Multi-objective Optimization of Condensation Heat Transfer inside Horizontal Tube using Particle Swarm Optimization Technique

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Abstract--In this paper, heat transfer coefficient and frictional pressure drop during condensation of refrigerant inside a smooth tube are formulated as a multi - optimization problem and solved using particle swarm optimization (PSO) algorithm. The objective of this work is to predict the optimum values of refrigerant mass flux and vapor quality for which there is a maximum condensation heat transfer and minimum flow frictional pressure gradient inside tube. The PSO algorithm was run for mass flux and vapor quality range from 100 kg/m^2-s to 250 kg/m^2-s and 0.1 to 0.8 respectively. The optimum refrigerant mass flux and vapor quality predicted by particle swarm optimization algorithm are 134.85 kg/m^2-s and 0.79 respectively.

Keyword- Condensation, Frictional pressure gradient, Heat transfer coefficient, Particle swarm optimization

I.   INTRODUCTION

Over the years, a number of experimental works have been done on condensation heat transfer and pressure drop for different refrigerants and tube geometries. The effect of design and operating parameters on heat transfer coefficient and pressure has been investigated by researchers. Heat transfer coefficient and pressure drop were determined using several empirical correlations. A little attention has been given on the optimization of heat transfer and pressure drop during the condensation of the refrigerant inside plain tube.

The heat transfer coefficient and pressure drop depend on many factors such as mass flux, vapour quality and properties of refrigerant, tube's geometry and orientation. The heat transfer coefficient and pressure drop are found to increase with the increasing mass flux and vapor quality of refrigerant. The condensation heat transfer inside tube is accomplished by pump work. The pump work is found to increase with pressure drop along the tube during condensation. The design and operating parameters are required to be optimized to achieve maximum heat transfer and minimum pressure drop.

Existing literature reveals that many researchers have used different optimization methods for the design optimization of heat exchangers. Artificial neural network (ANN) and genetic algorithm (GA) used to develop the empirical correlations for heat transfer characteristics during the downward flow of R-134a inside smooth and corrugated tubes [1, 2]. Heat transfer rate and pressure drop in shell and tube heat exchanger was optimized using genetic algorithm [3]. Genetic and particle swarm optimization (PSO) algorithms were employed for thermo economic optimization of shell and tube condenser [4]. Teaching-learning based optimization (TLBO) technique was used to optimize the cost, weight, effective of plate fin heat exchanger [5]. Effectiveness and cost optimization of the plate fin exchanger was performed using modified teaching-learning based optimization technique [6]. Heat transfer coefficient optimization during condensation of refrigerant inside smooth horizontal tube was done through TLBO [7].

The objective of the present paper is to predict the optimum value of refrigerant mass flux and vapor quality for which there is maximum heat transfer coefficient and minimum pressure drop during the condensation of R-245fa inside the plain horizontal tube using particle swarm optimization method.

II. PARTICLE SWARM OPTIMIZATION TECHNIUQE (PSO)

Particle swarm optimization (PSO) is a population based evolutionary optimization method proposed by Kennedy [8]. Figure 1 shows the flow chart of PSO. In particle swarm optimization algorithm the population is randomly initialized and moves in randomly selected directions through the search space and keeps record of the best previous position of itself and its neighbors as well. The population updates after each generation based on the best location achieved by a particle, Pbest, and other best location achieved by any other particle among the population, gbest, so far.

In particle swarm optimization the population is updated after each generation by changing the velocity and position of each particle towards Pbest and gbest. The velocity and position are updated according to equations 1 and 2.

$$v_i^{t+1} = w v_i^t + c_1 \, rand_1 \, [gbest_i^t - x_i^t] + c_2 \, rand_2 \, [gbest_i^t - x_i^t] \quad (1)$$

$$x_i^{t+1} = x_i^t + v_i^{t+1} \quad (2)$$

Where, i = population index

t = generation

x_i^t = position of ith particle.

v_i^t = velocity of ith particle.

$Pbest_i^t$ = best previous position of particle.

$gbest_i^t$ = the best particle among all particles in a swarm.

