Analysis of Thermal–Hydraulic Performance Assessment of a Compact Aluminum Plate-Fin Heat Exchanger

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Abstract

This article depicts the thermal–hydraulic assessment of a counter-flow type fluid–gas aluminum compact heat exchanger, with offset strip fin (OSF) geometry. Compact heat exchangers are important for cryogenic and high-efficient thermal systems because of their large surface-area density, as well as the possibility to gain a considerable value of heat transfer in small volumes. Tests were performed at different mass flow rates and two hot-side inlet temperatures of 66 °C and 96 °C. It is observed that heat-exchanger effectiveness increases with the mass flow rate; roughly vary between 0.70–0.92, as the temperature gradient increases at lower inlet temperatures. The overall UA further increased monotonically with the flow rate due to improved convection by repeated boundary-layer interruption of the OSF geometry. At higher mass flow rates, increases in form drag and vortex shedding led to a substantial rise in pressure drop. Comparison with established empirical correlations revealed good coincidence in the laminar and transition ranges and larger discrepancies for higher Reynolds numbers due to imperfections of fin manufacturing. In general, the OSF plate-fin heat exchanger showed good thermal effectiveness and impressive heat-transfer results, respectively revealing its potential for cryogenic or high-performance purposes.

Keywords: *compact heat exchanger; offset strip fin; effectiveness; overall heat transfer coefficient; heat transfer.*

1. Introduction

Over the past decade, work in compact plate-fin heat exchanger (PFHE) technology with offset strip fins (OSFs) has transitioned from predominantly empirical, to include high-fidelity CFD together with optimization and data-driven modelling. Heat exchangers are used for transferring thermal energy between two or more fluid streams separated by a solid boundary. They affect the performance, size and energy aspect of thermal equipment. CHX present a compact high surface area density (above 700 m^2/m^3) that guarantees a strong heat transfer performance in limited spaces. They are particularly important in cryogenic applications where gas liquefaction requires efficiencies higher than 85–90%. Plate fin heat exchangers (PFHEs) are commonly employed in compact heat exchanger design, owing to their relatively lower weight, modular construction and ability to accommodate several fluid features. OSFs are one of the highest effective fin profiles because they disrupt boundary-layer growth, improve mixing and provide high heat-transfer coefficients. But these advantages are in an exchange for much higher pressure drop.

Recent experimental works have focused on the development of realistic data for full-scale or representative PFHEs and OSF channels, subjected to different thermal and flow conditions. Gupta et al. [1] investigated the performance of a plate-fin heat exchanger with kinks and carried out experimental performance tests on it using AI models (ANN,

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ANFIS) to predict effectiveness and pressure drop. They found that AI models could predict thermal-hydraulic performance within 10–20% of measurements, suggesting that they can be utilized for quick design and performance estimation. Jiang et al. [2] conducted helium low-temperature experiments in PFHE channels for heat-transfer coefficients and pressure drop of multiple fin geometries. Their test database is important for validation of cryogenic applications, for which the thermo-physical data are strongly temperature dependent.

In aerospace and low-pressure environmental applications, Wan et al. [3] tested an offset-fin flat-tube heat exchanger at reduced ambient pressures and the heat- transfer rate and the pressure drop both decreased significantly with the reduction of pressure, which has some implications for altitude aircraft. Other experimental works are centered on the production of deviations. Yan et al. [4] found that it can be due to fin misalignment, brazing defects and surface roughness in the range of 8–15%, suggesting the need of accounting for manufacturing variations for performance prediction. The work presented in the last decade has obliged high-fidelity CFD simulations in both OSF and compact heat-exchanger studies. Doğan et al. [5] performed a comprehensive numerical study of OSF compact heat exchangers, and found that Colburn j factor and Fanning friction f factor are sensitive to fin spacing, strip length, and offset frequency. Their work suggested new performance correlations that applies to today's high compactness exchangers. Shahdad and Fazelpour [6] conducted the 3D CFD simulation using ANSYS Fluent (2020 R1) to investigate heat-transfer and pressure-drop performance of PFHE with different enhanced fin surfaces. The study demonstrated a strong thermal enhancement from boundary layer thinning and enhanced mixing, presenting novel CFD-validated correlations for PFHE designers. In a similar attempt, Li et al. [7] was is that they performed a numerical validation of optimized compact PFHE structures with the aid of modified or corrugated fin surfaces. They report significant enhancements in the overall heat-transfer coefficient and diminished pressure drop. A CFD study on a fin-by-fin model by Blecich et al. [8] visualized flow separation, enhancement of heat transfer zones and distribution of local friction for OSF geometry at a fairly high resolution. This fine comparison is useful for the second generation fin design tools.

