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Performance evaluation of air-cooled screw chillers at low part load ratios and outdoor temperatures in Dubai and measures to improve the performance

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Abstract

Air-cooled chillers are considered to be intensive energy consumers and account for over one-fourth of the energy consumption in air-conditioned buildings. In Dubai, for about 4-5 months, chillers operate at low part load ratios with a reduced coefficient of performance (COP) due to low outdoor temperatures. This paper evaluates the performance of air-cooled twin-circuit screw chillers serving a medium-scale commercial building in the month of January when the outdoor temperatures are low. The performance of the chillers are analysed in view of calculated parameters and optimal conditions obtained from the readings taken. The chillers are found to be operating poorly at low COPs. Measures to rectify the problems in the performance of the chillers and advanced chiller technologies that are currently under research are discussed.

Keywords: COP behavior, chiller plant control (CPC), condensing temperature control (CTC), variable speed condenser fans (VSF)

1. Introduction

Air-cooled chillers are generally considered to be energy intensive equipment in air-conditioned buildings in places with hot, humid and sub-tropical climates [1]. These chillers can account for over one-fourth of the total electricity consumption of commercial buildings [2]. Therefore, it is of utmost importance to optimize their efficiency or coefficient of performance (COP) in order to reduce the energy consumption and carbon footprint of commercial buildings. Many central cooling systems in air-conditioned buildings have multiple chillers operating in parallel to meet the varying cooling load requirements [3]. The energy performance or COP (cooling output over chiller power input) of chillers depends on the heat rejection medium, ambient conditions, compressor efficiency and the load carried by the chiller [4]. In Dubai, for about 4-5 months, chillers operate at low part load ratios with a reduced COP due to low outdoor temperatures and building cooling load requirements.

Air-cooled chillers are used to produce chilled water to air-side systems such as primary air fan coil systems, constant air volume systems and variable air volume systems in different zones of the building. An air-cooled screw chiller generally comprises a shell and tube evaporator, an air-cooled condenser with constant speed condenser fans and one or more refrigeration circuits in parallel depending on its size. Each refrigeration circuit includes one expansion valve and one or two constant speed twin-screw compressors [5]. Air-cooled screw chillers are operated under head pressure control (HPC) whereby the heat rejection airflow of the condenser is regulated by staging several groups of condenser fans [5]. The number of staged condenser fan groups is kept minimum in most operating conditions to enable the condensing temperature to remain between 45 °C and 50 °C [3]. The cooling capacity of the chiller is controlled by a modulating sliding valve in each compressor. The supply chilled water temperature has to

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be set at 7°C (in summer) and 6°C (in winter) to meet the de-humidifying capacity of the air-side equipment. The chiller will be loaded and un-loaded when the return chilled water temperature varies from the floating point which is 12.5°C. The cooling capacity of air-cooled screw chillers depends on the number of refrigeration circuits and not on the number of compressors in each refrigeration circuit [5]. Even though, it is widely known that the COP of air-cooled chillers is substantially lower than that of water-cooled chillers, they are still a popular choice as they are desirable in hot and arid regions where water resources are scarce or in sub-tropical regions where sea or fresh water is not readily available for comfort cooling [7]. Also, they are suitable for small to medium-scale commercial developments due to the ease of installation, simplicity of operation and lower installation and maintenance costs when compared to water-cooled chillers [8]. In this work relevant readings for performance evaluation are taken at the site of the chiller plant. Parameters and optimal conditions of the existing chillers are calculated based on the readings taken. Finally measures to improve performance of the chillers and advanced chiller technologies currently under research are discussed.

2. Observation and Readings

The chiller plant in consideration served a medium-scale commercial building in Dubai and consisted of three air-cooled twin-circuit screw chillers – Chiller-1 which is the redundant chiller, Chiller-2 which is the lag chiller and Chiller-3 which is the lead chiller. Chiller-3 started first and ran for 6 minutes. Chiller-3 and Chiller-2 then shared the load for 4 minutes. Then, Chiller-2 continued on for 2 minutes. Therefore, the total chiller plant operation per cycle was 12 minutes. Chiller-3 started again after 8 minutes and the cycle continued. Both the chillers had only one and the same refrigeration circuit in operation throughout.

