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Design of fuzzy system for vapour compression refrigeration system

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This paper presents fuzzy modeling of multi-input single output (MISO) control of vapour compression refrigeration cycles. Fuzzy model of two input parameters, mass flow rate, and evaporating temperature was developed to describe the significant dynamic of vapour compression cycles. To develop an effective model, fuzzy rule base was designed for MISO control in vapour compression system. The effect of input parameters on superheat through the developed fuzzy model was studied. Comparison of fuzzy and mathematical modeling showed that the MISO control could significantly improve the performance and energy efficiency of vapor compression refrigeration cycle.

Keywords: Vapour compression refrigeration system, Fuzzy modeling, Sugeno fuzzy inference system

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Introduction

Control of vapour compression system is a multidimensional problem. Minimization of energy consumption and the achievement of proper superheat control and capacity control is conflicting requirements. Fuzzy model is an important tool to understand system characteristics and improve the design of real plants and their control systems more precisely and accurately as compared to other modeling of refrigeration and air-conditioning systems. Fuzzy logic theory is one of the most innovative, active and fruitful areas of research for science and engineering applications, especially in industrial processes[1]. Fuzzy set theory is used to analyze engineering applications and to control difficult to control systems with great ease, which gives a considerable approximation of the parameters of mathematical functions[2,3]. A fuzzy system is emerging as a very strong tool to solve different kinds of problems in various application domains[4]. In vapour compression refrigeration system (VCRS), research has been undertaken to modify and to replace conventional control[5,6]. Advances in variable-speed drive technology opened up new possibilities for improving system performance and energy efficiency in vapor compression[5]. The compressor speed can be continually adjusted so as to modulate the heat exchanger capacity to match the actual thermal load. The speeds of fans can be altered to change the heat transfer rates across the heat exchangers. The opening of the expansion valve can be varied, so that refrigerant flow rate and pressure drop can be changed. VCRSs equipped with these variable-speed and variable-position drives have already been used for residential and commercial applications[7], but so far no substantial performance improvement. There are strong cross-couplings among various actuating inputs and performance outputs, such as evaporating temperature, condensing temperature and superheat[8]. A proper coordination among valve opening, fan speeds, and compressor speed will improve superheat behavior, withstanding external disturbances to a greater extent while effectively modulating system capacity.

VCRSs are inherently complex thermo-fluid systems that are generally equipped with multi-actuators and have to meet multi-performance requirements. Therefore, there is a potential for fuzzy control to significantly improve the control performance and energy efficiency of VCRSs. This paper presents a MISO control designed for regulating VCRS based on fuzzy model using sugeno fuzzy inference system and ANFIS under MATLAB environment. Input parameters taken are mass flow rate and evaporator temperature. The control objective was defined in order to improve the performance of the vapor compression cycle in terms of regulating the desired superheat.

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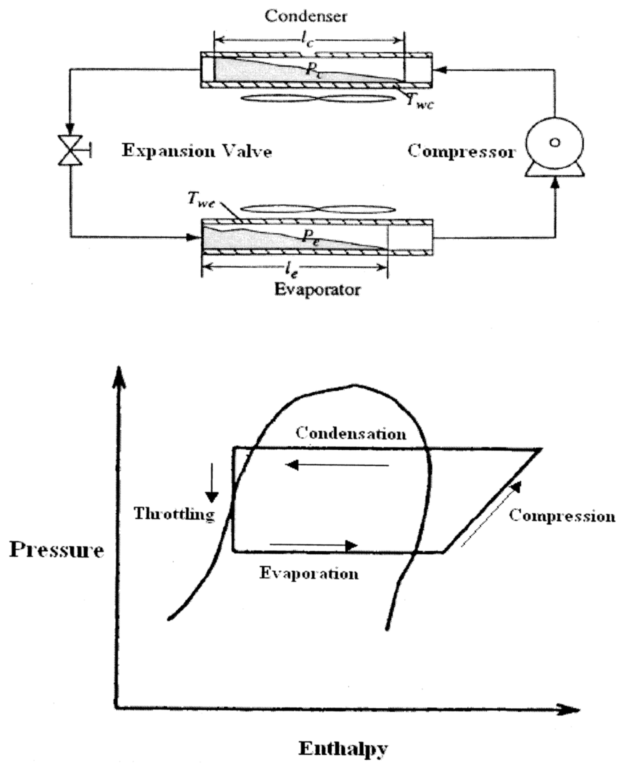

