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A novel exergy based charge optimisation for a mobile air conditioning system

An experimental study

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Abstract

In this paper, a novel exergy based charge optimisation technique is proposed instead of coefficient of performance (COP) based method for a mobile air conditioning system. The exergy destructions in each of the components in the system as a function of compressor speed and refrigerant charge level were estimated based on experimental data. It was found that the COP of the system decreases as the compressor speed increases, while the total exergy destruction in the system increases. The exergetic analysis of performance data showed that the percentage loss in the compressor, evaporator, condenser and thermostatic expansion valve lies in the range of 59.88–69.9%, 17.53–25%, 7.80–16.36% and 3.05–15.73%, respectively. Based on COP, the optimum charge varied with respect to compressor speed. The exergy based optimisation is found to be more reliable and consistently yielding a single optimum charge of 620 g irrespective of the compressor speed. The maximum exergetic efficiencies at 620 g were 43.32% and 38.15% for 900 rpm and 1800 rpm respectively.

Keywords Mobile air conditioning · Optimum refrigerant charge · Exergetic efficiency · COP · R134a · Exergy destruction

List of symbols

A/C	Air conditioning
COP	Coefficient of performance
DBT	Dry bulb temperature (°C)
DSC	Degree of subcooling
DSH	Degree of superheating
ED	Exergy destruction rate (kW)
EDR	Exergy destruction ratio
F	Fans
H	Heaters
h	Enthalpy (kJ kg^{-1})
HC	Hydrocarbon
HVAC	Heating, ventilation and air conditioning
IHX	Internal heat exchanger
IRDC	Integrated receiver dryer condenser
$\dot{m}$	Mass flow rate (kg s^{-1})
MAC	Mobile air conditioning
P	Pressure (bar)
PR	Pressure ratio
Q	Heat transfer (kW)
RH	Relative humidity (%)
RPS	Regulated power supply
$\dot{S}$	Entropy (kJ kg^{-1} K^{-1})
T	Temperature (°C)
TXV	Thermostatic expansion valve
VCRS	Vapour compression refrigeration system
VFD	Variable frequency drive
W	Work consumption (kW)
η	Efficiency (%)

Subscripts

comp	Compressor
cond	Condenser
d	Discharge
ele	Electrical power
evap	Evaporator
ex	Exergy
exp	Expansion device
gen	Generation
o	Dead state

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r   Refrigerant
ref   Reference state

Introduction

In an automobile, air conditioning system consumes 12–17% of the engine's total energy [1]. So it is important to reduce the energy consumption of the air conditioners in automobiles, which also reduces the exhaust emissions from the vehicles. Many researchers are working on new environment-friendly alternate refrigerants because of the Kyoto protocol [2] and Montreal protocol [3]. It is also very important to use energy-efficient components in the air conditioning systems to reduce the energy consumption [4].

The efficiency of the mobile air conditioning (MAC) system predominantly depends on the heat exchangers used. In the past laminated plate evaporators were used in the MAC systems but currently minichannel and microchannel heat exchangers are used for improving system performance [5, 6]. The use of minichannel heat exchangers in air conditioning system also reduces the refrigerant charge, which reduces the environmental impacts due to refrigerants. However, it has a negative impact on the pressure drop and power consumption [7–11]. The requirement for the design and analysis of MAC systems are more complicated when compared to the room air conditioning systems. The MAC system may lose its refrigerant charge over a period of time due to leaks through the flexible hoses and connections between the joints through sealing. The loss of refrigerant charge can reduce the system performance. In particular, the observable effects are reduction in cooling capacity, reduction in condenser pressure and also increase in the degree of superheating at evaporator outlet. This may cause overheating of the compressor and increase the power consumption [12].

The biggest challenge for MAC engineers is to use less power from the engine in order to make an efficient system and also to reduce the pollution effects from the engine. MAC engineers normally fill additional refrigerant charge so that the system can perform across a wide range of heat loads and vehicle operating conditions. It is essential to determine the optimum charge quantity of refrigerant for a MAC system in order to get maximum performance under any range of ambient conditions. For proper charge optimisation and efficient use of energy, a detailed thermodynamic analysis is required. Generally, first law-based analysis is used; however, it deals with conservation of energy and it does not give details of energy losses (due to irreversibilities) in the system or in each process. However, the second law-based thermodynamic analysis gives a better understanding of irreversibility. The second law analysis also determines the magnitude of exergy losses associated in each process and thereby it gives a proper direction for improving system performance. Many researchers have carried out the performance evaluation and charge optimisation in MAC systems using first law analysis [13–17]. In MAC systems, charge optimisation is usually done at some extreme operating conditions in a wind tunnel. But such operating conditions vary with different types of manufacturers in the world. Many researchers have conducted charge optimisation at idling speed. Nelson et al. [18] and Collins et al. [19] carried out charge optimisation for a MAC system with an evaporator air inlet temperature and RH of 43.3 °C and 40%, respectively, at a compressor speed of 1800 rpm. Lee et al. [20] conducted charge optimisation under evaporator air inlet conditions 30 °C DBT and 50% RH with a compressor speed of 1800 rpm. Qi et al. [9] performed a charge optimisation test at evaporator air inlet conditions of 35 °C and 40% RH with a speed of 1800 rpm. Wang et al. [21] studied the effects of refrigerant charge on the MAC system performance. The test was conducted at evaporator air inlet temperature and RH of 30 °C and 40% with compressor speed of 2000 rpm. According to SAE standards [22], air-side conditions were fixed at 35 °C and 40% relative humidity for both the condenser and the evaporator with compressor idling speed. Cummings et al. [23] and Zilio [16] were also used SAE standard for charge optimisation test.

For calculating actual system losses due to irreversibility, exergy based analysis is a useful tool, which was recommended by many authors [24–36]. Yataganbaba et al. [24] presented a theoretical model-based exergy analysis of R1234yf, R1234ze as a replacement for R134a in multi-evaporator system. They concluded that R1234ze has the highest exergy efficiency than R134a and R1234yf. It was observed that the greatest portion of exergy destruction took place in the compressor. Golzari et al. [25], Hepbasli [26], Aprea et al. [27] and Saravanakumar et al. [28] also confirmed that the majority of the losses occurred in the compressor, which decides the overall performance of the system. Qureshi et al. [29] experimentally investigated the effects of subcooling on the system performance. It was found that exergetic efficiency was improved up to 21% by increasing the subcooling from 5 °C to 8 °C, while cooling capacity also got increased by 0.5 kW. Bayrakci et al. [30] theoretically compared the energetic and exergetic performance of VCR system using R290, R600, R600a and R1270 as an alternatives to R134a and R22. They concluded that R1270 was the best choice for replacing R134a and R22 in terms of COP and exergy efficiency. They also reported that the exergy efficiency was decreased as evaporation temperature increases.

