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Performance Analysis and Optimal Design of Heat Exchangers Used in High Temperature and High Pressure System

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Abstract

A computational study for the optimal design of heat exchangers (HX) used in a high temperature and high pressure system is presented. Two types of air to air HX are considered in this study. One is a single-pass cross-flow type with straight plain tubes and the other is a two-pass cross-counter flow type with plain U-tubes. These two types of HX have the staggered arrangement of tubes. The design models are formulated using the number of transfer units (ε - NTU method) and optimized using a genetic algorithm. In order to design compact light weight HX with the minimum pressure loss and the maximum heat exchange rate, the weight of HX core is chosen as the object function. Dimensions and tube pitch ratio of a HX are used as design variables. Demanded performance such as the pressure loss (Δ P) and the temperature drop (Δ T) are used as constraints. The performance of HX is discussed and their optimal designs are presented with an investigation of the effect of design variables and constraints.

Key Words : Heat exchanger, Optimal design

Introduction

As the importance of global environmental problems is growing, efficient energy management becomes an urgent target in science and technology. In further detail, ACARE(Advisory Council for Aerospace Research in Europe) demands that air transportations must have the environment-friendly aero engine in which the amount of CO_2 and NO_x emissions should be 50% and 80%, respectively less than the present amounts [1] [2] [3].

HX is one of the major components common in a wide variety of thermal energy handling processes, such as conversion, transport, consumption and storage. Improvement of HX performance affects both directly and indirectly the performance of various devices and systems. Especially in the aerospace industries, these environmental issues and airlines require gas turbine manufacturers to produce environmentally friendly gas-turbine engines with lower emissions and improved specific fuel consumption. These requirements can be met by incorporating HX into gas turbines for intercooling and recuperation [4] [5] [6].

In order to satisfy this goal, the next-generation aero-engine should adopt a regeneration system with HX that compact and ultra light weight, high effectiveness, minimum pressure loss to maintain performance benefit, very high pressure & temperature capability, structural integrity to cope with large temperature difference, and low cost are required.

Hence, the object of this work is an optimal design and a performance analysis of high-performance HX used in a high temperature and high pressure system.

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Design Procedures

1. The Type of Heat Exchangers

In this work, we consider two types of air-to-air HX. One is a single-pass cross-flow type with straight plain tubes and the other is a two-pass cross-counter flow type with plain U-tubes. The core of HX consists of plain tubes with staggered arrangement, which is shown schematically in Fig. 1 and Fig. 2 where W, H and B are length, height and width of HX, respectively.

For HX performance analysis, a HX rating program has been developed for the thermal analysis of HX using the ε-NTU method. When inlet temperatures are given, ε-NTU method is more suitable than the LMTD method. To calculate the total heat transfer rate of the HX, the heat transfer correlations incorporated into the rating program are as follows;

