montreal Protocol. The hydrofluorocarbon (HFC) gases, which were initially developed as alternatives to CFC and HCFCs, have been found to have Global Warming Potential (GWP) and therefore the utilization of these gases will be banned in the near future (Riffat *et* al., 1997). Hence, research and development activities for refrigerants to reduce global warming potential (GWP) have increased considerably in the past years.

The studies on R744 which was used as a refrigerant at the beginning of 1900s especially on marine vehicles, has recovered and increased up to date because of the negative environmental effects of CFC, HCFC and HFCs, toxicity of ammonia and the flammability of hydrocarbons that are known to be the two other group of natural refrigerants.

Since the critical temperature of R744 is much lower than that of the other refrigerants, namely 31.06°C, the gas cooling process which occurs at a constant pressure but a variable temperature takes the place of the condensation process that occurs at both constant pressure and temperature. Therefore a transcritical refrigeration / heat pump system has to be constructed (Richter *et.* al. 2003). In addition to the transcritical properties, the high saturation pressures of R744 that could be as high as 10 to 20 times of the conventional refrigerants require challenging modifications on the system components such as the compressor, the heat exchangers and the expansion device(Yamaguchi *et*. al., 2011). These topics have been covered in literature (Cecchinato and Corradi, 2011) (Krinivasan *et*. al., 2010).

Some research on heat pumps, automotive air conditioning systems, industrial cooling with CO_2 as a refrigerant exists in the literature (Hongyun *et*. al. 2008) (Sung *et*. al. 2009). Most of the papers in the literature are related to industrial cooling systems, whose cooling capacities are over 1000 W (Yin *et.* al., 2001) (Tao *et*. al., 2009) (Sung et. al., 2009). Domestic type refrigerators use smaller compressor capacities between 100 W to 300 W. In order to simulate domestic type refrigerator R744 refrigeration cooling cycle electrical and calorimetric measurements are not available in the market and literature.

At the Yildiz Technical University Laboratory, a "calorimeter" test system has been constructed to test the performance of compressors. Such systems can be used to measure the refrigeration capacity and the coefficient of performance in an accurate manner by controlling the compressor inlet pressure, outlet pressure, ambient temperature, subcool and superheat temperatures.

2. EQUIPMENT

A calorimeter test system is basically a model of the vapor compression cycle for refrigerators. Listed equipment has been used to build up the calorimeter. Main parts of the vapor compression cycle listed below:

1. Compressor
2. Gas cooler
3. Evaporator
4. Expansion valve
5. Measurement equipment:
6. High pressure sensor
7. Low pressure sensor
8. Flow meter
9. Thermocouple

Experimental setup flow chart can be seen on Figure **1**. The experimental setup was built in Yildiz Technical University, at the Thermodynamics and Heat Transfer Division.

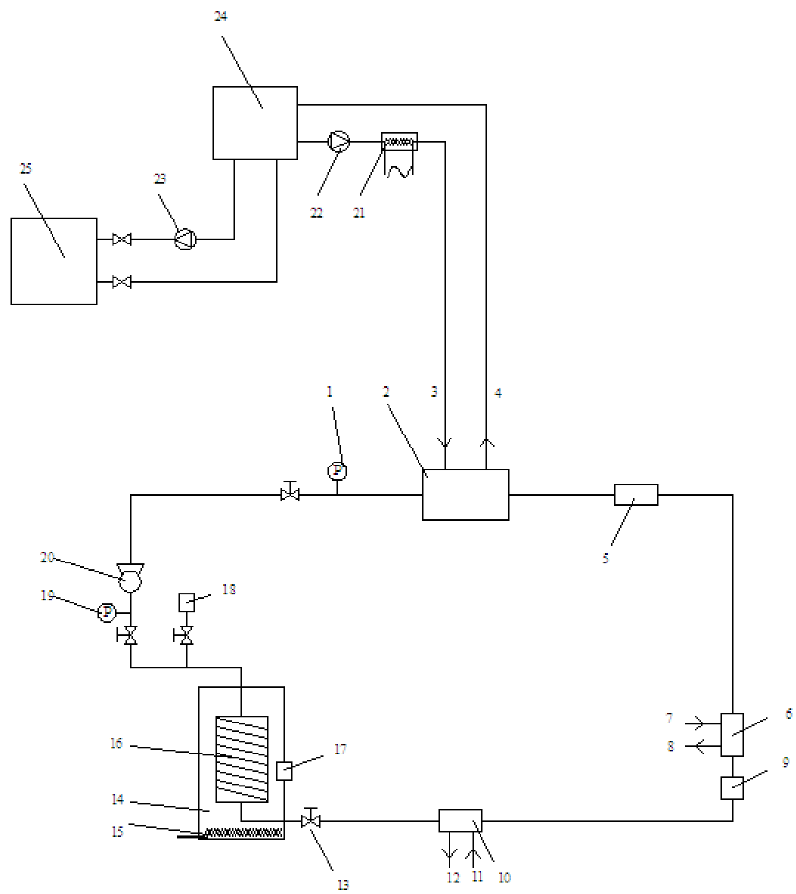

Figure 1: Experiment setup flow chart. (Kurtulus, 2011)

List of numbered items can be seen in Table **1**.

