Design and Development of Performance Rating Apparatus for Cold Room Refrigeration Unit Using Cooling-Load Balance Method

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Abstract. Refrigeration systems are composed of many components working together, and their refrigerating capacity depends upon these components constituting the refrigeration system. The design of the refrigeration system does not only consider cooling loads from products but also the operating conditions that change from time to time, especially the ambient temperatures due to global warming and climate change. Therefore, this work aims to develop the apparatus for performance rating of the cold room's refrigeration unit according to the ASHRAE standard test methods as well as the Eurovent certification programs. The principle of enthalpy change is used to determine the system performance, in general. In this work, the cooling-load balanced method is proposed, which shows a good refrigeration capacity estimation by balancing the cooling load with refrigeration capacity. In addition, the use of an ultrasonic flow meter for refrigerant flow rate measurement is also discussed for additional validation in future works.

Keyword. ASHRAE, Cooling-load balance, Cold room, Eurovent, HVAC, Ultrasonic flow meter

1 Introduction

Refrigeration is essentially a part of many supply chains as it is used in preservation of products; for example, perishable products like food and pharmaceutical products, which require low temperature to slow down the rate of spoiling and oxidizing. In the distribution process of these products from manufacturers until the last mile delivery, a cold room is a facility used to preserve these products. As different products require different storage temperatures, this facility is usually designed for specific products or compromised between a group of products.

The design of a cold room, in terms of refrigeration capacity, should consider the cooling load handled by the cold room, which is a function of the product to be stored, the environmental condition, etc. Since global warming and climate change affect the design parameters for cold room refrigeration unit selection, some design criteria used in practice are no longer valid. The refrigeration unit is composed of many components and the resulting refrigeration capacity is derived from them. At present, manufacturers of these components propose a selection software to mitigate the complexity of matching up these components. However, to avoid insufficient refrigeration capacity, many designers overdesign the system resulting in unnecessary first costs as well as an increase in operating costs of up to 50% due to higher energy consumption. The increase in unit energy consumption is primarily due to the power surges to turn the unit on or off, which can happen more often when the unit has greater refrigeration capacity than required. This compressor short-cycling problem also reduces the unit lifespan as it causes wear and tear on many components of the unit, i.e., more maintenance cost is expected.

The performance of the cold room has been studied in different aspects by many researchers, including the uniformity of temperature throughout the cold room [1–3], and heat and moisture transfers between the products and cold air inside the cold room [4–6]. The results from these studies reveal that the products had excessive weight loss, reduced shelf life or product quality. The temperature-time history within a cold room has been predicted by mathematical models [7] Furthermore, the CFD models are used to study the temperature and velocity distributions in the chamber [8].

Besides designing the refrigeration system to obtain the desired system capacity, selection of the components making up the refrigeration system is another issue. Some manufacturers overclaim their product performance, such as the heating or cooling coils. The small-size coils are claimed to have high capacity than the larger coils. This results in a competitive cost, but their performance remains unprovable.

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The design of this type of facility should then be verified by means of experiments to assure the system performance. In this work, the design of the apparatus used to determine the refrigeration unit performance is proposed, and the standard test methods of ASHRAE 33 and the European standard EN 328 used in the Eurovent certification program for coil performance rating are used as guidelines to ensure the measurement result reliability. The proposed method provides another alternative to determine the unit performance with lower-cost instruments.

2 Refrigeration Capacity Determination

The apparatus developed in this work is designed using the measurement principle proposed in the standard test methods for coil performance rating, where some instructions are followed, e.g., instrument accuracy, mounting, calculation method. The standards used mainly in this work are 1) ANISI/ASHRAE 33 – Methods of Testing Forced-Circulation Air-Cooling and Air-Heating Coils [9], 2) BS EN 328 Heat Exchanger – Forced Convection Unit Air Coolers for Refrigeration – Test Procedures for Establishing Performance [10]. There are other standards used as complements in measurements [11,12]. The refrigeration capacity from the refrigerant side according to the standard test method is determined from the product of the refrigerant mass flow rate and the difference in specific enthalpy between coil inlet and let, as follows:

$$Q = m \times (h_i - h_o) \tag{1}$$

where Q is the refrigeration capacity in kW, m is the refrigerant mass flow rate in kg/s, and h_i and h_o are the refrigerant specific enthalpy at the coil inlet and outlet, respectively.

The enthalpy of the refrigerant entering the coil h_i used in the calculation of the refrigeration capacity is the refrigerant enthalpy at the evaporator inlet, which is equal to the enthalpy at the condenser exit as in the ideal vapor compression cycle.

