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Practical Control Scheme of the Variable Speed Refrigeration System

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Abstract

With the improvement of standard of manufacturing process, oil cooling unit for manufacturing machine has been developed. A control system must be designed in order to keep oil temperature of the machine within a very restricted range and also to reduce energy consumption. In order to get the low deviation of the controlled temperature and the low efficiency, the on/off control scheme is gradually being replaced by a variable speed refrigeration system (VSRS) with an inverter driven compressor over recent decades.

This paper gives the flowchart to control the compressor speed and also the electronic expansion valve (EEV) aperture in oil cooling unit refrigeration system using R22 as the refrigerant. This control scheme has already tested in experiment apparatus with room temperature condition constant at 25°C and variable load condition at 4kW, 6kW, 7kW, 8kW and 10kW.

Key words: Variable speed, Control system, refrigeration system.

1. Introduction

The control variables of VSRS are mainly focused on the superheat and the refrigeration capacity. The control method of superheat is adjusting the opening angle of EEV to maintain the superheat at a constant level. And the method of refrigeration capacity control is consisted in varying the compressor speed to continuously match the compressor refrigeration capacity to the thermal load [2~3], In fact, there exist several drawbacks in case of traditionally designed controllers for such a system. It is known that a basic refrigeration cycle consists of a compressor, heat exchangers, and expansions valve. In fact, since all such components in the cycle are connected with various pipes and valves, they show inherent nonlinear characteristics in operational ranges. Hence, it is almost impossible to identify exact dynamic characteristics and develop complete dynamic model for the refrigeration systems. In the VSRS, not only the chamber temperature but also the superheat is changed with the variation of compressor frequency. It is the same as the case of the variation of EEV's opening angle. Hence, many studies focused on controlling the superheat or capacity and little of them studied up controlling both of them at the same time. It is noted that in the VSRS, the capacity and superheat can not be controlled independently because of interfering loops when the compressor speed and electronic expansion valve opening angle are varied simultaneously. This study is focused on couple control of refrigerant capacity and EEV on an oil cooling unit with R22 as the refrigerant.

2. Refrigeration cycle

Figure 1 shows a simple vapor compression refrigeration cycle on T-s diagram for different compression processes. The coefficient of performance is given by Eq. 1. The cooling effect and work of compression are formulated using Eq. 2 and Eq. 3 respectively.

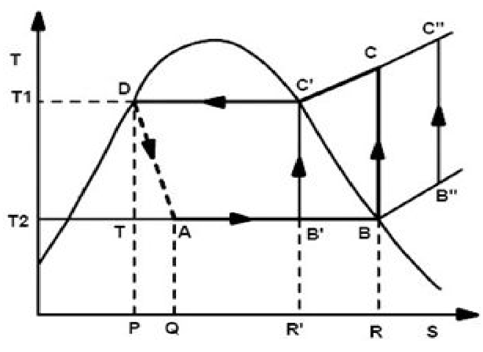

Figure 3. T−s diagram of refrigeration cycle

$$COP = \frac{Cooling\,Effect}{Work\ supplied} \quad (1)$$

$$q_2 = (h_B - h_A) \quad (2)$$

$$w = (h_C - h_B) \quad (3)$$

3. Design of controller

Changing the compressor frequency will change the refrigerant mass flow rate, and then it also changes the cooling capacity. In the experiment we have to measure the pressure, temperature and mass flow rate to get the empirical correlation between the compressor frequency and the cooling capacity. After the experiment has done, we will get the frequency-cooling capacity correlation formula with the output is linear equation

$$y = Ax + B \quad (5)$$

where y is the cooling capacity, x is the compressor frequency, A is the slope and B is the constant value got from experiment. By having this formula we have two benefits. The first benefit is we don't need to measure the mass flow rate to get the cooling capacity, this means we don't need to attached the pressure and refrigerant mass flow rate sensor that are very expensive. The second benefit is we can use this formula on logic control unit to control the cooling capacity.

a. Controlling cooling effect

In this study, the proportional gain is taken from linearization of the frequency range of the compressor from zero until maximum and the maximum cooling effect result. Fig. 2 shows the flowchart diagram of control system.