Fig. 1—Vapour compression system and its representation on *p-h* chart

Mathematical Modeling of Vapour Compression System

With respect to evaporator and condenser, VCRS has six state variables: length of the two-phase section in the evaporator *le*, evaporating pressure *Pe*, evaporator temperature *Te*, length of the two-phase section in the condenser *lc*, condensing pressure *Pc*, and condenser temperature *Tc* (Fig. 1). Output variable to be monitored and regulated is superheat *SH,* while mass flow rate and *Te* were used as the control inputs to VCRS. The standing assumptions were: one-dimensional fluid flow, negligible heat conduction along the axial direction of heat exchangers, invariant mean void fraction in the two-phase sections during a short transient, and negligible refrigerant pressure drop along the heat exchangers. In addition, it was also assumed that the storage capacitance for mass and thermal energy of all single-phase sections was negligible compared to the two-phase sections that dominated the heat exchanger dynamics. The last assumption implies that *SH* can be correlated to *Pe*, and the length of the superheated section, or equivalently *le*, in an algebraic equation.

Applying mass and energy balance principle to the evaporator, following mathematical model[10] can be realized for the evaporator in vapour compression cycle:

$$\rho_{le} h_{lge} A_e (1-\bar{\gamma}_e) \frac{dl_e}{dt} = -\dot{m}_v (h_{ie} - h_{ge}) - \alpha_{ie} \pi D_{ie} l_e (T_e - T_{re}) \quad \ldots (1)$$

The first term on the right-hand side of Eq. (1) corresponds to the energy storage rate due to the refrigerant flow (refrigerant entering the two-phase section with h_{ie} and exiting with h_{ge}). The second term on the right represents the heat transfer rate from the tube wall to the two-phase refrigerant. Combination of the two terms on the right corresponds to the net energy supply rate in the two-phase section; negative signs indicate that a decrease in the energy supply rate will result in a decrease in the evaporation rate and leads to an increase in l_e. Further, assuming that the vapor volume is larger than the liquid volume in the evaporator, and applying the mass and energy principles to the entire evaporator, the following expression holds for the vapor density:

$$A_e L_e \frac{d\rho_{ge}}{dt} = \dot{m}_v X_{ie} - \dot{m}_{com} + \frac{\alpha_{ie} \pi D_{ie} l_e (T_e - T_{re})}{h_{lge}} \quad \ldots (2)$$

The term on the left-hand side of Eq. (2) represents the rate of change of vapor mass in the evaporator, based on the assumption that the average vapor density is the saturated density ρ_{ge}. On the right-hand side of Eq. (2), first term is vapor mass flow rate entering the evaporator, second term is vapor mass flow rate leaving the evaporator, and last term is the vapor generation rate. Applying the chain rule to the derivative in Eq. (2) and replacing *Xie* by $(h_{ie} - h_{le})/hl_{ge}$, Eq. (2) can be rewritten as:

$$A_e L_e \frac{d\rho_{ge}}{dP_e} \frac{dP_e}{dt} = \dot{m}_v \frac{h_{ie} - h_{le}}{h_{lge}} - \dot{m}_{com} + \frac{\alpha_{ie} \pi D_{ie} l_e (T_e - T_{re})}{h_{lge}} \quad \ldots (3)$$

Applying the energy balance principle to the evaporator, the following equation results:

$$(C_p \rho A)_{we} \frac{dT_e}{dt} = \alpha_{ie} \pi D_{ie} (T_{re} - T_e) + \alpha_{oe} \pi D_{oe} (T_{ae} - T_e) \quad \ldots (4)$$

In Eq. (4), a uniform temperature was assumed throughout the evaporator tube wall, ignoring the influence of the superheated section.

Assuming that the axial conduction is negligible, superheat is expressed as follows:

$$SH = (T_{ae} - T_{re})\left[1 - \exp\left(-\frac{\alpha_{ioe}\pi D_{ie}(L_e - l_e)}{C_p\,\dot{m}_{com}}\right)\right] \quad \ldots\ (5)$$

where α_{ioe} is the equivalent heat transfer coefficient between the refrigerant and the ambient air at the superheated section.

