International Refereed Journal of Engineering and Science (IRJES)
ISSN (Online) 2319-183X, (Print) 2319-1821
Volume 3, Issue 1 (January 2014), PP. 29-31

Thermal performance analysis of cross-flow unmixed-unmixed heat exchanger by the variation of inlet condition of hot fluid

Dr. M K Chopra[1], Ramjee Singh Prajapati[2]
[1]*Dean and HOD Mechanical Engineering Department RKDF Bhopal (M.P.)*
[2]*PG Student Mechanical Engineering Department RKDF Bhopal (M.P.)*

Abstract:- In this paper the thermal performance analysis of a cross flow unmixed heat exchanger performed by the variation of volume fraction of hot fluid, which is a mixture of water and coolant. An experimental analysis has been performed on unmixed cross flow heat exchanger which is automobile radiator using air and water-coolant as fluids. Where air serves as cold fluid and water-coolant serves as a hot fluid. Automobile radiator was connected with variable speed petrol engine to vary the mass flow rate of air. Also the volume fraction of coolant varies from 0-60% for changing the mass flow rate of water-coolant mixture. Experimental results such as heat exchanger effectiveness, overall heat transfer coefficient, heat transfer rates have been calculated for assessing the performance of heat exchanger, for taking all readings i.e. mass flow rate of air and temperature of hot fluid we are using a anemometer and thermocouple respectively. The objective of this project is too determined at what percentage of water-coolant mixture the performance of unmixed cross flow heat exchanger deteriorates and at what percentage we obtain maximum heat exchange rate.

I. INTRODUCTION

Fin-and-tube heat exchangers have been employed in industrial application such as car radiator, air conditioning, refrigeration, cryogenics, heat recovery, marine and boiler economizer, as well as in many products available in the marketplace. One of the main problems of heat exchangers is the prediction of the transient behavior which occurs during start up and shut down or during non-stationary function. The control of systems incorporating heat exchangers is necessary to understand the response of the latter to the variations of flow rates and entering fluid temperatures.

The analysis for the response characterization and the effectiveness of heat exchanger is made for water-coolant mixture, in which we changing the amount of coolant in the mixture, for the safe operation of heat exchangers under steady and transient conditions.

Over the years, several articles and books on the design of fin-plate heat exchangers have appeared in the literatures. A comprehensive review of solution methods for determining effectiveness (e or P)–number of transfer units (NTU) relationships for two-fluid heat exchangers

With simple and complex flow arrangements is presented by Sekulic et al. [2]. The methods were categorized by the authors as: analytical methods for obtaining exact solutions, approximate methods, curve-fit to the results from the exact solutions, numerical methods, matrix formalism, and methods based on exchanger configuration properties, as the use of flow reversal symmetry of exchanger configurations. In conformity to the authors continuing efforts to design more efficient systems, more compact exchangers, or specific operating conditions may require effectiveness–NTU formulae for a new heat exchanger, not reported in the literature. Using some of these methods Pignotti and Shah [3] obtained effectiveness–NTU explicit formulas for new arrangement.

Nomenclature

A	Area	R	Heat capacity ratio
C	Heat capacity rate	T	Absolute temperature
C_h	Heat capacity rate of hot fluid	$T_{h,I}$	Temperature of hot fluid at inlet
C_c	Heat capacity rate of cold fluid	$T_{h,o}$	Temperature of hot fluid at outlet
C_{max}	Maximum heat capacity rate	$T_{c,i}$	Temperature of cold fluid at inlet
C_{min}	Minimum heat capacity rate	$T_{c,o}$	Temperature of cold fluid at outlet
C_p	Specific heat at constant pressure	ΔT	Local temperature difference
h	Specific enthalpy	ΔT_m	Mean temperature difference
NTU	Number of exchanger heat transfer unit	U	overall heat transfer coefficient
q	Heat transfer rate	U_m	Mean overall heat transfer coefficient
q_{max}	Maximum heat transfer rate	W	Mass flow rate of hot fluid
q''	Heat flux, Heat transfer rate per unit surface area	w	Mass flow rate of cold fluid
		x	Cartesian coordinate along x direction

Greek Notations

θ	Temperature difference	ε	Heat exchanger effectiveness
θ_1	Temperature difference on side 1	δ	Denotes difference
θ_2	Temperature difference on side 2		
θ_m	Log mean temperature difference		

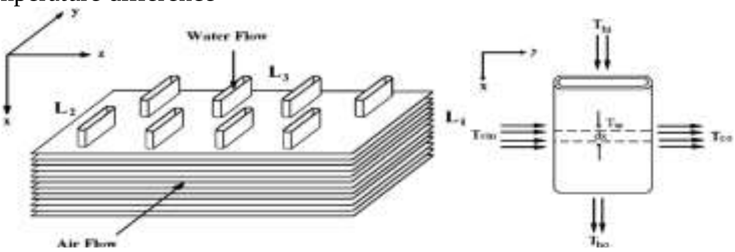

Fig 1 Schematic diagram of fin and tube (unmixed) heat exchanger

II. METHODOLOGY DEVELOPMENT

Governing Equation

The configuration of fin and tube heat exchanger used in analysis is shown in figure 1 in which the air is working as cold fluid and water coolant mixture is working as hot fluid. The governing equations presented in these sections are those developed for a cross flow heat exchanger. These are the basic equation applied for the numerical methodology and having assumptions that fluid is flow one dimensional and unsteady, there is no phase change in exchanger and axial conduction of fluid is neglected. It is considered that the tube side fluid is perfectly mixed in the tube cross section and external fluid is perfectly unmixed. The energy balance equation for hot and cold fluid is written as

$$dq = q'' \, dA = -C_h \, \Delta T_h = - C_c \, \Delta T_c \qquad \text{Eq. (1)}$$