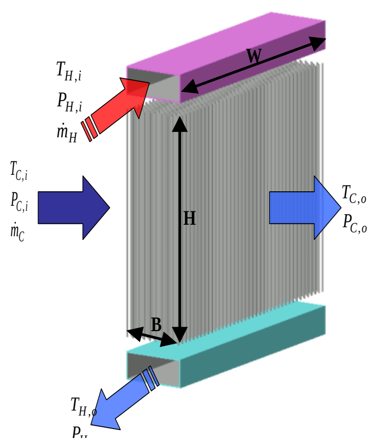

Fig. 1. Straight tube type heat exchanger

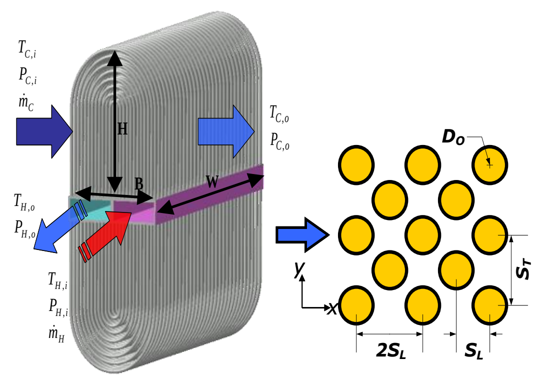

Fig. 2. U-tube type heat exchanger

Table 1. Equations for HX performance analysis of straight type HX

	Equation	
Heat Transfer	$h_H = Nu_H \dfrac{k}{D_i}$	(1)
	For laminar flow, $Nu_H = 3.66 + \dfrac{0.19[\mathrm{Re}\,\mathrm{Pr}(D_i/H)]^{0.8}}{1+0.117[\mathrm{Re}\,\mathrm{Pr}(D_i/H)]^{0.467}}$	(2)
	For turbulent flow, $Nu_H = \dfrac{(f/8)(\mathrm{Re}-1000)\mathrm{Pr}}{1+12.7\sqrt{f/8}\,(\mathrm{Pr}^{2/3}-1)}\left[1+\left(\dfrac{D_i}{H}\right)^{2/3}\right]$ where $f = (1.82\log_{10}\mathrm{Re}-1.64)^{-2}$	(3)
	$Nu_C = 1.04\mathrm{Re}_D^{0.4}\mathrm{Pr}^{0.36}\left(\dfrac{\mathrm{Pr}}{\mathrm{Pr}_w}\right)^{0.25}$ $\quad 1 \le \mathrm{Re} < 5\times10^2$ $Nu_C = 0.71\mathrm{Re}_D^{0.5}\mathrm{Pr}^{0.36}\left(\dfrac{\mathrm{Pr}}{\mathrm{Pr}_w}\right)^{0.25}$ $\quad 5\times10^2 \le \mathrm{Re} < 10^3$ $Nu_C = 0.35\left(\dfrac{X_t}{X_l}\right)^{0.2}\mathrm{Re}_D^{0.6}\mathrm{Pr}^{0.36}\left(\dfrac{\mathrm{Pr}}{\mathrm{Pr}_w}\right)^{0.25}$ $\quad 10^3 \le \mathrm{Re} < 2\times10^5$ $Nu_C = 0.031\left(\dfrac{X_t}{X_l}\right)^{0.2}\mathrm{Re}_D^{0.8}\mathrm{Pr}^{0.36}\left(\dfrac{\mathrm{Pr}}{\mathrm{Pr}_w}\right)^{0.25}$ $\quad 2\times10^5 \le \mathrm{Re} < 2\times10^6$	(4)
Pressure Loss	$\Delta P_H = \dfrac{1}{2}(\dot{m}_h)^2\left[v_i(K_c+1-\sigma^2)+2(v_o-v_i)+f\dfrac{H}{D_i}v_m - v_o(1-\sigma^2-K_e)\right]$, $\sigma = \dfrac{A_t}{A_h}$, $\dot{m}_h = \dfrac{\dot{M}_h}{A_t}$	(5)
	$f = \dfrac{64}{\mathrm{Re}}$ for laminar, $\quad f = (1.82\log_{10}\mathrm{Re}-1.64)^{-2}$ for turbulent	(6)
	$\Delta P_C = \mathrm{Eu}\dfrac{1}{2\rho}\dot{m}^2 z$	(7)

2. Heat Exchange Calculation for Straight Tube

In order to calculate heat exchange of HX, heat transfer coefficient of the flow inside tubes and the flow across tube bank are estimated by using empirical formulae. Table 1 shows equations for HX performance analysis of straight tube type HX.

– Hot–side (flow inside tube)

The Nusselt number (Nu) for the hot–side flow is estimated with eq. (1) and (2) which are Gnielinski correlation modified by Hausen [7]. According to Filonenko [8], friction factor f for turbulent tube flow is calculated with eq.(3). In the equation (1), (2) and (3), h_H, D_i, and H are the heat transfer coefficient, tube inner diameter, air thermal conductivity and tube length, respectively.

