MODELLING AND OPTIMIZATION OF DIRECT EXPANSION AIR CONDITIONING SYSTEM FOR COMMERCIAL BUILDING ENERGY SAVING

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ABSTRACT: This paper presents a comprehensive refinement of system modeling and optimization study of air-cooled direct expansion (DX) refrigeration systems for commercial buildings to address the energy saving problem. An actual DX rooftop package of a commercial building in the hot and dry climate condition is used for experimentation and data collection. Both inputs and outputs are known and measured from the field monitoring. The optimal supply air temperature and refrigerant flow rate are calculated based on the cooling load and ambient dry-bulb temperature profiles in one typical week in the summer. Optimization is performed by using empirically-based models of the refrigeration system components for energy savings. The results are promising as approximately 9% saving of the average power consumption can be achieved subject to a predetermined comfort constraint on the ambient temperature. The proposed approach will make an attractive contribution to residential and commercial building HVAC applications in moving towards green automation.

***Keywords**: Direct Expansion, Empirical Method, Energy Saving, Optimization*

1. INTRODUCTION

The increasing consumption of energy in buildings on heating, ventilating and air-conditioning (HVAC) systems has initiated a great deal of research aiming at energy savings. With the consolidation of the demand for human comfort, HVAC systems have become an unavoidable asset, accounting for almost 50% energy consumed in building and around 10-20% of total energy consumption in developed countries [1]. The direct expansion (DX) air conditioning plant is one of the main HVAC systems for different types of buildings. Its operation has a significant effect to the overall building energy consumption. DX air conditioning plants are simpler in configuration and more cost-effective to maintain than central cooling ones using chillers and cooling towers [2]. Therefore, these systems have a wide application in small to medium size buildings. The performance of DX systems can be improved by determining the optimal decision variables of the system to address the issues of energy savings and occupant comfort.

There are many studies on how the different control strategies can reduce the HVAC energy consumption. Also several studies have been developed for optimal control of HVAC systems. Braun et al. [3] proposed a methodology for determining the optimal control strategy for an HVAC system in which they showed that the energy consumption of a cooling plant can be represented in terms of the controlled and uncontrolled variables by a quadratic cost function. Their method allowed a rapid determination of near-optimal control setting over a range of conditions. Ahn and Mitchell [4] used this methodology for a central cooling plant and illustrated the optimally-set temperatures for supply air, chilled water and condenser water, to be selected such that energy consumption is minimized under a range of uncontrolled variables. Yao et al. [5] describes a mathematically optimal operations of a large cooling system based on an empirical model for chilled water cooling systems and showed that 10% energy savings were possible by applying the optimal model to a residential cooling system. Huh and Brandemuehl [8] suggested a method for optimization of a DX air conditioning system to minimize energy with a special emphasis on the control of humidity in commercial buildings. They showed that the most significant factors affecting the system operations are the interaction between the compressor capacity control and the supply air fan control.

The effect of different HVAC optimized control strategies on energy consumption in an actual typical air conditioning system was explored by Ghaddar et al [9]. The simulation results demonstrated that varying the supply air temperature and fresh air flow rate will result in a decrease of 11% of the total operational energy cost. Yan et al. [10] illustrated an adaptive optimal control model for building cooling and heating sources. They showed that in this model, a penalty function can be constructed to transform the constrained optimization problem into an unconstrained optimization problem, and that the energy consumption of the cooling system can be reduced by 7% by applying adaptive optimal control. Beghi and Cecchinato [11] considered the problem of interaction between the evaporator and electronic expansion valve to derive high-performance adaptive control algorithms, but it is necessary to consider the optimization of HVAC systems as a whole. Yao and Chen [12] developed a global optimization problem based on energy models of components, but their formulation is for central air conditioning systems.

In the context of energy optimization much research effort has been paid to central chilled water cooling plants, there are however a few published studies that have addressed the effects of optimal control on overall energy consumption by packaged rooftop DX air conditioning systems. Apart from the practical validation of these methods on real HVAC systems, especially on DX systems are also important.