The negative sign in this equation are a result of T_h & T_c decreasing with A(these temperature decreases with increases flow length); also, dq is the heat transfer rate from hot to cold fluid. In general for any isobaric change of state Eq. 1 should be replaced by

$$dq = -W \, dh_h = -w \, dh_c \qquad \text{Eq. (2)}$$

Here if the change of state is phase change, enthalpy differences should be replaced by enthalpies of phase changes either enthalpy of evaporation or enthalpy of condensation. The overall heat transfer rate equation on differential base for surface area dA is

$$dq = U \, (T_h - T_c)_{local} \, dA = U \, \Delta T \, dA \qquad \text{Eq. (3)}$$

Integrating equation 1^{st} and 2^{nd} together over the entire heat exchanger surface for specified inlet temperature will result in an expression that will relate all important operating variables and geometric parameter of heat exchanger.

$$q = \int C \, dT = C_h \, (T_{h,i} - T_{h,o}) = C_c \, (T_{c,o} - T_{c,i}) \qquad \text{Eq. (4)}$$

$$q = \int U \, \Delta T \, dA = U_m A \, \Delta T_m \qquad \text{Eq. (5)}$$

Numerical solution methodology

The proposed work is based on the governing equations used in this paper. First the mass flow rate and the temperature is measured with the help of anemometer and thermocouple respectively. Also find the temperature of water coolant mixture at the inlet and outlet of the radiator. Now, by varying the volume fraction of coolant in mixture determine the parameters again. Repeat this experiment until the volume fraction of coolant reaches upto 60% of the total water coolant mixture

NTU Effectiveness method

Due to limited access to experimental equipment and related references, we utilized ε-NTU method to compare the numerical solutions obtained in steady state with the results of this analytical method. The effectiveness (ε) of a heat exchanger is defined as the ratio of the actual heat transfer to the maximum heat transfer that could be theoretically obtained in the heat exchanger without any size limitations.

The maximum heat transfer rate can be expressed by

$$q_{max} = C_{min} \, (T_{hi} - T_{ci}) \qquad \text{Eq. (6)}$$

The effectiveness of the heat exchanger is as follows

$$\varepsilon = \frac{q_{act}}{q_{max}} \qquad \text{Eq. (7)}$$

Where q_{act} is the actual heat transfer rate in the heat exchanger. Therefore, for a known value of the effectiveness of a heat exchanger, the actual heat transfer rate can be determined as follows

$$q_{act} = \varepsilon \, C_{min} \, (T_{hi} - T_{ci}) \qquad \text{Eq.(8)}$$

The dimensionless parameter NTU (the number of transfer units), to specify the heat transfer characteristics of a heat exchanger, is defined as

$$\mathrm{NTU} = \frac{\mathrm{UA}}{\mathrm{C_{min}}} \qquad \text{Eq. (9)}$$

Where U is the overall heat transfer coefficient, A is the area of heat exchanger and

Here the effectiveness of heat exchanger is related to NTU as follows:

$$\varepsilon = \frac{1-\exp[-\mathrm{NTU}\ (1-\mathrm{R})]}{1-\mathrm{R}\exp[-\mathrm{NTU}\ (1-\mathrm{R})]} \qquad \text{Eq. 10}$$

III. RESULT AND DISCUSSION

- The experimental results in this work are presented in form of table 1.

Fluid temperature		Mass flow rate	Fluid temperature		Mass flow rate	q	ε
T_{ci}(°C)	T_{co}(°C)	w (kg/sec)	T_{hi} (°C)	T_{ho}(°C)	W(kg/sec)	kw	%
27	51	.279	113	105	.199	6.696	27.9
27	52.5	.333	113	104.5	.245	8.492	29.7
27	53.6	.386	113	104.1	.291	10.268	30.9
27	54.5	.466	113	103.8	.363	12.815	31.9
27	55.7	.519	113	103.4	.417	14.895	33.4
27	56.3	.599	113	103.2	.497	17.551	34.1
27	52.1	.626	113	104.5	.531	15.713	28.7

Table :1Experimental results

- By increasing the flow rate of cold fluid and increase in composition of coolant, it was found that the overall heat transfer coefficient was also increased**.**
- Similarly also found that when mass flow rate of cold fluid increases, the cold fluid outlet temperature will increases and when mass flow rate of hot fluid increases , the hot fluid outlet temperature will be decreases.

IV. CONCLUSION

Experiment was conducted on unmixed-unmixed cross flow heat exchanger with different flow rate and different composition of hot fluid. The effect of these parameters on outlet temperatures and overall heat transfer coefficient were studied. It was found that as the mass flow rate of cold fluid increases the cold fluid outlet temperature decreases, the mass flow rate of hot fluid increases with the hot fluid outlet temperature decreases. The performance of unmixed-unmixed heat exchanger deteriorates when percentage of coolant in water coolant mixture exceeds 50% because viscosity will be excessive the circulation of fluid will affect and heat transfer rate decreases. The graph, effectiveness as a function of NTU and heat capacity ratio verified numerically.

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