$rand_1$, $rand_2$ are two different random numbers in the range [0 1]

Local and global search is balanced by inertia weight 'w'. The high inertia weights permit travelling the large design space while small inertia weights permit travelling only nearby region of the design space by particles. The value of inertia weights is considered in the range [0.4 0.9]. The optimum value of inertia weights improves the performance of the algorithm. The acceleration coefficients c_1 and c_2 are recognized as cognitive and social parameters. The cognitive and social parameters pull each particle towards Pbest and gbest respectively. Small value of constants allows particles to travel far away from the target regions before being pulled back, while high value of constants results in sudden movement towards the target region [9]. Bergh F and Engelbrecht [10] have reported effect of c_1, c_2, and w, on various standard functions. In the present paper, particle swarm optimization algorithm was run considering the following parameters:

Number of particles = 50

Number if iteration = 100

Cognitive parameter (c_1) = 2

Social parameter (c_2) = 2

Inertia weights range [0.9 0.4]

III. PROBLEM FORMULATION FOR MULTI OBJECTIVE OPTIMIZATION

For optimum performance of the condenser, the heat transfer coefficient and pressure drop along the flow are required to be optimized. Multi-objective optimization of maximization and minimization functions together can be done considering a single objective function, as represented by Eq. 3.

$$\max f(z) = \frac{\omega \times f_1(z)}{f_1^*} - \frac{(1-\omega) \times f_2(z)}{f_2^*} \quad (3)$$

Where, combined objective function f (z) is a function of two functions $f_1(z)$ and $f_2(z)$ to bemaximized and minimized respectively. f_1^* , f_2^* are the maximum and minimum values of $f_1(z)$ and $f_2(z)$ when solved, as a single objective problem separately. Here ω is the weight assigned to objective functions.
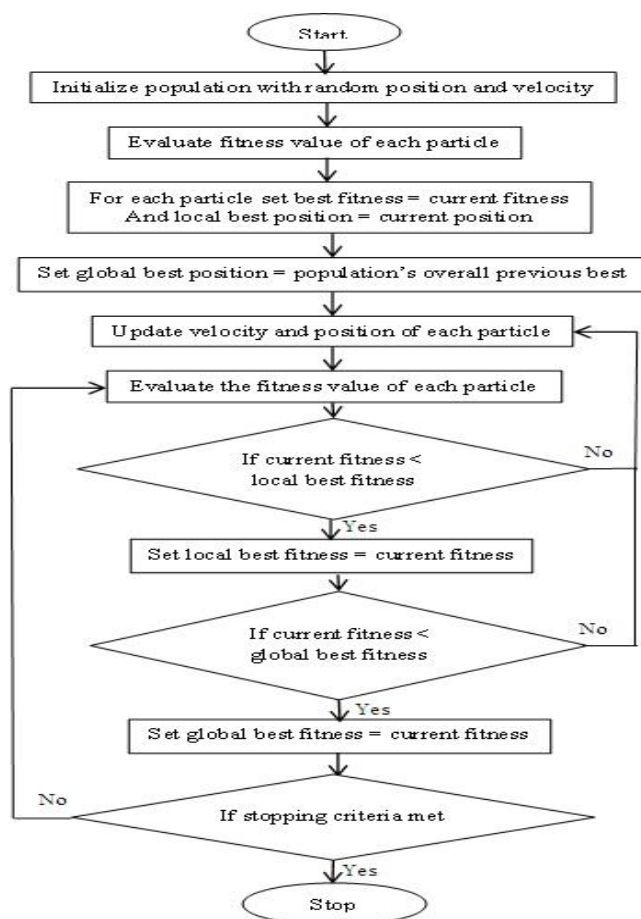
Fig. 1. Flow chart of particle swarm optimization algorithm.