This paper discusses the modeling of the controlled and uncontrolled variables of a DX cooling plant under operation conditions with the aim to obtain explicit expressions of decision variables to facilitate the derivation of optimal parameters for direct expansion air conditioning systems. Using this method, the overall energy consumption through a given DX system can be modeled by quadratic equations in terms of controlled and uncontrolled variables by a suitable regression analysis of experimental data. The controlled variables will then be selected so that energy consumption is minimized under a range of uncontrolled variables. In order to quantify and determine the optimal control variables of the DX system, a field test was conducted. A comparison of actual and optimal power consumption was conducted for one week in the summer time in a hot and dry climatic condition. The results show that the on-line implementation of the proposed decision variable set to determine the optimal system references could save energy by 9%, while maintaining the building comfort conditions.

2. MATHEMATICAL MODEL

The schematic block diagram of a DX air conditioning system is shown in Fig. 1. In the followings a combined theoretical-empirical modeling approach will be developed for these components.

2.1 DX Evaporator

The DX evaporator in the plant is of the rectangular finned tube type and heat exchange between refrigerant and air is assumed to be counter-flow. The cooling duty required for the air cooling and dehumidifying process is calculated using the difference between inlet and outlet air enthalpy. If the inlet and outlet air enthalpy respectively are $h_{ae,in}$ and $h_{ae,sp}$, M_{ae} is the evaporator fan air flow rate, then the required cooling duty Q_e, based on the principle of energy conservation is given by:

$$Q_e = M_{ae}\left(h_{ae,in} - h_{ae,sp}\right), \tag{1}$$

where the air enthalpy is evaluated by

$$h_{ae} = 1.005T_{db} + 0.001d_a(2500 + 1.84T_{db}). \tag{2}$$

In (2), T_{db} is the ambient dry-bulb temperature and d_a is the moisture content of air evaluated by

$$d_a = \frac{(2500 - 1.347T_{wb})d_{sa} - 1010(T_{db} - T_{wb})}{2500 + 1.84T_{db} - 4.187T_{wb}}, \tag{3}$$

where T_{wb} is the air wet-bulb temperature and d_{sa} is the saturated moisture content.

The input power of the evaporator fan, which is adjusted for part load operating conditions, can be expressed as a function of the airflow rate of the fan as follows:

$$P_{evapfan} = a_0 + a_1M_{ae} + a_2{M_{ae}}^2 + a_3{M_{ae}}^3, \tag{4}$$

where $P_{evapfan}$ is the evaporator fan input power and T_{sup} is the supply air temperature. Coefficients a_0 to a_3 are constant, which can be determined by curve-fitting of the experimental data. The fan air flow rate is a function of the supply air temperature, ambient dry-bulb temperature and cooling load.

2.2 Air-Cooled Condenser

The condenser in the plant is an air-cooled shell and of a low fin tube type. For a specific heat rejection capacity, the air flow rate of the condenser fan mainly depends on the air-cooled condenser and the enthalpy difference between air entering enthalpy $h_{ac,in}$, and air leaving enthalpy $h_{ac,out}$ as:

$$Q_{rej} = M_{ac}\left(h_{ac,out} - h_{ac,in}\right). \tag{5}$$

In (4), M_{ac} is the required ambient air flow rate for heat rejection in the condenser. The air-cooled condenser fans power consumption can be expressed as a function of the ambient air flow rate M_{ac}, which itself is based on the ambient air temperature:

$$P_{con\,fan} = b_0 + b_1 M_{ac} + b_2 {M_{ac}}^2 + b_3 {M_{ac}}^3, \tag{6}$$

where P_{confan} is the condenser fan input power. Again coefficients b_0 to b_3 are constant values and can be determined by curve-fitting the real data.

2.3 Variable Speed Rotary Compressor

The performance of the compressor under part load conditions depends on design parameters including the refrigerant mass flow rate m_r , the superheat temperature T_{sh} , the sub-cool temperature T_{cs} , the evaporator temperature T_e and the condenser temperature T_c. When the density of the refrigerant is independent on the superheat and sub-cool temperature, the compressor power input for a given refrigerant flow rate can be expressed as a function of the evaporator and condenser temperature as:

$$\begin{aligned} P_{comp} &= c_0 + c_1 T_e + c_2 T_c + c_3 {T_e}^2 + c_4 T_e T_c + \\ &\quad c_5 {T_c}^2 + c_6 {T_e}^3 + c_7 T_c {T_e}^2 + c_8 T_e {T_c}^2 + c_9 {T_c}^3, \end{aligned} \tag{7}$$

where c_0 to c_9 are constant parameters which can be determined by curve-fitting the experimental data using, e.g. the least square method. The rating procedure and the form of the equation are specified by ARI standard 540 [7]. The condensing temperature T_c is influenced by the ambient dry-bulb temperature and the heat rejection capacity of the condenser. In the same way, the evaporator temperature is influenced by the cooling capacity of the evaporator and the evaporator inlet air temperature. As a result, the effective variables for compressor's optimal setting are the system cooling capacity and the ambient dry-bulb temperature.