Table 1. Chiller data and specifications (ARI Standard Ratings) [12]

Refrigerant	HFC-134a
Rated supply chilled water temperature	6.7°C
Rated return chilled water temperature	12.2°C
Rated condenser air entering temperature	35°C
Rated cooling capacity	795 kW
Rated compressor power input	285 kW
Rated fan power input	15.6 kW
Rated cooler flow	34.2 l/s
Rated cooler water pressure drop	35.6 kPa
Rated COP	2.64
Number of screw compressors in each circuit	1
Number of condenser fans	8
Rated compressor voltage	400-420V
Rated condenser fan voltage	220-240V

Readings were taken on three consecutive working days in the month of January when the outdoor temperatures were low. Since the property served by the chiller plant is a commercial building, the number of people in the building and other factors that have an impact on the cooling load can be considered to be similar on all the three days.

Readings for Chiller-2 and Chiller-3 were taken at random time intervals but only when the chillers were operating alone and not sharing the building load. The outdoor temperature, chilled water supply and return temperatures were noted from the chiller display monitor. The chilled water inlet and outlet pressures were noted from the pressure gauges on the inlet and outlet water lines. For the compressor, current and voltage were noted on R, Y and B wires using a clamp-meter. For condenser fans, only current on the R, Y and B wires were noted using the clamp-meter. The voltage was assumed to be 230V as it was not possible to use the clamp-meter due to difficult accessibility of the wires. At a given time, about 2-3 minutes were taken to complete a set of readings during which each chiller can be considered to operate at a steady state.

Power input to compressors (in kilowatts):

$$P_C = \left[3V_{ph} I_{ph} \cdot \text{Power Factor} \right] \div 1000 \quad (1)$$

where I_{ph} is the phase current (A), V_{ph} is the phase voltage (V), and Power Factor = 0.95.

Power input to condenser fans (in kilowatts):

$$P_F = \left[IV\sqrt{3} \cdot \text{Power Factor} \right] \div 1000 \quad (2)$$

where I is the current flowing in the line (A), V is the assumed RMS voltage across R, B and Y wires = 230V, and Power Factor = 0.95 .

Total power input to chiller (in kilowatts):

$$P_{total} = P_C + P_F \quad (3)$$

The chilled water mass flow rate, M in kg/s was found using the pressure drop-cooler flow rate graph shown in Fig. 2 as the inlet and outlet pressures were noted while taking the readings at the site. The curve in the graph that was taken into account is 30GX250 [12]. The density of chilled water is taken as 1000 kg/m^3.

Chiller cooling output (in kilowatts):

$$Q = M\ C\left(T_r - T_s \right) \quad (4)$$

where M is the chilled water mass flow rate (kg/s), C is the specific heat capacity of chilled water = 4.193 kJ/kg. KT$_r$, and T_s is the return and supply chilled water temperature respectively ($^\circ$C).

Part load ratio of the chiller:

$$PLR = Q/Q_{total} \quad (5)$$

where Q_{total} is the total cooling capacity of the chiller = 795 kW.

Coefficient of performance of the chiller:

$$COP = Q/P_{total} \quad (6)$$

With the help of my industrial supervisor, optimal conditions of the existing chillers were discussed. It was assumed that the cooling output of each chiller when operating alone varies linearly from 200 kW to 280 kW as the outdoor temperature increased from 20°C to 28.5°C. The supply chilled water temperature was set at 6°C and the average COP in all operating conditions was assumed to be 2.64. The inlet and outlet chilled water pressures were taken to be the same as that of the readings and therefore the chilled water mass flow rate through each chiller was also the same. The return chilled water temperature was calculated using equation (4). The total power input to chiller was calculated using equation (6). The power input to the compressor and condenser fans was calculated by assuming that $P_C = P_{total}/1.1$ and $P_F = P_{total} - P_C$ [1].

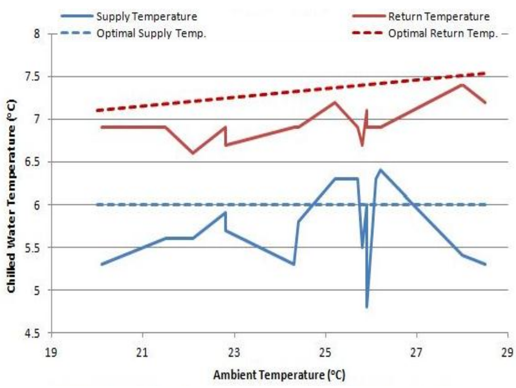

Fig. 1. Water temperature Vs-ambient temp of chiller-2.