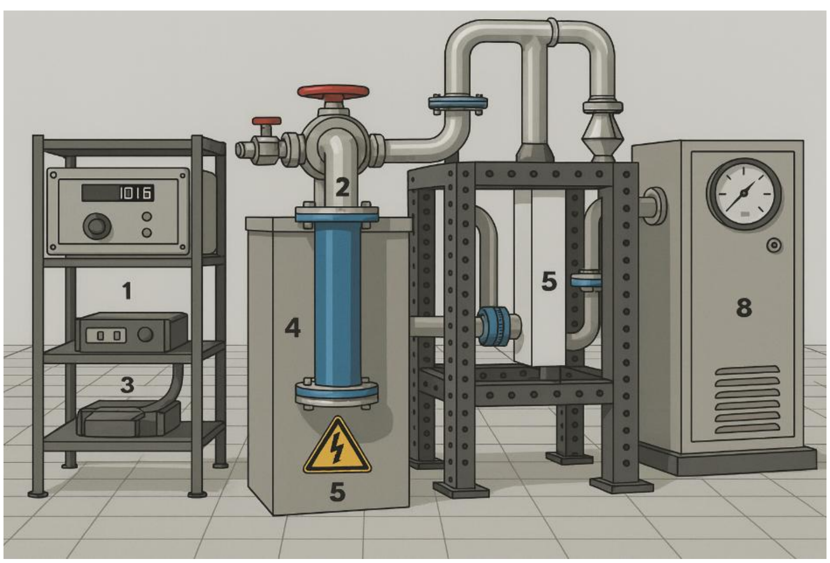
Mixed or hybrid fin geometries were also investigated by other researchers. Abbas and Mohammed [9] investigated various fin offsets and arrangements in PFHEs, and concluded that the hybrid fin arrangement might be superior to pure OSF configuration in some range of Reynolds numbers. Kim et al. [10] used CFD to investigate cross–counter-flow PFHEs in HRVs, and found that fin geometry and channel aspect ratio both have significant impact on thermal performance, as well as pressure drop. Jiang et al. [11] proposed a distributed-parameter cryogenic PFHE model considering longitudinal conduction, ambient heat-in-leak and temperature-dependent helium properties. Their results show that conduction and parasitic heat-in-leak can have a tremendous impact on effectiveness -particularly in the case of high-NTU exchangers. Borjigin et al. [12] numerically studied small scale longitudinal conduction effects and found that, the thin metal walls can transport sufficient axial heat in counter-flow PFHEs to degrade their performance. Saberimoghaddam et al. [13] studied the effect of parasitic wall conduction in cryogenic gas heat exchanger systems and pointed out that a disregard of conduction bias can yield systematic errors for performance prediction. Ranganayakulu et al. [14] investigated total conduction and maldistribution effects, illustrating that flow maldistribution aggravates the conduction-induced performance worsening in a PFHE. Fang et al. [15] also proposed nondimensional performance maps for cryogenic PFHEs which included cooling, conversion and total effectiveness measures to aid in design and optimization. Flow ViewModel Verification of the Hybridistic NTU–CFD Approach for both Cryogenic Plate-fin Systems and Non-cryogenic Ones Recently, a hybrid NTU–CFD prediction method for plate-fin exchangers was proposed by Cheng et al. [16], which contributed somewhere in between the traditional empirical correlations and full CFD simulation approaches for cryogenic as well as non-cryogenic systems.

