Isı Bilimi ve Tekniği Dergisi, 31, 1, 11-17, 2011
J. of Thermal Science and Technology

ISSN 1300-3615

AN EXPERIMENTAL INVESTIGATION OF THE EFFECT OF REFRIGERANT CHARGE LEVEL ON AN AUTOMOTIVE AIR CONDITIONING SYSTEM

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(Geliş Tarihi: 31. 08. 2009, Kabul Tarihi: 09. 03. 2010)

Abstract: During the last 25 years automotive air conditioning (AAC) systems have significant development introduced by the industry and research institutes in the world to minimize the global warming threat to the environment. This paper reports the results of a study on the performance of an AAC system with measuring the compressor driving speed and the refrigerant leakage. For this purpose an experimental set up is designed and constructed to investigate the system performance. Although, the manufacturer's recommended amount for the tests with R-134a as refrigerant was 750 g, the experiments were also carried out by selecting different amount of the same refrigerant charges to analyse the coefficient of performance (COP), the cooling capacity and the compressor power change with respect to the rotating speed of the compressor. The evaluation of experimental data revealed that the best cooling capacity was achieved at 500 g refrigerant charge. Although, while the charge level decreased 40% below or increased 20% above the 500g of the charge amount, cooling capacity loss increased up to 25% when optimum value of 500 g of the cooling refrigerant was utilized. The test results proved in each case that increasing the compressor driving speed cause almost a linear change in the corresponding power level. The test results also shown that COP of the cooling system was decreased effectively when the revolution speed increased for any specified charge amount of the refrigerant.
Keywords: Automotive Air Conditioning; COP; Refrigerant Charge; Compressor Speed

SOĞUTUCU AKIŞKAN MİKTARININ OTOMOTİV KLİMA SİSTEMİ PERFORMANSINA ETKİSİNİN DENEYSEL ARAŞTIRILMASI

Özet: Çevreyi tehdit eden küresel ısınmayı azaltmak amacıyla araştırmacı ve endüstri kuruluşları tarafından son 25 yılda otomotiv klima sistemlerinde (AAC) önemli gelişmeler sağlanmıştır. Bu çalışmanın amacı, kompresör hızı ve soğutucu akışkan kaybının bir AAC sistemi performansına etkisini incelemektir. Bunun için bir deney düzeneği kurulmuştur. Deney düzeneği için üretici firma 750 g soğutucu akışkan tavsiye etmesine rağmen kompresör hızına göre performans-katsayısı (COP), soğutma kapasitesi ve kompresör gücü değişimini analiz etmek için aynı soğutucu akışkanın farklı miktarları ile de deneyler gerçekleştirilmiştir. Yapılan deneysel gözlemler sistem için en iyi soğutma kapasitesinin 500 g soğutucu akışkan durumunda (optimum değer) elde edildiğini ve soğutucu akışkan miktarı 500 g'dan %40 oranında düşük veya %20 oranında yüksek tutulduğunda ise soğutma kapasitesinin %25'e kadar azaldığını göstermiştir. Test sonuçları ayrıca, soğutucu akışkan seviyesinin belirli bir miktarı için kompresör hızı arttıkça soğutma sistemi performans-katsayısının etkili bir şekilde azaldığını göstermiştir.
Anahtar kelimeler: Otomotiv Klima Sistemi, COP, Soğutucu Akışkan Şarj Miktarı, Kompresör Hızı

NOMENCLATURE

AAC	Automotive air conditioning
a,b	air flow area dimensions
CFC	chlorofluorocarbon
COP	coefficient of performance
h	specific enthalpy (kJ kg^{-1})
HCFC	hydrochlorofluorocarbons
m	refrigerant mass (kg)
$\dot{m}$	mass flow rate (kg s^{-1})
MAC	Mobile air conditioner
n	compressor speed (rpm)
$\dot{Q}$	cooling capacity (W)
δQ_c	Uncertainty
rpm	revolution per minute
T	temperature (ºC, K)
u	uncertainty
v	air velocity
$\dot{W}$	compressor power (W)

Subscripts

a	air
e	evaporator
c	condanser
in	inlet
out	outlet
r	refrigerant
1,.., 4	locations in refrigerant circuit as shown in Fig.1

INTRODUCTION

The air conditioning systems were realized in past as a luxury item concerning the automotive industry, but today it is considered to be the most essential and important equipment in the field. Regarding the automotive air conditioning systems in question, it is expected that they should be able to remove the heat produced in the passenger compartments of the cars as quickly as possible under any given environmental condition. The system must also be reasonably quiet while working. These requirements make the design and analysis of automotive air conditioning systems more complex when compared with their stationary counterparts.

