Accepted Manuscript

Research Paper

Experimental Investigation of a Mechanical Vapour Compression Chiller at Elevated Chilled Water Temperatures

Kyaw Thu, Jayaprakash Saththasivam, Bidyut Baran Saha, Kian Jon Chua, S. Srinivasa Murthy, Kim Choon Ng

PII: S1359-4311(17)32153-1

DOI: <http://dx.doi.org/10.1016/j.applthermaleng.2017.05.091>

Reference: ATE 10403

To appear in: *Applied Thermal Engineering*

Received Date: 31 March 2017

Revised Date: 12 May 2017

Accepted Date: 16 May 2017

Please cite this article as: K. Thu, J. Saththasivam, B.B. Saha, K.J. Chua, S. Srinivasa Murthy, K.C. Ng, Experimental Investigation of a Mechanical Vapour Compression Chiller at Elevated Chilled Water Temperatures, *Applied Thermal Engineering* (2017), doi: <http://dx.doi.org/10.1016/j.applthermaleng.2017.05.091>

This is a PDF file of an unedited manuscript that has been accepted for publication. As a service to our customers we are providing this early version of the manuscript. The manuscript will undergo copyediting, typesetting, and review of the resulting proof before it is published in its final form. Please note that during the production process errors may be discovered which could affect the content, and all legal disclaimers that apply to the journal pertain.

Experimental Investigation of a Mechanical Vapour Compression Chiller at Elevated Chilled Water Temperatures

Kyaw Thu^{1,2,*}, Jayaprakash Saththasivam³, Bidyut Baran Saha^{4,5,*}, Kian Jon Chua², S. Srinivasa Murthy⁶, and Kim Choon Ng⁷

¹ Kyushu University Program for Leading Graduate School, Green Asia Education Center
Interdisciplinary Graduate School of Engineering Sciences, Kyushu University,
Kasuga-koen 6-1, Kasuga-shi, Fukuoka 816-8580, Japan

² Department of Mechanical Engineering, National University of Singapore,
9 Engineering Drive 1, Singapore 117576, Singapore

³ Qatar Environment and Energy Research Institute (QEERI), Qatar Foundation, 5825 Doha, Qatar

⁴ International Institute for Carbon-Neutral Energy Research (WPI-I2CNER),
Kyushu University, 744 Motoooka, Nishi-ku, Fukuoka 819-0395, Japan

⁵ Mechanical Engineering Department, Kyushu University,
744 Motoooka, Nishi-ku, Fukuoka 819-0395, Japan

⁶ Interdisciplinary Center for Energy Research, Indian Institute of Science
Bangalore 560 012, India

⁷ Water Desalination and Reuse Center, King Abdullah University of Science and Technology,
Thuwal 23955-6900, Saudi Arabia

*Corresponding Authors' email: kyaw.thu.813@m.kyushu-u.ac.jp (KT),
saha.baran.bidyut.213@m.kyushu-u.ac.jp (BBS)

Abstract

The performance of a Mechanical Vapour Compression (MVC) chiller is experimentally investigated under operating conditions suitable for sensible cooling. With the emergence of the energy efficient dehumidification systems, it is possible to decouple the latent load from the MVC chillers which can be operated at higher chilled water temperature for handling sensible cooling load. In this article, the performance of the chiller is evaluated at the elevated chilled water outlet temperatures (7 – 17° C) at various coolant temperatures (28 – 32° C) and flow rates ($\Delta T = 4$ and 5° C) for both full- and part-load conditions. Keeping the performance at the AHRI standard as the baseline condition, the efficacy of the chiller in terms of compression ratio, cooling capacity and COP at aforementioned conditions is quantified experimentally. It is observed that for each one-degree Celsius increase in the chilled water temperature, the COP of the chiller improves by about 3.5% whilst the cooling capacity improvement

is about 4%. For operation at 17° C chilled water outlet temperature, the improvements in COP and cooling capacity are between 37 – 40% and 40 – 45%, respectively, compared to the performance at the AHRI standards. The performance of the MVC chiller at the abovementioned operation conditions is mapped on the chiller performance characteristic chart.

