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Performance comparison of Plain and Twisted Tube Heat Exchanger using Analytical and Experimental methods

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***Abstract* - This study presents a comprehensive performance comparison between plain tube and twisted tube heat exchangers, utilizing both analytical and experimental methods. The analytical approach includes mathematical modeling based on heat transfer equations, friction factor, and flow dynamics, allowing for a detailed theoretical prediction of thermal performance under varied operating conditions. Concurrently, an experimental setup is designed to evaluate real-world performance metrics, such as heat transfer rate, pressure drop, and effectiveness, for both heat exchanger types across a range of flow rates and fluid temperatures.**

Our findings indicate that the twisted tube heat exchanger achieves superior thermal performance compared to the plain tube design, due to enhanced fluid mixing and an increased surface area for heat exchange. The experimental results reveal a significant improvement in the heat transfer coefficient for twisted tubes, with up to a 30% increase under similar operating conditions, albeit with a higher pressure drop. Analytical results closely align with experimental data, validating the theoretical model and highlighting the benefits of twisted tube configurations for applications requiring high heat transfer rates. This study underscores the effectiveness of twisted tube heat exchangers in applications where thermal efficiency is critical and provides a robust framework for future optimization and scaling.

Index Terms— **Plain tube heat exchanger, Heat transfer enhancement, Thermal performance, Heat transfer coefficient, Energy efficiency.**

I. INTRODUCTION

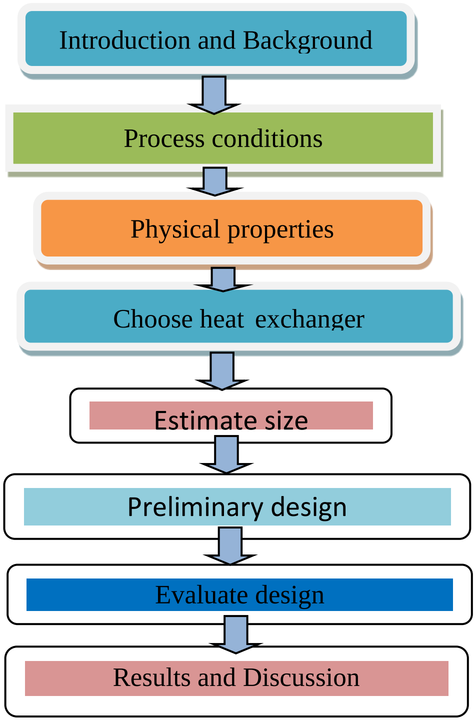
Heat exchangers are critical components in many industrial applications, including power generation, refrigeration, chemical processing, and HVAC systems, where efficient thermal management is essential for optimizing performance and reducing energy consumption. The efficiency of a heat exchanger is directly influenced by its design, specifically the tube geometry, which affects the rate of heat transfer, pressure drop, and overall thermal performance. Conventional heat exchangers with plain tube configurations are widely used due to their straightforward design and predictable thermal behavior. However, they often exhibit limitations in heat transfer efficiency, particularly under conditions where enhanced heat exchange is needed.

In recent years, twisted tube heat exchangers have gained attention as an alternative to plain tube designs due to their potential for higher heat transfer rates and improved thermal performance. The twisted tube configuration promotes secondary flow patterns and turbulence, enhancing fluid mixing and increasing the effective heat transfer surface area. This design modification can lead to a significant boost in thermal efficiency but may also introduce higher pressure drops, impacting the system's pumping power requirements and overall energy consumption.

This study aims to conduct a detailed performance comparison between plain and twisted tube heat exchangers using both analytical and experimental methods. The analytical approach involves developing mathematical models to predict thermal performance and pressure drop based on fluid properties, flow characteristics, and tube geometry. These models provide theoretical insights into the heat transfer mechanisms for each configuration, allowing for an initial evaluation of the potential benefits and trade-offs of using twisted tube designs.

An experimental setup is also developed to measure real-world performance parameters, including heat transfer coefficient, effectiveness, and pressure drop, across a range of flow rates and temperature conditions. By comparing analytical predictions with experimental data, this study seeks to validate the theoretical models and provide a comprehensive understanding of the performance dynamics of plain versus twisted tube heat exchangers. The findings of this research aim to inform design choices and operational strategies for heat exchanger applications where high efficiency is paramount, contributing to advancements in heat exchanger technology and energy-efficient systems.