Table 1: List of equipment

1- High pressure sensor	2- Gas cooler	3- Gas cooler cooling water inlet
4- Gas cooler cooling water outlet	5- Accumulator	6- Subcooler1
7- Subcooler 1 cooling water inlet	8- Subcooler 1 cooling water outlet	9- Flow meter
10- Subcooler 2	11- Subcooler 2 cooling water inlet	12- Subcooler 2 cooling water outlet
13- Expansion valve	14- Evaporator	15- Electrical heater
16- Cooling coil	17- Sight Glass	18- Charge/Vacuum
19- Low pressure sensor	20- Compressor	21- Gas cooler hot water tank
22- Pump 1	23- Pump 2	24- Water tank
25- Chiller		

Set up has been built with two cabins. One of them contains the compressor and the pressure sensors; the other cabin contains the heat exchangers inside. Both cabin temperatures are controlled with a PLC (Programmable logic controller) device. While testing the compressor performance, the temperatures of the cabins were held fixed.

Main cooling is supplied for the setup by a chiller with a capacity of 10 kW. Water cooling is needed for the cabin temperature control, the sub coolers and the gas cooler temperature controls. Item number 24 is buffer tank. The reason for using a buffer water tank is to prevent the temperature fluctuations of the cooling water, as the chiller turns on and off during operation. A precise temperature control is essential. Item number 21 is a hot water tank. Hot water tank is used to set the water temperature needed by the gas cooler. This device is controlled by its own temperature control device. Temperature control sensitivity is ± 0.1 ^{o}C

The compressor cabin can be observed in Figure **2**. The heat exchanger cabin can be seen in Figure **3**.