By analogy to the evaporator, the governing equations for the condenser were expressed as:

$$\rho_{lc} h_{lgc} A_c (1-\overline{\gamma}_c)\frac{dt_c}{dt} = \dot{m}_{com}(h_{lc} - h_{ic}) + \alpha_{ic}\pi D_{ic} l_c (T_{rc} - T_c) \quad \ldots\ (6)$$

$$A_c L_c \frac{\rho_{gc}}{dP_c}\frac{dP_c}{dt} = \dot{m}_{com} - \frac{\alpha_{ic}\pi D_{ic} l_c (T_{rc} - T_c)}{h_{lgc}} \quad \ldots\ (7)$$

$$(C_p \rho A)_{we}\frac{dT_c}{dt} = \alpha_{ic}\pi D_{ic}(T_{rc} - T_c) + \alpha_{oc}\pi D_{oc}(T_{ac} - T_c) \quad \ldots\ (8)$$

In Eq. (7), it was assumed that refrigerant leaves the condenser in the form of liquid. It should be noted that this study does not include the use of an accumulator or receiver. Therefore, whatever amount of refrigerant flows out of the evaporator will enter the condenser, and vice versa. Therefore, the total refrigerant charge was considered to be the sum of refrigerant stored in the evaporator and the condenser.

The refrigerant flow rate through the expansion valve was modeled as the following orifice equation:

$$\dot{m}_v = C_v a_v \sqrt{\rho_v (P_c - P_e)} \quad \ldots\ (9)$$

where C_v is the orifice coefficient. With the use of an electronic expansion valve, a_v was made a continually adjustable variable. For the compressor, it was assumed that compressor wall was well insulated from the ambient air. In general, the refrigerant flow rate through the compressor is dependent on compression ratio, compressor speed, and refrigerant density. That is

$$\dot{m}_{com} = f\left(\frac{P_c}{P_e}, \omega_c, \rho\right) \quad \ldots\ (10)$$

where f is given by the compressor performance map.

The relationship between the enthalpy at the compressor outlet h_{ic} (enthalpy at the condenser inlet) and the enthalpy at the compressor inlet h_{oe} (enthalpy at the evaporator outlet) is given by

$$h_{ic} = \frac{h^*_{ic}(P_e, P_c, h_{oe}) - h_{oe}}{\eta_{com}} + h_{oe} \quad \ldots\ (11)$$

where h^*_{ic} is the enthalpy at the compressor outlet if the compression process is isentropic, and η_{com} is the compressor coefficient given by the compressor performance map. The enthalpy at the evaporator outlet h_{oe} is dependent on l_e, P_e, and T_e. It was assumed that the enthalpy at the condenser outlet h_{oc} was equal to h_{lc}, the enthalpy of saturated liquid evaluated at P_c.

Eqs (1) through (11) represent a model that describes the essential dynamics of a vapor compression cycle.

Design of the Fuzzy Logic Expert System for VCRS

System Architecture

The task of converting crisp logic measurement into fuzzy variables is performed by fuzzification interface (Fig. 2). The inference engine does the processing, reaching at the end a "fuzzy" conclusion on whether a subsystem should be turned ON or OFF, and the defuzzification interface turns this fuzzy conclusion set to crisp logic decision, suitable to drive the actuator drive circuits. Fuzzy system retains the mass flow rate and evaporator temperatures within user defined limits and regulate the superheat of the system for minimum energy consumption.

Membership Functions

Design of fuzzy control system starts with establishing certain quantization levels for mass flow rate, evaporator temperature, and superheat along with membership functions corresponding to these quantization levels. This process defines appropriate fuzzy sets to be the basis for applying fuzzy logic. They serve as linguistic values to be assigned, respectively, to the fuzzy variables. Triangular membership functions are used, because they lead to very tractable systems, except for the superheat where