IV. EXPERIMENTAL SYSTEM

The particle swarm optimization algorithm results were validated with experimental data obtained from the experimental system shown in Figure 2. The experimental system is a concentric double pipe heat exchanger having two test sections of length one meter each. The inner and outer diameters of inner tube are 9.4 mm, 12.76 mmand that of the outer tube are 43 mm, 50 mm. The experiments were carried out according to the parameters shown in Table 1. A multichannel data acquisition system was incorporated with experimental system for data recording. R-245fa vapor and cooling water were flown in counter direction through the inner and outer tubes inside the test sections. The refrigerant vapor coming from the test section 2 flows through the post condenser where the entire amount of refrigerant vapour converted into liquid. This entire liquid refrigerant flows through the evaporator, made of stainless steel tube having inside diameter, outside diameter and length 16 mm, 17.5 mm and 3.6 m respectively. A step-down transformer incorporated with evaporator controls the vapor quality of refrigerant. A corioles mass flow meter was connected to control the refrigerant mass flow rate. Four T-type thermocouples were fixed at each six axial locations to quantity the accurate temperature of outer wall of the inner tube. The refrigerant pressure at the inlet and outlet of each test section was measured by pressure gauge. The pressure difference across the test section was measured using a pressure transducer. Condenser pressure was taken as the average refrigerant pressure of both test sections. Refrigerant properties were taken according to average condenser pressure.
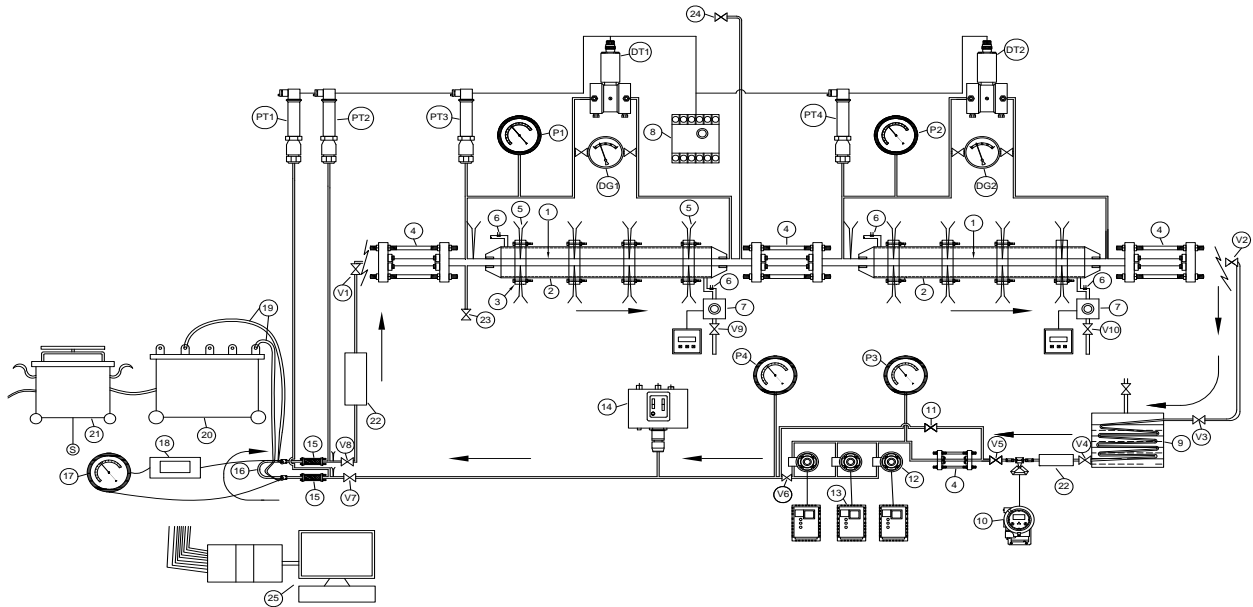
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Fig. 2. Schematic diagram of experimental system.

1	Test-section	9	Post-condenser	17	Voltmeter
2	Outer tube	10	Coriolis mass flow meter	18	Power analyzer
3	Flanged joint	11	By-pass valve	19	Cable
4	Visual section	12	Pump	20	Transformer
5	Thermocouple	13	Frequency controller	21	Variac
6	Thermopile	14	HP cut-out	22	Filter
7	Turbine flow- meter	15	Electrical isolator	23	Charging valve
8	24 V DC power supply	16	Evaporator	24	Purge valve

25 Data acquisition system
P Pressure gauge
DG differential pressure
DT pressure -transducer

Table 1. Operating parameters

Parameters	Value
Refrigerant mass flux (kg/m^{2}s)	100 - 250
Heat flux (kW/m^2)	7.5 – 20.5
Condensing pressure (kPa)	220
Condensing temperature (^{0}C)	35.95
Vapor quality	0.1 – 0.8

V. DATA COLLECTION AND DATA REDUCTION

The experimentation was carryout for the condensation of R-245fa with refrigerant mass fluxes 100, 150, 200 and 250 kg/m^2-s and vapor quality range from 0.1 to 0.8. The vapor quality at entry and exit of each test section was calculated through energy balance along the test condenser and evaporator. The test section vapor quality was considered as the average of its entry and exit values.