Several research works have been conducted in past in order to find the alternative refrigerants since the recognition of Chlorofluorocarbons (CFCs) and Hydrochlorofluorocarbons (HCFCs) as the type of refrigerants which cause a serious the depletion of ozone layer and hence trigger a great deal of global warming (Jung and Park; 1999; Al-Rabghi and Niyaz, 2000; Brown et al., 2002; Joudi et al., 2003; McCulloch et al., 2003; Tamura et al., 2005; Wongwises et al., 2006, Kim et al., 2007). It was then reported that it will be more benefitable to chose the natural refrigerants such as hydrocarbon, ammonia, carbon dioxide and water as an alternative refrigerant for most of the air conditioning systems. However, the flammability of hydrocarbons, sharp smell of ammonia, high vaporization pressure and low critical temperature of carbon dioxide restrict the use of these refrigerants considered. Another reason for the restriction not to use water in the cooling systems is that water needs a very high flow rate and operate satisfactorily only at the temperatures over 0°C.

No matter how leak-proof automotive air conditioning systems are built, refrigerant loss cannot totally be prevented. A research conducted for European Association of Car Manufacturers (ACEA) demonstrated clearly that refrigerant leakage in an AAC system was 10 g/yr (Clodic et al., 2007). Another research work proved that annual leakage amount for commercial cooling systems was 15-20% of the charged amount (Little, 2002). To solve the leakage problem in question various investigations were made in past and reported elsewhere. It is suggested in the literature that one way of reducing the refrigerant leakage is to use the indirect cooling system (Palm, 2007). However, indirect cooling systems are not employed widely in practice because of its some known intolerable disadvantages.

Refrigerant leakage during the use of CFC and HCFCs not only negatively affects the environment, but changes also the system performance. A research work on the effects of compressor revolution speed and refrigerant charge amount, on cooling capacity and COP in an automotive AC system revealed that while cooling capacity increases with increasing compressor revolution speed, COP decreased (Lee and Yoo, 2000; Kaynakli and Horuz, 2003). It is also stated that a 10% increase of refrigerant charge amount at a constant compressor revolution speed, results accordingly a certain amount of increase in the cooling capacity as well as in the COP of the air conditioning system. However, further increase in the refrigerant amount did not affect the cooling capacity at all but rather reduces the COP. The observations of system element losses against refrigerant charge amount at 0.77, 0.52 and 0.43 kg in an AAC system were made by Ratts and Brown, (2000). They determined that a 44% reduction in refrigerant charge amount causes a 26% increase in the total system performance. The losses in compressor, condenser, evaporator and the expansion valve were decreased in the order of 13%, 8%, 10% and 33% respectively with respect to a 44% reduction in the refrigerant charge amount. The effects of CO_2/propane mixture ratios in an AC system as well as their charge amounts on cooling performance were investigated by Kim et al., (2007). Their work proved that specific cooling capacity was linearly increased with increasing charge amount of the mixture. Besides, an automotive air conditioning system utilizing CO_2 has been studied by Liu at al., (2005) to investigate the effects of charge amount, oil type, evaporator output pressure, compressor revolution speed, air temperature and flow into both the condenser and the evaporator. They reported that the system performance were rather dependent on the refrigerant charge amount in the AC systems. A detailed literature survey about the reduction of refrigerant in AC systems and its effect on system performances were carried out by Poggi et al. (2008).

In this study, the performance changes in automotive air conditioning systems depending up on the refrigerant leakages were investigated. To observe the COP, cooling capacity and the compressor power; initially the manufacturer's recommended amount of 750 g of the R-134a type refrigerant charged into the system and the required tests were carried out. Then, differing amounts of same type of charge refrigerant were also charged into the test equipment and then the measurements were repeated.