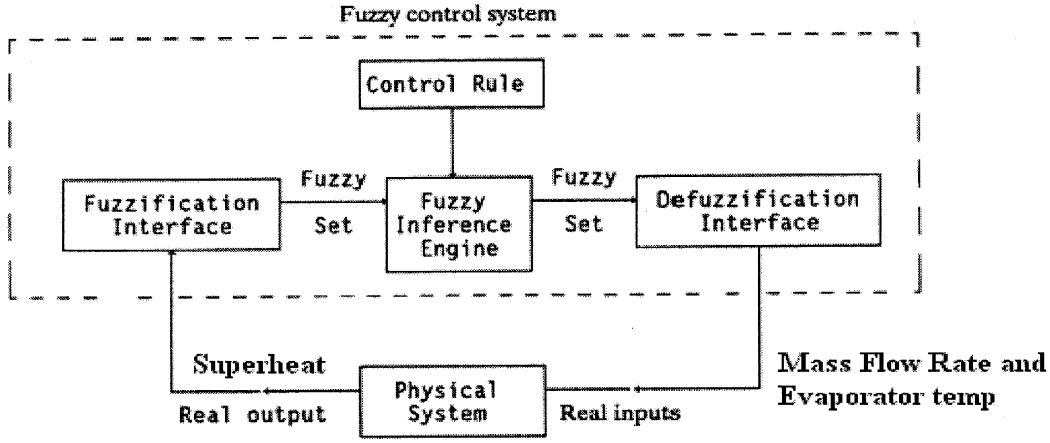

Fig. 2—Block diagram of fuzzy logic system and physical system

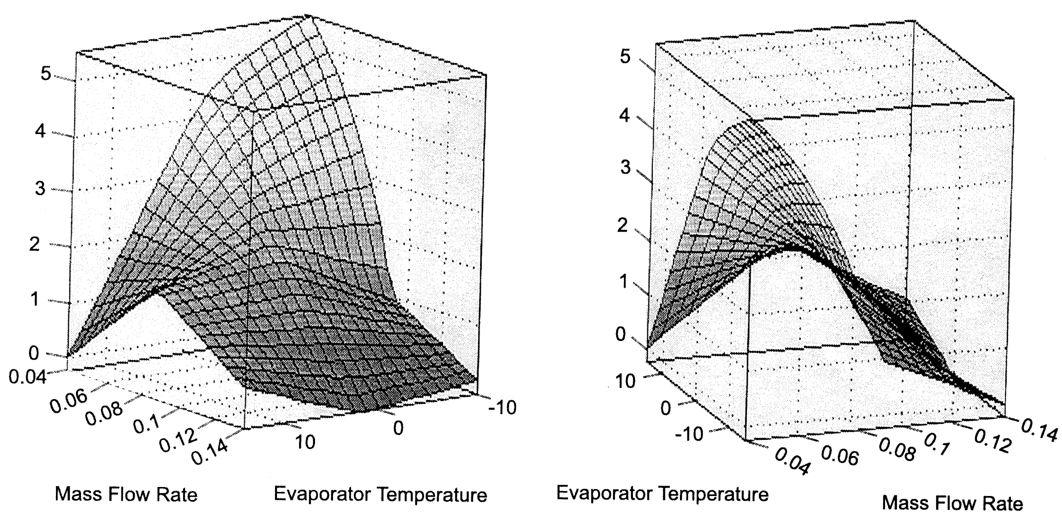

Fig. 3—Control surface for the fuzzy model from two different angles

membership function basing to Sugeno fuzzy inference system was used.

Fuzzy Control Algorithm and Control Surface

The good adequacy of the identified model allows synthesizing the control algorithm. In the undertaken model, two inputs (mass flow rate and evaporator temperature) and one output (superheat) has already been identified. By taking into account the number of membership functions and the constraint associated with VCRS, fuzzy base containing certain set of rules was designed from the expert knowledge and using ANFIS. This control algorithm as fuzzy rules generates geometric representation of model. This surface is known as control surface (Fig. 3) from different directions.

Results and Discussion

VCRS undertaken has been analyzed with mathematical modeling and fuzzy modeling. Evaporator temperatures (Fig. 4) and mass flow rates (Fig. 5) of refrigerant are having the reciprocal effect on the superheat at the entry of the compressor. Superheat is found to be smaller in case of fuzzy modeling analysis as compared to mathematical modeling. Lesser superheat value is always desirable in VCRSs. So new approach of fuzzy modeling is better than the mathematical modeling. It can establish more efficient control of superheat.

Conclusions

A fuzzy model has been developed to predict the thermal performance of VCRS. Comparison of fuzzy

