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ESTIMATION OF OVERALL HEAT TRANSFER COEFFICIENT AND EFFECTIVENESS IN A DOUBLE PIPE HEAT EXCHANGER OF A FOOD DRYING SYSTEM

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Abstract: The study explores the design and evaluation of the heat transfer parameters of a food drying system using waste heat. The system uses engine exhaust gas to preheat air, reducing energy consumption. The system was modeled using CATIA and Computational Fluid Dynamics (CFD) simulations using ANSYS Fluent. Mesh convergence studies ensured accuracy. The study found that the heat recovery system significantly improved the heat transfer rate of the drying process. Heat transfer utilization was quantified, revealing that the system could lower energy requirements, reduce operational costs, and improve sustainability. Implementing such systems also contributes to reducing carbon emissions and mitigating global warming. The obtained outcomes reveal a significant agreement between the conclusions drawn from the analytical and numerical methodologies.

Keywords: Food Dryer, Heat Exchanger, ANSYS 15, Waste heat.

List of symbols:

A	area
Cp	specific heat, J/kg-K
D	diameter, m
f	friction factor
h	heat transfer coefficient
k	thermal conductivity, W/m-K
L	length, m
Nu	Nusselt number
P_r	Prandtl number
Q	heat transfer rate, w
Re	Reynolds number
T	temperature, K
U	Overall heat transfer coefficient, W/m^2K
V	velocity, m/s

Greek Symbols

ρ	Density kg/m^3
υ	Viscosity kg/m-s
Δp	Pressure drops
ΔT	Temperature difference
ε	Effectiveness

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I INTRODUCTION

In recent years, the rising demand for energy efficiency and sustainability has directed significant attention toward waste heat recovery systems. These systems offer an effective way to harness energy that would otherwise be lost, contributing to improved operational efficiency and reduced environmental impact. Waste heat, often generated in industrial, automotive, and domestic applications, represents a largely untapped resource that can be repurposed to enhance energy systems and support sustainable development. Fathieh et al. (2015) [1] explored the effectiveness of heat exchangers by employing transient step change methods, highlighting their ability to optimize heat transfer processes. Their work underscores the importance of evaluating heat exchanger designs to improve energy utilization in industrial applications. Similarly, Wang et al. (2014) [2] investigated low-grade waste heat recovery using thermoelectric systems, demonstrating the feasibility of converting waste heat into useful energy for applications such as power generation and heating. These studies emphasize the potential of integrating advanced materials and methods to enhance waste heat recovery efficiency. Industrial processes often generate substantial amounts of waste heat, which can be repurposed for heating and cooling applications. Brückner et al. (2015) [3] identified the potential for such applications in industrial waste heat management, suggesting that innovative heat recovery technologies could play a critical role in reducing carbon footprints and improving energy systems. On a smaller scale, Khaled et al. (2015) [4] conducted an experimental analysis of waste heat utilization for water heating, showcasing the practical implications of waste heat recovery in everyday systems. HVAC systems also represent a significant opportunity for waste heat recovery, as explored by Ramadan et al. (2016) [5]. Their study demonstrated how waste heat from HVAC systems can be redirected to produce domestic hot water, illustrating the versatility and effectiveness of heat recovery in reducing energy consumption in residential and commercial buildings.

Finally, Chaudhar et al. (2012) [6] evaluated the performance of shell-and-tube heat exchangers, which are commonly employed in waste heat recovery systems due to their robustness and efficiency in transferring heat between fluids. Building upon these findings, this study focuses on utilizing a double pipe heat exchanger to recover waste heat from engine exhaust gases for food drying applications. Food drying is an energy-intensive process that traditionally relies on electricity or fossil fuels, leading to high energy costs and greenhouse gas emissions. By repurposing exhaust heat, this work aims to reduce the energy demand for food drying, enhance efficiency, and contribute to sustainable food processing practices. The results of this study could provide valuable insights into waste heat recovery applications across various industries. Mounika et al. (2016) [7] The heat transfer processes take place from the coolant to the tubes then from the tubes to the air through the fins. After the analysis is carried out, the heat transfer coefficient of air and ethylene glycol is estimated and further overall heat transfer coefficient is calculated.

