

Design and Fabrication of a Gasketed Plate Heat Exchanger: Performance Analysis and Comparison with Shell and Tube Heat Exchanger

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Abstract: *This paper presents the design, fabrication, and experimental performance analysis of a gasketed plate heat exchanger (GPHE) operating under counter-flow conditions using water as the working fluid on both sides. The design was based on standard thermodynamic criteria — hot side inlet at 60°C, outlet at 40°C, and cold side inlet at 32°C, outlet at 42°C — with mass flow rates of 0.33 kg/s and 0.65 kg/s respectively. The Log Mean Temperature Difference (LMTD) was calculated as 12.33 K. Heat exchanged on the hot side was 27.88 kW and on the cold side 27.17 kW, with an average of 27.52 kW. The effectiveness-NTU method yielded an effectiveness of 71.4% and NTU of 1.63. The paper further establishes that the gasketed plate heat exchanger offers superior performance, compactness, and sustainability compared to the conventional shell and tube heat exchanger (STHE) for similar duty conditions*

Keywords: LMTD, NTU, Plate Heat Exchanger, Counter-Flow, Effectiveness, Heat Transfer

I. INTRODUCTION

Heat exchangers are one of the most fundamental equipment in thermal engineering, widely used in industries such as food processing, chemical manufacturing, HVAC, and power generation. Among the various types available, the gasketed plate heat exchanger (GPHE) has gained increasing preference in low-to-medium pressure applications due to its compact size, high thermal efficiency, and ease of maintenance.

The shell and tube heat exchanger (STHE) has been the traditional choice for decades. However, with increasing focus on energy efficiency, space optimization, and sustainability, the GPHE presents clear advantages. The objective of this work is to design and fabricate a GPHE based on experimentally determined design criteria, analyse its thermal performance using LMTD and NTU methods, and compare it against a conventional STHE on key performance indicators.

II. LITERATURE REVIEW

Martin [1] established the generalized L  v  que equation for heat transfer in PHEs, which remains the standard reference for thermal design. Kumar [2] demonstrated that PHEs achieve heat transfer coefficients 3–5 times higher than STHEs for equivalent flow conditions. Shah and Focke [3] provided a comprehensive study on the effect of chevron angle on the Nusselt number and friction factor in corrugated plate exchangers, concluding that higher chevron angles improve heat transfer at the cost of increased pressure drop.

Gut and Pinto [4] developed a systematic framework for the configuration and sizing of GPHEs and confirmed their superiority in thermal compactness. Sund  n and Manglik [5] further explored enhanced surface geometries and their role in improving overall heat transfer coefficients. On the sustainability front, Hesselgreaves [6] highlighted that



compact heat exchangers significantly reduce material usage and energy consumption in industrial thermal management systems.

The literature consistently indicates that GPHEs outperform STHes in applications involving moderate pressures and temperatures, particularly where frequent cleaning is necessary or space is a constraint.

III. DESIGN CRITERIA AND FORMULAE

3.1 Working Fluid and Design Conditions

Water is used as the working fluid on both sides of the heat exchanger. The design operates in counter-flow configuration, which maximises the log mean temperature difference and therefore the driving force for heat transfer. It is noted that hot water is less dense than cold water, which is a consideration in practical flow arrangement and plate orientation.

Parameter	Hot Side (Water)	Cold Side (Water)
Inlet Temperature	60°C	32°C
Outlet Temperature	40°C	42°C
Density	986.5 kg/m ³	991.9 kg/m ³
Specific Heat Capacity	4.18 kJ/kgK	4.18 kJ/kgK
Volume Flow Rate	20 L/min	39.5 L/min
Mass Flow Rate	0.33 kg/s	0.65 kg/s

Table 1: Design Conditions for GPHE

3.2 Formulae Used

Mass flow rate: $\dot{m} = Q (L/min) \times \rho (kg/m^3) / 60000$

Heat transfer rate: $Q = \dot{m} \times C_p \times \Delta T$

LMTD (Counter-Flow): $LMTD = (\Delta T_1 - \Delta T_2) / \ln(\Delta T_1 / \Delta T_2)$

where $\Delta T_1 = T_{H,in} - T_{C,out}$ and $\Delta T_2 = T_{H,out} - T_{C,in}$

IV. CALCULATIONS AND RESULTS

4.1 Heat Transfer Rates

Hot Side: $Q = 0.33 \times 4.18 \times (60 - 40) = 27.88 \text{ kW}$

Cold Side: $Q = 0.65 \times 4.18 \times (42 - 32) = 27.17 \text{ kW}$

The small discrepancy of 0.71 kW ($\approx 2.6\%$) between the two sides is attributed to minor heat losses to the surroundings, which is acceptable in experimental work. The average heat exchanged is taken as 27.52 kW.