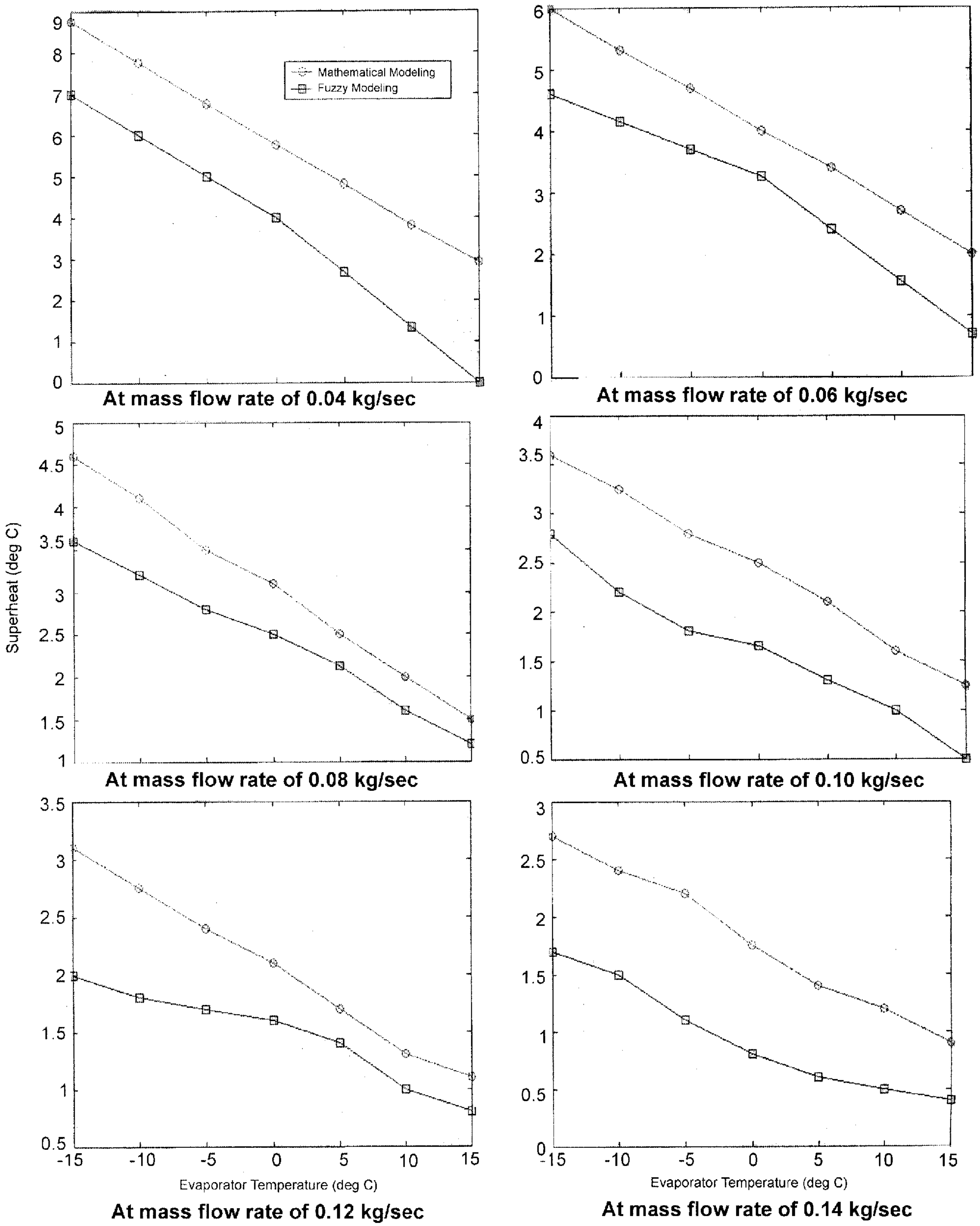

Fig. 4—Superheat versus evaporator temperature at different mass flow rates from mathematical and fuzzy modeling