Kishore et al. (2010) [8] The momentum and thermal eddy diffusivity characteristics are evaluated from the experimental heat transfer and pressure drop data. It is observed that the influence of nanoparticles in the base liquid water on the momentum eddy transport is not perceptible for the range of concentration considered. However, the presence of nanoparticles in water yielded enhancements in convective heat transfer. An improvised method for the evaluation of both eddy momentum and thermal diffusivities as a function of dimensionless velocity u+ and distance y+ is presented.

PK Sarma et al. (2010) [9] The present investigation deals with a differential formulation to estimate the eddy diffusivity together with the universal velocity for fully developed turbulent flows in a tube. The subsequent theoretical predictions of wall friction coefficients and Nusselt numbers are in reasonable agreement with the classical solution of Blasius wall friction coefficient and Dittus and Boelter correlation for heat transfer, $Nu=0.023\ Re^{0.8}\ Pr^{1/3}$, The prediction of Nusselt numbers from the eddy diffusivity expression satisfactorily agree with the Dittus and Boelter heat transfer correlation.

Kishore et al. 2001[10] The development of high-performance thermal systems has stimulated interest in methods to augment or intensify heat transfer rates. The performance of conventional heat exchangers can be substantially improved by a number of augmentation techniques. A good amount of research effort of both theoretical and experimental nature can be found in the literature defining the conditions under which an augmentation technique will improve heat and mass transfer rates.

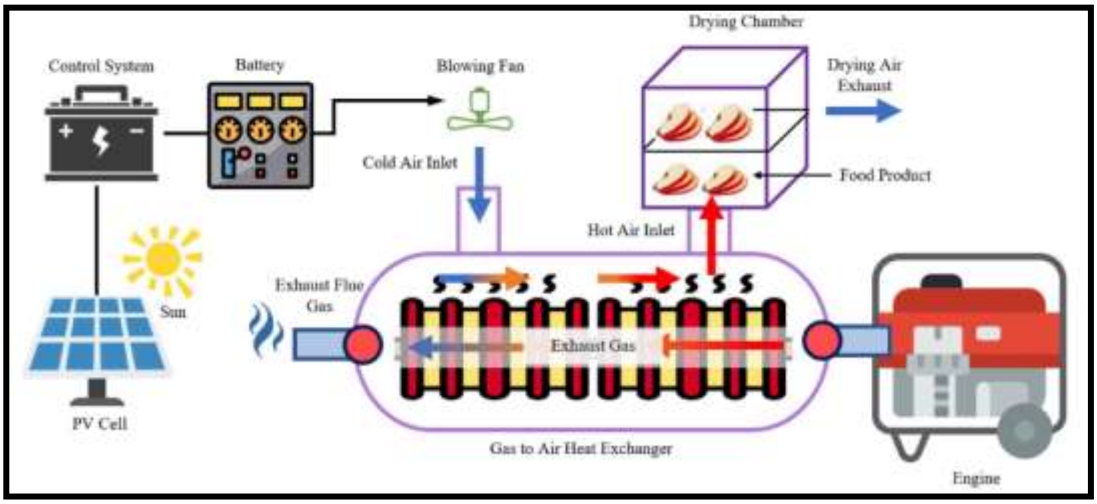

Fig: 1. food drying system from exhaust gas constructional details.

Table: 1. Available sources of waste heat

Several sources of waste heat (Masud et al.2019) [11]

Sources of waste heat	Exhaust temperature (K)
Diesel engine	573.15–873.15
Gas turbine	643.15–816.15
Ceramic kiln	473.15–573.15
Cement kiln	473.15–623.15/573.15–723.15
Brick kiln	873.15–973.15
Container glass melting	433.15–473.15/413.55–433.15
Boiler	503.15
Food industry	437.15
Conventional incinerator	1033.15

II CFD MODELLING

The configurations utilized in this investigation are chosen in accordance with the information gathered from the literature survey. The ANSYS Design Modeler is utilized to establish the geometric model, facilitating rapid modifications when needed. Initially, the ANSYS Workbench interface is initiated, unveiling a variety of solvers and analysis alternatives situated on the left side. The selection is made for flow analysis using FLUENT, encompassing tasks such as geometry manipulation, meshing, solving, analysis, and interpretation of results.

The geometry of the double-pipe heat exchanger was created using CATIA software. The design consisted of a concentric tube-in-tube arrangement, where the exhaust gases flowed through the inner tube and ambient air through the outer annulus. Key geometric parameters included:

- Inner tube diameter: 165 mm
- Outer shell diameter: 450 mm
- Length of the heat exchanger: 2000 mm

The inlet and outlet ports for both fluids were designed to ensure smooth flow transitions and effective heat transfer. The geometry was imported into ANSYS Fluent using the STEP file format for further CFD analysis.