Ratts et al. [31] experimentally analysed the effects of refrigerant charge on MAC system performance. They found that the exergetic efficiency was increased by 26%

with 44% (0.77 kg to 0.43 kg) reduction in refrigerant charge. A decrease in the refrigerant charge level from 0.68 kg to 0.54 kg, the losses in the compressor, the condenser, the evaporator and expansion valve decrease by 13, 8, 10 and 33%, respectively. Cho et al. [32] carried out performance and exergy analysis using R1234yf in a MAC system with internal heat exchanger (IHX). They concluded that the IHX improves the exergetic efficiency of R1234yf system by 1.5% to 4.6%. At lower speeds, exergy destruction was higher in the evaporator, whereas the exergy destruction in the compressor dominates the system performance at higher speeds. Kizilkan et al. [33] conducted an experiment to analyse the exergetic performance of refrigeration system with R404a by varying compressor frequency from 35 Hz to 50 Hz. They reported that maximum exergy destruction experienced in the compressor and expansion devices were increased as compressor speed increases; however, the losses in condenser and evaporator decrease. Sagia et al. [34] presented the effects of condensation and evaporation temperatures on exergetic efficiency of the vapour compression refrigeration (VCR) cycle. At constant evaporation temperature, the exergetic efficiency was decreased as condensation temperature was increased from 25 °C to 60 °C. However, the exergetic efficiency of the system attains a maximum of 45% at evaporator temperature of -25 °C for R22 refrigerant. Raveendran et al. [35] proposed a brazed plate condenser in a refrigeration unit with HC mixture (54.8% HC600a and 45.2% HC290) and R134a. The results showed that the proposed system with HC mixture has 4–7% higher exergy efficiency than that of R134a. Anand et al. [36] carried out an exergy analysis in a VCR system for different percentage of refrigerant charge. It was found that total exergy destruction was higher when the system charge was 100%, whereas it is low when charge is 25%.

According to the literature, exergy analysis in the field of mobile air conditioning systems was limited and many researchers have conducted charge optimisation using first law analysis. Moreover, to the best of the author's knowledge, no study has been performed related to the exergy-based charge optimisation in MAC system. In this work, the performance and exergy destruction of R134a mobile air conditioning system with minichannel evaporator and integrated receiver drier (IRD) condenser as a function of refrigerant charge quantity and compressor speed were studied. The optimal refrigerant charge quantity of the MAC system with respect to COP and exergetic efficiency was arrived.

Experimental

Experimental set-up

The MAC test rig is shown in Fig. 1 which consists of wobble plate compressor (160 cc), a parallel flow IRD condenser, an externally equalised thermostatic expansion valve and a minichannel evaporator. The wobble plate compressor is belt driven by a 3-phase 2HP electrical motor with a maximum speed of 3000 rpm. The motor is powered through a variable frequency drive (VFD) for changing the compressor speeds for various tests. The diaphragm of the TXV which regulates the valve opening is under balance against the evaporator pressure, the saturation pressure corresponding to the superheat temperature at evaporator exit and the spring tension. However in order to consider the frictional pressure drop in the evaporator, the pressure is tapped at the exit of the evaporator instead of the inlet. This is externally equalised TXV, which is commonly used in the MAC systems. Generally, evaporator capacity is controlled by a thermostat which cuts the power supply to the electromagnetic clutch when the required supply air temperature is achieved. However, in this present study thermostat was disconnected in order to conduct experiments at steady state. The details of MAC system components are tabulated in Table 1.

The air flow across the condenser can be fixed as required by varying the speed of the blower through a VFD. Similarly, the air flow through the evaporator (cooling coil) could be varied by using regulated power supply (RPS) through which the blower is operated. The cooling coil was fixed in its original HVAC panel and the same was placed in the calorimetric chamber. The chamber consists of electrical heaters and an ultrasonic humidifier to supply heat load to the system that maintains specified temperature and relative humidity (RH) in the chamber. All the refrigerant lines and calorimetric chamber were thermally insulated by polyurethane foam. The measurement locations of refrigerant-side temperatures and pressures are indicated in Fig. 1, and the dry bulb temperature (DBT) and RH of the air were measured across evaporator and inside the calorimeter. For measuring the refrigerant mass flow rate, a mass flow meter (Endress + Hauser promass 80) was fixed in the liquid line between condenser and thermostatic expansion valve (TXV). Air flow rate through the condenser and evaporator was calculated by measuring the velocity of air stream using vane-type anemometer. An optical tachometer (SPEEDS—HTM560) was used to measure the speed of the compressor. All the temperature and pressure sensors were logged into the computer by means of a data acquisition system. The accuracy of measuring parameters is listed in Table 2.
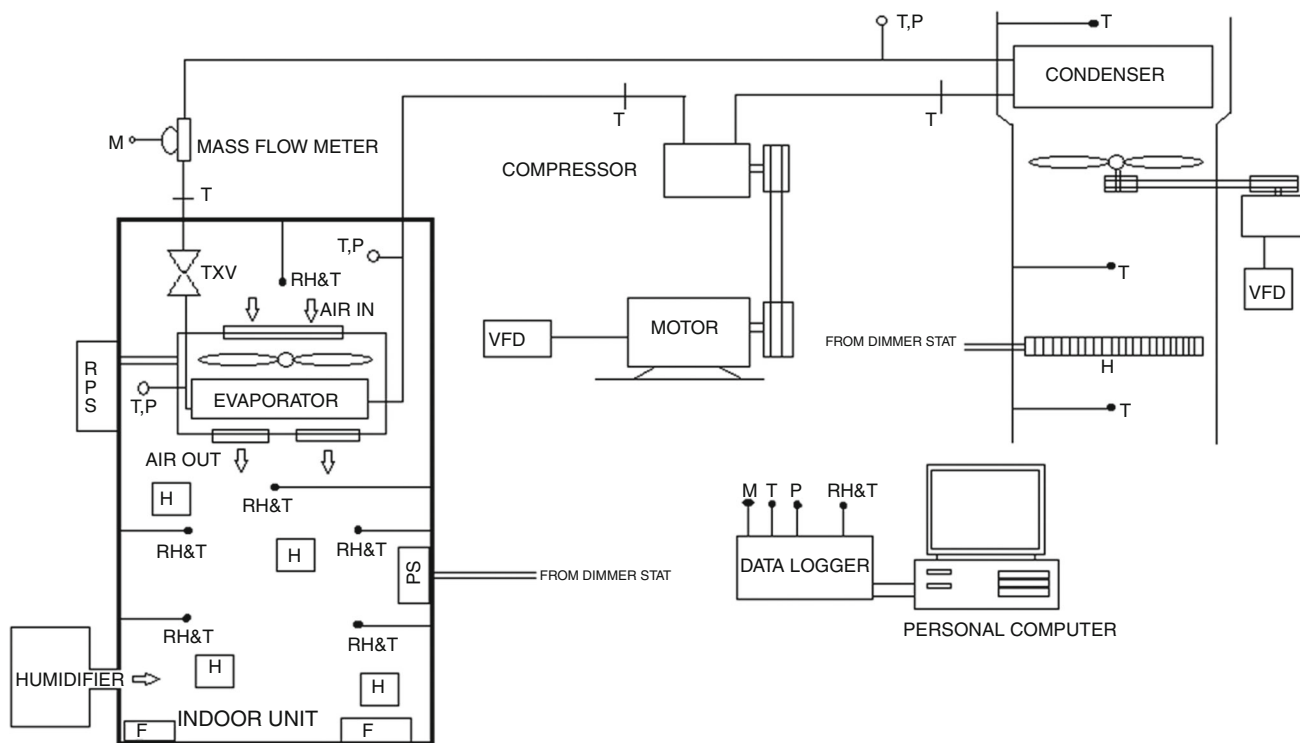
Fig. 1 Schematic diagram of MAC bench test rig

