DESIGN AND TEST OF A SMALL AMMONIA HEAT PUMP

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Abstract: Since synthetic refrigerants may cause environmental damages, by depleting the ozone layer or by contribution in global warming, many researchers, in recent years, have focused on the use of natural refrigerants such as ammonia to replace the synthetic ones. Although ammonia has been used in large refrigeration systems, its application in small units, say a small heat pump, is quite rare.

The work presented in this paper is design and test of a small water-to-water ammonia heat pump, providing about 7 kW heat, sufficient for space heating and tap water heating of a single-family house. To reduce the charge and to overcome the problem of accumulation of oil at the bottom of evaporator, a minichannel heat exchanger is used as evaporator. Oil miscible in ammonia is also used to facilitate the oil return to compressor. A permanent magnet motor together with an inverter is used to run the compressor. Supplying the 7 kW heat, the heat pump runs efficiently with heating capacity to compression work ratio of 5.1 at evaporation and condensation temperatures of -5 °C and 40 °C.

Key Words: ammonia, heat pump, refrigerant charge, minichannel heat exchanger, ground source

1 INTRODUCTION

Heat pump technology is widely used in Sweden with different applications; for example, only in district heating in 2008, approximately 9% of the supplied heat, around 4.5 TWh, was provided by heat pumps (SF 2008). More than 800000 heat pumps were sold in Sweden between the years 1994 and 2008 (Forsén 2009). This number is quite large compared to the number of heat pumps sold in other European countries, and also the share of small heat pumps installed in newly-built single-family houses is much larger than other types of heat pumps (WWF).

Commercially-built heat pumps found in Swedish market used for single-family houses usually have 5kW to 10 kW heating capacity, which typically covers about 90% of annual required heating and 60% of required heating power in the coldest days of the year. Of the about 800 000 heat pumps installed, about half are ground source heat pumps, extracting heat from 100m to 200m boreholes or ground surface soil to the evaporator of the heat pumps by the means of circulation of glycol or ethanol solutions at temperatures close to and below 0 °C. The heat is then transferred to the space heating system, having supply temperature of 35 °C to 50 °C or higher, and sanitary hot water, the temperature of which should exceed 60 °C periodically to thermally eliminate some microorganisms from water (Palm 2008). The most common refrigerant for ground source heat pumps in Sweden is

R407C. After R407C, R404A, R134a, and R290 are mostly used ones. For air-to-air heat pumps R410A is commonly used (Karlsson et al. 2003).

In view of the fact that most of the synthetic refrigerants, in case of leakage or release, are harmful to the environment by contributing in global warming or depleting stratospheric ozone layer, many research projects have been done recently to find alternative refrigerants posing no or negligible threat to the environment (Granryd et al. 2005). Among alternative refrigerants, ammonia, a natural refrigerant with zero Global Warming Potential (GWP) and Ozone Depletion Potential (ODP), can be a sensible choice.

An advantage of ammonia as an alternative for the synthetic refrigerants is that the emission of ammonia into the atmosphere, unlike almost all other refrigerants, is harmless to the environment in terms of its global warming potential and ozone depletion potential, which are both zero. Besides, it is biodegradable with atmospheric lifetime of few days (ASHRAE 2002).

In a comparison between 40 refrigerants to rank them based on their global warming potential (GWP, relative to CO_2 based on 100-year time horizon), ozone depleting potential (ODP, relative to R11), and atmospheric lifetime (ALT, number of years), Restrepo et al. (2008) have found only R1270 (propene, GWP_{R1270}=3, ODP_{R1270}=0, ALT_{R1270}=0.001) and R601 (n-pentane, GWP_{R601}=0, ODP_{R601}=0, ALT_{R601}= 0.01) less harmful than R717 (ammonia, GWP_{R717}=0, ODP_{R717}=0, ALT_{R717}=0.25) to the environment due to their shorter atmospheric lifetime. However, Calm and Hourahan (2001) have mentioned much higher GWP values for R1270 and R601, about 20 and 11 respectively based on 100-year time horizon, and atmospheric lifetime of ammonia is considered only few days in a position document approved by ASHRAE in 2002 and 2006 (ASHRAE 2002).