II. METHODOLOGY



III. MATHEMATICAL MODELING

Heat exchangers are critical components in thermal systems, designed to transfer heat efficiently between two or more fluids. Among the various types of heat exchangers, plain and twisted tube heat exchangers have garnered attention due to their contrasting characteristics. The mathematical modeling of these systems offers insights into their thermal and hydraulic performance, enabling predictions of efficiency and optimization under various conditions.

Parallel flow (NTU Method)
Formulae used;
i) Mean flow velocity; Vm

$$Vm = \frac{4\dot{m}}{\pi \rho D_i^2} \frac{m}{s}$$

ii) Reynolds number; Re

$$Re = \frac{\rho V_m D_i}{\mu};$$

For; Turbulent Flow-Re > 10^4
iii) Friction factor (for smooth tubes at turbulent flow); f

$$f = 0.184\, R_e^{-0.2}$$

ii) Nusselt number (for smooth tubes at turbulent flow); N_u

$$N_u = \frac{\frac{f}{8} * R_e * P_r}{1.07+[12.7*(\frac{f}{8})^{0.5}*(P_r^{0.67}-1)]} * (\frac{\mu_m}{\mu_w})^n$$

For;
Rough/Smooth tubes; n=0
Convective heat transfer co-efficient; h

$$h = \frac{kN_u}{D_i} \frac{W}{m^2K}$$

Heat Capacity values; C_{max} & C_{min}

$$C_x = \dot{m} * C_p \frac{W}{K}$$

Capacity ratio; C

$$C = \frac{C_{min}}{C_{max}}$$

Heat transfer rate: $\dot{Q}$max

$$\dot{Q}max = Cmin\ \Delta T\ KW$$

$$\Delta T = (Tt\ i/p - Ts\ i/p)\ K$$

The transfer on surface area: As
As = πDitL m2
Thermal Resistance; Rth

$$Rth = \frac{1}{UA} = \frac{1}{\pi L}(\frac{1}{h_i D_{it}} + \frac{\ln\frac{D_{ot}}{D_{it}}}{2k} + \frac{1}{h_o D_{ot}})$$

For parallel flow;

$$NTU = \frac{UA}{C_{min}} = \frac{1}{C_{min} * R_{th}}$$

The theoretical effectiveness for a parallel flow; (ϵ_{theory})

$$\epsilon_{theory} = \frac{1-e^{-NTU(1+C)}}{1+C}$$

The Analytical effectiveness for a parallel flow; ($\epsilon_{analytical}$)

$$\epsilon_{analytical} = \frac{T_{i_t} - T_{o_t}}{T_{i_t} - T_{i_s}}$$

T_{i_t}- Hot fluid input or initial temperature at tube side
T_{o_t}- Hot fluid output or final temperature at tube side
T_{i_s}- Cold fluid input or initial temperature at shell side
Comparing both theoretical and analytical effectiveness; ϵ

$$\epsilon = \frac{\epsilon_{theory} - \epsilon_{analytical}}{\epsilon_{theory}} \times 100$$

For Tube Side;
Tube side Mean flow velocity; Vmt

$$Vmt = \frac{4\dot{m}_t}{\pi \rho_t D_{it}^2}$$

Kinematic viscosity; ϑ_t

$$\mu_t = \vartheta_t * \rho_t \frac{Kg}{ms}$$

$$\rho_t = 982.28 \frac{Kg}{m^3}$$

$$\mu_t = 0.0005 \text{ Pa s}$$

$$\therefore \vartheta_t = \frac{\mu_t}{\rho_t} \frac{m^2}{s}$$

Reynolds number of the tube; Ret

$$Ret = \frac{\rho_t V_{mt} D_{it}}{\mu_t}$$

Friction factor (for smooth tubes at turbulent flow); f_t

From data hand book;

f_t = 0.184 $R_{et}^{-0.2}$

Nusselt number (for smooth tubes at turbulent flow); N_{ut}

$$N_{ut} = \frac{\frac{f_t}{8} * R_{et} * P_{rt}}{1.07 + [12.7 * (\frac{f_t}{8})^{0.5} * (P_{rt}^{0.67} - 1)]} * (\frac{\mu_m}{\mu_w})^n$$