Table 1 Components of MAC system

Items	Specifications
Compressor	Wobble plate type with 160 cc
Condenser	IRD condenser (590 × 444.8 × 16/mm)
Expansion device	Thermostatic expansion valve
Evaporator	Minichannel type (230 × 201 × 38/mm)

Table 2 Measured parameters and accuracy

Measuring parameters	Accuracy
Temperature	± 0.15/°C
Refrigerant pressure	± 0.25/%
Air-side pressure	± 5/Pa
Refrigerant mass flow rate	± 0.15/%
Compressor speed	± 0.05/%
Power meter	± 0.5/%

Test condition

In the present study, one typical condition is chosen for charge optimisation test with two compressor speeds as listed in Table 3.

Experimental procedure

Initially, the condenser blower was switched ON and the speed of the blower was regulated with the help of VFD to attain the required face velocity of air. The evaporator blower speed was regulated using RPS to attain the required air flow rate, and then, compressor was switched ON. The required compressor speed was achieved by using VFD, which controls the motor that drives the compressor.

Initially, 300 g of refrigerant R134a was filled in the MAC system. For each of the charge quantities in order to achieve the test conditions, the loads (sensible and latent heat) inside the calorimeter were regulated using a dimmerstat and the ultrasonic humidifier, respectively. The humidity was set to automatically maintain the RH inside through the feedback control system. Small fans are suitably placed in the cabin for maintaining uniform cabin temperature and RH.

By suitably adjusting the heat loads, the test conditions were achieved. After attaining the test condition, the system is in steady state for about 30 min. The following variables were logged in for another 30 min at 10-s interval: (1) refrigerant temperature and pressure across heat exchangers, TXV and compressor; (2) mass flow rate of refrigerant; (3) DBT and RH of air across condenser and evaporator; (4) air-side pressure drop across evaporator; and (5) compressor power. Then, the system charge quantity was increased from 300 g in steps of 30 g to arrive

Table 3 Test conditions for charge optimisation

Comp. speed/rpm	Evaporator coil inlet conditions		Condenser inlet conditions	
	Air flow rate	Temperature (DBT and RH)	Air face velocity	Temperature
900 (idling) 1800 (city limit)	420/m³ h⁻¹	35/°C and 40/%	2.5/m s⁻¹	35 °C

at the optimum charge level. The same procedure was repeated at each charge level. The entire procedure was carried out separately for both the compressor speeds.

Exergy analysis

According to the first law, the performance of VCR system as shown in Fig. 2 is depended on the overall COP which is defined as the ratio of cooling capacity to electrical power consumed by the compressor (Eq. 1).

$$\text{COP} = \frac{Q_{\text{evap}}}{W_{\text{ele}}} \tag{1}$$

where Q_{evap} and W_{ele} are the cooling capacity and electrical power consumptions by the compressor. These are calculated by using Eqs. 2 and 4:

$$W_{\text{comp}} = \dot{m}_{\text{r}}(h_2 - h_1) \tag{2}$$

$$W_{\text{ele}} = \frac{W_{\text{comp}}}{\eta_{\text{ele}}\eta_{\text{mech}}} \tag{3}$$

$$Q_{\text{evap}} = \dot{m}_{\text{r}}(h_1 - h_4) \tag{4}$$

whereas the heat rejection in the condenser is also calculated by using enthalpy difference across the condenser as follows,

$$Q_{\text{cond}} = \dot{m}_{\text{r}}(h_2 - h_3) \tag{5}$$

Another important parameter in VCR system is the volumetric efficiency of the compressor, which is calculated by using following equation:

$$\eta_{\text{vol}} = \frac{\dot{m}_{\text{r}} v_{\text{r.in}}}{\text{compressor displacement rate}} \tag{6}$$