MATERIAL AND METHOD

The schematic diagram of the experimental set-up used in this experimental investigation is shown in Fig. 1. It is a close refrigerant loop charged with R-134a as refrigerant and constructed by employing the original components of the air conditioning system of a commercially available driving car. The main air conditioning system comprises a finned tube condenser with evaporator, a swash plate compressor (Sanden SD 508) with 131.1 cc displacement volume, an a suitable magnetic clutch (25 W). The compressor pulley is driven by a 5.5 kW, 3-phase electric motor by the help of a belt-hoop system. A 7 kW, 3-phase COMMANDER inverter was employed to run the electrical motor. The evaporator and the condenser having wavy fin geometry are prepared by the combination of a finned aluminum plate and reliable copper tube. The cross sectional area of the evaporator and the condenser are 1.29x10^{-2} m^2 and 1.51x10^{-1} m^2, respectively. An orifice tube was also employed in the set-up as the expansion device.

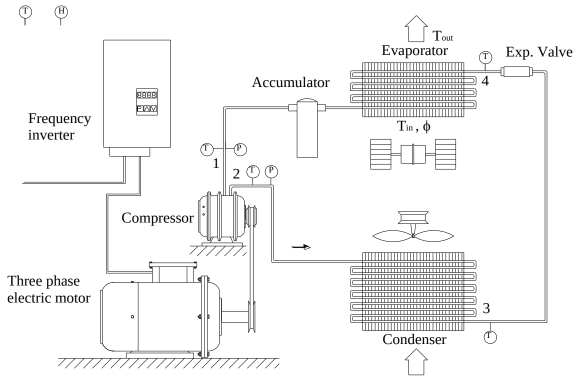

Figure 1. Schematic diagram of the experimental set-up.

Initally, the unknown cooling refrigerant in the AAC system was totally discharged prior to the tests and then recharged with the manufacturer's recommended amount of 750 g of R-134a type gas as the first charging refrigerant. The compressor was then run at this charge level of the gas for each speed of the 1000, 1500, 2000, 2500, 3000 and 3500 revolution per minute (rpm) respectively. The measurements were taken at each compressor speed after the compressor had been fully stabilized. Following this initial test, the above measurement procedure is repeated for differing amounts (300, 400, 500, 600, 700 and 800 g) of the same cooling refrigerant.

Under the ambient temperature and humidity conditions; low and high cooling system pressures, evaporator and condenser input-output temperatures, compressor revolution speed, air temperature blown out of the evaporator and evaporator air velocity were recorded.

Following the stabilization of the measuring set-up, nominal temperature was recorded in each case for the calculations and the compressor revolution speed was adjusted before each test via an inverter that is directly coupled to the electrical motor. Temperature measurements were made by employing K-type thermocouple and ADAM 4018+ ADAM-4520 Modules. Air velocity was measured by HD 2303.0 vane type Anemometer. The technical specifications of equipment used in the measurement system are given in Table 1.

Table 1. The technical specifications of the measuring system

Temperature	ADAM-4520 module (RS-232 to 422/485 Converter) ADAM-4018+(8 Channel Input) K-Type Thermocouple, 0 ~ 1370° C Sampling Rate 10 sample/sec
Air velocity	Delta Ohm Anemometer Vane Probes AP472 S4HT 0- 50 m/s Resolution 0.01 m/s
Pressure	Refco Manometer, Class 1.6

For computing the system performances, the following assumptions were made;

- The evaporator and the condenser pressure losses were assumed to be negligible.
- The entalphi changes in the expansion valve were negligible.

