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The Optimization Design of a Compact Integrated Precooler for Advanced Space Launcher Engines

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In recent years, aeroplane engines have evolved towards the direction of hypersonic flight, re-use and combined cycle propulsion. As one of the core components, air-hydrogen precooler plays a vital role in the development of liquefied air cycle engines. In order to solve the problem of rapid cooling of high-temperature high-speed air to cryogenic temperature in a very short period of time, the present study proposes a novel compact integrated precooler, which is made up of thin-walled fin and printed circuit plates. To examine the thermal-hydraulic performance of the novel compact integrated precooler design, two different types of conventional precoolers, i.e., shell-and-tube heat exchanger and printed circuit heat exchanger, are used for comparison in terms of volumetric power, power per mass unit and compactness. The results indicate that the comprehensive performance of the novel precooler is significantly improved.

1. Introduction

Recently, hypersonic transportation and routine access to space urgently demand the development of hypersonic airbreathing propulsion systems. The optimisation of the thermodynamic cycles of aero engines has always been in the main targets of engineering efforts for environmental and economic reasons (Missirlis et al., 2019). The safe, reliable, and sustained operation aircraft could extend and diversify space activities. As a representative, existing combined cycle engines mainly include Turbine Based Combined Cycle (TBCC), Rocket Based Combined Cycle (RBCC), Air Turbine Ramjet (ATR), three-combination engine and pre-cooled engine, etc (Tang et al., 2019). Figure 1 shows the types of combined cycle engine. The mentioned above engines combine different types of airbreathing or rocket engines, which give full play to their respective advantages. Combined cycle engines air-breathing mode pre-cooled inlet air can effectively improving performance and specific impulse. Due to the presence of the precooler, the cycle core engine can be isolated from the real flight conditions. Consequently, it can operate at even higher Mach numbers (Yu et al., 2019).

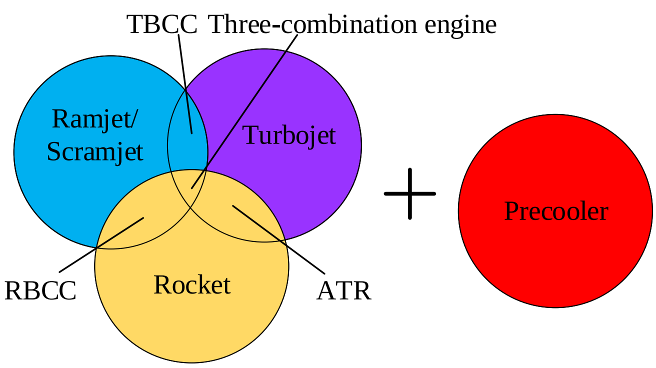

Figure 1: The types of combined cycle engine

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The most representative pre-cooled engine is Liquid Air Cycle Engine (LACE), as shown in Figure 2, which was proposed by Marquardt in the late 1950s (Mamoru et al., 1999), the high heat capacity liquid hydrogen is used to liquefy intake air in the precooler. The final liquefied air is utilized as the oxidiser and hydrogen to combust in the thrust chamber. The main advantages of LACE system include: First, the LACE take off without carrying oxidants can theoretically reduce the gross takeoff weight and dry mass. Second, using the same nozzle for air-breathing and rocket modes. Nevertheless, wang et al. (2014) also pointed out that LACE has icing and precooler design problem. As indicated, the successful implementation of these cycles relies critically on the development of high-performance precoolers (Murray et al., 1997).