In addition to direct impact of a refrigerant on global warming, due to its molecular structure when it is released to the atmosphere, it also may indirectly contribute in global warming, considering the energy needed for preparation of the refrigerant and efficiency and energy consumption of the cycle, in which the refrigerant is used. The indirect effect of a refrigerant, related to the energy consumption, has a significant share in total equivalent warming impact; it is, however, not easy to determine since it depends on many local factors such as the efficiency of the cycle, generation method of the consumed electricity, and refrigerant production processes (Calm 2006).

Although ammonia is considered a natural refrigerant since its molecules are found in nature, the ammonia used as refrigerant is produced through industrial processes, which may contribute to global warming (Hwang and Ohadi 1998). Campbell and McCulloch (1998) have estimated the equivalent amount of produced CO_2, assumed to be entirely released to the environment, per kilogram of manufactured ammonia, 2 $kgCO_2/kg$, and have compared to that of R134a, 6 to 9 $kgCO_2/kg$, R22, 3 $kgCO_2/kg$, R12, 3 $kgCO_2/kg$, Isobutane, 0.5 $kgCO_2/kg$, and cyclopentane, 1 $kgCO_2/kg$. However, energy consumption and CO_2 release in production process of refrigerants are insignificant compared to the consumed energy and equivalent CO_2 emissions during the lifetime of a heat pump or refrigeration cycle (Campbell and McCulloch 1998). Therefore, performance of the cycle which depends on the design of the cycle with regard to the working condition and thermodynamic and transport properties of the refrigerant, beside its GWP, is a decisive factor determining the total equivalent warming impact of a system.

Coefficient of performance of heat pump cycle (COP_1) of a cycle, with different refrigerants, modeled in Engineering Equation Solver (EES) software program (Klein 2009) with evaporation temperature of -5 °C, condensation temperature of 40 °C, superheating of 5K, subcooling of 5K, and assumed electrical motor efficiency of 80% is estimated and tabulated in Table 1. In calculating COP_1, power consumption of the auxiliary equipments is not taken into account. The refrigerants in this modeling are ammonia, currently used refrigerants in

domestic heat pumps in Sweden, which are R407C, R404A, R134a, R290, and R410A, and, for comparison, R22. In the model no pressure drop or heat transfer with ambient is taken into account, and volumetric efficiencies and isentropic efficiencies are estimated using the empirical correlations suggested by Granryd et al. (2005).

Table 1 Comparison between few refrigerants in a modeled cycle with evaporation temperature of -5 °C, condensation temperature of 40 °C, superheating of 5K, subcooling of 5K, and assumed electrical motor efficiency of 80%. (Pressure losses and heat transfer to ambient is neglected)

Refrigerant	Evaporation Pressure (kPa)	Condensation Pressure (kPa)	Pressure ratio	Critical temperature (°C)	Critical pressure (kPa)	vaporization latent heat (kJ/kg)	vapor specific heat (1) (kJ/kg.K)	$C_{p,liq}/h_{fg}$ (2) (1/ K)	vapor specific volume (3) (m^3/kg)	Compressor discharge temperature (°C)	heating capacity (kJ/kg)	COP_1 (4)
R22	421.9	1534	3.64	96.1	4989	209	0.725	0.006	0.0567	75.6	205	3.94
R290	406.1	1369	3.37	96.7	4247	382	1.733	0.008	0.1151	55.9	357	3.96
R134a	243.5	1017	4.18	101	4059	202	0.872	0.007	0.0848	63.1	196	3.52
R404A	518.6	1825	3.52	72.2	3735	170	0.941	0.010	0.0393	54.4	147	3.68
R407C	431.2	1628	3.78	86.8	4597	215	0.816	0.008	0.0559	65.4	197	3.73
R410A	678.1	2413	3.56	72.1	4925	229	1.070	0.008	0.0397	70.7	213	3.79
R717	354.9	1555	4.38	132.3	11333	1280	2.595	0.004	0.3548	134.2	1387	3.91

(1) saturated vapor at evaporation temperature
(2) ratio of specific heat of saturated liquid at condensation temperature to latent heat of vaporization at evaporation temperature
(3) superheated vapor at compressor inlet
(4) Power consumption of the auxiliary equipments is not taken into account.

Calculated COP_1 values in Table 1 for R717 (ammonia), R290 (Propane), and R22 are almost the same and higher than that of the rest of the refrigerants for the assumed working condition, which may be taken as an indicative of more efficient performance. However, the estimated values should be interpreted with care as some assumptions and simplifications have been made in modeling. For example, if pressure drops were introduced to the model, the COP_1 of ammonia cycle might become higher than the others since less pressure losses are expected in ammonia cycle due to its low molecular weight (Lorentzen 1988).