The parameters concerning the system performances were calculated as follows;

Cooling capacity of the evaporator ($\dot{Q}_e$) was calculated by using equation (1) given below;

$$\dot{Q}_e = \dot{m}_a \left(h_{in} - h_{out}\right) \tag{1}$$

Power ($\dot{W}$) given to the refrigerant in the compressor was obtained by equation (2);

$$\dot{W} = \dot{m}_r \left(h_2 - h_1\right) \tag{2}$$

Assuming that the heat received by the refrigerant is equal to the heat loss by the air passing through the evaporator, equation (1) can now be rewritten as follows;

$$\dot{Q}_e = \dot{m}_a \left(h_{in} - h_{out}\right) = \dot{m}_r \left(h_1 - h_4\right) \tag{3}$$

Hence, from equation (3), the refrigerant flow rate can be expressed as;

$$\dot{m}_r = \frac{\dot{Q}_e}{\left(h_1 - h_4\right)} \tag{4}$$

The coefficient of performance (COP) can be calculated from the following equation;

$$COP = \frac{\dot{Q}_e}{\dot{W}} \tag{5}$$

Here, the enthalpy changes according to the related temperature and pressure values were obtained using *RefUtil.exe programme* (Anon., 2009). A typical example of pridicting the required enthalpy is given on the p-h diagram of Fig. 2.

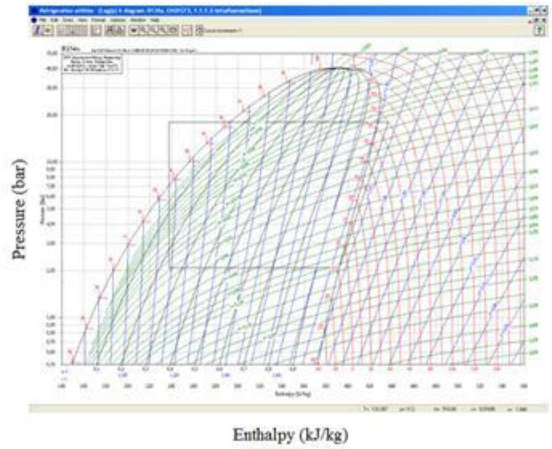

Figure 2. A typical measured experimental example for p-h diagram.

Mathematical models including the measured size, variables affecting measurement and uncertainty ratios of variables were developed in the literature for the uncertainty analysis purposes (Holman, 2001). Thus, the following well known equations may now be used for to calculate the uncertainty (δQ_c) of the cooling capacity (Q_e).

$$Q_e = f\left(v, a, b, T_{in}, T_{out}\right) \tag{6}$$

$$U\left(\frac{dQ_e}{Q_e}\right) = \frac{d\dot{Q}_e}{\dot{Q}_e} \tag{7}$$

$$\delta Q_c = \sqrt{\left(\frac{\partial Q_c}{\partial a}\right)^2 \delta a^2 + \left(\frac{\partial Q_c}{\partial b}\right)^2 \delta b^2 + \left(\frac{\partial Q_c}{\partial v}\right)^2 \delta v^2 + \left(\frac{\partial Q_c}{\partial T}\right)^2 \delta T^2 + \left(\frac{\partial Q_c}{\partial T}\right)^2 \delta T^2} \tag{8}$$

The accuracies of measurements together with the uncertainties in the calculated results are given in Table 2.

Table 2. The accuracies of the measurements and the uncertainties in the calculated results.

Measurements	Accuracy
Temperature	± 0.1%
Air velocity	± (0.2 m/s +1.0 %)
Calculated results	Uncertainty
Cooling capacity	± 5.28%

RESULTS AND DISCUSSION

Referring the experimental data, the variation of calculated cooling capacity depending on the compressor revolution speed for various amounts of refrigerants are given in Fig. 3. As seen in the related characteristic curves of Fig. 3 that the cooling capacity increases with increasing compressor revolution speed at all the refrigerant amounts tested. It is believed that an increase in the refrigerant flow circulation within the system and the increase in the compressor revolution speed were the main causes of the observed effect. It can also be seen from Fig. 3 that cooling capacity reaches its maximum level at a refrigerant amount of 500 g and decreases with any other amount of charging refrigerant. The cause of decrease in cooling capacity at lower amounts of refrigerant can be attributed to the flow of two-phase refrigerant from condenser to the expansion-valve. However, it is reported elsewhere (Farzad and O'Neal, 1991) that the causes of decrease in cooling capacity at higher amounts of refrigerants were the reduction in condensation area in the condenser and the worsened compressor volumetric performance due to the increased condensation pressure. Our findings are well agreed with these observations.