– Cold–side (flow across tube bank)

The Nusselt number for the flow across the tube bank is estimated with the correlation, eq. (4), given by Zukauskas [9]: X_t and X_l are transverse and longitudinal tube–pitch ratios, respectively, while Pr and Pr_w are Prandtl numbers evaluated at the bulk temperature and the tube wall temperature. The Reynolds number Re_D is based on the tube outer diameter and the mean velocity at the cross section of tube bank.

3. Pressure Loss Calculation for Straight Tube

– Hot–side (flow inside tube)

The pressure loss of flow inside tube is estimated with eq. (5) given by Kays and London [10]. K_c and K_e are inlet, outlet loss coefficient for a multiple tube HX core with abrupt–contraction entrance and abrupt–expansion exit, v_i, v_o and v_m are inlet, outlet and mean specific volume, σ is ratio of free–flow area to frontal area, respectively. Loss coefficient K_c and K_e are function of the contraction and expansion geometry as shown in Fig.3. The curves of Fig.4 provide us with estimates of entrance and exit pressure drop data [10].

– Cold–side (flow across tube bank)

The pressure drop of the flow across tube bank is a function of geometry, the number of tube rows in the bank z, the flow velocity u, and the physical properties of the fluid, which is shown in eq.(7) where Eu is Euler number. For staggered tube banks, the pressure drop (Euler number) curves for banks with many rows of tubes by Zukauskas are shown in Fig. 5 with tube pitch ratio as a parameter(X_t)[9]. Curves are fitted by inverse power series whose constants depend on the value of X_t and Reynolds number. These curve fitting values are used in performance analysis program.

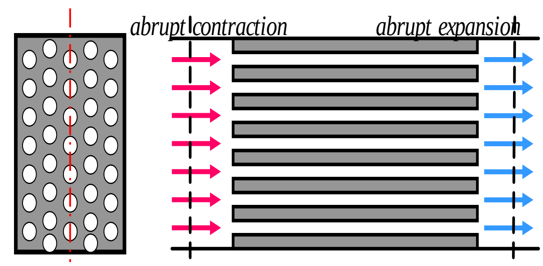

Fig. 3. Flow inside tube core

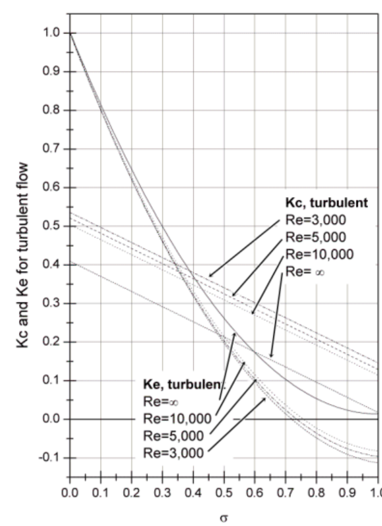

Fig. 4. Entrance and exit pressure–loss coefficients for a multiple circular–tube heat exchanger core with abrupt–contraction entrance and abrupt–expansion exit[10]

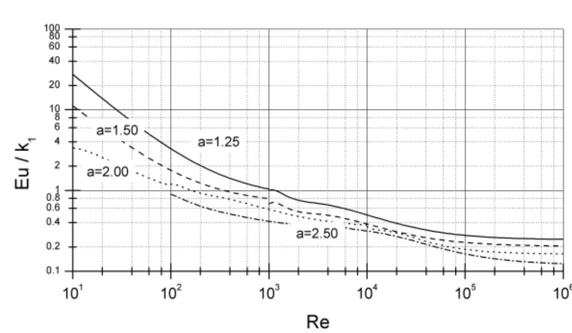

Fig. 5. Pressure drop of staggered banks as referred to the relative transverse pitch a (Euler number with, Re and tube pitch ratio)[9]

4. Heat transfer correlation for U-bend

The Nusselt number for the flow inside U-bend of HX is estimated with the correlation, eq. (8), given by Mashfeghian and Bell [11]. μ_b and μ_w are dynamic viscosity evaluated at bulk mean temperature and wall condition, respectively.

