Energy Savings Potential by Variable Speed Control of Compressor for Air-Conditioning Systems using PID Controller

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Abstract. PID controller strategy is presented for a comfort air-conditioning (AC) system concerning energy performance and indoor comfort requirements. This paper presents an algorithm developed based on PID controller that enable selected compressors to run at the appropriate operating speed. Measurements and computer interface system were designed and including software to implement the controller algorithms. The experiments were conducted with temperature set-points for the conditioned space of 20, 22 and 24^{o}C with internal heat loads 0 and 1000 Watt. The experiment results indicate that the proposed technique can save energy and thermal comfort in comparison with on/off control technique.

Introduction

With rising living standards and expectation for thermal comfort, air conditioning (AC) has gradually become a necessity. Consequently the increase use of AC has had a significant impact on the total amount of energy used. The design standards for AC are fundamentally based on the principle of maintaining thermal comfort. However, current practice seems to indicate oversizing of AC systems. Investigation by Yu [1] showed that 67% of the respondents claimed that they intentionally oversized AC design for about 10 to 15%.

The major reason for the reduction in energy consumption is the significant improvement in the performance of the compressor known as coefficient of performance (COP), which depends on the compressor speed. In the United States, energy usage of compressor in term of COP was 4.14 in the late 1970s and improved to 5.86 in 1996. In 1997, the lowest energy usage was 7.17 COP [2]. The operating frequency of the inverter of the AC should be at its lowest frequency to get the highest COP within the variable range of the compressor operating frequency when the system capacity matches the cooling load on the system [3]. The basic difference between variable speed AC and conventional AC systems is in the control of the system capacity. In variable speed AC the capacity of the AC system is matched to the load by regulating the speed of the compressor motor such that the capacity of the system tracks the load acting.

This paper is focused on energy saving using a PID controller. The main idea of designing the controller is to maximize energy saving and thermal comfort for an AC system application through variable speed drive control. The results of the PID controller will be compared to the on/off controller.

Coefficient of Performance

The COP of a refrigeration machine is the ratio of the energy removed at the evaporator (refrigerating effect) to the energy supplied to the compressor. The thermodynamic COP is defined as enthalpy ratio of compression work at isentropic and its refrigeration effect is an ideal thermodynamic cycle and is given by [3]:

$$\mathrm{COP} = \frac{(h_1 - h_4)}{(h_2 - h_1)} = \frac{Q_e}{W_{com}} \tag{1}$$

for the Carnot refrigeration cycle [4]:

$$\mathrm{COP}_{\mathrm{carnot}} = \frac{T_1(s_1 - s_4)}{(T_2 - T_1)(s_1 - s_4)} = \frac{T_1}{T_2 - T_1} \tag{2}$$

where h_1, h_2 (kJ/kg) are the enthalpy at the compressor inlet and outlet respectively, h_4 (kJ/kg) is the enthalpy at the evaporator inlet, Q_{c} (kJ/kg) is the refrigerating effect, W_{com} (kJ/kg) is the compression work, T_1 ($^{\mathrm{o}}$C) is the evaporating temperature, T_2 ($^{\mathrm{o}}$C) is the condensing temperature, s_1 (kJ/kg.K) is the entropy at the compressor inlet and s_4 (kJ/kg.K) is the entropy at the evaporator inlet.

PID Controller

The PID controller algorithm is the almost universally used control strategy. The PID control equation is defined by [4]:

$$u(t) = K_p e(t) + K_i \int e(t)dt + K_d \frac{de(t)}{dt} \tag{3}$$

The use of proportional control (P) requires just one variable to be selected, the proportional gain K_p, for the control system to satisfy the required dynamic behavior. The use of a proportional plus integral gains (PI) or proportional plus derivative gains (PD) controller requires the selection of two variables, K_p and K_i or K_p and K_d, respectively. With a PID controller three variables have to be selected: K_p, K_i and K_d. For a digital PID controller [4] the controller gains K_P, K_I and K_D can be determine from the analog controller gains using the following relationships:

$K_P = K_p$
$K_I = K_i \times \Delta t$
$K_D = K_d / \Delta t$

where Δt is the sampling time (minute).