Exergy analysis is a beneficial tool to evaluate the thermodynamic losses in each component of the system and the whole system and to determine exergetic efficiency of the system. This analysis can indicate the possibilities of the system and component improvements. Exergy is defined as the maximum work that can be produced by a process or a system or the useful work potential of a given amount of energy at a specified state. The available energy at exit of the process is always lesser than the exergy at inlet because of some undefined losses. This loss in available energy during a process occurs due to the entropy generation. This amount of energy loss is normally called as the exergy destruction or internal energy losses. The components with more exergy losses are having a greater potential for system improvements. The general exergy balance can be expressed in the following Eq. 7 [37]

$$\text{ED} = (\dot{\varepsilon}_{\text{in}} - \dot{\varepsilon}_{\text{out}}) \tag{7}$$

where ED is the exergy destruction rate and $\dot{\varepsilon}_{\text{in}} - \dot{\varepsilon}_{\text{out}}$ is the rate of total exergy transferred by the heat, work and the mass interaction during the process. It is also determined

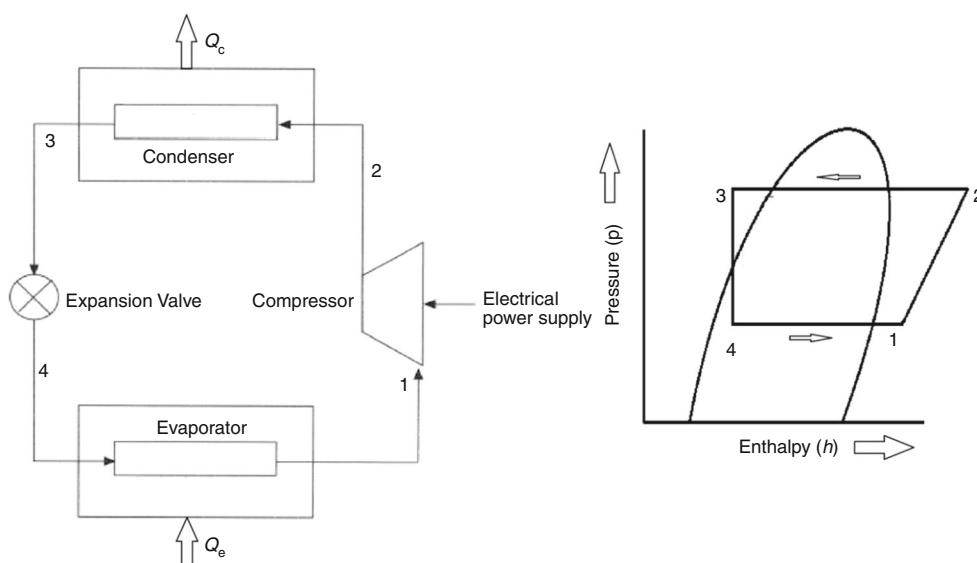**Fig. 2** Basic schematic and p–h diagram of VCR system

by calculation of entropy generation in a process using Eq. 8 [37]

$$ED = T_o \dot{S}_{Gen} \qquad (8)$$

where T_o is the dead-state temperature or environment temperature and $\dot{S}_{Gen}$ is the entropy generated during the process. Exergy destructions for major components of the VCR cycle as shown in Fig. 2 are as follows:

Compressor:

$$ED_{comp} = \dot{m}_r(h_1 - h_2) - \dot{m}_r T_0(s_1 - s_2) + \dot{W}_{ele} \qquad (9)$$

Condenser:

$$ED_{cond} = \dot{m}_r T_0(s_3 - s_2) + Q_{cond}\left(\frac{T_0}{T_{ref,out}}\right) \qquad (10)$$

$$ED_{cond} = -\dot{m}_r T_0(s_2 - s_3) - Q_{cond}\left(\frac{T_0}{T_{ref,out}}\right), \qquad (11)$$

$$\text{if } W_{ele} \text{ is positive}$$

Expansion device:

$$ED_{exp} = \dot{m}_r T_0(s_4 - s_3) \qquad (12)$$

Evaporator:

$$ED_{evp} = \dot{m}_r T_0(s_1 - s_4) - Q_{evap}\left(\frac{T_0}{T_{ref,out}}\right) \qquad (13)$$

The total exergy destruction:

$$ED_{total} = ED_{comp} + ED_{cond} + ED_{exp} + ED_{evap} \qquad (14)$$

The relative exergy destruction of the each component in the VCR system is defined as ratio of the individual component exergy destruction to the total exergy destruction in the system.

$$ED_{relative} = \frac{ED_{component}}{ED_{total}} \times 100 \qquad (15)$$

The exergetic efficiency is very useful to optimise the VCR system parameters and which is defined as:

$$\eta_{ex} = \frac{Q_{evap}\left(1 - \frac{T_0}{T_{ref,out}}\right)}{W_{ele}} = 1 - \frac{ED_{total}}{ED_{in}} \qquad (16)$$

The exergy destruction ratio (EDR) is also another evaluation concept of second law performance. The exergy destruction ratio is defined as the ratio of total exergy destruction to the exergy in the product [38]. Both exergetic efficiency and EDR give better understanding on the system performance than the first law-based COP.

$$EDR = \frac{ED_{total}}{Q_{evap}\left(1 - \frac{T_0}{T_{ref,out}}\right)} = \frac{1}{\eta_{ex}} - 1 \qquad (17)$$

Results and discussion

Performance of the system

Based on the experimental data obtained, the performance parameters, namely cooling capacity, work of compression and COP of the system, were estimated at all charge levels for the two compressor speeds considered. The performance variables observed, namely the refrigerant mass flow rate, the discharge pressure, degree of subcooling, degree of superheating, are plotted along with the estimated parameters, viz. cooling capacity, work of compression and COP in Fig. 3. As shown in Fig. 3a, the refrigerant mass flow rate and discharge pressure increase with the increase in charge quantity and compressor speed. In a typical MAC system, the refrigerant mass flow rate is regulated by the TXV according to the load sensing by degree of superheating at the evaporator outlet. The results indicate that up to a 500 g of charge quantity, the TXV just allows the refrigerant mass flow rate to drastically increase because it is fully opened due to the high degree of suction superheat as shown in Fig. 3b. It is also indicates that the degree of superheating decreases drastically as the refrigerant charge is increased further, while the degree of subcooling increases as the charge is increased to 500 g. The degree of superheating is significantly smaller (< 10 °C), and TXV is able to control the system refrigerant flow rate, which means that the system runs smoothly and the degree of superheating (DSH) at evaporator outlet slowly reaches the normal range of 5 °C. At the same time the degree of subcooling (DSC) at condenser outlet is above 10 °C as shown in Fig. 3b. This point is called minimum effective charge quantity of the system. But this need not be optimal charge level, and hence, the system was charged further to realise a plateau in DSC. At compressor speed of 900 rpm, the degree of subcooling at condenser outlet is almost the same between the range of 590–620 g and after 620 g; the DSC is suddenly increased about 1 °C. It also indicates that refrigerant mass flow rate is decreased after 620 g (Fig. 3a) which means the system attains its maximum liquid storing limit in the IRD condenser. Because of overcharging, the excess refrigerant gets stored in condenser and the discharge pressure increases, and due to these changes, the compressor pumping capacity reduced at the same rpm.