4.2 LMTD Calculation (Counter-Flow)

$\Delta T_1 = T_{H,in} - T_{C,out} = 60 - 42 = 18^\circ\text{C}$

$\Delta T_2 = T_{H,out} - T_{C,in} = 40 - 32 = 8^\circ\text{C}$

$LMTD = (18 - 8) / \ln(18 / 8) = 10 / \ln(2.25) = 10 / 0.811 = 12.33 \text{ K}$

Figure 1 below shows the temperature distribution of both fluids along the exchanger length in the counter-flow arrangement. The converging temperature profiles confirm a close terminal approach of 8°C at one end and 18°C at the other, which is characteristic of an efficient counter-flow design.



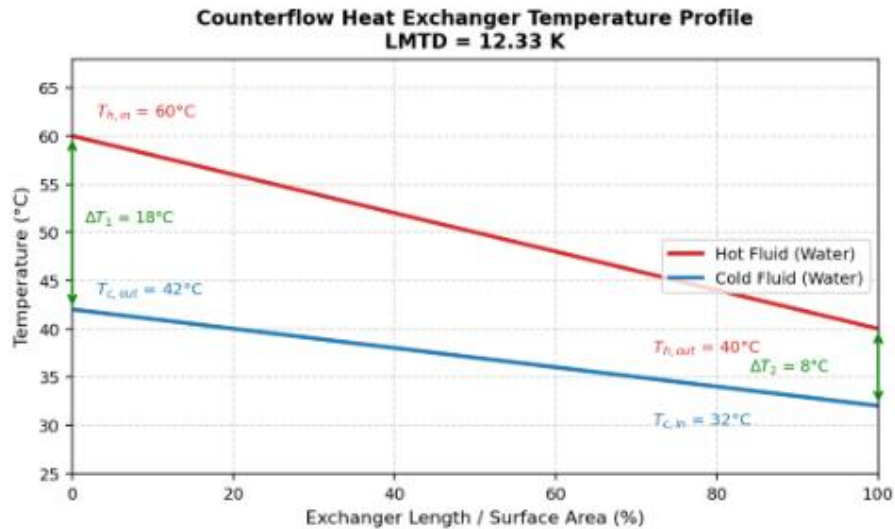


Figure 1: Counter-Flow Temperature Profile (LMTD = 12.33 K)

4.3 Effectiveness-NTU Method

The NTU-effectiveness method provides an independent performance assessment without requiring the outlet temperatures as primary inputs, making it particularly useful for design validation.

$C_{min} = \dot{m}_h \times C_p = 0.33 \times 4.18 = 1.379 \text{ kW/K}$ (Hot side)

$C_{max} = \dot{m}_c \times C_p = 0.65 \times 4.18 = 2.717 \text{ kW/K}$

Capacity Ratio: $Cr = C_{min} / C_{max} = 1.379 / 2.717 = 0.508$

$Q_{max} = C_{min} \times (T_{H,in} - T_{C,in}) = 1.379 \times (60 - 32) = 38.61 \text{ kW}$

Effectiveness: $\epsilon = Q / Q_{max} = 27.52 / 38.61 = 0.714 \rightarrow 71.4\%$

Using the counter-flow effectiveness-NTU correlation for $Cr = 0.508$, the Number of Transfer Units (NTU) evaluates to 1.63. Figure 2 shows the effectiveness-NTU curves for various capacity ratios, with the operating point of the present exchanger clearly marked.

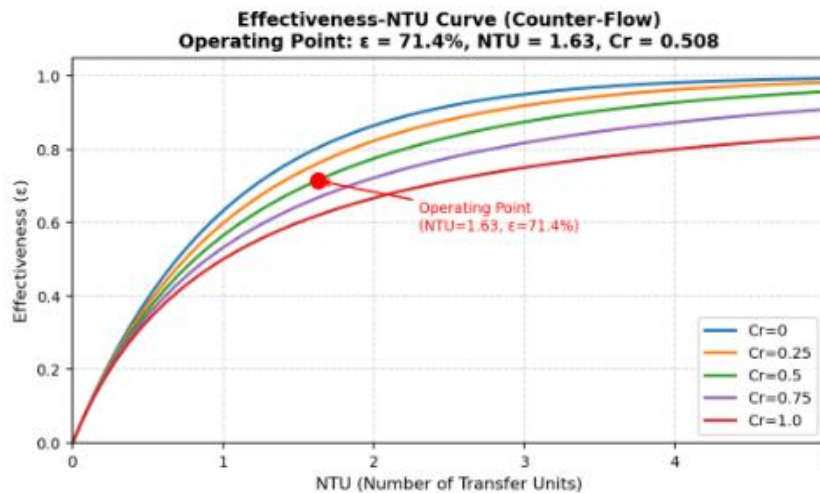


Figure 2: Effectiveness-NTU Curve (Counter-Flow) — Operating Point at $\epsilon = 71.4\%$, NTU = 1.63, Cr = 0.508