The output of the digital controller can be expressed by:

$$u_P(t) = (K_P \times e(t)) \tag{4}$$

$$u_{PI}(t) = [K_P \times e(t)] + [K_I \times (\sum_{i=1}^{n} e(t-i) \times \Delta t)] \tag{5}$$

$$u_{PD}(t) = [K_P \times e(t)] + \left[K_D \times \left(\frac{\Delta e(t)}{\Delta t} \right) \right] \tag{6}$$

$$u_{PID}(t) = [K_P \times e(t)] + [K_I \times (\sum_{i=1}^{n} e(t) \times \Delta t)] + \left[K_D \times \left(\frac{\Delta e(t)}{\Delta t} \right) \right] \tag{7}$$

where

$$e\ (t) = \textit{setpoint temperature}\ (t) - \textit{measured temperature}\ (t)$$
$$\Delta e = e(t) - e(t-1)$$

Experiment Apparatus and Method

Fig. 1 shows the points of measurement for temperature and pressure represented by T and P. Points T_1 and T_2: input and output temperatures at compressor, T_3 and T_4: input and output temperatures at condenser, T_5 and T_6: input and output temperatures at evaporator, and T_7-T_{11} are measurements for the room temperature. Points P_1 and P_2: input and output pressures at compressor, P_3 input pressure at condenser, and P_4 input pressure at expansion valve. Temperatures were measured by T type thermocouple and ICs temperature sensors. Pressures were obtained using Bourdon type gauges. Those locations on the high-pressure side ranged from 0 to 30 bar by 1 bar scales. The low-pressure side ranged from 0 to 10 bar by 0.2 bar scales. Electrical energy consumption was measured from the motor through PCI-1711/PCLD-8710 interface to the computer. Analog filter was used to separate the desired signals from unwanted interference or noise. The experiments were conducted for two different conditions: on/off and PID controller with 20, 22 and 24^{o}C temperature setting and internal heat loads: 0 and 1000 W.

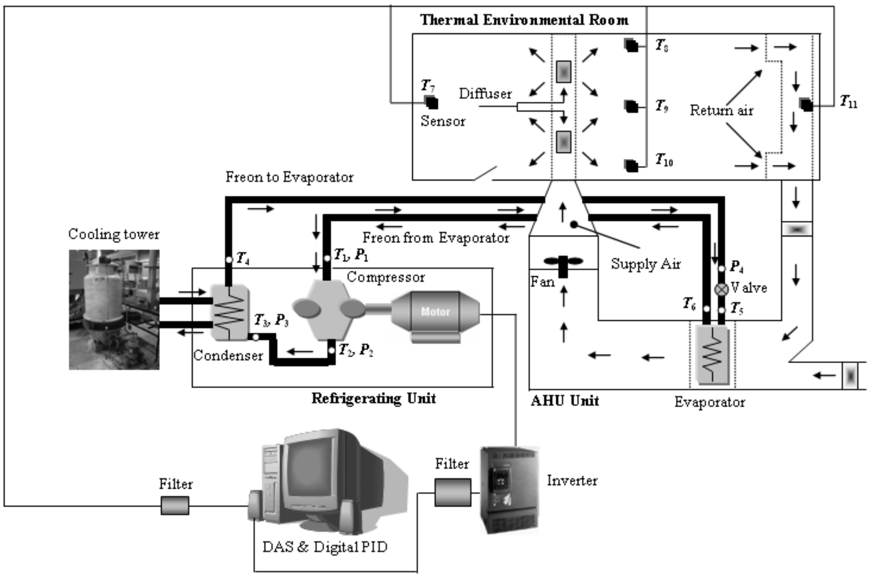

Fig. 1 The experimental setup

Energy Saving

The energy saving calculated is expressed in terms of saving in percentage unit, based on the difference between energy consumed using on/off control and energy consumed using PID controller. The equations are given as:

$$\text{Energy saving} = \frac{\text{(On/Off energy) - (PID energy)}}{\text{(On/Off energy)}} \times 100 \qquad (8)$$

Results and Discussion

Room Temperature Distribution. Fig. 2 shows the motor speed and temperature responses at various temperature settings. Initially the motor was set to run at the maximum speed (50 Hz). Maintaining the compressor motor speed at the maximum, the room temperature decreases as time increases. Referring to the setpoint temperature, the controller minimizes the error between the setpoint and the room temperatures. The figures show that the motor speeds drop abruptly as the room temperatures reach the setpoint. This action is taken to allow a fast heat recovery to the room until the measured temperature is equal to the setpoint temperature. The controller manipulates the motor speed until the room temperature is at or close to the setpoint temperature. The internal heat load affects the room temperature and the speed of the motor. Increasing the internal heat load results with a longer time to reach the setpoint temperature, and the motor speed decreases as the room temperature reaches the setpoint.