Keywords: mechanical vapour compression chiller; refrigerant R134a; elevated chilled water temperature; coefficient of performance

Introduction

From the landmark introduction of the modern air conditioner by Wills Carrier in 1902 [1–3], the mechanical vapour compression (MVC) systems have been the backbone of the heating ventilation and air conditioning, (HVAC) industry especially in building sector. However, significant amount of energy is utilized in the building sector, for instance, in the U.S., 41% of primary energy consumption is accounted for this sector in 2010 whilst air-conditioning for heating and cooling is about 48% of the building energy consumption [4–6]. A vapour compression refrigeration cycle essentially consists of an evaporator, a compressor, a condenser and an expansion device to pump the heat from a low-temperature reservoir to a higher one using electrical work input [7–9]. In air-conditioning application, the space to be cooled is the low-temperature reservoir whilst the ambient is the higher temperature reservoir [10,11]. The refrigerants employed in the MVC systems have evolved from CFCs and HCFCs to environmental-friendly versions such as CO₂, Hydro-Fluoro-Olefin 1234yf etc. [12–15]. With technological advancements, the best available energy consumption figure for the water-cooled chiller is reported to be about 0.47 kW/ton for full-load whilst 0.38 kW/ton for the Integrated Part Load Value or the IPLV [16]. A higher COP can be attained by narrowing the temperature difference between the heat sink and heat source. Conventionally, comfort air conditioning is achieved by the removal of moisture (latent load) and lowering of the temperature of air (sensible load) simultaneously in a common exchanger to achieve suitable supply air conditions. Such a cooling and dehumidification process limits the efficiency of a chiller. The removal of moisture by dew point condensation at the evaporator is the major factor contributing to the power consumption. By decoupling dehumidification load of a chiller using a separate dehumidifier, the MVC system can be operated at elevated chilled water temperature for improved efficiencies [17].

With the emergence of the advanced and efficient dehumidification systems, it is possible to decouple the latent load from the electric-driven air conditioners [18–23]. It is reported that the theoretical COP of an advanced dehumidification system can be from 9 to 17 with corresponding compression ratios of 1.5 to 3 [17,24] whilst the membrane dehumidification system using a sweep gas configuration can provide the dehumidification efficiency more than 200% [25]. Thus, the mechanical vapour compression chillers can be operated at the elevated chilled water outlet temperatures to handle the sensible load only. Operating at the higher evaporating temperatures for the same coolant temperature

reduces the pressure ratio of the compressor. Thus, significant savings in the energy consumption are realized in addition to cooling capacity improvement. The performance of a chiller is normally assessed in accordance with the guidelines by the AHRI standards whilst works on the chiller performance analysis and models are abundantly available in the literature with various types of refrigerants [26–29]. However, these works mainly focussed on the operating conditions suitable for the dehumidification process where a typical chilled water outlet temperature is about 7° C [26]. No work is available on the performance of the MVC chillers at operating conditions only for the sensible load handling i.e., the performance at elevated chilled water temperature. In this paper, the performance of the chiller, in terms of the Coefficient of Performance (COP) and the cooling capacity, is evaluated experimentally at the elevated chilled water outlet temperatures (7 – 17°C) using various coolant temperatures (28 – 32° C) and flow rates ($\Delta T = 4$ and 5° C) for both full- and part-load conditions.

Experimental setup and procedures

A water-cooled MVC chiller with cooling load and temperature rating facilities is constructed for performance evaluation tests. In the rating facilities, the required set points are achieved by mixing water from the cooling tower, the evaporator and the condenser using PID controller for the supply and bypass valves whilst Table 1 lists the operating conditions. The AHRI Standard 551/591 is adopted for the chiller rating where the standard conditions are 7° C and 0.0478 l/(s·kW) for the chilled water and 30° C for cooling water [4]. Table 2 furnishes details on the measuring instruments installed on the test facility. The facility can give the control accuracy of $\pm 0.1^\circ \text{C}$ and $\pm 0.45 \text{ LPM}$. Figure 1 gives the pictorial view of the test facility.