By following the standard test method, the instruments for measuring temperature, pressure, and refrigerant flow rate are required to assess the system capacity. On the other hand, the proposed method requires mainly the instrument used in the determination of the test chamber cooling load, i.e., the temperature sensor. Therefore, the cost of the instrument is greatly reduced.

2.1 Cooling load simulation

The cooling-load balance method proposed in this work is to balance the cooling load handled by the refrigeration system with the system refrigeration capacity. Once the thermal steady operation is established, the system capacity is equal to the cooling load handled by the system. This way, the additional cooling load is generated by heaters on top of the cooling load of an empty test chamber.

Therefore, the cooling load of this empty cold room is first estimated, then subtracted from the system refrigeration capacity to determine the additional cooling load generated by heaters. The parameters used to estimate the cooling load of this empty cold room are shown in Table 1.

Table 1. Parameters for empty cold room cooling load estimation.

Inside room dimension (width x length x height, m)	3.35 × 4.58 × 2.58
PU panel thickness (m)	0.1
PU thermal conductivity (W/m•K)	0.025
Room temperature (ºC)	-18
Ambient air temperature (ºC)	25
Air specific heat (J/kg•K)	1005
Heat from unit cooler fans (W)	293.33

Note that the parameters in Table 1 were chosen to determine the maximum heater capacity. Thus, the ambient air temperature as low as 25 ºC was used. The heat from unit cooler fans was calculated from 2 160-W fans operating 22 h a day. Note also that only transmission and equipment loads were taken into account since there were no loads from air change and products as the room was empty and kept closed during the test. The transmission load to the empty chamber can be calculated using the Fourier's law, as follows.

$$Q_{tran} = kA_s(T_{amb} - T_{room})/L \tag{2}$$

where Q_{tran} is the heat conduction through the chamber walls, k is the thermal conductivity of the wall panel, A_s is the cross-sectional area perpendicular to the heat conduction direction, L is the chamber wall thickness, and T_{amb} and T_{room} are the ambient and room temperatures, respectively. The room and ambient temperatures used in transmission load calculation can be measured in real-time in order to monitor the current cooling load.

The resulting cooling load of this empty test chamber was 1.29 kW. From the system refrigeration capacity of 5.12 kW, the additional cooling to be generated by heaters was 3.63 kW. In this work, 3 1.5-kW heaters were used as they were in stock of the industrial sponsor. Consequently, 4 cooling load configurations are available on this apparatus, as shown in Table 2.