Another point to be considered in interpreting the COP_1 values in Table 1 is that the rather low temperature of the discharged refrigerant vapor from compressor is not enough for producing sanitary hot water except for ammonia cycle. It implies that, to produce sanitary hot water, higher condensation pressure is needed for the refrigerants other than ammonia which leads to lower COP_1 values for them than the tabulated ones.

In addition to being benign to the environment, ammonia, as a refrigerant, has favorable properties. Its low molecular weight, 17.03 kg/kmol, results in high vaporization latent heat, low pressure drop, desirable heat transfer properties, and less losses in throttling. With high vaporization latent heat of the refrigerant smaller pipes and fittings can be used (Lorentzen 1988). Furthermore, its small liquid specific heat compared with latent heat of vaporization, the smallest in Table 1, reduces the losses in the expansion process. Besides, it has low cost and is not sensitive to water contamination (Dincer 2003).

In this project a small heat pump with about 7 kW heating capacity at -5 °C and +40 °C evaporation and condensation temperatures is designed and built to work with ammonia as

refrigerant. The heat pump is expected to produce enough heat to keep a single-family house warm in central Sweden and to provide tap hot water for the house.

Since ammonia is flammable and toxic in high concentrations, the heat pump is designed to keep the refrigerant charge low to reduce the risk of fire or poisoning in case of unwanted release of refrigerant to the surroundings. The compact design of the heat pump helps reducing the refrigerant charge. Besides, considering the limited space normally reserved for installation of a heat pump in a house, a 0.6×0.6 m^2 space, the compact design of the heat pump is necessary.

2 METHOD OF ATTACH)

Using Engineering Equation Solver (EES) (Klein 2009) software program, a simplified model of the heat pump is developed to estimate the performance of the heat pump and to predict the working parameters, used in sizing different components and pipes. In order to simulate the real working conditions of a heat pump, installed in a single-family house with bedrock as heat source, a brine circuit with electrical heaters is used as heat source, and a water circuit serves as heat sink.

A scaled three-dimensional model is drawn to design the heat pump and the other components, which are normally placed in the same package in commercially-built heat pumps, compact enough to fit into a 0.6×0.6 m^2 space. Compactness of the heat pump also helps to reduce the refrigerant charge.

The designed heat pump is built in Applied Thermodynamics and Refrigeration Laboratory, Department of Energy Technology, Royal Institute of Technology (KTH), Sweden. The constructed heat pump is used to perform experiments.

3 SYSTEM DESIGN

Figure 1 shows the schematic diagram of the configuration used for laboratory experiments. The ammonia vapor leaving the compressor is cooled in a separate heat exchanger called desuperheater in order that the heat from gas cooling is utilized for heating domestic hot water. The heat taken from condensation in the condenser is to be used in hydronic space heating system as well as preheating tap water; however, in the experiments performed in laboratory the heated water is drained. The water leaving the condenser is expected to be around 40°C, which is almost the design condensation temperature of the heat pump. The water temperature after the desuperheater is to be 60°C. To control the water temperature in the heat sink circuit, two 3-way valves and PID controllers are used to regulate the amount of water passing through the desuperheater and the amount of drained water. Positions of the 3-way valves are determined based on the sensed temperature of water after condenser and after desuperheater. To model borehole heat exchanger, the brine is heated by electrical heaters in the heat source loop. The volumetric flow rates of water and brine are measured by Brunata HGQ1-R0-184/1B0R24 and Brunata HGS5-R0-184/1B0R24.