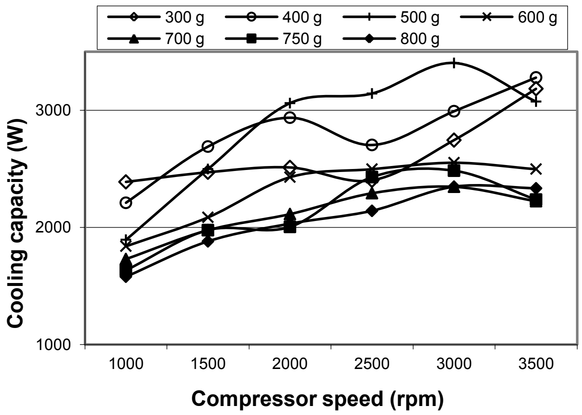

Figure 3. Effect of the compressor revolution speed on the cooling capacity in terms of refrigerant charge.

Although, a 500 g of refrigerant was found to be the best choice to charge the AAC system concerning the cooling capacity, the manufacturer's recommended amount was 750 g (Anon., 1994). The reason for this may be to maintain a good working performance for the system even if some unexpected refrigerant loses take place due to some possible system malfunctions.

The highest computed cooling capacity was found to be about 3400 W at a 500 g refrigerant charging amount (may be regarded as its optimum value) at a speed of 3000 rpm of the compressor.

The test result shown that the cooling capacity of the system was decreased 25% at the same compressor speed and increased up to 20% (600 g) over its optimum value. However the cooling capacity of the system was decreased by 12% and 19% according to a 20% (400 g) and 40% (300 g) decrease in the optimum refrigerant amount respectively. Hence, the test results demonstrate clearly that higher refrigerant charging amounts influence more rapidly on the cooling capacity change than that of the lower values.

The variation of compressor power against compressor revolution speed for differing refrigerant charges is given in Fig. 4. It can be seen from the related characteristic curves of Fig. 4 that, up to the 500 g of the optimum value of the refrigerant used in the cooling system increases the compressor power level accordingly. Any refrigerant amount above the optimum value however, decreases the compressor power level.

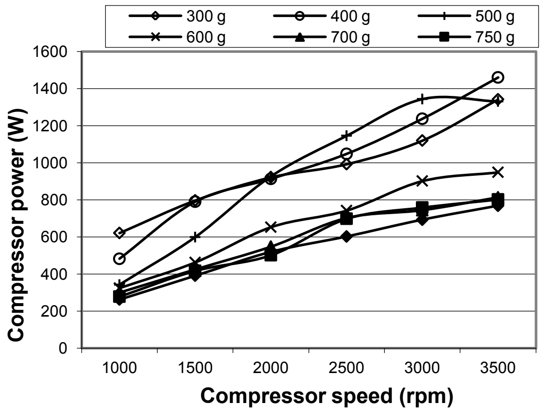

Figure 4. Effect of the compressor revolution speed on the compressor power in terms of refrigerant charge.

The characteristic curves in Fig. 4 demonstrate clearly that power given to the compressor is almost linearly changing with the compressor revolution speed at all charging amounts of the refrigerant. The characteristic curves of Fig. 4 indicate that the power given to the compressor takes its maximum values in the case of 400 g and 500 g of the charging refrigerants respectively.

The maximum value of the compressor power was found to be approximately 1460 W at a refrigerant amount of 400 g when the compressor speed was 3500 rpm. The cause of reduction in the power level over 500 g charging amount of the refrigerant may be attributed in this case to both decrease in compression ratio and the volumetric performance at extreme charge levels (Grace et al., 2005). Another important point to be mention here that the power driven from the network by the inventor was not stable. During the experimental tests, it was varified that the driven power level was reached its maximum value which was approximately 4.5 times greater than that of the compressor power level under consideration.

The variation of COP depending on the compressor revolution speed as well as on the charge of refrigerant can be seen in Fig. 5 and Fig. 6 respectively.