Even more recently studies provide a refinement on thermal–hydraulic modelling and optimization and compact plate-fin exchangers. Jung et al. [17], established new heat transfer and friction correlations for the OSF tubes as a function of the offset ratio using detailed 3D CFD, thus updated j–f data for modern OSF geometries. Petrović et al. [18] carried out a topology writing process, CFD and experiments to optimize and validate a plate-fin heat exchanger with the topology-optimized fins, showing that atypical fin arrangements can beat common designs while being manufacturable. Ye et al. [19] investigated the influence of plate-fin geometry parameters: fin thickness, fin height, porosity and axial length on the performance of a cascade refrigerator system, associating directly PFHE geometrical configuration to COP, exergy destruction and entropy generation. Aktas et al. [20] PMFHE with offset-strip fins on the cooling side and wavy fins on the air side using MgO–water nanofluid was experimentally studied, significant increase in thermal and hydrodynamic re-sistance were observed against a base fluid. In parallel, Zou et al. [21] provided an overview of machine learning for heat exchanger modelling, illustrated that data-driven methodologies can be used as a complement to CFD and traditional correlations in compact heat exchanger design and performance prediction.

Following is a detailed experimental investigation of the OSF PFHE. The study aims to identify thermal-hydraulic performance parameters and compare them to developed correlations for cryogenic and high efficiency applications. The present study bridges these gaps by experimentally characterizing a full-size helium brazed aluminum counter-flow PFHE with offset strip fins under balanced air-flow to address the above-discussed issues. Performance, overall thermal conductance and pressure drop are obtained over a mass flow rate range; results are correlated and corrected for longitudinal wall conduction with classical correlations. This represents an important experimental benchmark for OSF based cryogenic PFHEs, while the extent to which standard design correlations reflect true behaviour of manufactured cores is also made clear.

2. Experimental Setup

The experimental test facility for the thermal–hydraulic testing of the compact heat exchanger is presented in Fig. 1. The arrangement comprises control and power unit, flow control valve, electric heater module insulated vertical test section with the heat exchanger and pressure accountability cabinet. The rigs offers ability to provide controlled heating, controlled flow and its measurement as well as precise measurement of temperature and pressure parameters in the test section. The experimental apparatus works like a closed-cycle heating flow system and is constructed to deliver conditioned air to the test coil, whilst monitoring its thermal behavior. The block diagram of the system can be presented as in Figure 2.



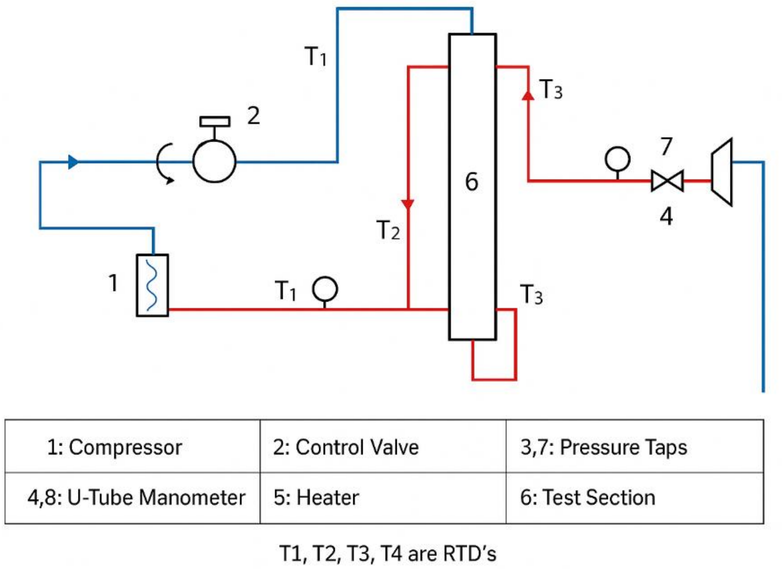


Figure 1 Schematic diagram of the test experimental-setup

Table 1 Description elements of the test rig.

No.	Component	Description / Function
1	Control & Power Unit	Houses the electronic controller, switches, power supply, and instrumentation used to operate the heater and monitor outputs.
2	Main Flow Control Valve	Regulates the mass flow rate entering the test section and controls inlet conditions.
3	Heater Safety Panel	Electrical hazard-protected heater enclosure containing heating elements.
4	Heated Vertical Tube	Blue-coated heated pipe delivering hot air/gas from the heater to the test section.
5	Insulated Test Section / Heat Exchanger Column	Experimental core containing the heat exchanger, fully insulated to reduce heat loss.
6	Upper Stainless Steel Manifold	Directs heated fluid into the test section and distributes flow uniformly.
7	Lower Return Manifold	Collects exiting fluid from test section and channels it back into the loop.
8	Pressure Monitoring Cabinet	Houses pressure gauge and related measurement electronics for system pressure monitoring.