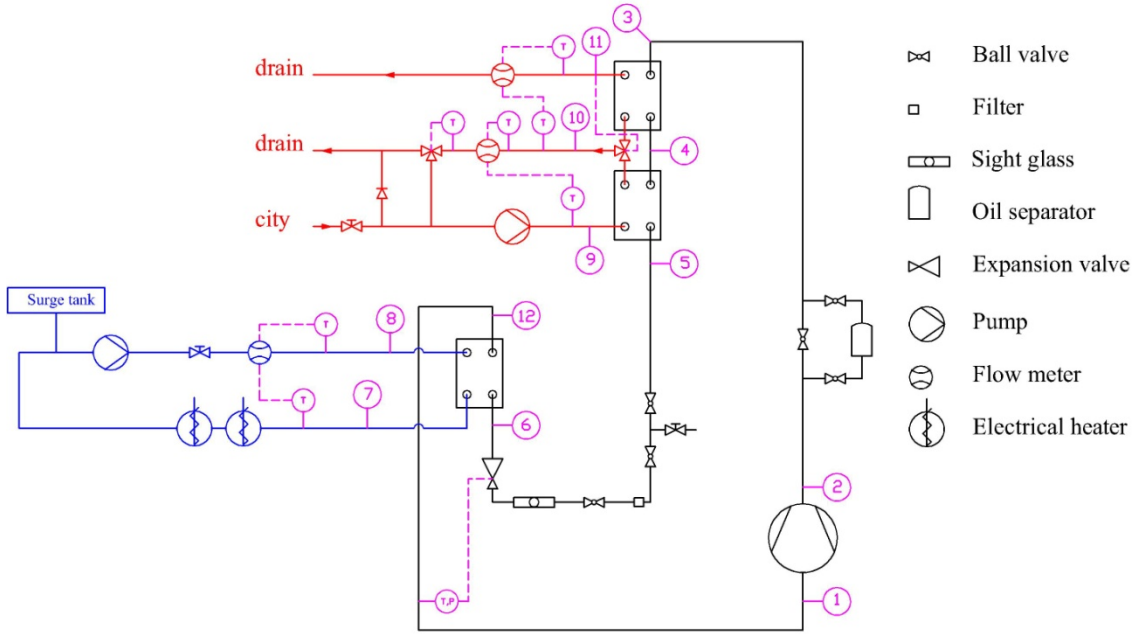


Figure 1 Schematic drawing of the heat pump, heat sink, and heat source (tested in laboratory)

An open-type reciprocating HKT-GOELDNER compressor, O 12 3 DK100, shown in Figure 2, is used to run the heat pump. The compressor is driven by a 6-pole permanent magnet electrical motor, TG-drives TGT6-2200-20-560/T1P. The rotation speed of the motor is controlled by the means of a frequency inverter, OMRON V1000 45P5. The electrical power is measured before the inverter by an active power transducer, Eurotherm E1-3W4. The used oil for compressor, FUCHS RENISO GL 68, is miscible in ammonia.



Figure 2 The compressor of the heat pump, GOELDNER O 12 3 DK100

The desuperheater and the condenser, ALFANOVA27-10H and ALFANOVA52-20H plate heat exchangers, Alfa Laval products, are entirely made of stainless steel. The electronic expansion valve, Carel E2V09BS000, is also compatible with ammonia. To reduce the charge and to overcome the problem of accumulation of oil at the bottom of evaporator, an aluminum minichannel heat exchanger is used as evaporator. All of the pipes, connections, and valves in contact with ammonia are stainless steel. The main criterion of sizing the steel pipes is the minimum flow velocity needed to return the oil to compressor.

A high efficiency variable speed Grundfos pump, CME3-3 A-R-A-E AVBE, is used to circulate the brine, ethanol 25% wt, through the evaporator and the electrical heaters, modeling borehole heat exchanger in laboratory experiments; the power consumption of the pump is measured by the means of a Eurotherm E1-1W0 power transducer. Another high efficiency variable speed pump, WILO Stratos ECO 25/1-5 BMS, is used to circulate water through the desuperheater, the condenser, and the rest of the heat sink circuit.

Signals from the measurement instrument are acquired by Agilent 34970A, Data Acquisition/Switch Units. The data acquisition unit is set up and run by a software program written in Agilent VEE programming language (AT 2010).

Figure 3 is a picture of the completed test facility. For the test reported in this paper, the heat pump is charged with less than 0.2 kg of ammonia.



Figure 3 The completed test facility

4 SOFTWARE MODEL OF THE HEAT PUMP

To predict the performance of the heat pump, a simple software model by the means of EES has been developed. The model is run for the conditions of the test reported in this paper, which are evaporation temperature and condensation temperature of T_{evap}= -5.2°C and T_{cond}= 40°C while the superheating and subcooling are ΔT_{SH}= 3.7K and ΔT_{SC}= 6.5K. The volumetric efficiency, η_s, is assumed 71.2% based on the data provided by the manufacturer of the compressor. The isentropic efficiency of η_k= 71.6% is found from the empirical correlation suggested by Granryd et al. (2005). The pressure drops are not taken into account in the model.