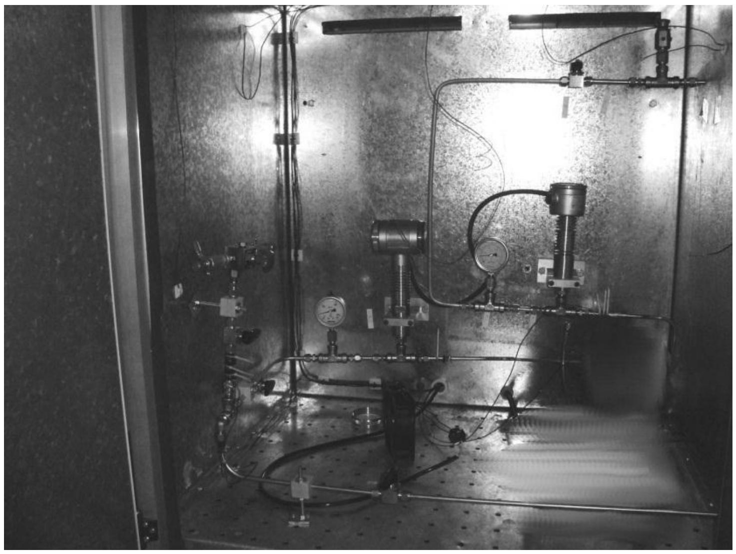

Figure 2: Compressor cabin

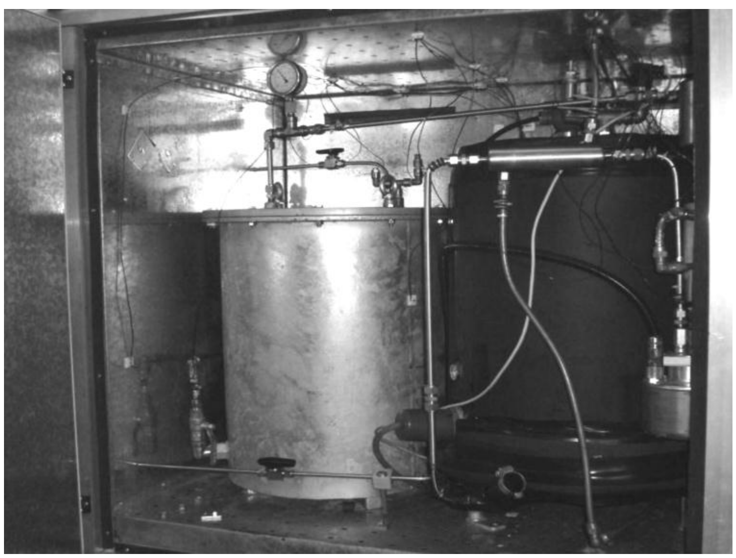

Figure 3: Heat exchangers cabin

3. MEASUREMENT EQUIPMENT and TEST CONDITIONS

Two different pressure sensor used on test system. Both sensors are suitable to use with R744. Sensors maximum allowed working pressure is 230 bar. High pressure sensor's measuring span range is 0 to 150 bar and have 2% accuracy. Low pressure sensor's measuring span range is 0 to 30 bar and have 1% accuracy. For temperature measurements T type thermocouple used. Thermocouples installed on every inlet and outlet of installed devices listed in Table 1. PLC is used to monitor the tests, temperature control and save measurement data. PLC programmed for PID (Proportional/Integral/Derivative) control mode for temperature which provided sensitive temperature control. Siemens flow meter used for flow rate measurements. Flow meter accuracy is 1%.

For different gas cooler pressure (between 70 / 110 bar) and evaporation temperatures (between -10 ^oC / -30 ^oC) the compressor cooling capacities are being measured. 25 tests run with the calorimeter. For every evaporation temperature, 5 different gas cooler pressure set and system performance measured. Evaporation pressure (temperature) is set by using manual metering expansion valve.

4. MATHEMATICAL MODEL

Mass flow rate m [kg/h] and compressor input power W_{ip} [Watt] mathematical model generated by using measured data. Used general equation can be seen in equation 1. Same kind of equation is presented in ANSI Standard ARI 540.

$$\mathbf{X = a_1P^2T^2 + a_2P^2T + a_3PT^2 + a_4PT + a_5P^2 + a_6T^2 + a_7P + a_8T + a_9} \quad 1$$

Where P is gas cooler working pressure [Bar] and T is evaporation temperature [C]. Mathematical model coefficients are listed in Table 2. Coefficients found by using least square method described in Chapra and Canale (2005). Same mathematical model used for both mass flow rate and compressor power input.

Table 2: Equations coefficients

Coefficients	X=m	$X=W_{ke}$
a_1	$-9{,}44\times10^{-8}$	$-3{,}62\times10^{-4}$
a_2	$5{,}66\times10^{-5}$	$6{,}50\times10^{-2}$
a_3	$-6{,}72\times10^{-5}$	$0{,}18\times10^{0}$
a_4	$3{,}15\times10^{-2}$	$-32{,}86\times10^{0}$
a_5	$-8{,}82\times10^{-3}$	$-2{,}83\times10^{0}$
a_6	$1{,}04\times10^{-2}$	$-23{,}24\times10^{0}$
a_7	$-3{,}59\times10^{0}$	$1430{,}26\times10^{0}$
a_8	$-4{,}99\times10^{0}$	$4157{,}40\times10^{0}$
a_9	$591{,}63\times10^{0}$	$-180768{,}72\times10^{0}$

5. RESULTS

When enough data collected from the calorimeter, a mathematical model developed to determine both flow rate and compressor input power as shown in Equation (1). For different gas cooler pressure, flow rate experimental (Exp.) and calculated (Calc.) results shown in Figure 4.