Tube side Convective heat transfer co-efficient; h_t

$$h_t = \frac{k_t N_{ut}}{D_{it}} \quad \frac{W}{m^2 K}$$

For Shell Side;

i) Shell side Mean flow velocity; Vms

$$\text{Vms} = \frac{4\dot{m}_s}{\pi \rho_s (D_{is}^2 - D_{ot}^2)}$$

$$\dot{m}_s = \frac{V_{ms} * \pi * \rho_s * (D_{is}^2 - D_{ot}^2)}{4} \quad \frac{Kg}{s}$$

ii) Kinematic viscosity; ϑ_s

$$\mu_s = \vartheta_s * \rho_s \frac{Kg}{ms}$$

iii) Reynolds number of the Shell; Res

$$\text{Res} = \frac{\rho_s V_{ms} (D_{is} - D_{ot})}{\mu_s};$$

Hence the flow is TURBULENT

Friction factor (for smooth tubes at turbulent flow); f_s

$$N_{us} = \frac{\frac{f_s}{8} * R_{es} * P_{rs}}{1.07 + [12.7 * (\frac{f_s}{8})^{0.5} * (P_{rs}^{0.67} - 1)]} * (\frac{\mu_m}{\mu_w})^n$$

Shell side Convective heat transfer co-efficient; h_s

$$h_s = \frac{k_s N_{us}}{(D_{is} - D_{ot})} \quad \frac{W}{m^2 K}$$

Capacity values; C_{max} & C_{min}

Capacity of tube; Ct

Ct = $\dot{m}$t $*$ Cpt

Capacity of Shell; Cs

Cs = $\dot{m}$s $*$ Cps

5.3) Capacity ratio; (C)

$$C = \frac{C_{min}}{C_{max}}$$

The mass heat transfer rate; $\dot{Q}$max

$\dot{Q}$max = Cmin ΔT KW

$\dot{Q}$max = Cmin (Tt i/p – Ts i/p) KW

v) The transfer on surface area; As

As = π $*$(DitL) m2

v) Thermal Resistance; (Rth)

$$\text{Rth} = \frac{1}{UA} = \frac{1}{\pi L}\left(\frac{1}{h_t D_{it}} + \frac{\ln\frac{D_{ot}}{D_{it}}}{2k} + \frac{1}{h_s D_{ot}}\right) \frac{K}{W}$$

vi) For Parallel flow;

$$\text{NTU} = \frac{UA}{C_{min}} = \frac{1}{C_{min} * R_{th}}$$

vii) The theoretical effectiveness for a parallel flow; ($\in_{theory}$)

$$\in_{theory} = \frac{1 - e^{-NTU(1+C)}}{1+C}$$

viii) The Analytical effectiveness for a parallel flow; ($\in_{analytical}$)

$$\in_{analytical} = \frac{T_{i_t} - T_{o_t}}{T_{i_t} - T_{i_s}}$$

ix) Comparing both theoretical and analytical effectiveness; $\in$

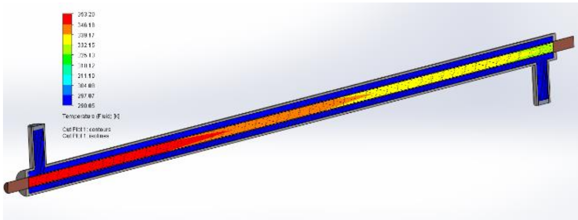
$$\in = \frac{\in_{theory} - \in_{analytical}}{\in_{theory}} \text{ x } 100$$

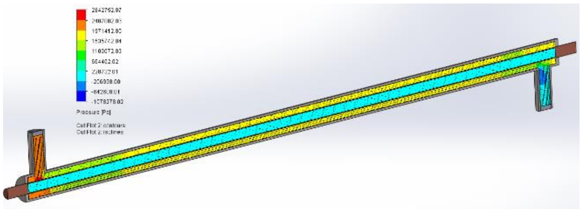
TWISTED TYPE HEAT EXCHANGE:

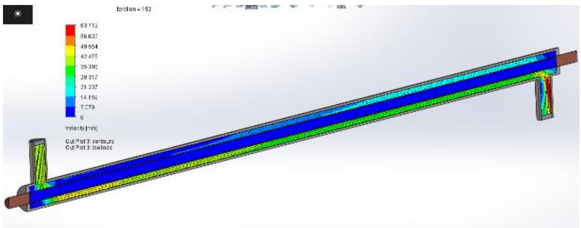
TUBE SIDE	SHELL SIDE
Fluid (water) at 353.2 K;	Fluid (water) at 290.2 K;
$C_{pt} = \rho\omega = 4180.2 \frac{J}{Kg\,K}$	$C_{ps} = \rho\omega = 4186.4 \frac{J}{Kg\,K}$
ρ_t=990.09 $\frac{Kg}{m^3}$	ρ_s= 998.1 $\frac{Kg}{m^3}$
μ_t= 0.0006 Pa s	μ_s= 0.0011 Pa s
P_{rt}=4.03	P_{rs}=7.49
K_t=0.6354 $\frac{W}{m\,K}$	K_s=0.5943 $\frac{W}{m\,K}$
$\dot{m}_t$ = 0.2 $\frac{Kg}{s}$	$\dot{m}_s$ = x $\frac{Kg}{s}$
V_{mt} = x $\frac{m}{s}$	V_{ms} = 40 $\frac{m}{s}$

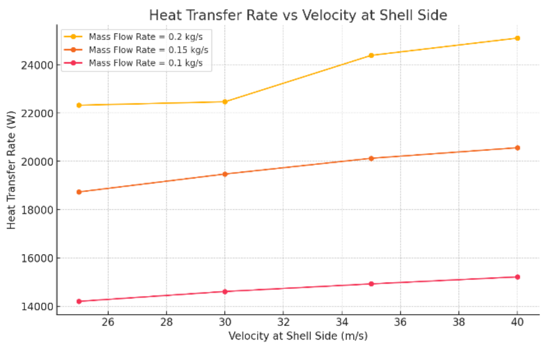
IV. CFD ANALYSIS AND EXPERIMENTAL VALIDATION

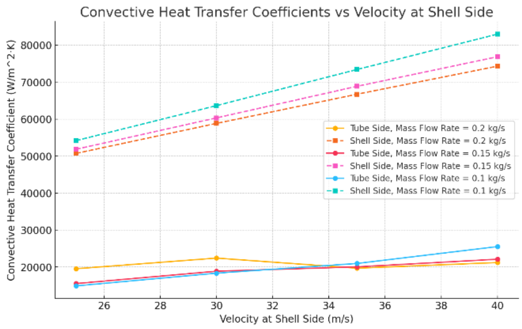
PLAIN TYPE TUBE AND SHELL HEAT EXCHANGER





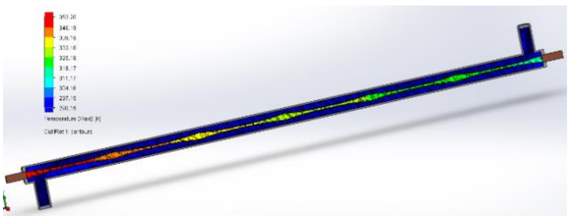


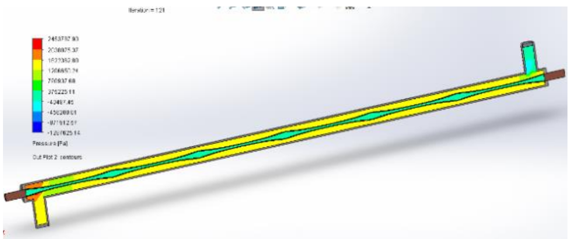


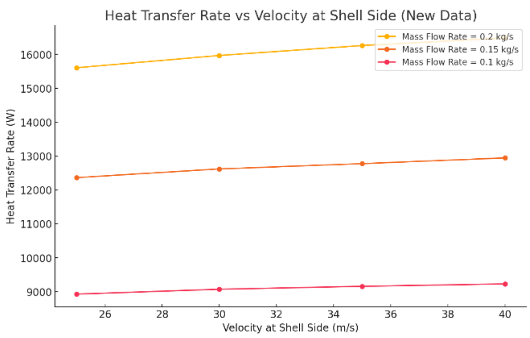


The graphs above display the following:
Heat Transfer Rate vs. Velocity at Shell Side
As the velocity at the shell side increases, the heat transfer rate also increases for all mass flow rates. Higher mass flow rates (e.g., 0.2 kg/s) exhibit higher heat transfer rates compared to lower flow rates.
Convective Heat Transfer Coefficients vs. Velocity at Shell Side.
Both tube-side and shell-side convective heat transfer coefficients increase with increasing shell-side velocity.
The shell-side coefficient is consistently higher than the tube-side coefficient across all flow rates.
For each mass flow rate, the tube-side and shell-side coefficients exhibit a noticeable increase as velocity rises.