Figure 3c shows the variations of cooling capacity and the compressor power consumption with respect to charge quantity and compressor speed. It indicates that the cooling capacity and compressor power increase with compressor speed and charge quantity. The compressor power is increased up to 98% as compressor speed is increased from 900 to 1800 rpm. The cooling capacity is increased with
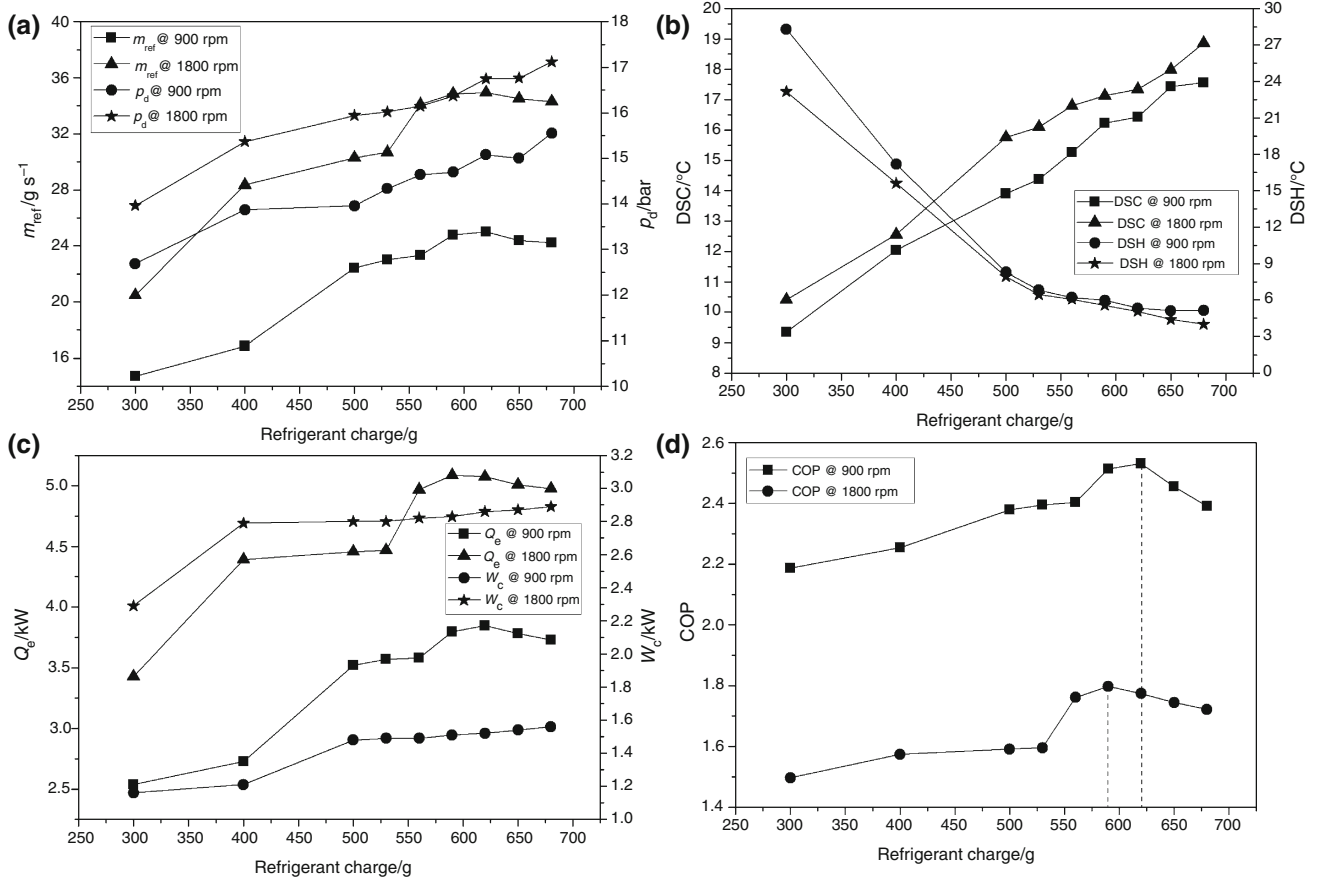
Fig. 3 Effects of refrigerant charge on **a** refrigerant mass flow rate and discharge pressure, **b** degree of subcooling and degree of superheating, **c** cooling capacity and compressor power, **d** COP

charge quantity up to 620 and 590 g at 900 and 1800 rpm, respectively, beyond that it starts decreasing.

In Fig. 3d, the change in COP with respect to charge quantity and compressor speed shows that the maximum COP of the system occurs with charge quantity of 620 g at compressor speed of 900 rpm. This means after the 620 g, the rise in compressor power consumption outweighs the rise in cooling capacity. This point is called optimum charge quantity of the system. Therefore, the optimal charge amount for the system is to be 620 g at compressor speed of 900 rpm. Similar trends occur at the compressor speed of 1800 rpm also, and the optimal charge of the system is obtained at 590 g. At this optimum point for 1800 rpm, COP of the system is decreased by 40.7% with respect to 900 rpm. It is clearly observed that at higher rpm, the rise in cooling capacity does not commensurate with the higher power consumed by the compressor; hence, the optimum charge arrived based on COP varies with respect to speed. In this paper, we propose alternatively to arrive at optimum charge with respect to exergetic efficiency.