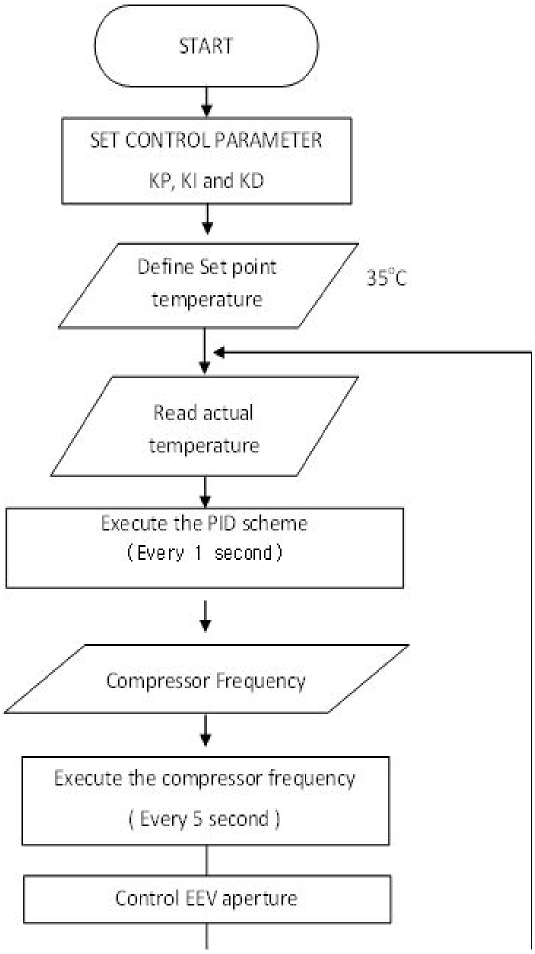

Figure 4. Flowchart of control system

In the Laplace domain, the PID equation to control the compression frequency is taken from Eq 6 [1].

$$U(s) = K\left(1 + \frac{1}{T_I s} + \frac{T_D s}{1 + \frac{T_D s}{N}}\right)E(s) \qquad (6)$$

Manipulating the equation to discrete time results the proportional, integral, derivative respectively shown in Eq 7 until Eq 9.

$$U_p(k) = K_p e(k) \qquad (7)$$

$$U_I(k) = U_I(k-1) + \frac{K_p T}{T_1} e(k) \qquad (8)$$

$$U_D(k) = K_p N(e(k) - e(k-1)) + \exp\left(\frac{-TN}{T_D}\right) U_D(k-1) \qquad (9)$$

Where U(s) and E(s) respectively is the Laplace output and error, K is the proportional gain, TI is the integral, or reset, time, TD is the derivative time, and N is a constant for the filter in the derivative portion

b. Controlling superheat

There are two basic schemes for sensing superheat. True superheat is a pressure-temperature relationship, specific to each refrigerant. When electronically derived, pressure-temperature (P-T) superheat requires the use of a pressure transducer, a temperature sensor, and a pressure-temperature table or equation.

A simpler but less accurate measure of superheat is the two-temperature method. In the two-temperature method, the temperature is sensed at the inlet and at the outlet of the evaporator. An advantage to two-temperature superheat is cost; pressure transducers are far more expensive than thermistors. Additionally, two-temperature superheat works with any refrigerant without re-programming. The temperature difference between the two sensors will indicate superheat no matter what the pressure-temperature relationship of the refrigerant.

The main disadvantage of the two-temperature method is the uncertainty that the inlet sensor is located properly. For the two-temperature superheat method to be accurate, the inlet sensor must be located in a position that has saturated refrigerant present at all times. Failure to find, or use, the proper location can lead to poor control or compressor damage.