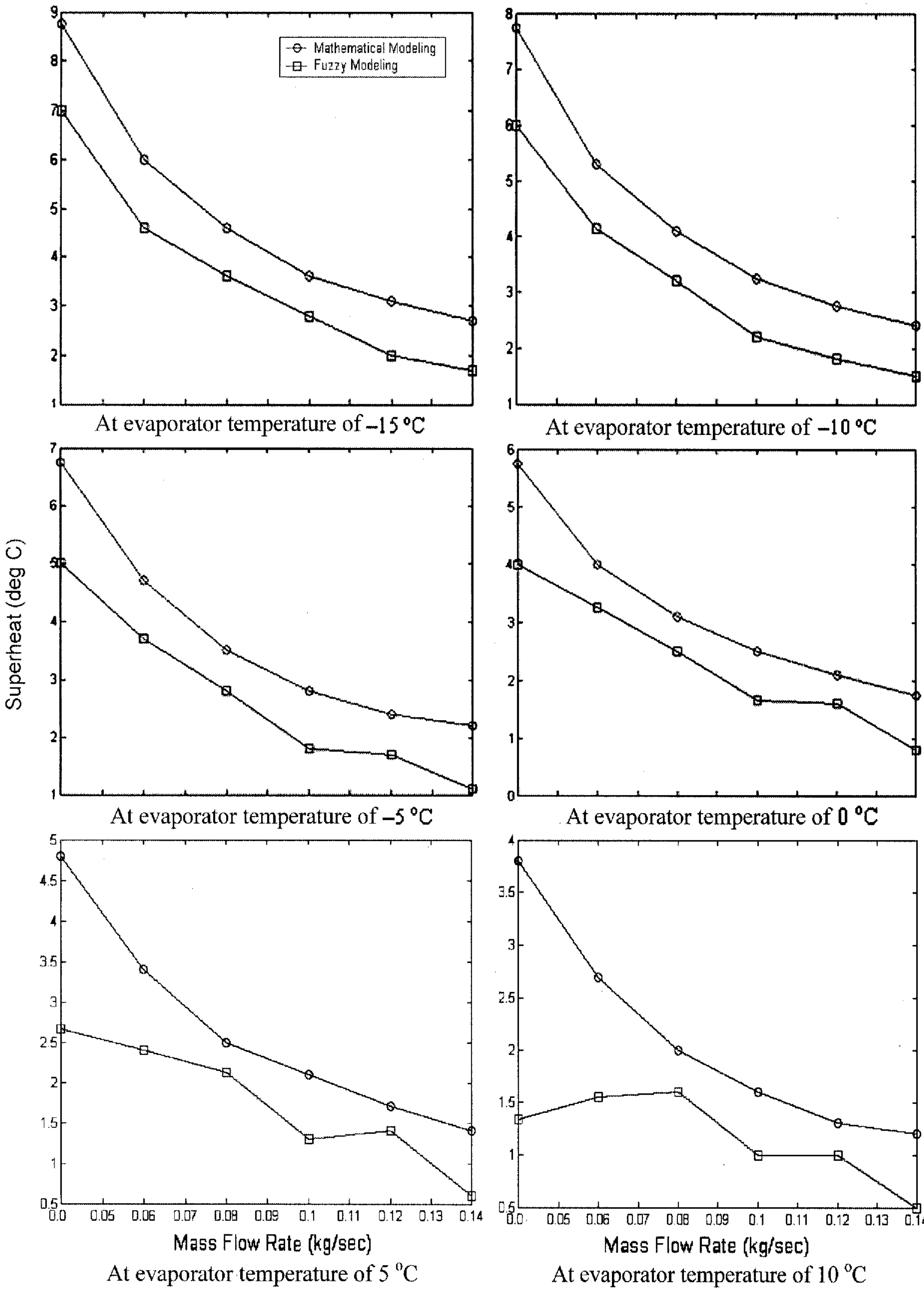

Fig. 5—Superheat versus mass flow rate at different evaporator temperatures from mathematical and fuzzy modeling

model and the mathematical model shows that fuzzy model has better precision and generalization ability. This model is giving lesser superheat value for the same range of evaporator temperature and mass flow rate. Control based on fuzzy modeling will be more efficient as it improves the superheat at the entry of compressor in VCRS. It is also evident that problems hard to be treated by mathematical modeling can be solved in this way with more ease.

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Nomenclature

A	Cross-sectional area
$ABCD$	Matrices
a	Valve opening
C_p	Constant pressure specific heat
C_v	Orifice coefficient
D	Heat exchanger tube diameter
G(s)	Model transfer function
H	Refrigerant enthalpy
J	Cost function
K(s)	Controller transfer function
K, r	Parameters in weighting matrices
L	Length of the heat exchanger
u	Refrigerant internal energy
y	Outputs
$\underline{\sigma}$	Minimum singular value
$\overline{\sigma}$	Maximum singular value
ω	Compressor speed
ρ	Refrigerant density
α	Heat transfer coefficient
$\overline{\gamma}$	Mean void fraction
l	Length of the two phase section
m	Mass flow rate
P	Pressure
QR	Weighting matrices
q	Heat transfer rate from tube wall to refrigerant
SC	Subcool
SH	Superheat
T	Temperature
t	Time
u	Control inputs
x	State variables
e	Evaporator
l	Liquid
g	Gas
i	Inlet or inside
o	Outlet or outside
r	Refrigerant
v	Valve
w	Tube wall