The heat transfer coefficient of each test section was calculated by applying energy balance between refrigerant vapor and water using Eq. 4. The overall heat transfer coefficient for each test run was considered as the average of two test sections.

$$h = \left[\frac{\Pi D_i \, l \, (T_s\text{-}T_{wo})}{m_w C_{pw}(T_{c,o}\text{-}T_{c,in})} - \frac{D_i \, \ln(D_o/D_i)}{2 \, k_w} \right]^{-1} \quad (4)$$

The two phase flow frictional pressure drop during condensation of R-245fa inside a plain horizontal tube was calculated by Eq. 5.

$$\Delta P_{fri} = \Delta P_{tot} - \Delta P_{mom} \quad (5)$$

Where ΔP_{fri}, ΔP_{tot} and ΔP_{mom} are frictional pressure drop, total pressure drop measured by pressure gauge and momentum pressure drop respectively. The calculation of the momentum pressure drop was made according to Eq.6.

$$\Delta P_{mom} = G^2 \left\{ \left[\frac{(1-x)^2}{\rho_L(1-\alpha)} + \frac{x^2}{\rho_G \alpha} \right]_{out} - \left[\frac{(1-x)^2}{\rho_L(1-\alpha)} + \frac{x^2}{\rho_G \alpha} \right]_{in} \right\} (6)$$

Here 'α' is the void fraction calculated as suggested by Steiner.

VI. RESULTS AND DISCUSSION

A. Heat transfer coefficient

Figure 3 represents the effect of refrigerant mass flux and vapor quality on the heat transfer coefficient during condensation of R-245fa inside a plain horizontal tube. As could be seen from the figure, heat transfer coefficient increases with rise in vapor quality and mass flux. The maximum heat transfer coefficient 3.27kW/m^{2}K was reported at mass flux 250 kg/m^{2}s and vapor quality 0.8. The rise in heat transfer coefficient is due to high turbulence induced in the condensate film at high mass flux and low thermal resistance offered to condensation heat transfer coefficient by thin vapor film on tube wall at high vapour quality [11, 12]. Experimental data of heat transfer coefficient were compared with several well-known correlations to predict the heat transfer coefficient during condensation of R-245fa inside a smooth horizontal tube. The Dobson's correlation [13] predicted the present heat transfer coefficient data within an error range of ± 20% as shown in Figure 4. The Dobson's correlation of heat transfer coefficient for plain flow inside a horizontal tube is given by Eq. 7.

$$h = 0.023 \times Re^{0.8} \times Pr^{0.4} \times \left[\frac{k}{D_i} \right] \times \left[1 + \frac{2.22}{X_{tt}^{0.89}} \right] (7)$$

Where Xtt is known Martinelli parameter computed as according to equation 8.

$$X_{tt} = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_G}{\rho_L} \right)^{0.5} \left(\frac{\mu_L}{\mu_G} \right)^{0.1} (8)$$
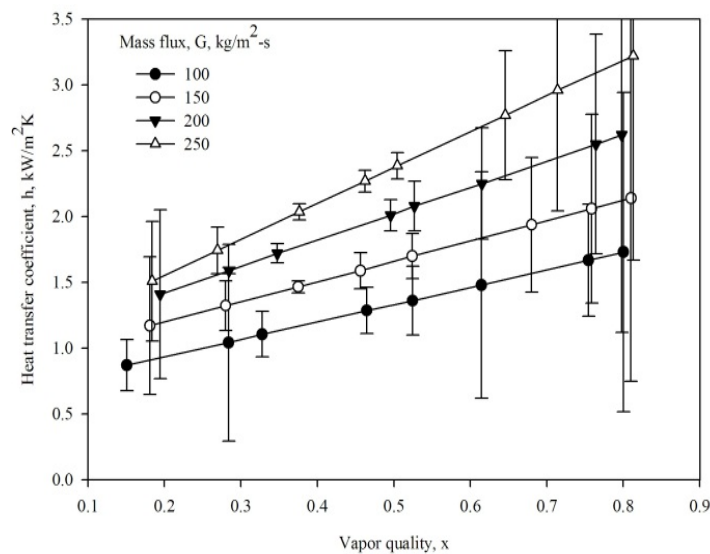