TWISTED TYPE SHEEL AND TUBE HEAT EXCHANGER







The graphs above represent the analysis of the new dataset:
Heat Transfer Rate vs. Velocity at Shell Side:
For all mass flow rates, the heat transfer rate increases with velocity at the shell side.
The heat transfer rate is highest for a mass flow rate of 0.2 kg/s compared to lower mass flow rates.
Convective Heat Transfer Coefficients vs. Velocity at Shell Side:
The convective heat transfer coefficients at both tube and shell sides show varying trends.
For the tube side, the coefficient generally increases with velocity but exhibits variability.
For the shell side, the coefficient fluctuates significantly, particularly for lower mass flow rates, indicating dynamic changes in flow and heat transfer characteristics.

V. RESULTS AND DISCUSSION

The analysis of the heat exchanger's performance, based on the provided data, highlights the influence of tube-side mass flow rate (m) and shell-side velocity on heat transfer characteristics. The following results and discussions are drawn:
The heat transfer rate increases consistently with higher shell-side velocity for all mass flow rates. The highest heat transfer rates were observed at a mass flow rate of 0.2 kg/s, with values ranging from 15,607 W (at Vs=25 m/s to 16,485 W (at Vs=40 m/s).
An increase in shell-side velocity enhances turbulence and heat transfer, leading to better thermal performance. Larger mass flow rates result in higher heat transfer due to increased fluid energy available for transfer.
For a mass flow rate of 0.2 kg increases steadily with velocity, peaking at 44,304 W/m²·K. Lower mass flow rates show lower values, with ranging between 38,123 and 114,563 W/m²·K.
The trend suggests that higher flow rates and velocities lead to improved heat transfer coefficients due to greater fluid turbulence.
The shell-side coefficient exhibits significant variability, especially for 0.15 kg/s. These variations may be attributed to complex flow dynamics and localized turbulence in the shell-side region. Higher mass flow rates lead to smoother trends indicating improved thermal efficiency under steady operating conditions.
The output temperature decreases with increasing shell-side velocity across all mass flow rates.This trend indicates enhanced heat dissipation with increased shell-side velocity.

Exhibits the highest heat transfer rates and relatively stable convective heat transfer coefficients. Suitable for applications requiring high heat removal rates.

Exhibits lower heat transfer rates but higher variability indicating potential inefficiencies due to fluctuating flow patterns. Suitable for low-energy systems where moderate heat transfer is sufficient.

The results demonstrate a clear relationship between shell-side velocity and heat transfer performance:

Effect of Mass Flow Rate: Higher tube-side mass flow rates generally result in improved performance metrics due to higher energy and turbulence within the system

The variability in shell-side heat transfer coefficients suggests that shell-side flow dynamics should be optimized for consistent performance.

In conclusion, the heat exchanger performs optimally at higher mass flow rates and moderate shell-side velocities, making these conditions ideal for efficient thermal performance. Further experimental and computational studies can refine the observed trends and validate these findings.

VI. CONCLUSION

The comparative performance analysis of the plain and twisted tube heat exchanger highlights the significant influence of tube-side mass flow rate (m) and shell-side velocity (Vs) on heat transfer characteristics. Key conclusions drawn from the study are as follows:

Heat Transfer Performance:

The heat transfer rate (Q) improves consistently with increasing shell-side velocity (Vs) across all mass flow rates.

The highest heat transfer rates were observed for the highest mass flow rate (m=0.2 kg/s), indicating that larger mass flow rates enhance energy transfer capacity.

Convective Heat Transfer Coefficients:

Tube-side convective heat transfer coefficients (h_{in}) increased steadily with both higher velocities and mass flow rates, driven by improved fluid turbulence.

Shell-side coefficients (h_{out}) exhibited variability, particularly for lower mass flow rates, reflecting the complex flow dynamics within the shell.

Output Temperature:

The output temperature (h_{out}) decreased with increasing shell-side velocity, demonstrating the enhanced heat dissipation capability of the system at higher velocities.

Performance Optimization:

Higher tube-side mass flow rates combined with moderate shell-side velocities provide the best thermal performance, offering a balance between heat transfer efficiency and system stability.

The study demonstrates the potential of twisted tube heat exchangers to outperform plain tube designs under certain operating conditions. Future work could involve advanced experimental and numerical studies to further understand the complex flow dynamics and refine heat exchanger design for specific industrial applications.

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