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Direct optimal control of valve openings in heat exchanger networks and experimental validations

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ABSTRACT

Heat exchanger networks (HENs) optimizations benefit the energy conservation, with many set point strategies controlling the operations indirectly by the help of detail control techniques. Based on the newly proposed thermal resistance-based optimization method and the flow resistance analysis, we obtain the physical models of heat exchangers, pumps and pipelines with adjustable valves. Combining these models, we introduce a direct optimal control strategy for adjustable valves in HENs to obtain the optimal valve openings directly. On this basis, we take a HEN with two heat exchangers as an example to validate the proposed control strategy. Groups of experiments and the results illustrate the optimal valve openings obtained from the proposed strategy lead a lower power consumption of all the pumps than that of any other alternative experiment, which indicates the potentials to reduce power consumption by the optimal control of valves. Finally, further experiments with optimal valve openings under different required heat transfer rates demonstrate the universality of the newly proposed control strategy.

1. Introduction

Heat exchanger networks (HENs) are one of the main components and consume a large proportion of energy in many engineering fields, such as power plants and chemical industries. HEN optimization is significant for energy conservations, where the optimal control of adjustable valves is a simple and efficient way to reduce the energy consumption of HENs [1].

In recent years, the optimal control of adjustable valves usually relies on the pressure differential set points control strategy. For instance, Wang and Burnett proposed a pressure set point control method for an indirect water-cooled chilling system [2], and Lu et al. raised an optimization method for a heat, ventilation and air condition (HVAC) system with the duct differential pressure set point [3]. Besides, the pressure differentials of other set points also work in the optimal operating of HENs, such as the pressure differentials of secondary variable speed pumps [4,5], the static pressures of fans [6], and the pressure drops through heat exchangers [2]. However, the direct control parameters in a HEN are usually the openings of valves, rather than the pressure differentials of some set points. Therefore, with these optimal set points, it is unavoidable to control the valve openings by seeking the help of some control strategies, such as the PID controllers [3,4,7], the direct digital control (DDC) strategies [8], the online control strategies [9], and the feedback [10] or self-turning control strategies [11,12].

Therefore, the aforementioned methods virtually consist of two separate steps. One is to get the optimal set points to satisfy the requirements, and the other is to achieve these set points by regulating the openings of valves through control strategies. These two sequential steps separate the influences of the characteristics of valves and pipelines out of the system global performance, and consequently narrow down the range of optimization results artificially. What's more, the control strategies require more or less setting time [13,14]. The set points are only controlled close to but not indeed the optimal values, incessantly varying within a range, which is influenced by the performance of controllers [15,16].

In order to directly obtain the optimal control parameters, Chen et al. [17] theoretically provided a thermal resistance-based method for HEN optimization, which gives physical relations to link the operating parameters, i.e. valve openings and pump operating frequencies, directly to the system requirements, such as required heat transfer rates and surrounding temperatures, and the performances of each component, including heat exchanger thermal conductances, pump and pipeline characteristics. Based on these physical relations and the system requirements, the method of Lagrange multipliers transfered the optimization into a mathematical extremum problem, which can be solved theoretically to provide the optimal operating parameters of variable speed pumps in HENs.

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Nomenclature

A	area, m^2
a	characteristic parameter of pump
b	dynamic coefficient of head loss
c	valve opening
c_p	constant pressure specific heat, J/(kg K)
D	diameter, m
d	equivalent dynamic head loss coefficient
f	Darcy friction-factor
H	head loss, m
g	acceleration of gravity, 9.8 m/s^2
h	installation height, m
h_f	Moody-type friction loss, m
h_m	minor loss, m
K	minor loss coefficient
k	heat transfer coefficient, W/(m^2 K)
L	length, m
m	mass flow rate, kg/s
P	power consumption, W
Q	heat transfer rate, W
R	thermal resistance, K/W; uncertainty function
R^2	coefficient of determination
S	cross-sectional areas of pipe, m^2
T	temperature, K
V	volume flow rate, L/min
v	fluid velocity, m/s
α, β, γ	Lagrange multipliers
δ	deviation
ξ	flow arrangement factor of heat exchanger
Π	Lagrange function
ρ	density, kg/m^3