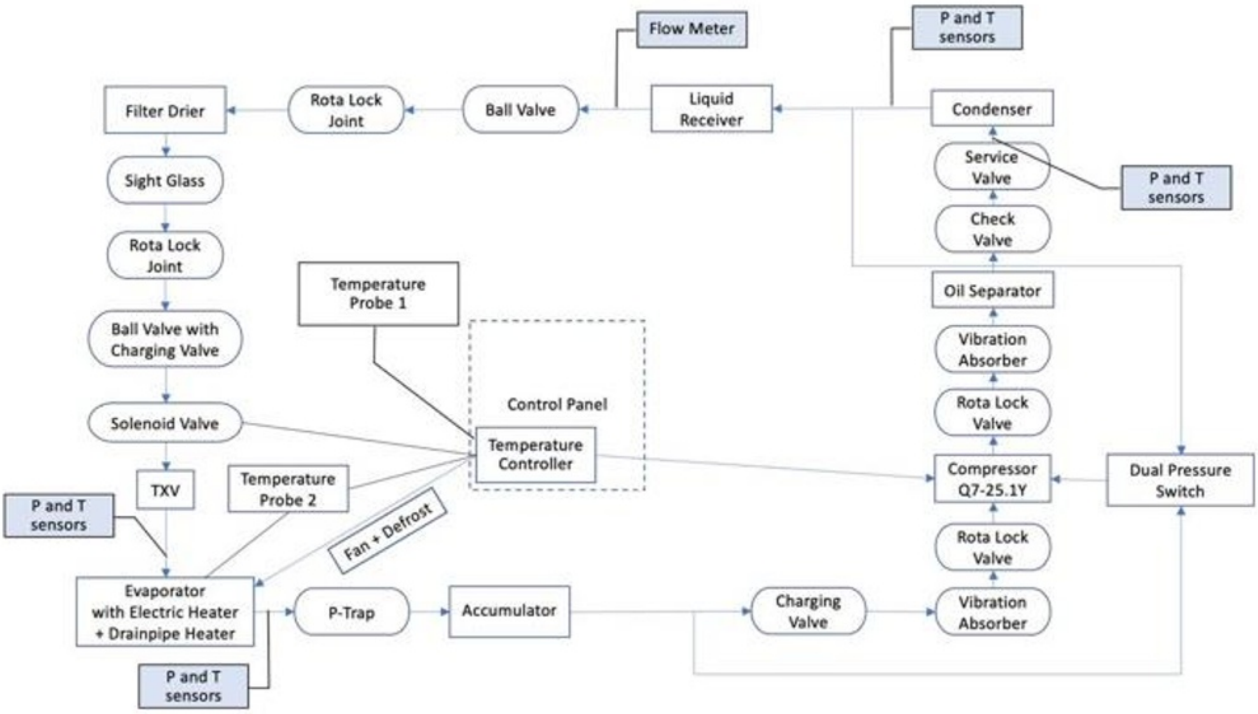

Fig. 1. Refrigeration system diagram.

Table 2. Cooling load configurations.

Configuration	Number of active heaters	Cooling load (kW)
1	0	1.29
2	1	2.79
3	2	4.29
4	3	5.79

2.2 Test chamber

The test chamber used in this work is developed in the cold room with the dimension of 3.2 × 4.8 × 2.4 m. The walls of this chamber are 4-inch PU panels, which are suitable for applications with low temperatures up to -20 ºC.

Fig. 2. Test chamber constituted from 4-inch PU panels.

2.3 Refrigeration unit

The refrigeration system using R404a has been designed, and the room temperature under test is -18 ºC, where the evaporating and condensing temperatures are -25 ºC and 48 ºC, respectively. The compressor used in this system is the Frascold Q7-25.1Y, which provides 5.12 kW of refrigeration capacity at the previously mentioned operating condition as estimated by the Frascold Selection Software.

In order to monitor the state of the refrigerant, additional instruments were installed in the refrigerant circuit, as shown by the blue balloons in the system diagram in Figure 1.

2.4 Instruments and measurements

The calculation of refrigeration capacity according to the standard test method is calculated as shown in equation (1) as a function of refrigerant mass flow rate and refrigerant enthalpies. These quantities are measured as instructed in the standards [9,10], where the flow rate is directly measured by an instrument and the refrigerant enthalpy is determined from the measured temperatures and pressures. Although the proposed method does not require the pressure and flow meter instruments, these instruments were installed in this apparatus for more validation in future works. The details of each measurement are covered in the following subsections.

2.4.1 Temperature measurement

The temperature sensors used in measurements must be in direct contact with the refrigerant and have a minimum accuracy of ±0.6 ºC. The guidelines for sensor calibration and consistency check must be followed as stated in [9,11].

The temperature measuring sensor used in this work is the ifm TT9281 with the accuracy of ±0.15 ºC installed with the thermowell E37603, as shown in Figure 2. There were 4 temperature sensors in total installed in the positions as shown in Figure 3; at the condenser inlet and exit and the evaporator inlet and exit. The connecting pipe between the sensors and refrigerant line has no size requirements and is the same size as the refrigerant line pipeline was used. The refrigerant lines shall be insulated adjacent to the temperature sensors. Moreover, before any test run, all temperature sensors should be checked for equivalent readings at static conditions. In the case where equivalent reading cannot be attained, the reading positions should be arranged to allow a single sensor to be interchanged between the inlet and outlet positions.

2.4.2 Pressure measurement

The pressure sensors must be in direct contact with the refrigerant and have a minimum accuracy of ±0.1 ºC equivalent to a change in saturation pressure. The guidelines for sensor calibration and consistency check must be followed as stated in [9,12].

The pressure measuring sensors used in this work is the Gems Sensors 3100 series with an accuracy of ±0.25%, as illustrated in Figure 4. There were 4 pressure sensors installed in the positions as shown in Figure 3; at the condenser inlet and exit and at the evaporator inlet and exit. The connecting pipe between the sensors and the refrigerant line has no size requirements and is the same size as the refrigerant line pipeline was used.

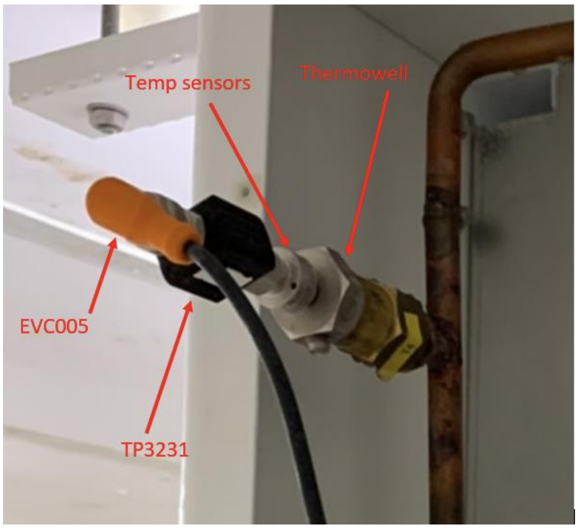

Fig. 3. Temperature measurements using ifm TT9281 coupled with thermowell E37603, transducer TP3231, and cable EVC005.