Fig. 1:

A data acquisition protocol was developed using the LabVIEW environment where the acquired data are processed using thermophysical properties of the refrigerant R134a. The steady state conditions were maintained at least 45 minutes for each set point. The superheat and sub-cooling were maintained at $6.5 \pm 0.5^\circ \text{C}$ and $3.5 \pm 0.5^\circ \text{C}$, respectively. The cooling capacity and the heat rejection at the condenser are calculated as,

$$q_{\text{Evap}} = \dot{m}_{\text{Ch}} \times c_p \times (T_{\text{Ch,In}} - T_{\text{Ch,Out}}), \quad (1)$$

$$q_{\text{Cond}} = \dot{m}_{\text{Cw}} \times c_p \times (T_{\text{Cw,Out}} - T_{\text{Cw,In}}), \quad (2)$$

where q_{Evap} is the cooling capacity in kW, $\dot{m}_{Ch}$ is the chilled water flow rate, c_p is the specific heat capacity, $T_{Ch,In}$ and $T_{Ch,Out}$ are the inlet and out temperatures of the chilled water. Similarly, q_{Cond} is the heat rejection at the condenser in kW, $\dot{m}_{Cw}$ is the coolant flow rate to the condenser, $T_{Cw,In}$ and $T_{Cw,Out}$ are the inlet and out temperatures of the coolant. The energy balance of the system is evaluated as,

$$EB = (q_{Evap} + P_{Com} - q_{Cond}) / q_{Cond} \quad (3)$$

Here EB is the energy balance and P_{Com} is the compressor power which is directly acquired using a power meter. The coefficient of performance (COP) of the system is calculated as,

$$COP = \frac{q_{Evap}}{P_{Com}} \quad (4)$$

Results and Discussion

The performance of the MVC chiller is evaluated at both full- and part-load conditions. Tests on the AHRI standard conditions were first carried out as the baseline performance before performing tests at elevated chilled water temperature conditions. Steady state conditions were maintained at least 45 minutes for each set point. Four sets of test conditions i.e., (#1) at $\Delta T = 5^\circ \text{C}$ for the condenser at full-load, (#2) at $\Delta T = 4^\circ \text{C}$ for the condenser at full-load, (#3) at $\Delta T = 5^\circ \text{C}$ for the condenser at part-load (45 Hz for the compressor) and (#4) at $\Delta T = 5^\circ \text{C}$ for the condenser at part-load (40 Hz for the compressor). For each set of test conditions, the temperature of the cooling water to the condenser is varied from 28°C to 32°C . The data analysis and processing is carried out using Mathematica 10.3 and all the data presented here are within 95% confident level. The quality of the data and the merit of the experimental facilities are first evaluated using the energy balance of the system for various operating conditions. Fig. 2 shows the energy balance of the chiller for various cooling loads operating at the aforementioned conditions. It is observed that the energy balance of the system is less than 0.03 showing the stability, consistency and accuracy of the measurements.

Fig. 2.

Figure 3 illustrates the performance of the MVC chiller at elevated chilled water temperatures in terms of variations in the compression ratio, the cooling capacity, the coefficient of performance (COP) and the chiller characteristic map. The COP and cooling capacity for various chilled water and

coolant temperatures are quantitatively presented here. It is noted that both the COP and the cooling capacity linearly increase with the increase in the chilled water outlet temperature whilst lower coolant temperatures yield better performance.

Fig. 3.

Similarly, the performance of the chiller at $\Delta T = 4^\circ \text{C}$ for the condenser coolant at full-load is shown in Fig. 4, at $\Delta T = 5^\circ \text{C}$ for the condenser coolant at part-load (45 Hz for the compressor) is given in Fig. 5 whilst Fig. 6 depicts the performance at $\Delta T = 5^\circ \text{C}$ for the condenser coolant at part-load (40 Hz for the compressor) operation. Here, similar performance improvement trend with the increase in the chilled water outlet temperature is recorded for all the test conditions. It is noted that the COP increase of the system is significantly higher for the part-load operation i.e., (40 Hz compressor) where the COP value of 5.12 is detected for the 17°C chilled water inlet temperature with the reduction in the cooling capacity. Such high COP may be attributed to the reduction in the compression ratio which varies between 2.6 and 3.5. Moreover, the improvement in the specific heat transfer area per unit cooling load for such operating conditions provides further boost for the COP improvement.