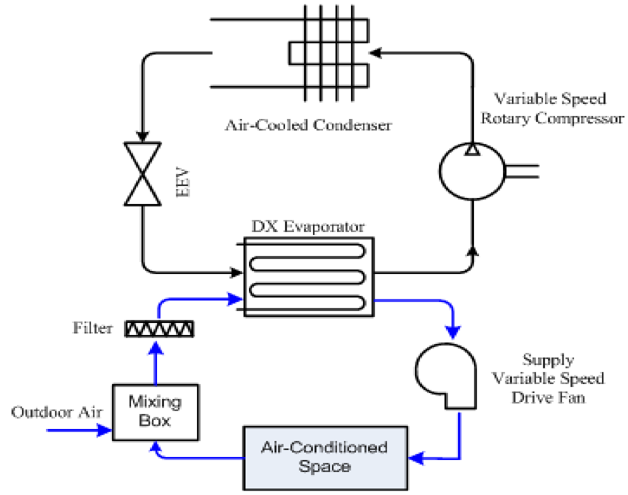

Fig. 1 Block diagram of DX air conditioning system.

2.4 Electronic Expansion Valve (EEV)

The expansion valve is a refrigerant flow control device that adjusts and controls the quantity of liquid refrigerant entering the evaporator, and thus regulates the refrigerant superheat temperature leaving the evaporator. The most effective variable for optimal operation control of EEV is the refrigerant mass flow rate while the cooling load is considered as an uncontrolled variable. The refrigerant flow through the EEV is represented by an orifice equation and its mass flow rate is calculated by [6]:

$$m_r = C\left(\frac{\pi D^2}{4}\right)\sqrt{2\rho(P_i - P_o)}\,, \tag{8}$$

where D is the orifice diameter, ρ is the refrigerant density, P_i is the upstream pressure, P_o is the downstream pressure and C is the mass flow coefficient, which is a function of the EEV geometric parameters and the refrigerant thermodynamical properties calculated by [6]:

$$C = 1.1868 \times 10^{-13}\left(\frac{(P_i - P_o)\sqrt{A}}{\sigma_i}\right)^{-1.4347}\left(\frac{d_i\sqrt{\rho P_i}}{\mu}\right)^{3.6426}. \tag{9}$$

In (10), σ_i is the surface tension, μ is the dynamic viscosity, A is the orifice area and d_i is the inlet diameter.

3. PROBLEM FORMULATION

The optimization problem is formulated through the determination of controlled variables, to minimize the objective functions subject to constraints. Problem variables for the direct expansion air conditioning system under investigation are included supply air temperature, refrigerant mass flow rate, total cooling load and ambient dry-bulb temperature.

The objective function is to minimize of the overall power consumption of the whole system with controlled variables being m_r and T_{sup}. The power consumption obtained from the compressor, evaporator fan, and condenser fans is given as:

$$Min \;\; P_{total} = P_{comp} + P_{evapfan} + P_{con\,fan} \, , \quad \textit{Subject to constraints.} \tag{10}$$

A quadratic form for this objective function for optimization is proposed. This function represents the total system's instantaneous power consumption P_{total}, in terms of the controlled and uncontrolled variables and consists of fifteen coefficients that can be determined empirically using real operation data and a regression technique.

$$\begin{aligned} P_{total} &= d_0 + d_1 m + d_2 m^2 + d_3 m T_{sup} + \\ & d_4 m Q_e + d_5 m T_{db} + d_6 T_{sup} + d_7 {T_{sup}}^2 + \\ & d_8 T_{sup} Q_e + d_9 T_{sup} T_{db} + d_{10} Q_e + d_{11} {Q_e}^2 + \\ & d_{12} Q_e T_{db} + d_{13} T_{db} + d_{14} {T_{db}}^2 . \end{aligned} \tag{11}$$

To fit the function to the real test data, the regression can be done using the MINITAB statistical software.