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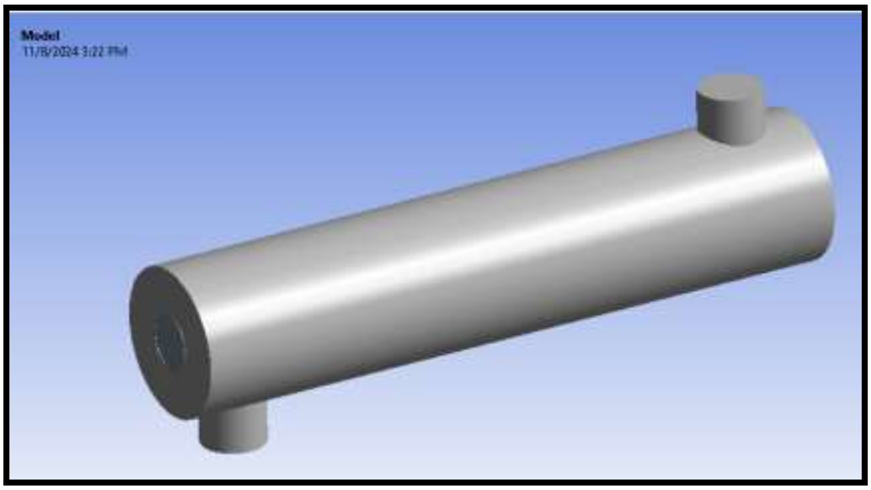

Fig: 2. Geometry model of double pipe heat exchanger.

After the model is prepared, the next phase entails generating the mesh. The material for the complete domain is specified as "FLUENT," and each boundary within the domain is identified for partitioning, with the goal of forming a tetrahedral mesh arrangement. By performing a right-click on the "mesh" choice, the creation menu becomes accessible, and the meshing process is initiated through this selection.
Meshing of the fluid domain was performed using tetrahedral elements to ensure a high-resolution grid suitable for complex flow patterns. The mesh was refined in critical regions, such as near the inlet and outlet ports, to capture gradients in velocity and temperature accurately.
The mesh consisted of:

- Nodes: 358,833
- Elements: 204,991

A mesh independence study was conducted to ensure the simulation results were not influenced by mesh density.

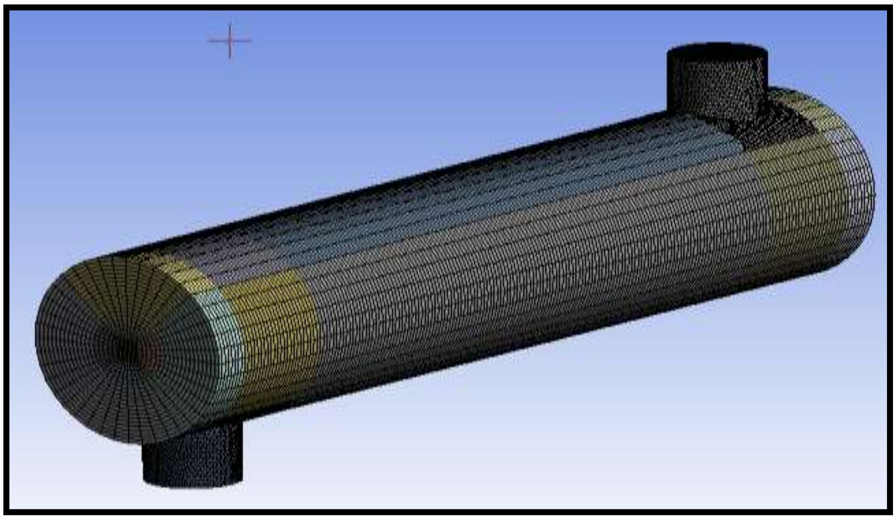

Fig: 3. Meshing model of double pipe heat exchanger

The following boundary conditions were applied:

- Inlet 1 (Exhaust gas): Velocity of 25 m/s, temperature of 600°C
- Inlet 2 (Ambient air): Temperature of 25°C, varying velocity based on Reynolds numbers ranging from 157,846 to 229,576.
- Outlets 1 and 2: Pressure outlet with standard atmospheric pressure.