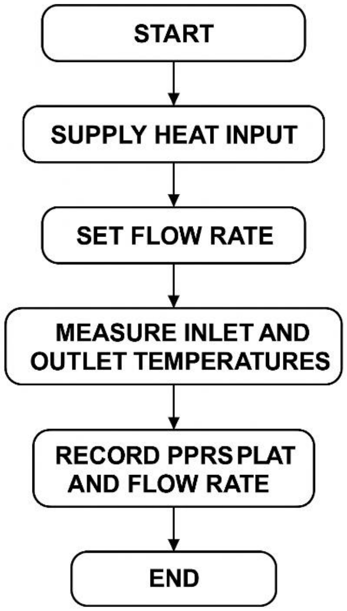


Figure 2 Experimental procedure.

2.1 Data Reduction

Data reduction converts raw experimental measurements into thermal–hydraulic performance parameters such as heat transfer rate, effectiveness, overall heat-transfer coefficient (U), Colburn factor (j), and Fanning friction factor (f).

1. Determination of Mass Flow Rate

$$Q = C\, C_a \sqrt{(2\, g\, H\, \rho)} \quad (1)$$

Where C is the area constant, C_a is the discharge coefficient, H is the manometer head difference, ρ is density, and g is gravitational acceleration.

2. Heat Transfer Rate

$$Q_h = \dot{m}_h C_{p,h} (T_{h,in} - T_{h,out}) \quad (2)$$

$$Q_c = \dot{m}_c C_{p,c} (T_{c,out} - T_{c,in}) \quad (3)$$

The smaller of Q_h and Q_c is taken as the actual heat-transfer rate.

3. Heat Capacity Rates

$$C_h = \dot{m}_h C_{p,h} \quad (4)$$

$$C_c = \dot{m}_c C_{p,c} \quad (5)$$

$$C_{min} = \min(C_h, C_c) \quad (6)$$

$$C_{max} = \max(C_h, C_c) \quad (7)$$

$$C_r = C_{min} / C_{max} \quad (8)$$

4. Heat Exchanger Effectiveness

$$\varepsilon = Q / [C_{min} (T_{h,in} - T_{c,in})] \quad (9)$$

5. Overall Heat Transfer Coefficient (U)

$$\Delta T_{lm} = [(T_{h,in} - T_{c,out}) - (T_{h,out} - T_{c,in})] / \ln[(T_{h,in} - T_{c,out})/(T_{h,out} - T_{c,in})] \quad (10)$$

$$U = Q / (A_{ht}\, \Delta T_{lm}) \quad (11)$$

6. Reynolds Number

$$Re = (G\, D_h) / \mu \quad (12)$$

7. Colburn Factor (j)

$$j = [h / (C_p\, G)]\, Pr^{(2/3)} \quad (13)$$

8. Fanning Friction Factor (f)

$$f = [\Delta P\, (2\, \rho\, D_h)] / [4\, L\, G^2] \quad (14)$$

9. Performance Ratio (j/f)

$$\text{Performance} = j / f \quad (15)$$

2.2 Uncertainty Analysis

Data reduction is the process of converting raw experimental measurements to thermal–hydraulic performance parameters, such as heat transfer rate, effectiveness (ε), overall heat-transfer coefficient (U), Colburn factor (j) and Fanning friction factor (f). For a calculated parameter $R = f(x_1, x_2, ..., x_n)$, the uncertainty is:

$$\delta R = \sqrt{[(\partial R/\partial x_1\, \delta x_1)^2 + (\partial R/\partial x_2\, \delta x_2)^2 + \ldots + (\partial R/\partial x_n\, \delta x_n)^2]} \quad (16)$$