$$Nu_{H,u} = 0.0285\,\mathrm{Re}^{0.81}\,\mathrm{Pr}^{0.4}\left(\frac{x}{D_i}\right)^{0.046}\left(\frac{R}{a}\right)^{-0.133}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} \quad (8)$$

$$\text{for } 4.8 \le \frac{R}{a} \le 26,\ 10^4 \le \mathrm{Re} \le 3\times10^5,\ 0 \le \frac{x}{D_i} \le \frac{\pi R}{2a}$$

5. Pressure loss correlation for U-bend

While the flow passes U-bend of tube, it loses pressure in addition to the pressure loss caused by straight tube. The total pressure drop in a bend is the sum of the frictional head loss due to the length of the bend, head loss due to curvature, and head loss due to excess pressure drop in the down stream pipe because of the velocity profile distortion.

We can estimate the pressure loss in U-bend by following equations with total loss coefficient defined by Ito [12]. De, a and R in eq.(9) and (10) are Dean number, inner radius of tube and curvature radius of U-bend, respectively.

$$\Delta P_{H,u} = K_u \frac{\rho u_m^2}{2} \quad (9)$$

$$\text{for laminar},\quad K_u = \frac{f_c L}{D_h}$$

$$\text{where},\ f_c = \begin{cases} 5\,\mathrm{Re}^{-6.5}(R/a)^{-0.175} & \text{for } 50 < De \le 600 \\ 2.6\,\mathrm{Re}^{-0.55}(R/a)^{-0.225} & \text{for } 600 < De \le 1400 \\ 1.25\,\mathrm{Re}^{-0.45}(R/a)^{-0.275} & \text{for } 1400 < De \le 5000 \end{cases}$$

$$\text{for turbulent},\ 2\times10^4 < \mathrm{Re} < 4\times10^5$$

$$K_u = \begin{cases} 0.00873B\, f_c(R/a) & \text{for } \mathrm{Re}(R/a)^{-2} < 91 \\ 0.00241B\,\mathrm{Re}^{-0.17}(R/a)^{0.84} & \text{for } \mathrm{Re}(R/a)^{-2} > 91 \end{cases} \quad (10)$$

$$\text{where} \begin{cases} f_c = (R/a)^{-0.5}\left[0.00725 + 0.076\left\{\mathrm{Re}(a/R)^2\right\}^{-0.25}\right] \\ B = 1 + 116(R/a)^{-4.52} \quad \text{for } \phi = 180^\circ \text{ (deg.)} \end{cases}$$

6. Optimal design method

In order to design compact light weight HX with the minimum pressure loss and the maximum heat exchange rate, the weight of HX core is chosen as the object function, which is described as the ratio of tube material volume to reference volume. Dimensions and tube pitch ratio of the HX are used as design variables. In the case of U-tube type HX optimization, a design variable to optimize curvature radius of the smallest U-tube is added. Hot-side and cold-side pressure loss (ΔP_H, ΔP_C) and hot-side temperature drop (ΔT_H) are used as constraints. Design parameters for optimization are shown specifically in Table 2.

Table 2. Design Parameters for optimization

Parameter		Description	Range
Design Variables	x_1	tube pitch ratio (transverse direction), $X_t = S_t/D_o$	$1.0 \le x_1 \le 2.5$
	x_2	tube pitch ratio (longitudinal direction), $X_l = S_l/D_o$	$1.0 \le x_2 \le 2.5$
	x_3	tube outer diameter, $D_o/D_{o,ref}$	$1.0 \le x_3 \le 2.0$
	x_4	height of HX (tube length), H/H_{ref}	$0.5 \le x_4 \le 2.0$
	x_5	length of HX, W/W_{ref}	$0.5 \le x_5 \le 2.0$
	x_6	width of HX, B/B_{ref}	$0.5 \le x_6 \le 2.0$
	x_7^*	margin from the center line of HX, M/D_o (radius of the smallest curvature of U-tube bend)	$0.5 \le x_7 \le 2.0$
Object Function	F	tube material volume / preference volume	$F \le 1.0$
Constraints	ΔT_H	temperature difference of the flow inside tube	$220.0K \le \Delta T_H$
	ΔP_H	pressure loss of the flow inside tube	$\Delta P_H \le 0.6\%$
	ΔP_C	pressure loss of the flow across tube bank	$\Delta P_C \le 4.0\%$