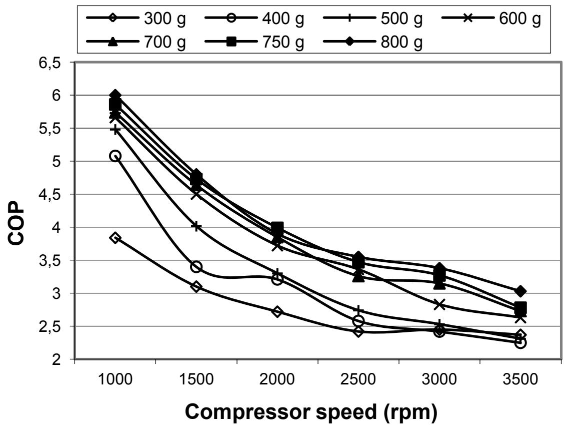

Figure 5. Effect of the compressor revolution speed on the COP in terms of refrigerant charge.

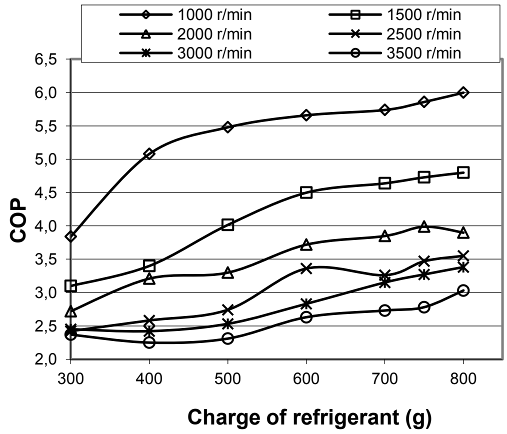

Figure 6. Effect of refrigerant charge on the COP in terms of the compressor revolution speed.

The characteristic curves of Fig. 5 clearly indicate that COP is decreasing considerably from its maximum level in all cases with an appropriate increase in the compressor revolution speed. The reason for this may be attributed to an increase in the power level of the compressor according to an appropriate increase in the cooling capacity. Furthermore as seen in Fig. 6, an increase in the refrigerant amount also results a corresponding increase in the COP of the system.The highest COP values of the system were recorded at a charging refrigerant amount of 800 g. The calculated result proved that the COP had a value of 6.0 for an 800 g refrigerant at a compressor speed of 1000 rpm. It was predicted however that the value of the COP value dropped down approximately 3.03 when the compressor speed increased up to 3500 rpm. It is also seen that the lowest COP (2.31) of the system achieved at a speed of 3500 rpm when the charging refrigerant amount was 500 g. Finally, it can be evaluated from the related characteristic curves of Fig. 5 and Fig. 6 that the cooling capacity reaches its maximum value when the system is charged with the optimum amount (500 g) of the refrigerant at a speed of 3000 rpm of the compressor. Under these conditions, it is observed that the COP of the system dropped down to its least value of 2.53. Hence, It can be concluded promptly here that the test results given in this work exhibit throughly a good agreement with the appropriate results reported elsewhere (Lee et al., 2000; Wongwises et al., 2006; Hosoz and Direk, 2006; Wang et al., 2006).

CONCLUSION

Effects of charge level and compressor revolution speed on an automotive air conditioning system performance were investigated in this study. It is believed that the change in the refrigerant charge amount as well as the compressor driving speed revolution in the practical work reported here make this study more attractive among the similar works undertaken in the literature.
The test results clearly proved that refrigerant amount influenced more effectively on cooling capacity much more than the compressor revolution speed. The best cooling capacity was obtained at 500 g refrigerant

charge level. It is found that the cooling capacity was influenced in this case by at the most by 25%.

In case of employing 400 g and 500 g of refrigerant amounts in to the cooling system, the compressor power exhibits its highest level in both cases and the cooling capacity under these conditions attains its maximum value. It should also be noted that here that increasing the compressor driving speed, cause almost a linear change in the corresponding power level.

Finally, the attention may be drawn to the point that although the manufacturer's recommended charge level for the refrigerant employed was 750 g, the test results demonstrated that the maximum cooling capacity was obtained with the dominant value of 500 g of the refrigerant under consideration. Thus, it can be argued here that even if some refrigerant losses take place in the cooling system due to some unavoidable leakage sources. It is obvious that the working performance of the system can still be properly maintained.

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