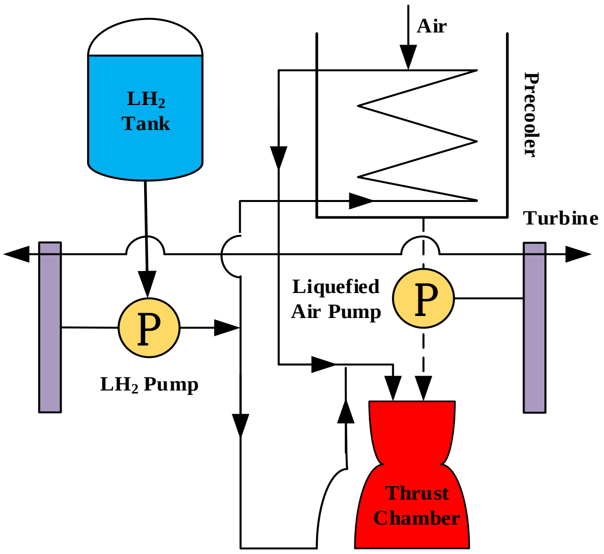

Figure 2: A basic model of LACE system (Mamoru et al., 1999)

Hendrick et al. (2009) selected and designed the shell-and-tube heat exchanger (STHE) as the LACE precooler. The STHE have flexible configuration but fundamentally cannot be used to achieve a compact design where space, volume, shape and weight are at a premium. So STHE are not an appropriate type of construction for being considered. A printed circuit heat exchanger (PCHE) is chosen as a precooler due to high compactness, small volume and low leakage. The main constraint is the high overall quality, which seriously affects its application in the aerospace field. For these reasons, combining the characteristics of LACE precooler design parameters, the present study proposes a novel compact integrated air-hydrogen precooler (IAHP), which is made up of thin-walled fin and printed circuit plates. This structure combines the advantages of conventional plate-fin heat exchangers (PFHE) and PCHE. In order to prove the feasibility and thermal-hydraulic performance of the IAHP design, two different types of conventional precoolers, i.e., STHE, and PCHE are used to compare with the IAHP in terms of volumetric power, power per mass unit and compactness.

2. Plant layout description

The designed novel IAHP is composed of two types of fluid channels, i.e., etched channels and plate-fin channels, as shown in Figure 3. The two types of fluid channels are alternately stacked to achieve heat exchange of cold and hot fluid. The high-pressure small-flow hydrogen flows in the etched channels, which are formed by two identical rectangular channel printed circuit plate through diffusion welding. Etched channels have the features of high heat transfer and strong pressure capacity, especially suitable for high-pressure and small-flow working flow. The channels usually have an equivalent diameter with 0.5 ~ 2 mm. The low-pressure large-flow air flows in the plate-fin channels, which are formed by ultra-thin fins. The channels usually have an equivalent diameter with 0.5 ~ 1.5 mm. This kind of flow path can greatly reduce the pressure loss and weight, increase system compactness and heat transfer capability. The 6061 aluminium alloy is chosen as the precooler processing material (Hendrick et al., 2009), which has comprehensive advantages in terms of processability, weldability, plating and corrosion resistance.

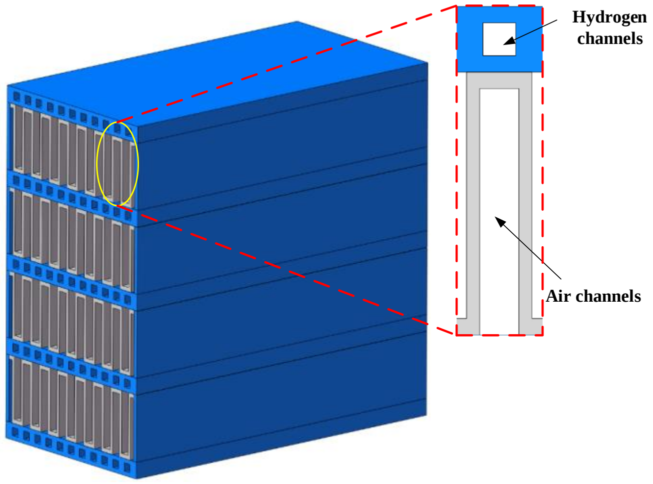

Figure 3: Heat transfer channels of the novel IAHP

3. Thermal design and optimisation algorithm

Heat exchanger thermal design is the main factor affecting heat exchanger performance, the selection of the thermal design method will directly determine the accuracy of the heat exchanger design results. It is notable that a large number of highly interdependent geometric and operational variables, which often show trade-offs. The Genetic Algorithm (GA) can omit the tedious procedures of manual optimisation design and achieve efficient calculation and optimisation.