Table 2 Summary of overall uncertainty

Parameter	Uncertainty (%)	Notes
Heat-transfer rate ($\dot{Q}$)	±3–5%	Dominated by mass flow and temperature difference uncertainties
Effectiveness (ε)	±4–7%	Strongly affected by inlet temperature measurement accuracy
Reynolds number (Re)	±2–3%	From mass-flow measurement uncertainty
Colburn j-factor	±5–8%	Affected by uncertainty in h and Pr
Fanning friction factor (f)	±6–10%	Highly sensitive to pressure-drop measurement

3. Results and Discussion

Variation of effectiveness with mass flow rate is plotted in Figure (3). It is observed that effectiveness increases with mass flow rate in both the cases. First the hot effectiveness increases, then it remains almost constant for some mass flow rates, and finally it rises. However, by using the two lines one can conclude that from experimental effectiveness said value obtained when high hot inlet temperature is achieved is higher than effectiveness value attained at high temperature and low hot inlet temperature. So we can conclude that with the rise in hot inlet temperature effectiveness increases.

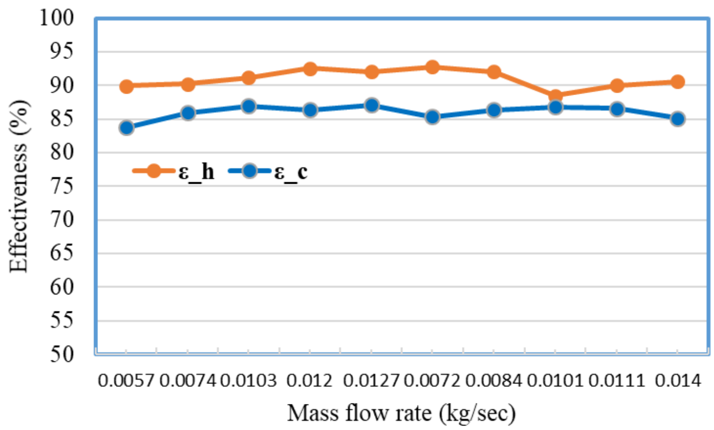


Figure 3 Variation of effectiveness with mass flow rate.

Variation of case average HTC is shown with mass flow rate in Figure 4. It is observed that the overall heat transfer coefficient increases as mass velocity increases. It is because with higher $\dot{m}$, the Reynolds number also becomes higher and hence Colburn factor (j), which is directly proportional to heat transfer coefficient increases so overall thermal conductance increases. The variation of hot and cold temperature difference is shown in Figure 5. The number of heat transfer unit is variant with mass flow rate as shown in Figure 6.

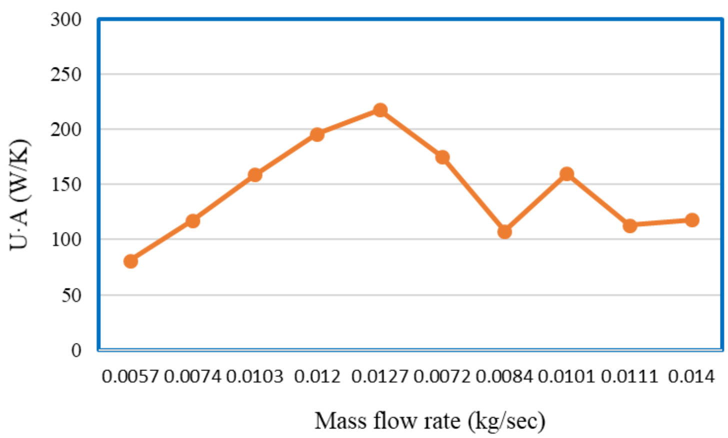


Figure 4 Variation of Overall heat transfer coefficient with mass flow rate.