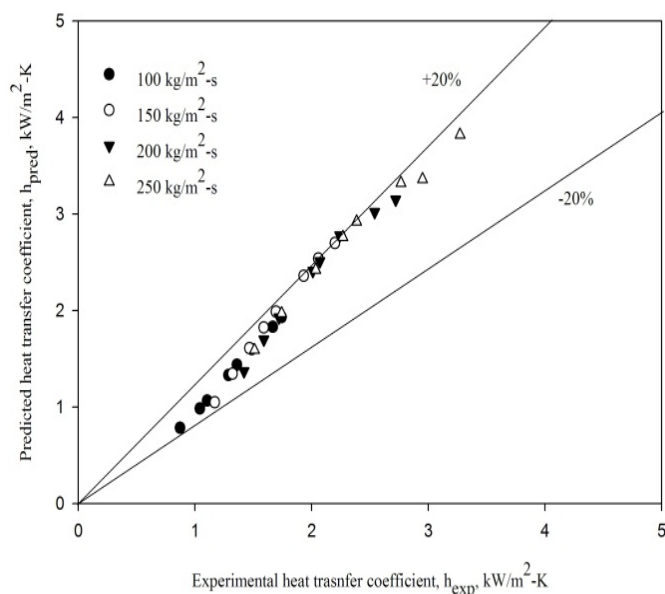Fig. 3. Effect of mass flux and vapor quality on the heat transfer coefficient.

Fig. 4. Comparison of experimental heat transfer coefficient with that predicted by Dobson's correlation.

B. *Frictional pressure drop*

Figure 5 shows the effect of refrigerant mass flux and vapor quality on the frictional pressure drop along the flow during condensation of R-245fa inside aplain horizontal tube. As could be seen from the figure, frictional pressure drop increases with mass flux and vapor quality of refrigerant. Increasing mass flux produces greater two-phase velocity resulting in a greater frictional pressure drop. This effect is more evident in the high vapor quality region where the pattern changes into fully developed annular region flow. The increase in frictional pressure drop with vapor quality at any mass flux is due to changes in flow pattern with increasing vapor quality.
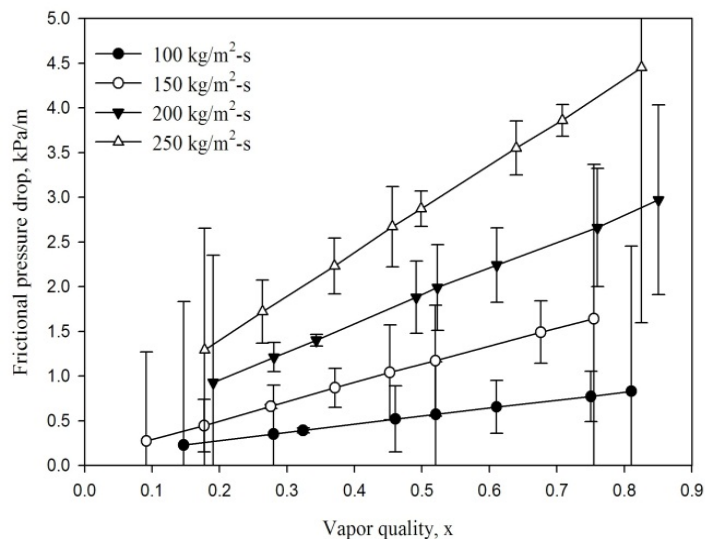

Fig. 5. Variation in frictional pressure drop with mass flux and vapor quality.

The present frictional pressure drop data were compared with some well-known two phase flow frictional pressure drop correlations [14, 15, 16]. Fig. 6 represents the comparison of experimental frictional pressure drop with that of predicted by above mentioned correlations. As could be observed from the figure all correlations predicted the data beyond an error range of +15%.
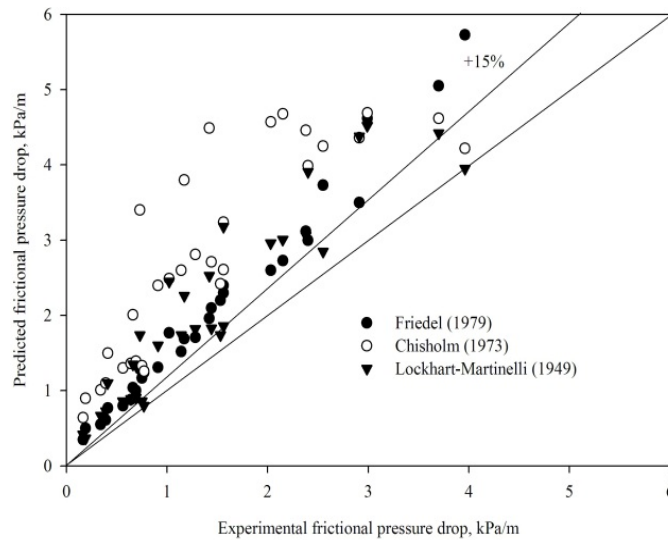
Fig. 6. Comparison of experimental frictional with different correlations.