3.1 Optimizing Conditions

Once the empirical coefficients in (13) have been obtained, the first-order derivatives with respect to the controlled variables namely the supply air temperature and refrigerant mass flow rate, are divided for the system optimizing conditions. The following two equations must be solved in order to calculate the optimal set-points:

$$\frac{\partial P_{total}}{\partial m_r} = 0, \qquad \frac{\partial P_{total}}{\partial T_{sup}} = 0 \, . \tag{12}$$

The second order derivatives of the objective function with respect to these controlled variables must be positive to define a local minimum control.

3.2 Constraints

Equality and inequality constraints should be taken into account during the optimization process. which are listed as follows:

Constraint 1. The building cooling load Q_b must be less than the cooling capacity of the DX plant:

$$Q_b < Q_e . \tag{13}$$

Constraint 2. The interactions between the evaporator, air-cooled condenser and the compressor formulate a constraint as:

$$P_{comp} + Q_e = Q_{rej} \, . \tag{14}$$

Constraint 3. The supply air temperature is restricted by

$$10°C \le T_{sup} \le 20°C . \tag{15}$$

Constraint 4. The refrigerant mass flow rate through the DX air conditioning system should be within its limitation:

$$m_{r,min} \le m_r \le m_{r,max} . \tag{16}$$

Constraint 5. The comfort ranges for the indoor air temperature T_z and relative humidity RH_z during occupied periods are given by:

$$\begin{aligned} & 22°C \le T_z \le 26°C \\ & 40\%°C \le RH_z \le 60\% \, . \end{aligned} \tag{17}$$

4. CASE STUDY

The proposed optimization process is applied on the existing DX system. The air flow rate in an evaporator and condenser is controlled by the variable speed drive (VSD) fans. The refrigerant mass flow rate is controlled by the electronic expansion valve.

4.1 Experimental Rig

In order to obtain an optimal set of control variables, a data base with information about the system performance under various operating conditions must first be gathered. For this purpose, a real-world test data were collected in testing conditions. High precision sensors/transducers were used for measuring all operating variables. Manometers were used for obtaining supply air flow rate. The temperature sensor for supply air is of platinum resistance type (accuracy: $\pm 0.1°C$). The refrigerant mass flow rate passing through the EEV is measured by a Coriolis mass flow meter (accuracy:$\pm$0.25°C). The ambient temperature was measured by digital thermometer (precision: $\pm$0.8°C). Powers of components were measured by a digital ac/dc power clamp multimeter (precision:$\pm$3.5%). Therefore a total fifty-six points of system power consumption and other variables were measured for each fifteen-minute period. The building sensible and latent cooling load, were calculated from monitoring data. After gathering the initial set of real test variable values, the MINITAB statistical software [14] was used to calculate the coefficients of the quadratic cost function. The regression process explains the coefficients in terms of variations in the actual overall system energy consumption. The R-squared value of the model was 0.93, indicating a good fit. These coefficients for the described D.X rooftop package are:

$$
\begin{array}{lll}
d_0 = -84.27 & d_1 = 3682 & d_2 = 10 \\
d_3 = 49.33 & d_4 = 13.13 & d_5 = -24.69 \\
d_6 = 4.07 & d_7 = 0.02 & d_8 = -0.028 \\
d_9 = -0.16 & d_{10} = -4.63 & d_{11} = -0.07 \\
d_{12} = 0.03 & d_{13} = 2.19 & d_{14} = 0.03.
\end{array}
\tag{18}
$$

Therefore, the set of control function in terms of the cooling load and ambient dry-bulb temperature are obtained by solving the first-order controlled variable derivatives:

$$
\begin{aligned}
m_{r,opt} &= 0.0028T_{db} + 0.0059Q_e - 0.0881, \\
T_{sup,opt} &= 0.5002T_{db} - 0.2683Q_e + 7.501.
\end{aligned}
\tag{19}
$$

Since the sign of the second order derivatives of control variables are positive, the control function defines a local minimum control point:

$$
\frac{\partial^2 P_{total}}{\partial {m^2}_r} = 20 > 0, \qquad \frac{\partial^2 P_{total}}{\partial {T^2}_{sup}} = 0.04 > 0\,. \tag{20}
$$

The control functions are plotted over the anticipated range of cooling loads Q_e, shown in Fig. 2. This figure illustrates that the optimal supply air temperature should be lowered as cooling load increases while the refrigerant flow value should be increased at a larger value of the cooling load. To calculate the total power consumption, the optimal set-point values of the $T_{sup,opt}$ and $m_{r,opt}$ were input into the cost function. The results obtained for the actual reference system and the system with optimal operating points are