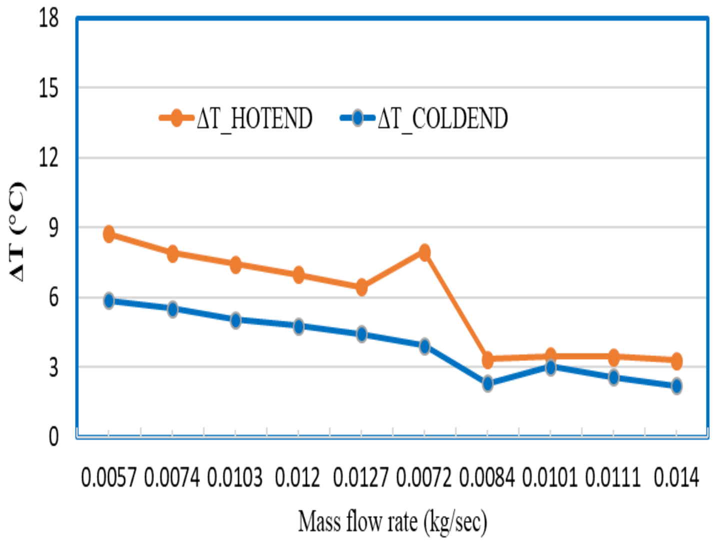


Figure 5 Variation of col and hot temperature difference with mass flow rate.

Fig.7 illustrates the pressure drop in heat exchanger with respect to the mass flow rate and compare also the experimental and theoretical pressure drop. Pressure drop is observed to increase with mass flow rate for all cases. But the experimental pressure drop is much higher than the theoretical pressure drop as in theoretical calculations, we have not considered the pressure losses occurring in piping's and also manufacturing deviations and head losses.

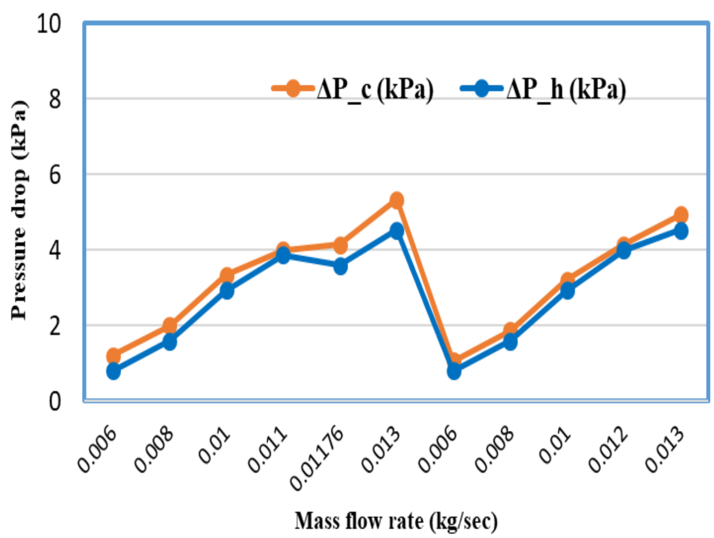


Figure (7) Variation of pressure drop and mas flow rate.

4. Conclusions

The hot test is performed in order to find the performance parameters of the existing ply fin heat exchanger at various mass flow rates with different temperatures. Average effectiveness of 90% is achieved. In both the cases, it is observed that effectiveness and overall thermal conductance enhances with mass flow rate It is also seen that hot fluid effectiveness increases versus flow of the fluid and agrees by 6%when compared with the efficiencies predicted from other correlations reported and that computed using Aspen simulation software. Pressure drop also increases with the increase in mass flow rate and here experimental values are higher than theoretical results because losses in pipe and manufacturing irregularities have been neglected. There exists a maximum value mass flow rate for which exhaust gas temperature is at the lowest level, considering a specific hot inlet temperature. The hot and cold effectiveness of heat exchanger at this point as difference between the two is minimum and we also have minimum imbalance.

5. Conflict of Interest

The authors declare that they have no conflict of interest.

6. Funding Declaration

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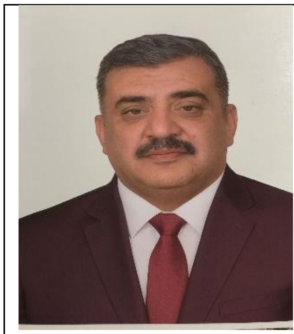	Ahmed Farhan Abdul, Senior Chief Engineer at the Ministry of Oil, Central Refineries Company (Al-Daura Refinery), holds a Bachelor's degree in Marine Mechanical Engineering from the Maritime College in Basra, obtained in 1992. He worked for one year at the Oil Exploration Company in 1993, and then at the Oil Tankers Company until 2008. From 2008 to 2025, he worked at the Central Refineries Company/Al-Daura Refinery in the Maintenance Department/Fixed Equipment Section, specializing in pipework, welding, and tank maintenance.