Subscripts

c	cold fluid
d	dynamic
e	evaporator
exp	experiment value
fit	fitted value
h	hot fluid
hx	heat exchanger
i	inlet
m	mixing process
o	outlet
p	pressure differential
r	refrigerant
ref	referenced value
s	static head
t	total
w	water

On this basis, this paper provides an effort to directly obtain the optimal openings of each adjustable valve for a practical HEN, avoiding the inconvenient control strategies with intermediate set points. Based on the physical models of heat exchangers, pumps, and pipelines, we first determined the thermal conductances of heat exchangers and the characteristic parameters of pumps and pipelines in a HEN by a series of experiments. With these physical models, utilization of the thermal resistance-based optimization method directly offered different optimal valve openings under different system requirements. Experimental measurements of the HEN performance illustrate that the optimized valve openings indeed lead to the lowest total power consumptions of HEN under a certain system requirement. Further experiments with the optimal valve openings also give the minimum power consumptions of all the pumps under different required heat transfer rates of HEN.

2. Experiment facilities and measurement instruments

Fig. 1 is the sketch of an experimental HEN studied in this paper, which is also applied in Ref. [18]. It consists of two counter-flow plate heat exchangers, three adjustable valves, three variable speed pumps, a thermostatic hot water tank, a chiller, and pipelines wrapped up with thermal insulating materials. It is worth pointing out that, all the variable speed pumps operate in their individually fixed frequencies in all the experiments, where the valve openings are optimized for energy consumption.

The working fluids in Loops 1 and 3 are both water, and that in Loop 2 is a refrigerant R142b. The pumps drive the working fluids to circulate in each loop, transferring heat in the thermostatic hot water tank through heat exchanger 1, heat exchanger 2, finally to the evaporator of the chiller. Three turbine volume flow-meters with an accuracy ± 0.5% of the full scale 20 L/min are utilized to measure the fluid flow rates in three loops. Three differential pressure transducers with an accuracy ± 0.2% of the full scale 350 kPa are employed to measure the pressure differentials of each VSP. A pressure gauge with the full scale 1.8 Mpa is equipped in Loop 2 to monitor the refrigerant absolute pressure. T type copper-constantan thermocouples (Produced by Omega Engineering) with an accuracy ± 0.2 °C serve to test the working fluid temperatures in each measurement points shown in Fig. 1, where two thermocouples are placed in each point for accuracy. The subscript w and r represent water and refrigerant, i and o mean the inlet and outlet of heat exchangers, and 1, 2 and e stand for heat exchanger 1, heat exchanger 2 and the evaporator, respectively. In addition to the measuring instruments, the experiment system also contains a data acquisition and control system (DACS) to log the measured data, including volume flow rates, pressure differentials and temperatures, and control the components, such as the openings of valves and the water temperature in the tank. Fig. 2 is the photo of the above experiment facility [18].

3. Physical models of each component

3.1. Physical models of heat exchangers

For the counter-flow heat exchangers, i.e. Heat Exchangers 1 and 2, the entransy dissipation-based thermal resistances are [19]

$$R_{hx1} = \frac{(T_{w1,i} + T_{w1,o}) - (T_{r1,i} + T_{r1,o})}{2Q} = \frac{\xi_1}{2}\frac{\exp[(kA)_1\xi_1] + 1}{\exp[(kA)_1\xi_1] - 1}, \quad \xi_1 = \frac{1}{m_1 c_{p,1}} - \frac{1}{m_2 c_{p,2}}, \tag{1}$$

$$R_{hx2} = \frac{(T_{r2,i} + T_{r2,o}) - (T_{w2,i} + T_{w2,o})}{2Q} = \frac{\xi_2}{2}\frac{\exp[(kA)_2\xi_2] + 1}{\exp[(kA)_2\xi_2] - 1}, \quad \xi_2 = \frac{1}{m_2 c_{p,2}} - \frac{1}{m_3 c_{p,3}}, \tag{2}$$

where R is the entransy dissipation-based thermal resistance, ξ is the flow arrangement factor of a heat exchanger, k is the heat transfer coefficient, A is the heat transfer area, m is the mass flow rate, and c_p is the constant pressure specific heat. The subscripts $hx1$ and $hx2$ represent Heat Exchangers 1 and 2, and the numbers 1,