3.1 Segmented LMTD method and GA optimisation

The widely used conventional thermal design methods are the Log-Mean Temperature Difference (LMTD) and ε-NTU methods (ε-NTU Method). These methods often assume the working fluid physical properties as constants. In fact, during the pre-cooling process, the physical properties of the working fluid fluctuate greatly in a very short time. Zhang et al., (2018) proposed the traditional LMTD method could bring huge calculation error in high-temperature design. In order to accurately calculate the heat transfer rate and pressure drop of the precooler, a segmented LMTD method is proposed. The heat exchanger is divided into finite segments based on the temperature variation range. Every segment is considered to be a small heat exchanger. Initial conditions and connection conditions are used to obtain design results. In order to study the actual thermal-hydraulic performance of the novel IAHP, the present study selected detailed operating parameters applied in the LACE for thermal design (Hendrick et al., 2009). The design conditions of the precooler are shown in Table 1.

Table 1: Design parameters of IAHP (Hendrick et al., 2009)

	Parameter	Unit	Value
Hydrogen side	Flow rate	kg/s	3.0
	Inlet temperature	°C	-64
	Outlet temperature	°C	171
	Inlet pressure	kPa	1,600
	Pressure drop	kPa	400
Air side	Flow rate	kg/s	52.8
	Inlet temperature	°C	206
	Outlet temperature	°C	90
	Inlet pressure	kPa	148
	Pressure drop	kPa	15

The detailed design results of IAHP are shown in Table 2. Since the air flow is much larger than the hydrogen flow, the plate-fin structure is chosen as the air channel. The fin height is much larger than the fin pitch to increase the flow cross-sectional area and reduce the air flow rate. The design can reduce the pressure loss. For PCHE, the detailed design results of PCHE are shown in Table 3. In the structure layout, two hydrogen channels are corresponding to one air channel, so the cross-sectional area of the air channel is about ten times than that of the hydrogen channel. These structural optimization designs help the both side fluids to be in an appropriate pressure range. The detailed design results of the STHE are shown in the work of Hendrick et al. (2018).

Table 2: Design results of IAHP

	Parameter	Unit	Value
Hydrogen side	channel diameter	mm	0.16
	channel pitch	mm	0.41
	plate thickness	mm	1.16
Air side	Fin pitch	mm	0.36
	Fin height	mm	2.30
	Fin thickness	mm	0.10
	Volume	m^3	0.24
	mass	kg	345.98

Table 3: Design results of PCHE

	Parameter	Unit	Value
Hydrogen side	channel diameter	mm	0.10
	channel pitch	mm	0.35
	plate thickness	mm	0.35
Air side	channel diameter	mm	0.45
	channel pitch	mm	0.7
	plate thickness	mm	0.7
	Volume	m^3	0.35
	mass	kg	674.27

The GA is a search heuristic algorithm based on genetics, mutation, selection and crossover, which has the characteristics of highly parallel, random, global search and adaptive. The present study combines the thermal design process with the GA and develops an optimisation algorithm based on MATLAB and NIST database. The thermal design and GA optimisation process is illustrated in Figure 5.