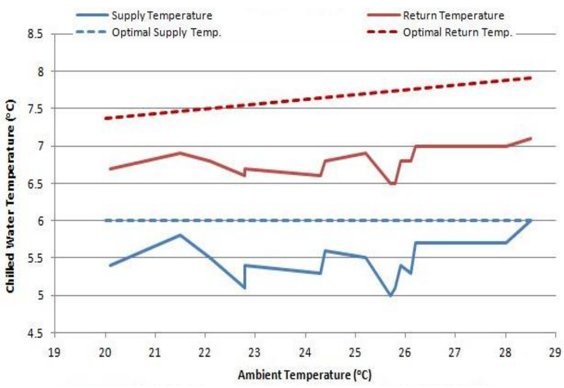

Fig. 2. Water temp Vs ambient temp of chiller-3.

3. Data Analysis and Discussions

In Fig. 1 and Fig. 2, it can be seen that the supply chilled water temperature is varying. In fact, it should be set at a constant temperature of 6°C. Return chilled water temperature can vary depending upon the cooling load in the building. The curve is generally expected to increase as the ambient temperature increases but there are a lot of irregular variations. The optimal behaviour of the parameters is shown in the graphs. The problem can be rectified by calibrating the Chilled Water Temperature set-point controller correctly and then testing it.

In Fig. 3, it can be seen that the cooling output of both the chillers are varying irregularly. Generally, the cooling output increases as the ambient temperature increases. Also, the cooling output of Chiller-3 is lower than that of Chiller-2 when in fact, they should provide nearly the same cooling output. The problem can be rectified by adding or re-filling new refrigerant in the chillers in order to maintain the optimum refrigerant pressure and provide sufficient cooling.

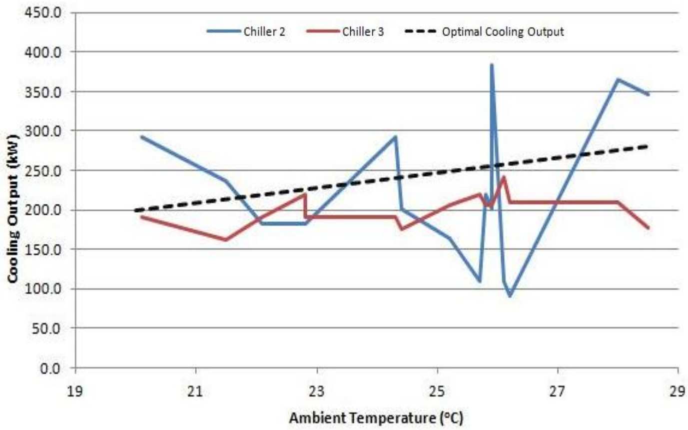

Fig. 3. Cooling output-ambient temperature graph.

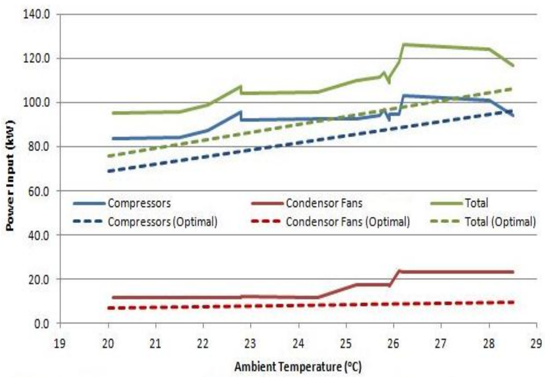

Fig 4. Power input Vs ambient temperature chiller 2.

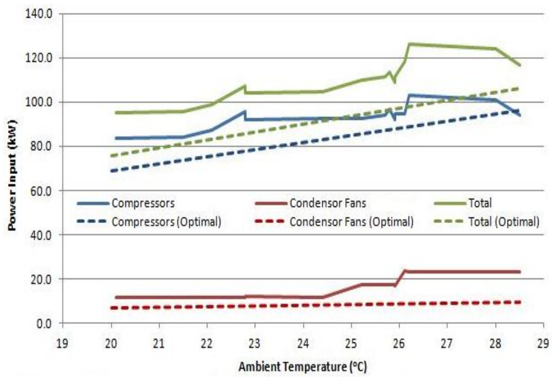

Fig 5. Power input Vs ambient temperature chiller 3.