Parameter	Value
LMTD	12.33 K
Heat Exchanged (Average)	27.52 kW
Capacity Ratio (Cr)	0.508
Effectiveness (ϵ)	71.4%
NTU	1.63

Table 2: Summary of Thermal Performance Results

V. GPHE VS SHELL AND TUBE HEAT EXCHANGER: COMPARATIVE ANALYSIS

5.1 Thermal Performance

The overall heat transfer coefficient (U) in a GPHE typically ranges from 3000–7000 W/m²K for liquid-liquid duties, whereas in a STHE it is generally between 500–1500 W/m²K under comparable conditions [2]. This difference arises because the corrugated plate geometry in GPHEs induces high turbulence even at low Reynolds numbers, significantly enhancing the convective heat transfer coefficient. For the same heat duty (27.52 kW), a GPHE therefore requires a substantially smaller heat transfer area than a STHE.

5.2 Compactness and Material Efficiency

A GPHE achieves a surface area density of up to 250 m²/m³ compared to only 50–100 m²/m³ in a STHE [3]. The compact plate arrangement allows the entire exchanger to occupy a significantly smaller footprint, directly translating to reduced raw material consumption and lower fabrication cost.

5.3 Maintenance and Cleanability

One of the most practical advantages of a GPHE is its fully dismantlable construction. Plates can be individually removed, inspected, and cleaned without special tooling. In a STHE, cleaning the tube bundle requires chemical descaling or mechanical rodding, both of which involve higher downtime and cost. This is particularly important in food, dairy, and pharmaceutical applications where hygiene standards are strict.

5.4 Sustainability and Energy Efficiency

The higher heat transfer efficiency of a GPHE means less heat is wasted to the environment, directly reducing energy consumption per unit of heat transferred. The compact size also means less insulation material is required. The GPHE's modular design allows plates to be added or removed to scale capacity without replacing the entire unit, extending equipment life and reducing industrial waste.

The counter-flow arrangement further maximises thermal recovery. In the present experiment, a close temperature approach of 8°C was achieved at one end — matching this in a STHE would require a significantly longer and more expensive unit.

5.5 Comparison Summary

Parameter	GPHE	STHE
U (W/m ² K)	3000–7000	500–1500
Surface Area Density (m ² /m ³)	Up to 250	50–100
Cleanability	Easy (dismantable)	Difficult



Temperature Approach	As low as 1°C	~5–10°C
Scalability	Modular (add plates)	Full unit replacement
Operating Pressure	Up to 25 bar	Up to 300+ bar
Sustainability	High	Moderate

Table 3: GPHE vs STHE – Parameter Comparison

VI. LIMITATIONS AND FUTURE SCOPE

The primary limitation of a GPHE is its operating pressure range. Due to the gasketed construction, it is generally not recommended for pressures above 25 bar or temperatures beyond 180°C, limiting its use in high-pressure steam or petrochemical applications where a STHE would be more appropriate. In the present fabricated prototype, gasket leakage at elevated temperatures was identified as a potential concern, consistent with published literature.

Future work can explore semi-welded or fully-welded plate heat exchangers to extend the operating range. Use of nanofluids as the working medium could further enhance the heat transfer coefficient and overall effectiveness beyond the 71.4% achieved here.

VII. CONCLUSION

A gasketed plate heat exchanger was successfully designed and fabricated based on experimentally established design criteria. The LMTD for counter-flow arrangement was 12.33 K, average heat duty was 27.52 kW, effectiveness was 71.4%, and NTU was 1.63 — confirming the design meets the intended thermal duty. The temperature profile graph (Figure 1) visually confirms the counter-flow advantage, while the ϵ -NTU graph (Figure 2) places the operating point clearly within the expected performance envelope for $Cr = 0.508$.

The comparative analysis clearly establishes that a GPHE is thermally superior, more compact, easier to maintain, and more sustainable than a conventional shell and tube heat exchanger for similar process conditions. The only trade-off is in maximum operating pressure, which limits its applicability in extreme industrial conditions.

Conflicts of Interest

The author declares no conflict of interest regarding the publication of this paper.

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