For 1500 rpm rotation speed of the shaft of the compressor, the EES software model predicts swept volume flow of V_s= 0.0026 m^3/s= 9.36 m^3/hr, refrigerant's mass flow rate of m_r = 0.0052 kg/s, heating capacity of Q_1= 7.38 kW, cooling capacity of Q_2= 5.77 kW, and compression work done on the refrigerant of E_k= 1.61 kW. Shares of condenser and

desuperheater from the heating capacity are anticipated to be Q_{cond}= 5.90 kW and Q_{desup}= 1.48 kW respectively. The ratio of heating capacity to compression work is calculated 4.6.

5 TEST RESULTS

After constructing the test facility, the heat pump has been run to detect and fix possible problems. The complete list of measured and calculated values is presented in Table 2.

Energy balance of heat exchangers yields mass flow rate of refrigerant of 0.0050 kg/s. The swept volume flow at rotation speed of 1500 rpm is about 9.36 m^3/hr, so the mass flow rate of refrigerant can be estimated at 0.0052 kg/s, 4% higher than the values given by the energy balance over heat exchangers.

The isentropic efficiency of the compressor, η_k, is calculated from the measured data to be 82.2%, which is 10.6% higher than the value predicted by the EES model. If the mass flow rate value given by energy balance over heat exchangers is used, the volumetric efficiency of the compressor is found 69.3%, which is slightly less than the value estimated based on the data given by the manufacturer of the compressor, 71.2%.

The compression work done on the refrigerant, the cooling capacity, and the heating capacity are calculated E_k= 1.38 kW, Q_2= 5.44 kW, and Q_1= 7.03 kW. By dividing compression work by the electrical power consumption of the compressor η_{tot}, covering the losses in the inverter, electrical motor, and compressor, is found 75.6%. The COP_1, 3.2, is defined as the ratio of the heating capacity to the sum of electrical power consumption of the compressor and the pump in heat source loop. COP_{1k}, 5.1, is defined as the ratio of the heating capacity to the compression work done on the refrigerant.

Table 2 Steady operation of the test facility (indices for temperatures and flow rates are in accordance with Figure 1)

T_1 (°C)	-1.5	**T_{cond} (°C)**	40.0	**η_{tot} (%)**	75.6
T_2 (°C)	127.5	**T_{evap} (°C)**	-5.2	**$LMTD_{desup}$(K)**	13.7
T_3 (°C)	126.6	**V_8 (m^3/hr)**	1.636	**$LMTD_{cond}$ (K)**	1.95
T_4 (1) (°C)	40.3	**V_{10} (m^3/hr)**	0.626	**$LMTD_{evap}$ (K)**	6.43
T_5 (°C)	33.5	**V_{11} (m^3/hr)**	0.054	**UA_{desup} (W/K)**	90.43
T_6 (°C)	-5.2	**Q_{desup} (kW)**	1.242	**UA_{cond} (W/K)**	2972
T_7 (°C)	-0.3	**Q_{cond} (kW)**	5.788	**UA_{evap} (W/K)**	845.7
T_8 (°C)	2.6	**Q_1 (kW)**	7.03	**U_{desup} (3) (W/m^2-K)**	452.2
T_9 (°C)	32.4	**Q_2 (kW)**	5.437	**U_{cond} (3) (W/m^2-K)**	3237
T_{10} (°C)	39.8	**E_{elm} (kW)**	1.827	**U_{evap} (3) (W/m^2-K)**	1057
T_{11} (°C)	59.9	**E_{pump} (kW)**	0.359	**ATD_{cond} (K)**	0.17
T_{12} (°C)	-3.8	**E_k (kW)**	1.38	**ATD_{evap} (K)**	5.1
ΔT_{SC} (K)	6.5	**m_r (2) (kg/s)**	0.005	**ε_{desup}**	0.994
$\Delta T_{SH, int}$ (K)	1.4	**V_s (m^3/s)**	0.0026	**ε_{cond}**	0.936
$\Delta T_{SH, ext}$ (K)	2.3	**η_k (%)**	82.2	**ε_{evap}**	0.375
P_{cond} (bar)	15.5	**η_s (%)**	69.3	**ΔP_{source} (bar)**	1.24
P_{evap} (bar)	3.5	**COP_1**	3.2		
P_{cond}/P_{evap}	4.45	**COP_{1k}**	5.1		

(1) Corrected for enthalpy and pressure at point 4
(2) From energy balance over heat exchangers
(3) Assumed to be constant all over the heat exchanger

The performance of the heat exchangers is also investigated by calculating logarithmic temperature difference (LMTD), overall heat transfer coefficient (U), approach temperature difference (ATD), and effectiveness (ε). The calculated values are included in Table 2.