The effects of refrigerant charge on volumetric efficiency and pressure ratio of the compressor are illustrated in Fig. 4. It is observed that the volumetric efficiency of the compressor decreases when the compressor speed is increased from 900 to 1800 rpm, while the pressure ratio increases up to 5.4. The reduction in volumetric efficiency
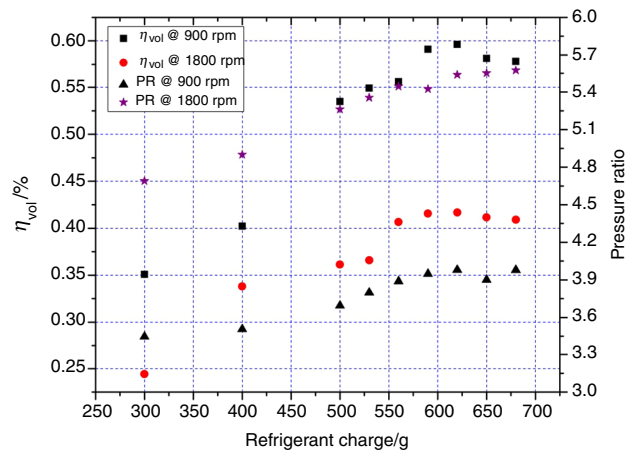

Fig. 4 Effects of refrigerant charge on compressor volumetric efficiency and pressure ratio

is due to the higher pressure ratio in the compressor at 1800 rpm when compared to that of 900 rpm. For both the compressor speeds of 900 rpm and 1800 rpm, the maximum volumetric efficiencies, viz. 59.62 and 41.63%, respectively, were experienced with 620 g of refrigerant charge only. The volumetric efficiency is dependent on the rpm of the compressor for a specific charge quantity which keeps on increasing until 620 g for both the speeds and then decreases. However, from the view point of energy efficiency the work of compression also should be considered along with the desired effect that is the cooling capacity where exergy efficiency can be considered. This is demonstrated in the next section.

System exergy analysis

The exergy efficiency of the system is dependent on the energy losses in the individual components. The exergy destruction rate in each component is discussed below. The exergy destruction rate in compressor as a function of refrigerant charge and the compressor speed is shown in Fig. 5. As speed increases from 900 to 1800 rpm, the exergy destruction in compressor increases up to 136%. For the lowest charge of 300 g, the exergy destructions in the compressor are 0.719 and 1.46 kW, while the maximum exergy destructions occur at the maximum charge of 690 g which are 0.831 and 1.604 kW at 900 and 1800 rpm, respectively. The exergy destruction in the compressor increases as the refrigerant charge increased from 300 to 500 g for both the compressor speeds which is due to the increase in discharge pressure and pressure ratio as shown in Figs. 3a and 4, respectively. After 500 g of charge, the exergy destruction rate reduces up to a charge level of
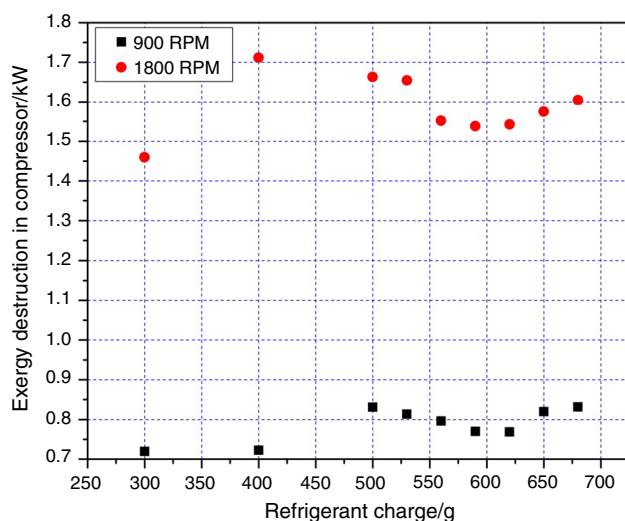
620 g, due to reduction in DSH at compressor inlet which increases the volumetric efficiency of compressor as well as reduces the work of compression. Beyond 620 g charge, exergy destruction starts increasing in both the conditions because of excess refrigerant accumulation in the IRD condenser, which causes increase in discharge pressure of the compressor, thus reducing the volumetric efficiency as shown in Fig. 4, respectively. This is clearly depicted in the reduction in mass flow rate of the refrigerant plotted in Fig. 3a.

The influence of refrigerant charge and compressor speed on the exergy destruction in the condenser is shown in Fig. 6. It indicates that the exergy loss in the condenser increases with the increase in refrigerant charge and the compressor speed. As the compressor speed increases from 900 to 1800 rpm, the exergy destruction rate increases from 45.55 to 98.92% for the range of charge quantity considered. The increase in refrigerant charge increases the condensing temperature leading to higher irreversibilities and the exergy destruction $\left[Q_c \left(1 - \frac{T_0}{T_{ref,out}} \right) \right]$ increases.

The effect of refrigerant charge and compressor speed on the exergy destruction in TXV is shown in Fig. 7. It indicates that the exergy destruction in the TXV increases as compressor speed increases from 900 to 1800 rpm by 77. 2–133.02%. Figure 7 shows that the exergy destruction increases as the charge increases from 300 to 530 g and beyond that it starts decreases. The maximum loss occurs at 530 g, while the minimum loss occurs at 680 g for both the compressor speeds. The exergy loss is higher at higher rpm; it is to be noted that the exergy destruction in the TXV depends on the combined effects of refrigerant mass flow rate and the quality of refrigerant at inlet which are depended on the charge quantity.