Fig. 4

Fig. 5

Fig. 6

As mentioned above, the performance of the MVC chiller generally improves at elevated chilled water temperatures. The improvement in the COP of a typical reversible cycle i.e., Carnot COP for refrigeration system is shown in Fig. 7 where it is noted that the COP can be almost twofold if the chilled water temperature is raised to 17°C from 7°C with slightly non-linear behaviour. Keeping performance at the AHRI standard conditions as baseline conditions, the performance improvement by increasing the chilled water inlet temperature in terms of the COP is given in Fig. 8. It is experimentally verified that the COP of the chiller improves around 3–3.5% for each one-degree Celsius increase in the chilled water temperature. It is observed that this improvement is slightly significant at the part-load conditions.

Fig. 7

From the energy balance or the first law point of view, the increment in the COP of the chiller at elevated chilled water temperature is quite trivial due to the reduction in the compressor work from

the smaller compression ratio. However, higher refrigerant flow rate is required in raising the chilled water temperatures to compensate higher heat transfer processes at elevated chilled water temperatures. Thus, the increase in the cooling capacity is expected at those operation conditions. Fig. 9 depicts the percentage improvement in the cooling capacity of the MVC chiller at elevated chilled water temperatures. It is noted that with each one-degree Celsius increase in the chilled water temperature, the cooling capacity of the chiller improves by about 4%.

Fig. 8

If the latent load of the MVC chiller is decoupled to the dehumidifier, the chiller can be operated at elevated temperatures to handle only the sensible load. By operating the chiller at higher chilled water temperature such as 17° C, the performance of the chiller improves remarkably, i.e., the COP increment is about 37 – 40% whilst the cooling capacity improves by 40 – 45%, most importantly, using the same heat exchanger inventory and hardware.

Fig. 9

Finally, the performance of the MVC chiller at elevated chilled water outlet temperatures for various coolant temperatures, flow rates ($\Delta T = 4$ and 5°C) at both full- and part-load conditions are depicted on the chiller performance characteristics map as shown in the Fig. 10. Significant improvement in the COP of the system is recorded for operations at part-load conditions showing the general improvement trend by increasing the chilled water temperatures.

Fig. 10.

Conclusions

The performance of the mechanical vapour compression (MVC) chiller is evaluated at elevated chilled water inlet temperatures for handling the sensible heat alone. The performance at AHRI standard was first assessed as the baseline condition. It is observed that the chiller performance improves both in terms of the COP and the cooling capacity at elevated chilled water temperatures which can be utilized in handling the sensible load if the latent load is removed by a dehumidifier. It is experimentally verified that for each one-degree Celsius increase in the chilled water temperature, the COP of the chiller improves by about 3.5% whilst the cooling capacity improvement is about 4%. By operating the chiller at higher chilled water temperature such as 17° C, the performance of the chiller improves remarkably, i.e., the COP increase is between 37 – 40% whilst the cooling capacity improves between 40 – 45%, using the same heat exchangers and other hardware. These results

emphasize the superiority of the hybrid air conditioning system with the dehumidifier and the MVC system.

Acknowledgements

The authors gratefully acknowledge the Kyushu University Program for Leading Graduate School, Green Asia Education Center and the National Research Foundation (NRF), Singapore, under the research grant (R-265-000-466-281) for their financial support to conduct this study.