In Fig. 4 and Fig. 5, it can be seen that the power input to compressor and condenser fans are showing normal behaviour. An increase is seen as the ambient temperature increases, even though the cooling output varies irregularly. Also, for the compressor, condenser fans total power inputs are higher than the respective optimal conditions. The cooling output of Chiller-3 is lower than that of Chiller-2 but its total power input is comparatively higher thereby resulting in lower COP. This may be due to it being the lead for a significantly long time thereby requiring maintenance. The following measures can be taken to rectify the problem:

- The bearings in the compressors and condenser fans should be checked for wear and tear and replaced if they are worn out.
- Sometimes, windings in the compressor motor might get too hot and the residue reacts with the oil in

the motor to form an acidic compound, thereby affecting the compressor and the entire refrigeration circuit [5]. Oil samples from the motor need to be sent to the lab to check for contamination. If the oil is contaminated, the motor windings should be replaced, the refrigeration circuit should be flushed and new refrigerant and oil should be added in the chillers.

- The chiller plant should be efficiently sequenced for winter and low load conditions.
- One chiller can provide the entire cooling output required during the 12 minute cycles for the whole day. The other chiller can provide for the cooling output required the next day and this cycle can be repeated. This will allow the chillers to operate at a higher part load and COP throughout the cycle and avoid unnecessary energy wastage and wear and tear due to the constant switching on-off of the chillers.
- For a particular day that a chiller operates, only one and the same refrigeration circuit should be in operation. The next time the same chiller is in operation, the other refrigeration circuit can provide the cooling output required and this cycle can be repeated. This will allow both the refrigeration circuits to be in operation rather than stressing one refrigeration circuit throughout.

4. Advanced Chiller Technologies

4.1. Chiller plant control (CPC)

CPC is considered to be a sub-system of Building Management System (BMS) and is provided by BMS suppliers. However, these suppliers without the specific heating, ventilation and air conditioning (HVAC) knowledge have great difficulty to provide an appropriate control system for optimizing the performance of the chiller plant. With the rapid development of chiller technology, CPC can now be carried out by stand-alone systems enabling the performance of chiller plants to be optimized based on the particular characteristics of the chillers [5].

CPC can monitor various operating parameters of chillers such as supply and return chilled water temperatures and running amperage of the compressor to name a few. They can identify chillers operating at base, peak and swing, and also allow for soft-start application. They can provide advanced chiller sequencing control thereby enabling the chiller plant to run with the minimum required number of chillers at optimum efficiency. They can provide reliable chiller plant operation by switching over to a standby chiller automatically to replace a problem chiller. They are also a useful analytical tool for operators to identify any deteriorated component of chillers through the provision of regular diagnostic reports. If the identified deteriorated component can be serviced or replaced prior to chiller failure, the reliability of the chiller plant can be increased. Provision and installation of CPC can cost around US$65,000 which is reasonable considering the life-cycle of 10-15 years of a chiller plant [5]. CPC is available in the market for the existing chiller plant at the site.

4.2. Condensing temperature control (CTC)

Currently, air-cooled chillers are operated under Head Pressure Control (HPC) whereby the heat rejection airflow of the condenser is regulated by staging condenser fan groups. The number of staged condenser fan groups is kept at a minimum to enable the condensing temperature to float between 45^{o}C and 50^{o}C in order to maintain the minimum evaporator and condenser pressure differential at 60-80 psi to ensure sufficient lubrication to the compressors [5]. Under HPC, the COP of air-cooled chillers reduces considerably at low part load ratios and outdoor temperatures [7].

The COP of air-cooled chillers can be increased by raising the evaporator temperature or reducing the condensing temperature. However, when the supply chilled water temperature is set at 6°C and 7°C for winter and summer conditions respectively, the potential for raising the evaporator temperature is limited as it varies within a narrow interval of 3-5^{o}C over the entire range of chiller load conditions [5].

HPC can be replaced with CTC whereby the compressor electric demand reduces considerably by allowing the condensing temperature to float closely above any given outdoor temperature by staging maximum number of condenser fans in order to provide the required heat rejection airflow. The considerable reduction in compressor electric demand always exceeds the resulting increase in condenser fan electric demand [7]. Under CTC, the chiller COP could increase by 2.3-115.4% depending on the load conditions and outdoor temperatures. At outdoor temperatures as low as 15 °C, chillers can operate at a COP of above 5.0 at full-load [9].

CTC is still under research as chiller manufacturers experience difficulties in maintaining a suitable evaporator and condenser pressure differential under this technology. The development of magnetic bearing might be able to solve this problem. Also, thermostatic expansion valves have to be replaced with electronic expansion valves [5].