Fig. 5 Exergy destruction in compressor as a function of refrigerant charge and compressor speed
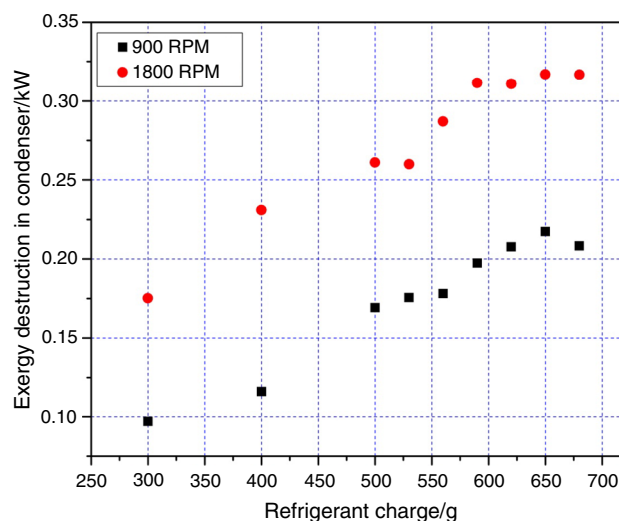

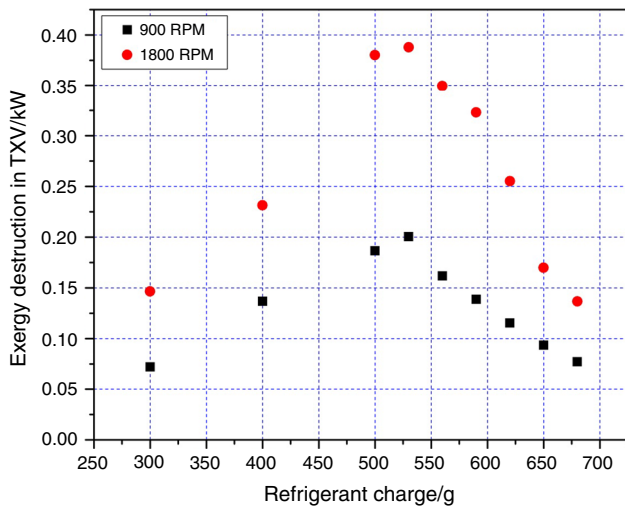Fig. 6 Exergy destruction in condenser as a function of refrigerant charge and compressor speed

Fig. 7 Exergy destruction in thermostatic expansion valve (TXV) as a function of refrigerant charge and compressor speed

The effect of refrigerant charge and compressor speed on the exergy destruction in evaporator is shown in Fig. 8. The exergy destruction in the evaporator increases with the increase in compressor speed and decreases with the increase in charge quantity up to 560 and 590 g at 900 and 1800 rpm, respectively. With the increase in refrigerant charge, the evaporation pressure, temperature and mass flow rate of refrigerant are increased. The increase in evaporator temperature reduces the temperature difference between evaporator and indoor air which reduces the exergy destruction $\left[Q_e \left(1 - \frac{T_0}{T_{ref,out}} \right) \right]$. The increase in mass flow rate enhances the heat transfer by reducing the length of superheating region in the evaporator. The maximum and minimum loss occurring in the evaporator at 300 and 560 g at a speed of 900 rpm are 0.221 and 0.283 kW,
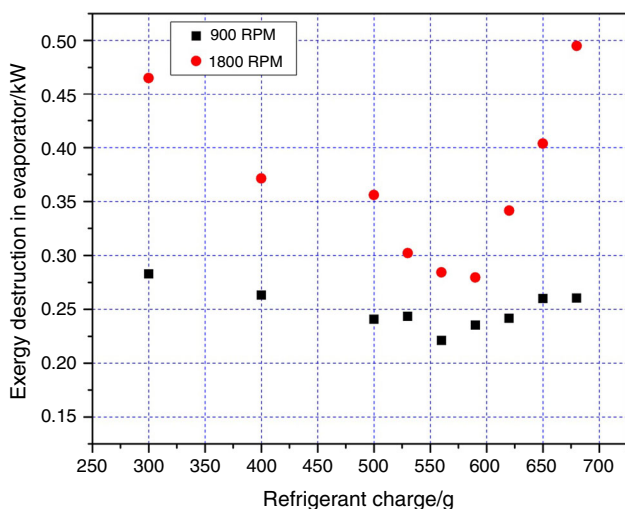
respectively. At a compressor speed of 1800 rpm, the maximum and minimum exergy destructions are experienced at 680 and 590 g which are 0.495 and 0.284 kW, respectively. Exergy destruction in the evaporator is increased by 18.85–90.11% as compressor speed increases from 900 to 1800 rpm.

The effects of refrigerant charge quantity on the total exergy destruction in the system at the compressor speeds of 900 and 1800 rpm are the algebraic sum of all the individual component losses as shown in Fig. 9. The result indicates that the total exergy destruction is increased as refrigerant charge increases from 300 to 500 g, which means increase in refrigerant charge causes more of refrigerant to flow and increases the heat dissipation. Beyond 500 g, the degree of subcooling and degree of superheating reach a combination that corresponds to minimum effective charge quantity which results in the reduction in total exergy destruction. However, when the charge quantity increases above 620 g, the excess charge accumulation in the IRD condenser results in excessive exergy destruction. As the compressor speed increases from 900 to 1800 rpm, the total exergy destruction is increased in the range of 77.31–105.45% for the considered range of charge quantity.

The effects of refrigerant charge on the exergetic efficiency and exergy destruction ratio (EDR) are shown in Fig. 10. The exergetic efficiency decreases as compressor speed increases from 900 to 1800 rpm, whereas the EDR increases. The maximum exergetic efficiency attained for both the compressor speeds of 900 and 1800 rpm is 43.32 and 38.15%, respectively, at 620 g. Similar studies conducted by Cho et al. [32] reported that the exergetic efficiency of the MAC system with R134a was reduced by 53.3% as speed increased from 800 to 2500 rpm.
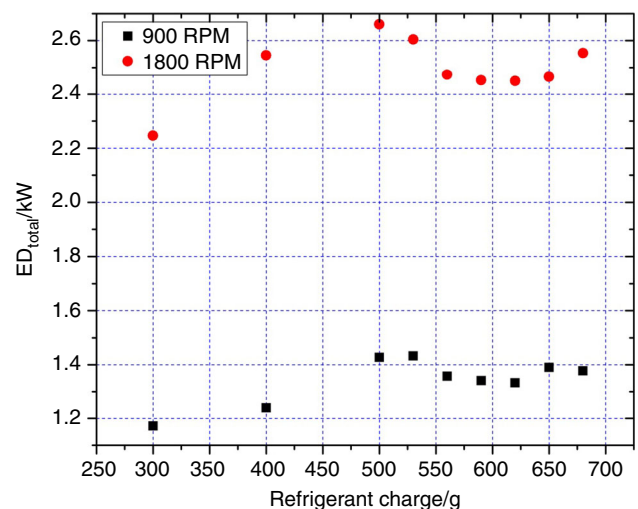

Fig. 8 Exergy destruction in evaporator as a function of refrigerant charge and compressor speed