References

- [1] C. Hempstead, W.E. Worthington, *Encyclopedia of 20th-century technology*, Routledge, 2005.
- [2] W.H. Carrier, *Method of humidifying air and controlling the humidity and temperature thereof.*, 1914.
- [3] W.H. Carrier, *Apparatus for treating air.*, 1906.
- [4] B.R. Hughes, H.N. Chaudhry, S.A. Ghani, A review of sustainable cooling technologies in buildings, *Renew. Sustain. Energy Rev.* 15 (2011) 3112–3120. doi:10.1016/j.rser.2011.03.032.
- [5] Heating and cooling no longer majority of U.S. home energy use - Today in Energy - U.S. Energy Information Administration (EIA), (n.d.).
[https://www.eia.gov/todayinenergy/detail.php?id=10271&src=< Consumption Residential Energy Consumption Survey \(RECS\)-f2](https://www.eia.gov/todayinenergy/detail.php?id=10271&src=< Consumption Residential Energy Consumption Survey (RECS)-f2) (accessed March 31, 2017).
- [6] 2011 Buildings Energy Data Book, (2012).
- [7] G. Ding, Recent developments in simulation techniques for vapour-compression refrigeration systems, *Int. J. Refrig.* 30 (2007) 1119–1133. doi:10.1016/j.ijrefrig.2007.02.001.
- [8] N.Q. Minh, N.J. Hewitt, P.C. Eames, N.J. Hewitt, P.C. Eames, Improved Vapour Compression Refrigeration Cycles: Literature Review and Their Application to Heat Pumps, in: *Int. Refrig. Air Cond. Conf.*, Purdue, 2006.
- [9] J. Saththasivam, K.C. Ng, Predictive and diagnostic methods for centrifugal chillers, in: 2008 Winter Meet. New York City, New York Am. Soc. Heating, Refrig. Air-Conditioning Eng. Inc., New York City, NY, 2008: pp. 282–287.
- [10] H.T. Chua, K.C. Ng, J.M. Gordon, Experimental study of the fundamental properties of reciprocating chillers and their relation to thermodynamic modeling and chiller design, *Int. J. Heat Mass Transf.* 39 (1996) 2195–2204. doi:10.1016/0017-9310(95)00335-5.
- [11] J.M. Gordon, K.C. Ng, H.T. Chua, C.K. Lim, How varying condenser coolant flow rate affects chiller performance: thermodynamic modeling and experimental confirmation, *Appl. Therm. Eng.* 20 (2000) 1149–1159. doi:10.1016/s1359-4311(99)00082-4.
- [12] V.W. Bhatkar, V.M. Kriplani, G.K. Awari, Alternative refrigerants in vapour compression refrigeration cycle for sustainable environment: a review of recent research, *Int. J. Environ. Sci. Technol.* 10 (2013) 871–880. doi:10.1007/s13762-013-0202-7.

- [13] M. Beshr, V. Aute, V. Sharma, O. Abdelaziz, B. Fricke, R. Radermacher, A comparative study on the environmental impact of supermarket refrigeration systems using low GWP refrigerants, *Int. J. Refrig.* 56 (2015) 154–164. doi:10.1016/j.ijrefrig.2015.03.025.
- [14] I. Sarbu, A review on substitution strategy of non-ecological refrigerants from vapour compression-based refrigeration, air-conditioning and heat pump systems, *Int. J. Refrig.* 46 (2014) 123–141. doi:10.1016/j.ijrefrig.2014.04.023.
- [15] P.A. Domanski, Evolution of Refrigerant Application, in: *Int. Congr. Refrig.*, Milan, Italy, 1999.
- [16] U.S. Environmental Protection Agency, Update of Guidelines to Include Procurement of Energy Efficient Products Required by the Energy Policy Act of 2005, (2005).
https://www.epa.gov/sites/production/files/2015-09/documents/ae_addendum1_508.pdf (accessed March 31, 2017).
- [17] J. Woods, Membrane processes for heating, ventilation, and air conditioning, *Renew. Sustain. Energy Rev.* 33 (2014) 290–304. doi:10.1016/j.rser.2014.01.092.
- [18] T.D. Bui, Y. Wong, K. Thu, S.J. Oh, M. Kum Ja, K.C. Ng, I. Raisul, K.J. Chua, Effect of hygroscopic materials on water vapor permeation and dehumidification performance of poly(vinyl alcohol) membranes, *J. Appl. Polym. Sci.* 134 (2017) n/a--n/a. doi:10.1002/app.44765.
- [19] T.D. Bui, F. Chen, A. Nida, K.J. Chua, K.C. Ng, Experimental and modeling analysis of membrane-based air dehumidification, *Sep. Purif. Technol.* 144 (2015) 114–122. doi:10.1016/j.seppur.2015.02.019.
- [20] W. Yuan, B. Yang, B. Guo, X. Li, Y. Zuo, W. Hu, A novel environmental control system based on membrane dehumidification, *Chinese J. Aeronaut.* 28 (2015) 712–719. doi:10.1016/j.cja.2015.04.016.
- [21] S.-M. Huang, L.-Z. Zhang, Researches and trends in membrane-based liquid desiccant air dehumidification, *Renew. Sustain. Energy Rev.* 28 (2013) 425–440. doi:10.1016/j.rser.2013.08.005.
- [22] X. Han, X. Zhang, L. Wang, R. Niu, A novel system of the isothermal dehumidification in a room air-conditioner, *Energy Build.* 57 (2013) 14–19. doi:10.1016/j.enbuild.2012.10.039.
- [23] S. Bergero, A. Chiari, Experimental and theoretical analysis of air humidification/dehumidification processes using hydrophobic capillary contactors, *Appl. Therm. Eng.* 21 (2001) 1119–1135. doi:10.1016/S1359-4311(00)00107-1.
- [24] NanoAir - Dais Analytic, (n.d.). <http://www.daisanalytic.com/applications/nanoair/> (accessed May 4, 2017).
- [25] P. Scovazzo, A.J. Scovazzo, Isothermal dehumidification or gas drying using vacuum sweep dehumidification, *Appl. Therm. Eng.* 50 (2013) 225–233. doi:10.1016/j.applthermaleng.2012.05.019.