The present experimental data were used to develop an empirical correlation based on Cavallini [17] to predict frictional pressure drop during condensation of R-245fa inside plain horizontal tube. The Eq. 9 is the modified correlation of frictional pressure drop for condensation inside a smooth horizontal tube. The experimental data predicted by Eq. 9 are within an error band of -5 % to +15%, as shown in Figure 7.

$$\left(\frac{dp}{dz}\right)_f = \varphi^2{}_{LO} \times 2 \times f_{LO} \times \frac{G^2}{D_i \times \rho_L} \ (9)$$

Where,

$$f_{LO} = 0.034 \times \left(\frac{G \times D_i}{\mu_l}\right)^{-0.25} (10)$$

$$\varphi^2{}_{LO} = Z + 3.595 \times F \times H \ (11)$$

$$Z = (1-x)^2 + x^2 \times \left(\frac{\rho_L}{\rho_G}\right) \times \left(\frac{\mu_G}{\mu_L}\right)^{0.2} (12)$$

$$F = x^{0.9525} \times (1-x)^{0.414} \ (13)$$

$$H = \left(\frac{\rho_L}{\rho_G}\right)^{1.132} \times \left(\frac{\mu_G}{\mu_L}\right)^{0.44} \times \left(1 - \frac{\mu_G}{\mu_L}\right)^{3.542} (14)$$
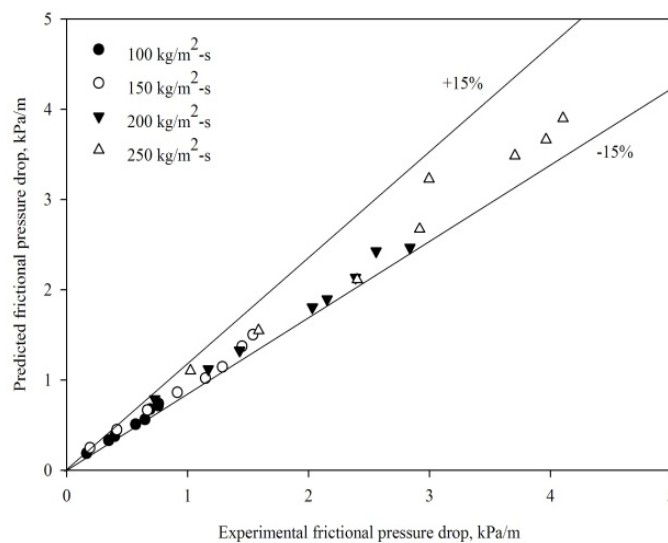

Fig. 7. Comparison of experimental frictional pressure drop with that predicted by Eq. 9

C. Validation of particle swarm optimization algorithm (PSO)

The particle swarm optimization algorithm was executed for the maximization of heat transfer coefficient during condensation of R-245fa inside a smooth horizontal tube. The Eq. 7 was considered as a maximization objective function. The algorithm was run initially executed at each mass flux 100, 150, 200, and 250 kg/m^{2}s. Vapor quality of refrigerant was varied from 0.1 to 0.8 for each mass flux. The optimized results were compared with experimental, as shown in Table 2. The PSO algorithm was again run for the same maximization function with mass flux and vapor quality varying between 100 to 250 kg/m^{2}s and 0.1 to 0.8 respectively. Table 3 shows the comparison of PSO predicted and experimental. As could be observed from tables 3 and 4, PSO predicted values are very close to the experimental.

Table 2. Comparison between PSO and experimental results.