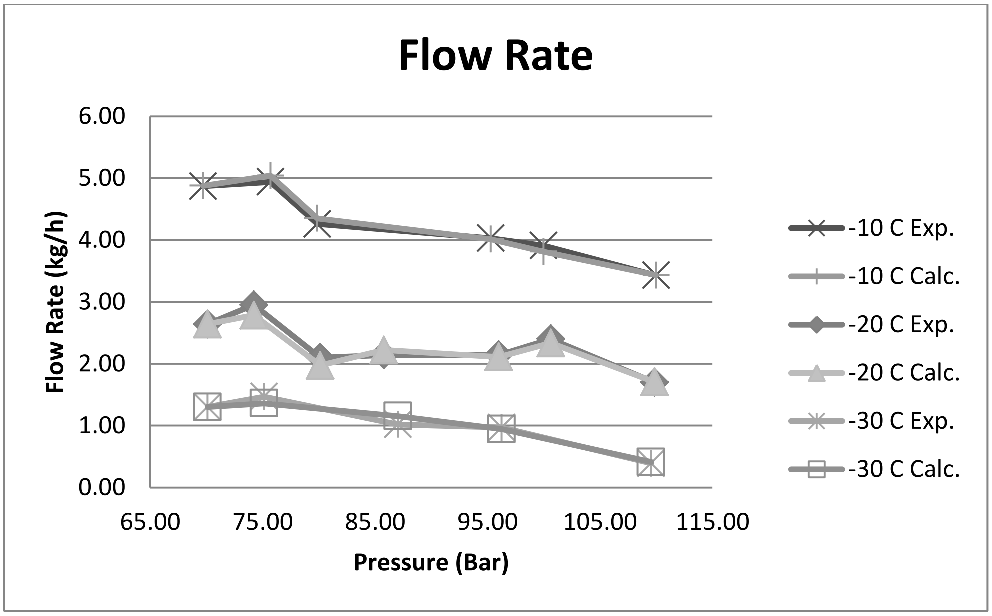

Figure 4: Flow rate results (Kurtulus, 2011)

Legend shows tested evaporation temperatures. -10 C, -20 C, -30 C are evaporation temperatures which can be using metering valve by controlling compressor inlet pressure. Over all mass flow rate at -10 C evaporation temperature is 3,98 kg/h, at -20 C evaporation temperature it is 1,90kg/h and at -30°C evaporation temperature it is 0,91 kg/h. As it seen in Figure 4, mass flow rate decreases by working evaporation temperature. Using developed mathematical formula, mass flow rate can be predicted within maximum 11.48% error (85 bar / -30 ^{o}C).

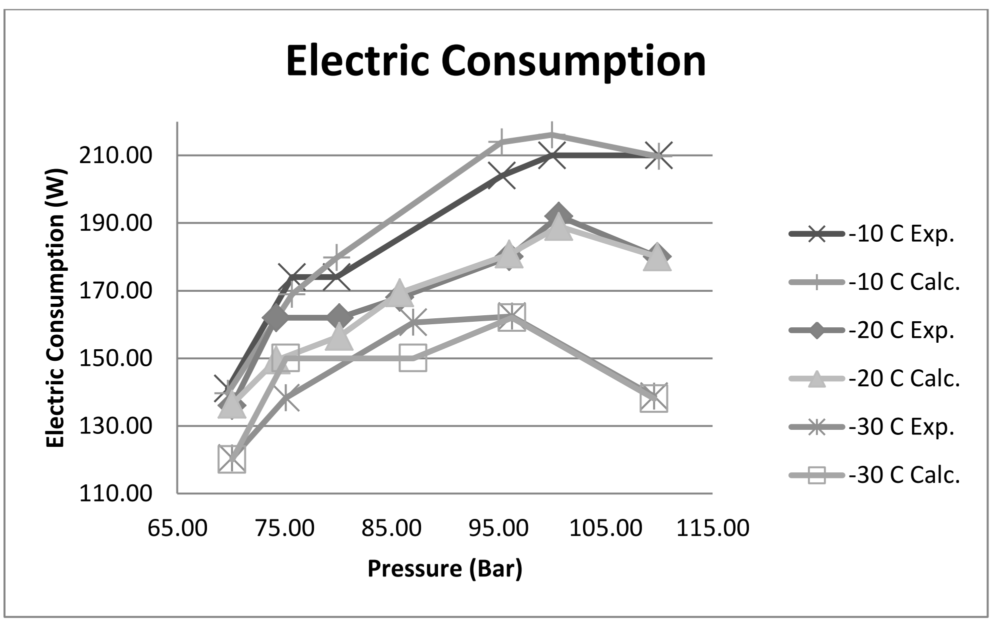

Figure 5: Compressor input power results (Kurtulus, 2011)

Figure 5 shows input power varies with gas cooler working pressure. Using developed mathematical formula for compressor input around 75 bar gas cooler working pressure calculated results gives maximum errors. We believe that working close to critical point experiment results might have been affected. We found our mathematical formula for can predict compressor input power within maximum 8.45% error (75 bar / -30 ^{o}C).

6. CONCLUSION

In this study, a commercially available R744 compressor was tested to get flow rate and compressor power input data. Using collected data a mathematical model developed to determine mass flow rate and compressor input data. Experimental results and calculations shows that in range of gas cooler working pressure from 70 bar to 110 bar and evaporation temperature from -10 ^{o}C to -30^{o}C mathematical model can be used to estimate R744 flow rate and input power for this specific compressor model. Calculated maximum error for mass flow rate is 11.48% and for compressor input is 8.45%. We believe that developed mathematical formulation can be used for simulation model on a household refrigerator R744 as refrigerant.

NOMENCLATURE

P	Gas cooler working pressure. [bar]
T	Evaporator temperature [C]
m	Mass flow rate [kg/h]
W	Compressor input power [W]

Subscripts

ip	Input power.

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