- [26] 2015 Standard for Performance Rating of Water-chilling and Heat Pump Water-heating Packages Using the Vapor Compression Cycle, (n.d.).
ahrinet.org/App_Content/ahri/files/STANDARDS/AHRI/AHRI_Standard_551-591_SI_2015_with_Errata.pdf (accessed March 31, 2017).
- [27] Schultz Ken, Kujak Steve, TEST REPORT #7, System Drop-In Tests of R134a Alternative Refrigerants (ARM-42a, N- 13a, N-13b, R-1234ze(E), and Opteon™ XP10) in a 230-RT Water-Cooled Water Chiller, (n.d.).
http://www.ahrinet.org/App_Content/ahri/files/RESEARCH/AREP_Final_Reports/AHRI_Low-GWP_AREP-Rpt-007.pdf (accessed March 31, 2017).
- [28] Saththasivam, Predictive and Diagnostic Methods for Vapor Compression Chiller, 2010.
- [29] J.M. Gordon, K.C. Ng, Cool Thermodynamics: The Engineering and Physics of Predictive, Diagnostic and Optimization Methods for Cooling Systems, Cambridge International Science Publishing, 2000.

List of Figures

Fig. 1: The pictorial view of the water-cooled MVC experimental facility.

Fig. 2. Energy balances of the MVC chiller, (a) at $\Delta T = 5^\circ \text{C}$ for the condenser at full-load, (b) at $\Delta T = 4^\circ \text{C}$ for the condenser at full-load, (c) at $\Delta T = 5^\circ \text{C}$ for the condenser at part-load (45 Hz for the compressor) and (d) at $\Delta T = 5^\circ \text{C}$ for the condenser at part-load (40 Hz for the compressor).

Fig. 3. The performance of the MVC chiller at $\Delta T = 5^\circ \text{C}$ for the condenser at full-load and elevated chilled water temperatures (a) Compression ratio, (b) Cooling capacity, (c) COP and (d) Chiller characteristic map.

Fig. 4. The performance of the MVC chiller at $\Delta T = 4^\circ \text{C}$ for the condenser at full-load and elevated chilled water temperatures (a) Compression ratio, (b) Cooling capacity, (c) COP and (d) Chiller characteristic map.

Fig. 5. The performance of the MVC chiller at $\Delta T = 5^\circ \text{C}$ for the condenser at part-load (45 Hz compressor speed) and elevated chilled water temperatures (a) Compression ratio, (b) Cooling capacity, (c) COP and (d) Chiller characteristic map.