4.3. CTC with variable speed condenser fans (VSF)

When variable speed control is applied to condenser fans, each of them can operate at lower speed with much reduced electric demand while maintaining the condensing temperature at its set-point. This further improves the COP than when the chiller operates under CTC alone [7].

To successfully implement CTC with the use of variable speed condenser fans, the optimum set-point condensing temperature should be determined based on the chiller part load ratio together with the outdoor temperature rather than on the outdoor temperature alone. This enables the COP of chillers to increase by 4.0-127.5% depending on the load conditions and outdoor temperatures [7].

4.4. Observations in present case

The expected COP behaviors in Fig. 6, Fig. 7 and Fig. 8 were plotted using the part load performance curves of the chiller model in [7] and [9]. It was mentioned in [3], which uses the same chiller model in [7] and [9] that these part load performance curves are generally applicable for air-cooled screw chillers rated at 703-1406 kW with a similar configuration. As the chillers present at the site are rated at 795 kW and have a similar configuration as the modeled chiller, these part load performance curves were referred.

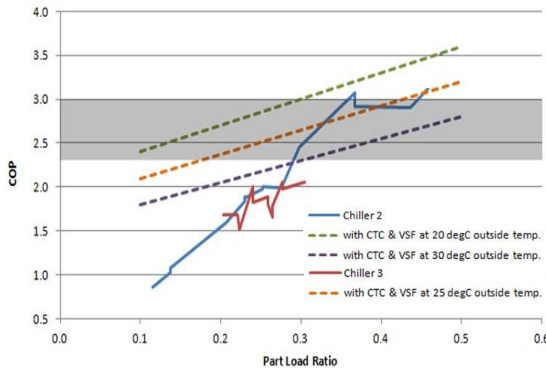

Fig. 6. COP behaviour with CTC.

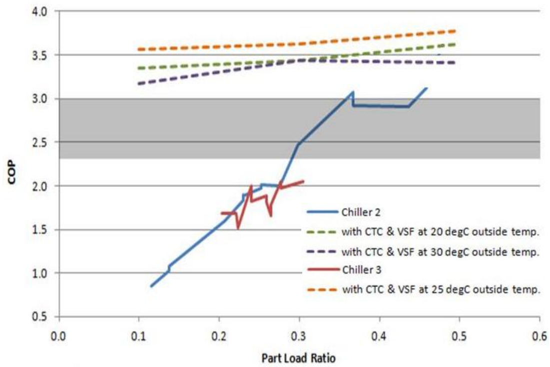

Fig. 7. COP behaviour with CTC and VSF.

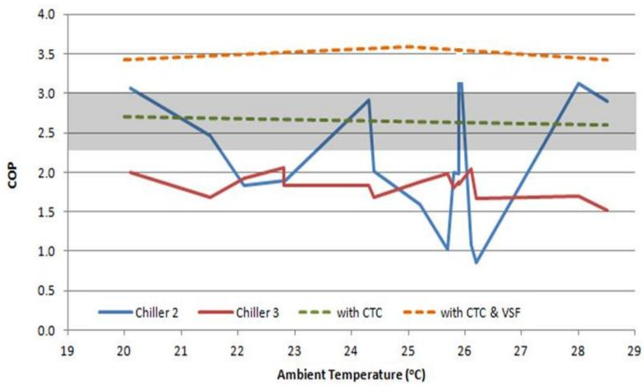

Fig. 8. Comparison of both above behaviours.

In Fig. 6 and Fig. 7, it can be seen that, at low part load ratios and outdoor temperatures, significant improvements in COP can be achieved if CTC alone or CTC with VSF is implemented to the existing chillers. In Fig. 8, it can be seen that the application of CTC with VSF can further improve the COP in a comparison.

5. Conclusions

Relevant readings were taken at the site of the chiller plant and important parameters were calculated based on these readings. Optimal conditions of the existing chillers were calculated and discussed. The performances of the chillers were discussed by analyzing the readings taken, calculated parameters and optimal conditions. The chillers were found to be operating poorly with low COPs and measures to rectify the problems in the performance of the chillers were discussed. Also, advanced chiller technologies such as Chiller Plant Control (CPC), Condensing Temperature Control (CTC) and Variable Speed Condenser Fans (VSF) were discussed. Expected COP behaviors due to the implementation of CTC and CTC with VSF to the existing chillers at the site were studied. It was found that the COP of the chillers increases significantly at all operating conditions due to the implementation of these technologies thereby indicating the importance of research being carried out in these areas.

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