2.4.3 Refrigerant flow rate measurement

According to the ASHRAE 33 [9], there are 3 methods used to measure the flow rate of refrigerant, i.e., 1) liquid flow meter with an accuracy of ±1% of the quantity measured, such as colioris mass flow meter or similar, 2) water condenser method, and 3) calibrated compressor method.

Fig. 4. The Gems Sensors 3100 series are used to measure refrigerant pressure intrusively.

The flow meter position should be between a liquid receiver and a refrigerant control device. Sight glasses at the upstream and downstream of the flow meter should be used to observe the state of refrigerant for any undesirable flashing of liquid refrigerant into vapor. If the vapor exists downstream of the flow meter due to the pressure drops across measurement instrument, a subcooler ahead of the flow meter should be used. The liquid line subcooling entering the flow meter should not be less than 2.8 ºC. Moreover, the standard also suggests measuring the temperature of the liquid refrigerant at the flow meter exit for the degree of subcooling and the oil separator at the compressor discharge to return the oil to the crankcase.

The use of liquid flow meter seems practical for industrial application since it is ready to use after appropriate installation. However, the instrument types proposed by ASHRAE are expensive. The more cost-effective approach using an ultrasonic flow meter for refrigerant flow rate measurement will be discussed later in future works.

2.4.4 Data logging and monitoring system

The software interface developed on Matlab is used to record data from measurement instruments, where the National Instrument USB-6211 was used as a multifunction DAQ device, which provides sufficient settling times at high scanning rates. It also offers high-speed streaming of data across USB. The measured data from all sensors are shown on the software interface and used in post-processing for enthalpy calculation, real-time cooling load, etc. The refrigeration system control box with indicator lights for active devices, the load heater switch, and the emergency stop button are shown in Figure 5.

The software interface developed in this work was intentionally focused on the refrigeration capacity determination. However, the apparatus was equipped with instruments for measuring temperature, pressure, humidity, and flow rate of the refrigerant. Thus, various indexes, such as time-based rate of energy consumption, compressor running time, deviation of the system P-h diagram from the design point, and so on, can be shown for system enhancement benefits.

The data recording system box, and the software interface are as shown in Figures 6, and 7, respectively.

Fig. 5. The refrigeration system control box.

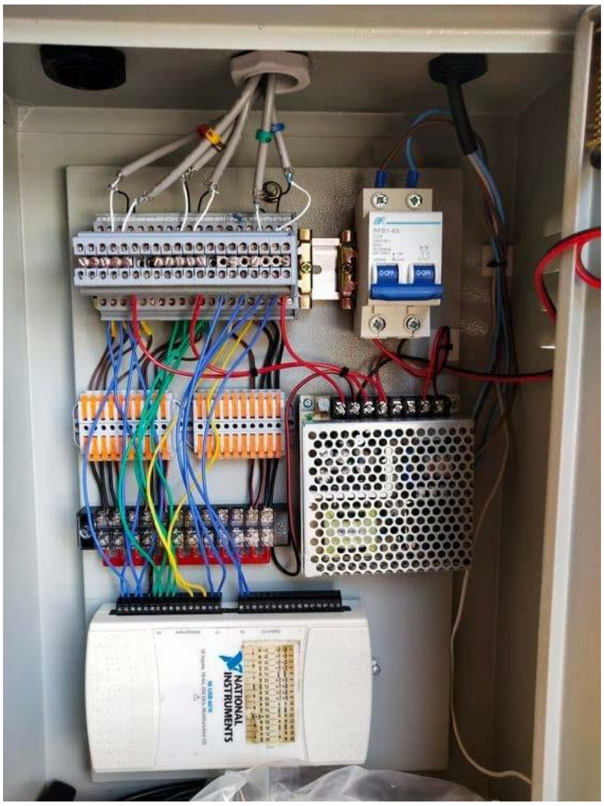

Fig. 6. The data recording system with NI USB-6211.

2.5 Refrigerant enthalpy determination

The thermophysical properties of volatile refrigerants can be obtained from ASHRAE Handbook-Fundamentals [13], REF PROP database from NIST, or the refrigerant manufacturer. However, in the case where these properties are monitored in real time, they should be calculated from mathematical models as a function of the measured temperatures and pressures.