As indicated by the performance curves of the pump in the heat source loop, for the flow rate of V_8= 1.636 m^3/hr and power consumption of E_{pump}= 0.359 kW the pressure drop in the heat source loop is estimated ΔP_{source}= 1.2 bar.

6 DISCUSSION

With regard to the value calculated for η_k, the compressor works efficiently; nevertheless, the electrical power needed to run the compressor is rather high, affecting the COP_1 adversely. This high power consumption is apparently due to losses in the electrical motor and the frequency inverter.

Another factor that lowers the COP_1 is the electrical power consumption of the pump installed in the heat source loop. When the electrical heaters are replaced by a borehole heat exchanger, the pressure drop will increase significantly leading to higher pump power consumption. To face less pressure drop, a plate heat exchanger can be installed instead of the minichannel heat exchanger. However, this may lead to problems with oil return. Another possibility is to redesign the minichannel heat exchanger to decrease the pressure drop on the water side. Moreover, the series of piping fittings on the water side can be replaced by smooth bends, and larger pipes can be used. To have low pressure losses, it is also helpful to keep the flow rate of the brine low as long as the satisfactory exchange of heat takes place in the borehole heat exchanger and evaporator.

Compared with the results of the EES software model the ratio of heating capacity to compression work done on the refrigerant, COP_{1k}= 5.1, is higher than the predicted value, 4.6. The reason for higher Q_1/E_k can be the higher isentropic efficiency of the compressor than the expected value, which results in lower discharge temperature and enthalpy, and therefore lower compression work. The ratio of the heating capacity to compression work, Q_1/E_k, of the current prototype, 5.1, is higher than that of its pre-prototype reported by Palm (2008), which is less than 4.8 for the reported pressure ratios of 4.17 and higher. The better performance can be a result of employing more efficient compressor and miscible oil.

Considering the high approach temperature difference and the low overall heat transfer coefficient and effectiveness of the evaporator, it is obvious that the performance is not as expected. As the performance of this heat exchanger operating with propane as refrigerant has been proven to be substantially better (Fernando et al. 2004), it seems that the miscible oil has not completely solved the problem of oil return. Replacing the minichannel heat exchanger with a plate heat exchanger is not expected to solve the problem, and would also increases the required refrigerant charge. Thus, there is a need for further experiments on different heat exchangers to reach the best compromise.

If the overall energy balance of the heat pump is investigated, the inward flow of energy into the system is 0.21 kW more than the outward flow of the energy from the system. This difference, about 3% compared to the total inward or outward flows of energy, can be the result of combination of the effect of the errors in measurements and unexpected inward flow of energy to the system. It is only from the expansion valve to the intake of the compressor that the temperature below ambient temperature makes the heat transfer to the system possible. Nevertheless, these areas are insulated well except some parts of the line between evaporator and compressor. The poor insulation over that line is because of limited space around the pipes. However, the heat transfer to the system through that line from the

ambient, which causes external superheat of 2.3 K, is only 0.032 kW. Therefore, the error in the overall energy balance is mostly due to measurement errors.

7 CONCLUSIONS AND FUTURE WORKS

A water-to-water heat pump, as a prototype, has been designed, constructed, and tested. The enough heating capacity, 7kW, and the compact design of the heat pump make it suitable for space heating and tap water heating of single-family houses. For the safety reasons in domestic applications, through compact design and using minichannel heat exchanger as evaporator, the heat pump works with low amount of refrigerant, less than 0.2 kg. Use of efficient mechanical components has resulted in favorable performance of the heat pump with ratio of heating capacity to compression work of 5.1.

The losses in the inverter and motor, affecting COP_1 adversely, are needed to be determined and reduced. Efficiency of the inverter can be found through measurement of its output power. Further improvement of the performance of the heat pump is possible by reducing the pressure drop in the heat source loop and by improving the effectiveness of the evaporator by redesigning the heat exchanger.

In addition to investigating the ways of improving the coefficient of performance of the heat pump, to analyze the test results comprehensively it is necessary to conduct more experiments on the test facility in different working conditions.

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