Mass flux (kg/m^{2}s)	Vapour quality		Heat transfer coefficient (kW/m^{2}K)	
	Experimental	PSO	Experimental	PSO
100	0.8	0.792	1.75	1.89
150	0.8	0.797	2.15	2.42
200	0.8	0.785	2.54	2.92
250	0.8	0.796	3.27	3.66

Table 3. Comparison of optimized results

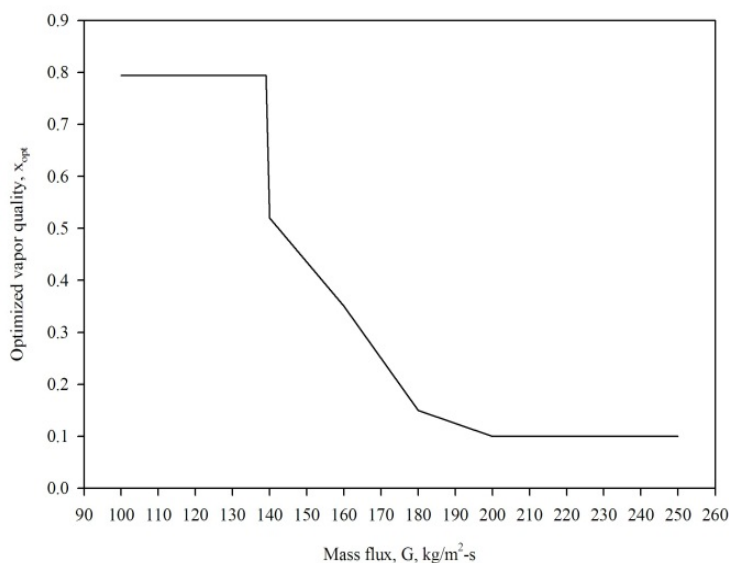| Parameters	Experimental	PSO
Mass flux of R-245fa (kg/m^{2}s)	250	248.21
Vapor quality	0.8	0.775
Optimum heat transfer coefficient (kW/m^{2}K)	3.27	3.75

Fig. 8. Variation in optimum vapor quality with mass flux.

D. Optimization of heat transfer coefficient and frictional pressure drop

A single multi-objective optimization problem was formulated as according to Eq. 3 for the maximization of heat transfer coefficient and minimization of fractional pressure drop. The PSO algorithm was initially for multi-objective optimization problem at constant refrigerant mass flux between 100 kg/m^{2}s to 250 kg/m^{2}s with step size 1 and vapor quality was varied from 0.1 to 0.8 for each mass flux. Fig. 8 represents the optimized refrigerant vapor quality for each mass flux. As could be seen, up to the mass flux 139 kg/m^{2}s optimized vapor quality predicted by PSO is 0.794. And as the mass flux increased beyond 139 kg/m^{2}s optimized vapor quality went on decreasing and became constant (0.1) on and after 200 kg/m^{2}s. The algorithm was run again considering the mass flux and vapour quality as variables. The mass flux and vapour quality were varied from

100 to 250 kg/m^{2}s and 0.1 to 0.8 respectively. The optimum refrigerant mass flux and vapor quality predicted by PSO was 134.85 kg/m^2-s and 0.79 respectively.

VII. CONCLUSIONS

On the basis of above results following inferences may be drawn pertaining to the condensation heat transfer and frictional pressure drop during condensation of R-245fa inside a smooth horizontal tube and their optimization using particle swarm optimization technique.

1. Refrigerant mass flux and vapor quality have greater influence on the heat transfer coefficient and frictional pressure drop during condensation. Heat transfer coefficient and frictional pressure drop along the flow inside tube increase with increasing mass flux and vapor quality.

2. The particle swarm optimization technique was effectively applied to multi-objective optimization of heat transfer coefficient and frictional pressure drop. Refrigerant mass flux below 140 kg/m^{2}s can provide maximum heat transfer and minimum pressured drop during condensation inside plain tubes. The values of optimum mass flux and vapor quality predicted by particle swarm optimization are134.85 kg/m^{2}s and 0.79 respectively.

3. The particle swarm optimization algorithm may also be used for the multi-objective optimization of design and operating parameters during swirling flow condensation of refrigerants inside tubes.

LIST OF SYMBLOS

C$_P$Specific heat(kJ / kg K)

DTube diameter (mm)

G   Mass flux (kg/m^2.s)

h   Heat transfer coefficient (kW / m^2 K)

kThermal conductivity (W / m K)

x   Vapor quality

μViscosity (μPa-s)

ρDensity (kg/m^3)

DIMENSIONLESS NUMBERS

Re   Reynolds number

Pr   Prandtl number

Xtt Martinelli parameter

SUBSCRIPT

L   liquid

G   gas

LO   liquid only

B   bottom

R   right

i   inner

o   outer

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