Fig. 6. The performance of the MVC chiller at $\Delta T = 5^\circ \text{C}$ for the condenser at part-load (40 Hz compressor speed) and elevated chilled water temperatures (a) Compression ratio, (b) Cooling capacity, (c) COP and (d) Chiller characteristic map.

Fig. 7 Percentage improvement in the COP of the reversible cycle (Carnot COP) for different chilled water temperatures.

Fig. 8. Percentage improvement in the COP of the MVC chiller, (a) at $\Delta T = 5^\circ \text{C}$ for the condenser at full-load, (b) at $\Delta T = 4^\circ \text{C}$ for the condenser at full-load, (c) at $\Delta T = 5^\circ \text{C}$ for the condenser at part-load (45 Hz for the compressor) and (d) at $\Delta T = 5^\circ \text{C}$ for the condenser at part-load (40 Hz for the compressor).

Fig. 9. Percentage improvement in the cooling capacity of the MVC chiller, (a) at $\Delta T = 5^\circ \text{C}$ for the condenser at full-load, (b) at $\Delta T = 4^\circ \text{C}$ for the condenser at full-load, (c) at $\Delta T = 5^\circ \text{C}$ for the condenser at part-load (45 Hz for the compressor) and (d) at $\Delta T = 5^\circ \text{C}$ for the condenser at part-load (40 Hz for the compressor).

Fig. 10. The performance of the MVC chiller at elevated chilled water outlet temperatures for various coolant temperatures, flow rates ($\Delta T = 4, 5^\circ \text{C}$) at both full- and part-load conditions.

List of Tables

Table 1. Summary of test conditions.

Table 2. Details of the components of the MVC test rig.

Table 1. Summary of test conditions.

Parameter	Range
Chilled water outlet temperature ($^\circ \text{C}$)	7.0 – 17.0
Cooling water inlet temperature ($^\circ \text{C}$)	28.0 – 32.0
Load profile	Full load – Part load
Coolant ΔT ($^\circ \text{C}$)	4 and 5

Note: The test standard is according to AHRI Standard 551/591.

Table 2. Details of the components of the MVC test rig.

No.	Name of the component	Specifications
1.	MVC System	Nominal capacity 10.0 kW
2.	Compressor	COPELAND Scroll compressor, model ZR81 KC-TFD-522, with a nominal 6.8 horsepower
3.	Evaporator	SUS304, shell and tube type, 4 passes, refrigerant in tube-side, water in shell-side
4.	Condenser	SUS304, shell and tube type, 4 passes, refrigerant in shell-side, water in tube-side
5.	Expansion device	TXV (EMERSON TXV, model HFES 8HC), EXV (EMERSON PWM EX-2 EXV with EC2-39 Controller)
6.	Refrigerant flow meter	KROHNE, Model 29/RR/M9/es; Range: 36-360 L/hr; Accuracy: 1.6% F.S
7.	Energy meter	HIOKI, Model 3184, measures the electrical power consumption of the whole chiller system, with an accuracy of $\pm 0.01 \text{ kW}$

8.	Variable speed drive	VLT2875, 7.5 kW, 10 HP, 16 A
9.	Pressure transducers	Gems Sensors Pressure Transducer Model:1200BGB1001A3UA/1200BGB2501A3UA; Output : 4-20mA, two wire, Range: 0-10/25 barG, Accuracy: 0.5 % F.S
10.	Temperature sensors	4 wire RTD, PT100, 1/10 DIN
11.	Control valve and actuator	CV216 RGA DN25 TA-MC100, 0,15 or 0,5 VDC, S3-50% ED c/h 1200 EN 60034-1
12.	Water flow meters	Siemens, MAG3100 (sensor), MAG6000 (Transmitter) (0.2% +/- 1 mm/s of rate)

Highlights

- Chiller performance improvement
- Experimental evaluation of the chiller performance at elevated chilled water outlet temperatures
- COP increases 37 – 40% and cooling capacity 40 – 45% using same hardware at 17° C outlet chilled water
- Design data of MVC system for sensible cooling