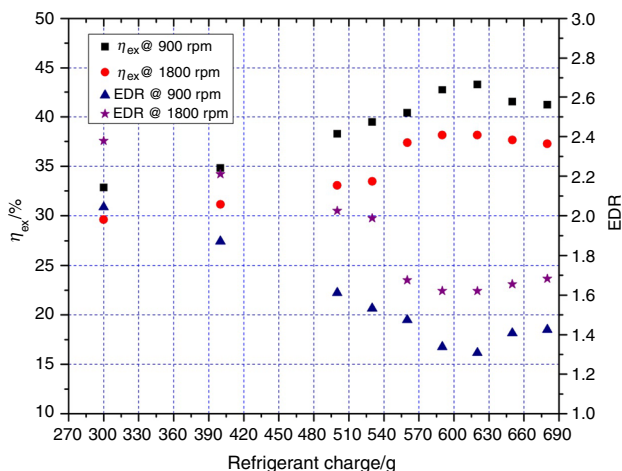Fig. 9 Effects of refrigerant charge on total exergy destruction of the system

Fig. 10 Effects of refrigerant charge on the exergetic efficiency and EDR

Apera et al. [39] also confirmed that the exergetic efficiency was decreased from 35 to 22% as compressor frequency was increased from 30 to 50 Hz. Thus based on exergetic efficiency, the optimum charge is the same irrespective of the compressor speed, and hence, it is a better approach than arriving optimum charge based on maximum COP that varies with compressor speed. The drop in exergetic efficiency due to rise in compressor speed varies in the range of 9.33–15.22% as the charge quantity varied from 500 to 690 g, while EDR increases in the range of 13.62–29.68%.

The effects of compressor speed on the exergy destruction through individual component loss and the relative exergy destruction of each component in the MAC system at the optimal refrigerant charge of 620 g are shown in Fig. 11a, b, respectively. The results indicate that the compressor loss accounting for the largest percentage of the total exergy destruction is 57.65% followed by evaporator, condenser and TXV as 18.13, 15.57 and 8.65%, respectively, at 900 rpm. Similar trends are experienced as compressor speed was increased to 1800 rpm. Such an observation was also reported by Apera et al. [40]. As compressor speed increases from 900 to 1800 rpm, the amount of loss occurring in the compressor, condenser, TXV and evaporator is increased by 100.6, 49.6, 113.33 and 41.22%, respectively.

When comparing the relative exergy destruction (ED$_{relative}$) with the increase in compressor speed, the ED$_{relative}$ of compressor and TXV is increased by 5.31 and 1.76%, respectively, whereas the ED$_{relative}$ of condenser and evaporators is decreased by 2.88 and 4.19%, respectively. Reduced relative exergy destruction in the condenser and evaporators is due to more refrigerant flow rate which leads to better heat transfer performance in the heat exchangers. The exergy destruction produced in the
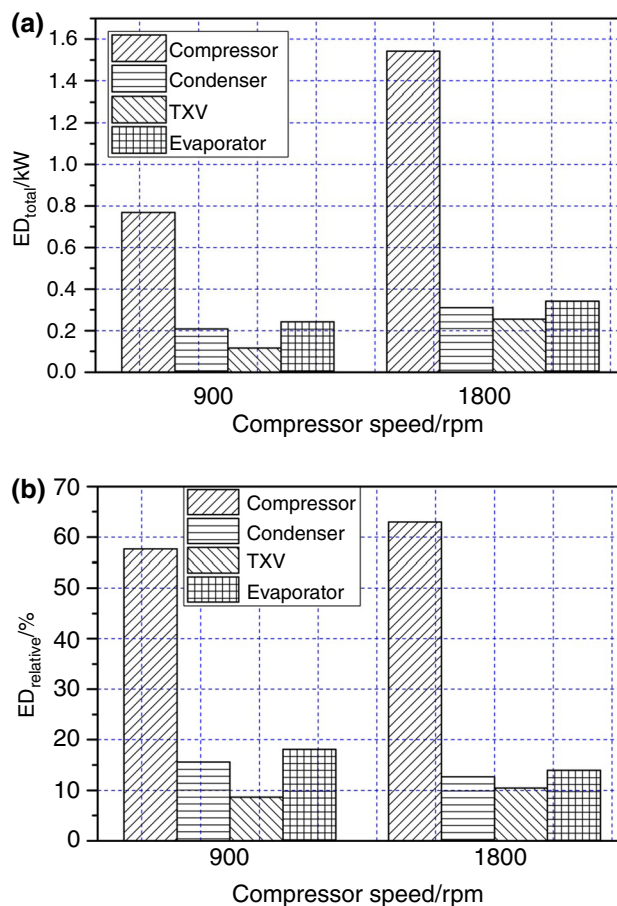

Fig. 11 Effects of compressor speed on exergy destruction at the optimal charge: **a** absolute loss, **b** relative loss with respect to total loss

thermostatic expansion valve mostly depends on the inlet condition and mass flow rate of the refrigerant.

Conclusions

Charge optimisation and the exergetic losses in an R134a mobile air conditioning system have been experimentally analysed for two compressor speeds of 900 rpm (idling speed) and 1800 rpm (city limit speed), and the following conclusions were made:

- According to the energy analysis, the optimal charge quantities to maximise system COP were 620 and 590 g for 900 and 1800 rpm, respectively.
- The exergy-based analysis proved that to be more reliable and consistent yielding a single optimum charge is of 620 g for both speeds that maximise exergetic efficiency.
- The greatest fraction of losses that had taken place in the compressor followed by evaporator, condenser and

TXV were 57.65, 18.13, 15.57 and 8.65%, respectively, at 900 rpm.

- For both the compressor speeds of 900 and 1800 rpm, the maximum volumetric efficiencies were experienced at 620 g of refrigerant charge, viz. 59.62 and 41.63%, respectively.

- As the compressor speed increases, the exergetic losses in the compressor and TXV increased, while losses in the condenser and evaporators decreased.

- Maximum exergetic efficiencies of 43.32 and 38.15% were experienced at 620 g for both the compressor speeds of 900 and 1800 rpm, respectively.

- When the compressor speed increases, the exergetic efficiency of the system reduced by 11.9% because of the higher EDR that increased by 24%.

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