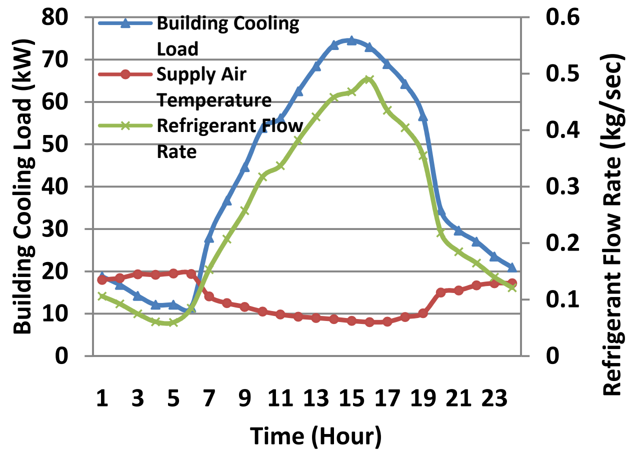

Fig. 2 Optimal control variables

compared in Fig. 3. It can be seen from this figure that the average power consumption by using optimal set-point values is nearly 9% less than that of the commonly-used control strategy.

We recall next that ASHRAE standard 55 recommends 25°C and 50-60% relative humidity for summer comfort conditions [13]. Figure 4 shows the profile of the indoor air temperature and relative humidity. Results show that the minimum, maximum and average values of the indoor temperature after optimization are respectively 21.8°C, 27.3°C and 23.6°C. These corresponding values for the indoor relative humidity are 49%, 58% and 53%. Therefore, the indoor temperature and humidity will be in the comfort ranges.

5. CONCLUSION

In this paper, we have addressed the modeling and optimization problem of a direct expansion air conditioning plant to target energy savings in a commercial building. Explicit relationships of the HVAC process characteristics with respect to the controlled and uncontrolled variables under operating conditions are established by using experimentally-collected data and empirically-based regression. The power consumption of the DX air conditioning system can then be expressed as a function of these variables in a proposed quadratic equation to facilitate the derivation of maximum energy saving conditions. A case study of a commercial building in hot and dry summer time shows the effectiveness of the proposed method. This methodology is promising in

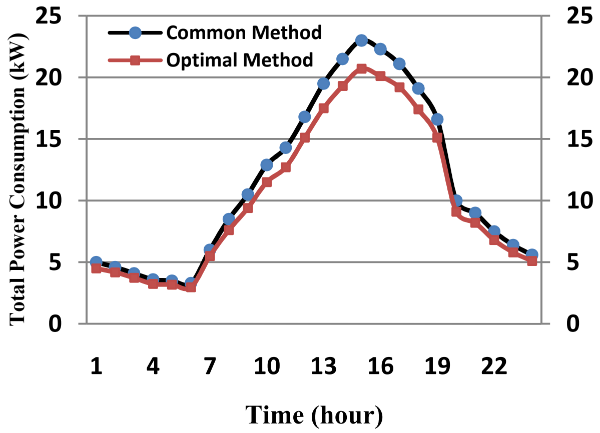

Fig. 3 Power consumption compression

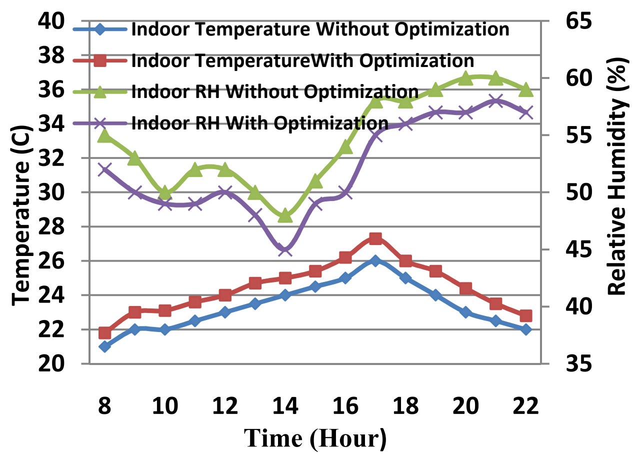

Fig. 4 Temperature and Relative Humidity

moving towards smart HVAC systems of the direct expansion air conditioning type for small and medium buildings.

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