(* : U-tube only)

With the above set of equations, design parameters are optimized by using the genetic algorithm with optimization software package, *iSIGHT*. Gradient-based algorithm has some problems when the objective function takes discontinuous distribution. The genetic algorithm is more suitable in this condition, because this algorithm is specialized in global optimization problems using whole area searching method [13].

Results

We figured out the correlations between each parameter and determined proper range of design parameters using the design of experiments (DOE) and approximation model (RSM, Response Surface Model).

Figure 6(a) shows that the volume of HX increases as the dimensions (x_4 , x_5 , x_6) increase and decreases as the pitch ratios (x_1 , x_2) increase. Tube outer diameter (x_3) does not have considerable effect on the volume of HX.

It is shown that the hot-side temperature difference is inversely proportional to the tube outer diameter (x_3) in fig. 6(b). The hot-side pressure drop decreases as the length (x_5) and the width (x_6) of HX increase in fig. 6(c). Figure 6(d) represents the cold-side pressure drop is more affected by the transverse pitch ratio (x_1) and the length of heat exchanger (x_4) than by other parameters.

Figure 7(a) through (d) show the interactions of design variables. The interaction of pitch ratios (x_1 , x_2) is shown in fig. 7(a), which indicates that the object function may exist as an optimized result within the range of $1.7 \le x_1 \le 2.2$ and $1.0 \le x_2 \le 1.8$.

Figure 7(b) is about the interaction of pitch ratio and tube diameter. As the effect of each parameter on the object function already shown in Fig. 6, even though the change of tube diameter couldn't have much influence over the object function, it is observed that the tube diameter changes linearly with the respect to the pitch ratio.

Figure 7(c) represents the interaction of pitch ratio (x_1) and tube length (x_4). The object function decreases when the pitch ratio increases and the dimension (HX height, length, width) decreases simultaneously. In fig. 7(d), as mentioned above regarding Fig. 6, it is shown that the object function changes linearly not with the diameter, but rather with HX dimension.

After the design space and rough estimate of the optimal design which can be used as a starting point for numerical optimization, a feasible range of object function and a calculating range of the design variables were determined as shown in Fig. 2. Then the optimal design was performed by genetic algorithm with iSIGHT.

Inlet conditions for flow of HX and parameters for the genetic algorithm are described in table 3. It took 15,000 iterations in 6 hours to obtain an optimized result for each case.

Optimal design results of a straight tube type HX and a U-tube type HX are compared to results of foreign partners in table 4. Optimal design results of a straight tube type HX and a U-tube type HX are shown in table 4.

In the case of a straight tube type HX, the object function is 0.95 with satisfying all of constraints in feasible range shown in table 2. In the case of a U-tube type HX, the object function is 1.05 with satisfying all constraints except hot-side pressure condition.

As a result, it was found that the volume, and thus the weight, of tube material of U-tube type HX will was larger than that of straight-tube type HX by about 10.5% under the same constraints. To put it more simply, it means the U-tube type HX will become heavier than the straight tube type HX under the same constraints.