Because of these factors, it is unlikely that retrofitting EEVs with two-temperature control to existing systems in the field will ever become practical.

4. Experiment apparatus

Fig. 4 and Fig. 5 respectively show the experiment apparatus and the real photo of the apparatus. The compressor speed is controlled by inverter frequency and the superheat is controlled by EEV aperture. The compressor normal frequency range is 30Hz-60Hz. and the EEV aperture full open is 512 Step from fully closed.

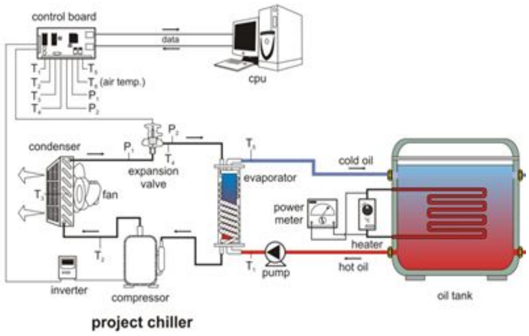

Figure 5 Experiment schematic

Figure 6 Experiment apparatus

5. Results and discussions

Based on experiment result have done with load variation from 4 kW until 10 kW, fig. 5 shows the controlled temperature. The Normal load is about 7 kW - 12 kW, below those load, the compressor will run under below 30 Hz. Hence the deviation of the constant temperature is high as shown in fig. 6. In normal load condition the compressor is never turned off, therefore the cooling effect of 8 kW and 10 kW are almost same with deviation value 0.2°C. And for abnormal condition, the lower load has higher deviation. This is difference is caused by the different on/off time of the compressor.

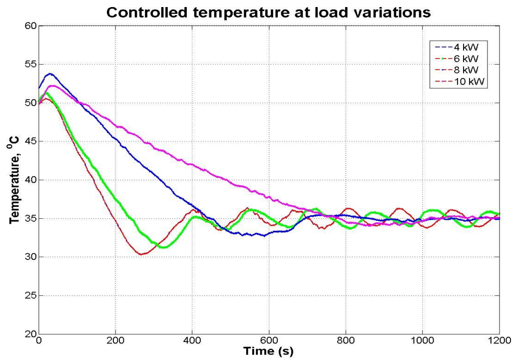

Figure 7 Controlled temperature

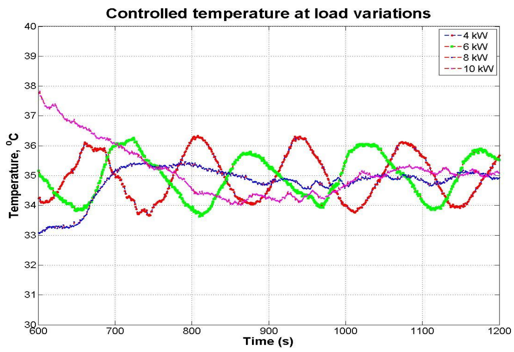

Figure 8 Temperature Deviation

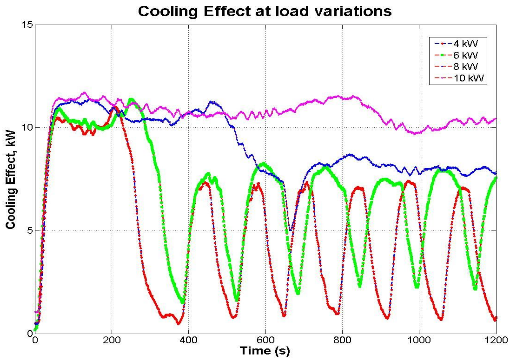

Figure 9 Cooling effect

6. Conclusions

The control system of the variable speed refrigeration system has made and using linearization cooling capacity as the proportional gain in PID control results deviation around 0.2^{o}C in normal load, and has higher deviation in abnormal load. The increased deviation is caused by thee compressor has a bandpass range of frequencies.

7. References

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