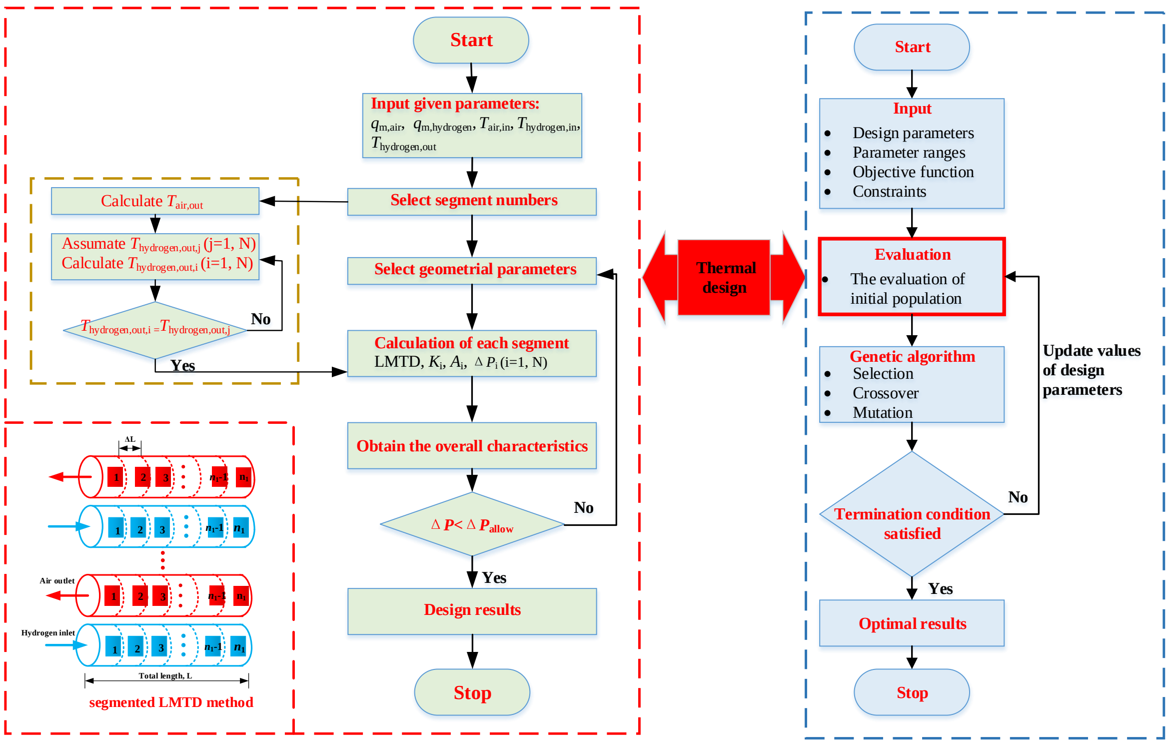

Figure 5: The thermal design and GA optimisation process

3.2 Basic equations

Equation of heat balance is as:

$$Q = q_{\mathrm{air}} \cdot c_{\mathrm{p,air}} \cdot (T_{\mathrm{air,out}} - T_{\mathrm{air,in}}) = q_{\mathrm{H_2}} \cdot c_{\mathrm{p,H_2}} \cdot (T_{\mathrm{H_2,out}} - T_{\mathrm{H_2,in}}) \tag{1}$$

where q_{air} and $q_{\mathrm{H_2}}$ are the mass flow rate of air and hydrogen, $c_{\mathrm{p,air}}$ and $c_{\mathrm{p,H_2}}$ are the specific heat at constant pressure of air and hydrogen, $T_{\mathrm{air,in}}$ and $T_{\mathrm{air,out}}$ are the inlet and outlet temperature of the air, and $T_{\mathrm{H_2,in}}$ and $T_{\mathrm{H_2,out}}$ are the inlet and outlet temperature of hydrogen (Qian, 2002).

The overall heat transfer rate *Q* is given by:

$$Q = UA\Delta T_{\mathrm{m}} \tag{2}$$

where ΔT_{m} is a log-mean temperature difference, and *U* is the overall heat transfer coefficient based on heat transfer area *A*, which is defined as:

$$\frac{1}{U} = \frac{1}{h_{\mathrm{air}}} + \frac{\delta}{\lambda} + \frac{1}{h_{\mathrm{H_2}}} \tag{3}$$

$$\Delta T_{\mathrm{m}} = \frac{\Delta T_{\max} - \Delta T_{\min}}{\ln \dfrac{\Delta T_{\max}}{\Delta T_{\min}}} \tag{4}$$

where δ is the hydraulic diameter between hot and cold channels, λ is a thermal conductivity of aluminium alloy 6061, h_{air} is convective heat transfer coefficient on the air side, $h_{\mathrm{H_2}}$ is convective heat transfer coefficient on hydrogen side, and $\Delta T_{\max}$ is the maximum temperature difference on one side, $\Delta T_{\min}$ is minimum temperature difference on the other side.