Table 3. Calculating condition for optimization

Flow Initial Condition	$T_{H,i}$	*1002 K*
	$T_{C,i}$	326 K
	$P_{H,i}$	55.7 bar
	$P_{C,i}$	1.34 bar
	$\dot{m}_H$	1.436 kg/s
	$\dot{m}_C$	1.446 kg/s
Genetic Algorithm	Population	150
	Generation	100
	Cross over rate	1.0
	Mutation rate	0.02

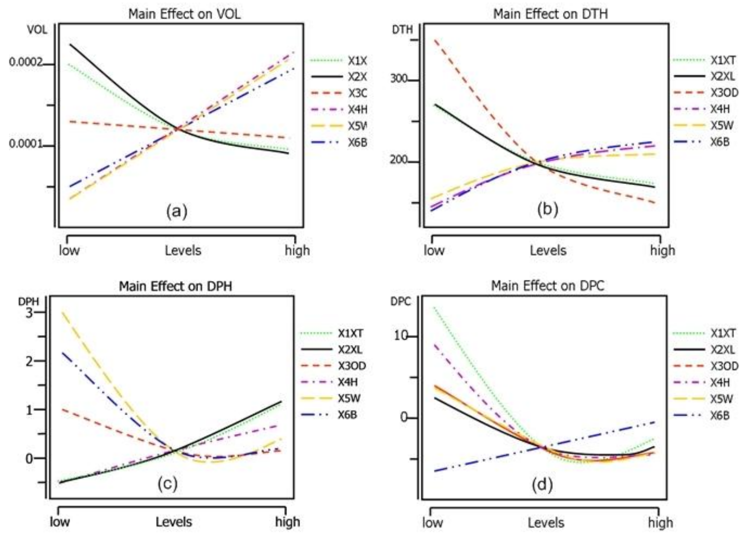

Fig. 6. Main effects on output

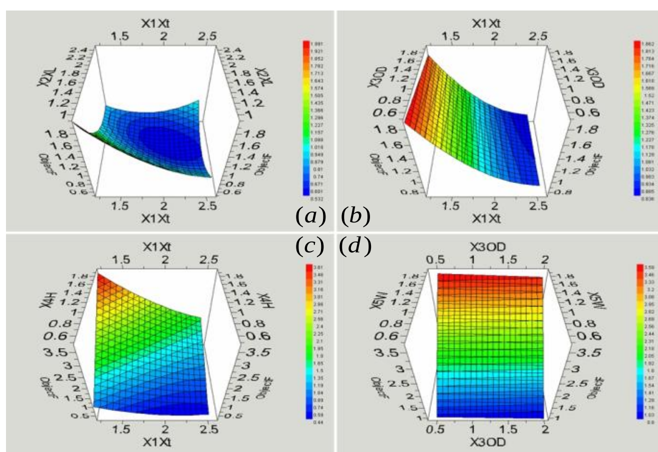

Fig. 7. Variable interactions

Table 4. Optimal design results

Parameter		*Ref. HX*	*Optimized*	
			Straight	U-tube
Optimized design variables (dimensionless)	x_1	2.00	1.87	2.0
	x_2	1.12	1.05	1.09
	x_3	1.00	1.00	1.00
	x_4	1.00	0.97	1.13
	x_5	1.00	1.00	0.94
	x_6	1.00	0.87	1.08
Performance	ΔT_H	216.20K	220.38K	228.40K
	ΔP_H	0.60 %	0.60 %	0.69 %
	ΔP_C	3.08 %	4.00 %	3.24 %
Object Function	*F*	1.00	0.95	1.05

Conclusion

We have developed HX rating programs for the thermal analysis and optimal design of HX using the ε −NTU (HX effectiveness − the number of transfer units) method. It supplies the object function during optimization process to design a HX for a high temperature and high pressure system.

A single−pass cross−flow type with straight plain tubes and a two−pass cross−counter flow type with plain U−tubes were designed and optimized with the genetic algorithm by *iSIGHT*. During the optimization process, the effect of design variables and constraints was investigated by DOE and RSM.

Because of disadvantage in the pressure loss of flow inside tube, U−tube type HX was not optimized in feasible region. U−tube type HX does not satisfies imposed requirements due to high pressure loss. In this study, it is predicted that a U−tube type HX would be heavier than a straight−tube type HX by about 10.5% under the same constraints.

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