Fig. 7. The software interface developed on Matlab.

Many mathematical models for determining thermophysical properties of refrigerants have been developed and published, but the one that is used by the REF PROP software from NIST is the model proposed by Lemmon *et. al.* [14–17], which works well for the refrigerant mixture used in this work.

The property calculation using the mentioned models could be time and resource consuming due to the equation complexity, and could not be appropriate for real-time monitoring, where many properties need to be recalculated. The remedy for this problem is to pre-calculate the properties for specific working temperatures and pressures and use linear regression from these pre-calculated values.

3 Results and discussions

After the refrigeration system has been charged by R404a, the refrigeration unit was turned on for a day in order to remove heat from the air inside the test chamber as well as heat from the chamber itself. Once the cold room was ready, the cooling loads have been applied to the system starting with the configuration 3 in Table 2. In this case, the cooling load was estimated to be 4.29 kW, which is less than the system refrigeration capacity of 5.12 kW. The system was cycling between on and off cycles as the room temperature varied between the cut in and cut off temperatures. At this cooling load condition, the results showed that the system refrigeration capacity is greater than 4.29 kW.

Then, the configuration 4 for cooling load of 5.79 kW was applied. This time, the system was overloaded, and could not reach the setpoint temperature of -18 ºC. Moreover, the room temperature kept rising. From this cooling load configuration, the system refrigeration capacity is less than 5.79 kW.

From the obtained results, we can conclude that the system refrigeration capacity lies between 4.29 kW and

5.79 kW, which is in good agreement with the value predicted by the software from the compressor manufacturer of 5.12 kW. The difference between these upper and lower limits is too large, and the results can be refined by reducing the cooling load increment. For the cooling-load balance method proposed in this work to determine the refrigeration capacity, the relevant error from this method could be originated from the empty cold room cooling load estimation due to the actual thermal conductivity of the chamber panels, which is able to degrade over time, and the estimation of transmission load from the floor of the test chamber.

In the future works, the adjustable load heater can be coupled with the temperature controller to establish the thermal steady operation. Furthermore, the cooling load of the empty cold room could be improved by calculating the real time transmission load from the measured room and ambient temperatures. The outer surface temperature of the test chamber floor can be measured to estimate more reasonable transmission load from the floor.

To measure the refrigeration capacity following the standard test method, the transit-time ultrasonic flow meter can be used to measure the refrigerant volume flow rate. The commercially available ultrasonic flow meters are easy to use and not an intrusive measurement. Despite its sufficient accuracy of about ±1%, the measurement accuracy could be lower if the flow is biphasic, as in the case of refrigerant flow. Thus, the measurement positions should be where the refrigerant is subcooled liquid or superheated vapor, for instance, at the condenser exit or the compressor exit. However, for the refrigerant flow, the speed of sound could be low, and the flow usually occurs in a small pipe. This could lead to an error on commercial ultrasonic flow meters, in which the traveling distance of the ultrasonic wave is pre-calculated and cannot be adjusted by the user. This leads to the wrong results and needs some post-processing for correction, which will be studied in the future work.

The use of an ultrasonic flow meter can also be used to determine the actual vapor quality of the refrigerant entering the evaporator, since it is the volume flow rate measurement. In conjunction with a mass flow meter, e.g., a colioris flow meter, the vapor quality of refrigerant can be calculated from the ratio of the volume flow rate to the mass flow rate, which is the specific volume of the liquid-vapor refrigerant mixture [18]. This information about the vapor quality at the evaporator inlet can be used to determine more accurate enthalpy of the refrigerant entering the evaporator, which will result in more accurate refrigeration capacity determination.

4 Conclusions

The cooling-load balance method has been proposed in this work, where the cooling load was additionally generated by heaters inside the test chamber to match the refrigeration capacity of the system under test. Once the thermal steady operation is established, the refrigeration capacity of the system is equal to the heat generated by heaters plus the cooling load of the empty test chamber. The cooling load generated in this work was incrementally adjusted, and thus the resulting refrigeration capacity was obtained in the form of a range, which is in agreement with the value predicted by the software from the compressor manufacturer. The result obtained from the proposed method can be refined by using the adjustable load heater in conjunction with a temperature controller to establish the thermal steady operation as proposed in the previous section. From this study, the proposed method showed a good potential in refrigeration capacity determination with a lower cost of measuring instruments compared with instruments suggested by ASHRAE and EN standard test methods. In addition, the use of an ultrasonic flow meter in the refrigeration capacity determination is also reviewed and compared with the refrigerant flow rate measurement requirements of the standard test method.

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