Convective heat transfer coefficients on both sides, h_{air} and $h_{\mathrm{H_2}}$, are calculated as:

$$h = \frac{Nu\lambda}{D_{\mathrm{h}}} \tag{5}$$

where λ is the thermal conductivity, and D_{h} is the hydraulic diameter.

For the hydrogen side, the global Nusselt number *Nu* is given by (Maylavarapu, 2011):

$$Re \le 2{,}300 \qquad Nu = 4.089 \tag{6}$$

$$2300 < Re < 5{,}000 \qquad Nu = 4.089 + \frac{Nu_{\mathrm{G|Re-5{,}000}} - 4.089}{5{,}000 - 2{,}300}\left(Re - 2{,}300\right) \tag{7}$$

$$Re \ge 5{,}000 \qquad Nu = \frac{\dfrac{f}{2}\left(Re - 1{,}000\right)Pr}{1 + 12.7\left(Pr^{2/3} - 1\right)\sqrt{\dfrac{f}{2}}} \tag{8}$$

where *Pr* is the Prandtl number, and $Nu_{\mathrm{G|Re\text{-}5000}}$ is the Nusselt number from the Gnielinski correlation evaluated at Reynolds number of 5,000.

For the air side, the heat transfer factor *j* and Fanning friction factor *f* are determined by (Qian, 2002).

$$\ln j = 0.103109\left(\ln Re\right)^2 - 1.91091\left(\ln Re\right) + 3.211 \tag{9}$$

$$\ln f_{\mathrm{f}} = 0.106566\left(\ln Re\right)^2 - 2.12158\left(\ln Re\right) + 5.82505 \tag{10}$$

The global Reynolds number Re is calculated:

$$Re = \frac{\rho V D_{\mathrm{h}}}{\mu} \tag{11}$$

where ρ , V , μ are density, area-averaged velocity in the inlet section, dynamic viscosity.

4. Performance comparison and discussion

In order to compare the three types of precoolers, the parameters in terms of compactness, volumetric power and power per mass unit of these precoolers are listed in Table 4. The designed results indicate that the IAHP shows excellent comprehensive performance. Under the condition of the same heat exchange and pressure drop, the compactness of IAHP is 1.61 times than that of PCHE, and 2.57 times than that of STHE. The volumetric power of IAHP is 1.46 times than that of PCHE, and 4.85 times than that of STHE. The power per mass unit of IAHP is 1.96 times than that of PCHE, and 1.70 times than that of STHE.

Table 4: Comparison of designed results for three kinds of structure precooler

Parameter	Unit	PCHE	STHE	IAHP
Volume	m^3	0.35	1.16	0.24
Weight	kg	674.27	579.15	345.98
Power	MW	10.5	10.5	10.5
Compactness	m^2/m^3	2449	1530	3940
Volumetric power	MW/m^3	30.11	9.06	44.00
Power per mass unit	kW/kg	15.70	18.13	30.77

5. Conclusions

The present study proposes a novel type of IAHP. Two different types of conventional precoolers, i.e., STHE, and PFHE are selected for comparison. Under the same working condition, the compactness of IAHP is 1.61 times than that of PCHE, and 2.57 times than that of STHE. The volumetric power of IAHP is 1.46 times than that of PCHE, and 4.85 times than that of STHE. The power per mass unit of IAHP is 1.96 times than that of PCHE, and 1.70 times than that of STHE. In brief, the IAHP shows excellent comprehensive performance.

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