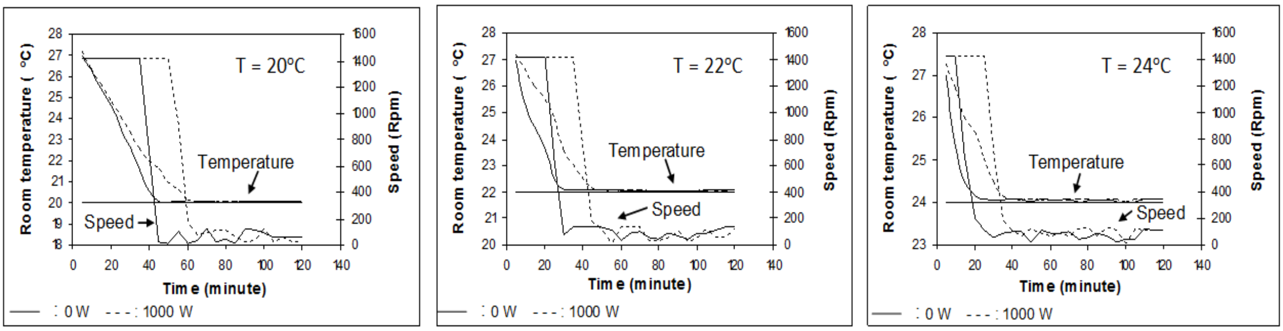

Fig. 2 Motor speed and temperature responses

Coefficient of Performance. Fig. 3 illustrates the relationship between the COP of actual and Carnot with frequency. The higher the frequency the smaller is the value of COP (actual and Carnot). The average values of the actual and Carnot COP were found to be 3.05 to 4.34 and 6.88 to 11.39. A high COP at the lower frequency is mostly due to the small compressor power consumption compared with that at higher frequencies. When the compressor power consumption increases, COP decreases with the increase of the compressor frequency.

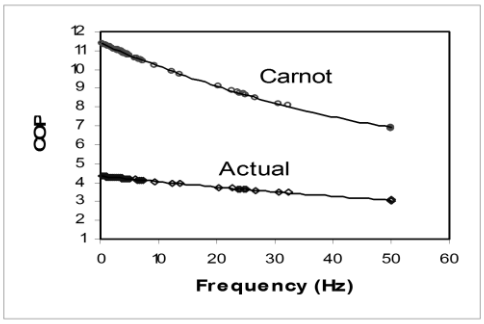

Fig. 3 The actual and Carnot COP

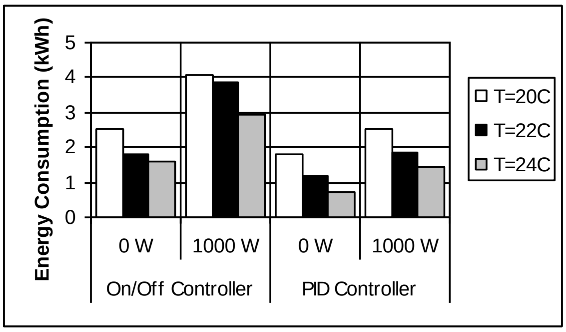

Fig. 4 The energy consumption distribution

Energy Consumption and Energy Saving. Fig. 4 shows the energy consumption at various internal heat loads for PID controller mode. Energy consumption was calculated once the motor starts using the relationship given by Eq. (8). The results indicate that the higher the internal heat load the higher is the energy consumption and PID controller results with smallest energy consumption. The energy consumption was calculated by multiplying the power consumption of the motor with the actual operating hours given by Eq. (8). The power and energy consumption were calculated every five minutes.

Fig. 5 shows the energy saving for PID controller. The figure shows that the energy consumption varies with different internal heat loads. The higher the energy consumption the smaller is the energy saving. For the two hour continuously used, energy savings achieved by running the system at various internal heat loads (0 and 1000 W) are 28.46 to 55.61%.

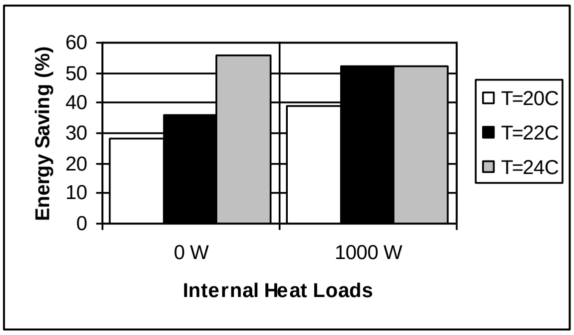

Fig. 5 Energy saving

Summary

The controllers i.e. on/off and PID has been developed to control the motor speed of the compressor in order to maintain the room temperature at or close to the setpoint temperature. The PID controller that produced satisfactory solution in term of energy saving for the room has been employed. The impacts of controller on the performance of the system, the room temperature and energy consumption have been analyzed experimentally. Energy analysis shows that PID controller gives the highest saving in comparison with on/off controller. The main outcome of this study shows that using variable speed compressor and choosing suitable control strategy, it is possible to control the space temperatures with significant energy saving.

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