The flow was modeled as turbulent, and the k-ε turbulence model was chosen for its robustness and efficiency in handling industrial-scale flows. The simulation also highlighted minimal pressure drops, ensuring system efficiency. Variations in Reynolds numbers demonstrated the impact of flow dynamics on the overall heat transfer coefficient and frictional losses.

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CFD modelling proved to be a powerful tool in optimizing the design and performance of the heat exchanger, confirming its viability for waste heat recovery in food drying applications.

III MATHEMATICAL MODELLING

Hydraulic Diameter (D_h):

The hydraulic diameter is determined using D_h = Do - D_i, where Do is the outer diameter and D_i is the inner diameter. It is used to calculate the Reynolds number and other flow-related parameters.

$$D_h = D_0 - D_I \qquad \text{Eq (1)}$$

$$A = \frac{\pi}{4} \times D^2 \qquad \text{Eq (2)}$$

$$m = \rho A V \qquad \text{Eq (3)}$$

$$Q = m c_p \Delta T \qquad \text{Eq (4)}$$

Reynolds number:

$$R_e = \frac{\rho V D}{\vartheta} \qquad \text{Eq (5)}$$

Prandtl number:

$$P_R = \frac{\vartheta . C_p}{K} \qquad \text{Eq (6)}$$

Nusselt number:

$$Nu = 0.023 (R_e)^{0.8} (P_r)^n \quad n=0.3 \text{ for heating} \quad n = 0.4 \text{ for cooling} \qquad \text{Eq (7)}$$

Heat Transfer Coefficient (h):

The convective heat transfer coefficient h is determined using the Nusselt number correlation for turbulent flow, Nu = 0.023xRe^0.8 x Pr^0.3 for heating, followed by calculating h = Nu x k/D, where k is the thermal conductivity of the fluid and D is the tube diameter. This coefficient governs the rate of heat transfer at the tube surface

$$h = \frac{Nu . k}{d} \qquad \text{Eq (8)}$$

Friction Factor (f):

The friction factor f for smooth tubes is calculated using f = 0.184 / Re^{0.2}. This value is used to estimate the pressure drop in the system due to fluid flow resistance.

$$f = \frac{0.184}{(R_E)^{0.2}} \qquad R_e \geq 2300 \text{ for smooth tubes} \qquad \text{Eq (9)}$$

Pressure Loss:

$$\Delta p = \frac{f L \rho V^2}{2D} \qquad \text{Eq (10)}$$

Overall Heat Transfer Coefficient:

$$\frac{1}{U_0} = \frac{1}{\left(\frac{r_0}{r_i}\right)\frac{1}{h_i} + \frac{r_0}{k}\ln\left(\frac{r_0}{r_i}\right) + \frac{1}{h_0}} \qquad \text{Eq (11)}$$

$$(\Delta T)_m = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln\left(\frac{T_1 - t_2}{T_2 - t_1}\right)} \qquad \text{Eq (12)}$$

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Heat Transfer Area (A):

The heat transfer area is calculated using the inner diameter of the tube and its length. This area defines the surface through which heat exchange occurs between the exhaust gas and ambient air.

$$A=\pi DL \qquad \text{Eq (13)}$$

Heat Transfer Rate (Q):

The heat transfer rate Q is calculated using the formula Q = U A Delta Tm, where U is the overall heat transfer coefficient, A is the heat transfer area, and Delta Tm is the mean temperature difference between the hot and cold fluids. This quantifies the total heat exchanged in the system.

$$Q = UA\left(\Delta T\right)_m \qquad \text{Eq (14)}$$

$$C = \frac{C_{\min}}{C_{\max}} \qquad \text{Eq (15)}$$

Number of Transfer Units (NTU):

The NTU is calculated as NTU = UA/C, where U is the heat transfer coefficient, A is the heat transfer area, and C is the heat capacity rate. This dimensionless quantity helps assess the efficiency of the heat exchanger.

$$NTU = \frac{UA}{C_{\min}} \qquad \text{Eq (16)}$$

Effectiveness(ε):

The effectiveness (ε) of a heat exchanger is a measure of its ability to transfer heat relative to the maximum possible heat transfer under ideal conditions.

$$\varepsilon = \frac{1-Exp\left[-N\left(1-C\right)\right]}{1-CExp\left[-N\left(1-C\right)\right]} \qquad \text{Eq (17)}$$

IV RESULTS AND DISCUSSION

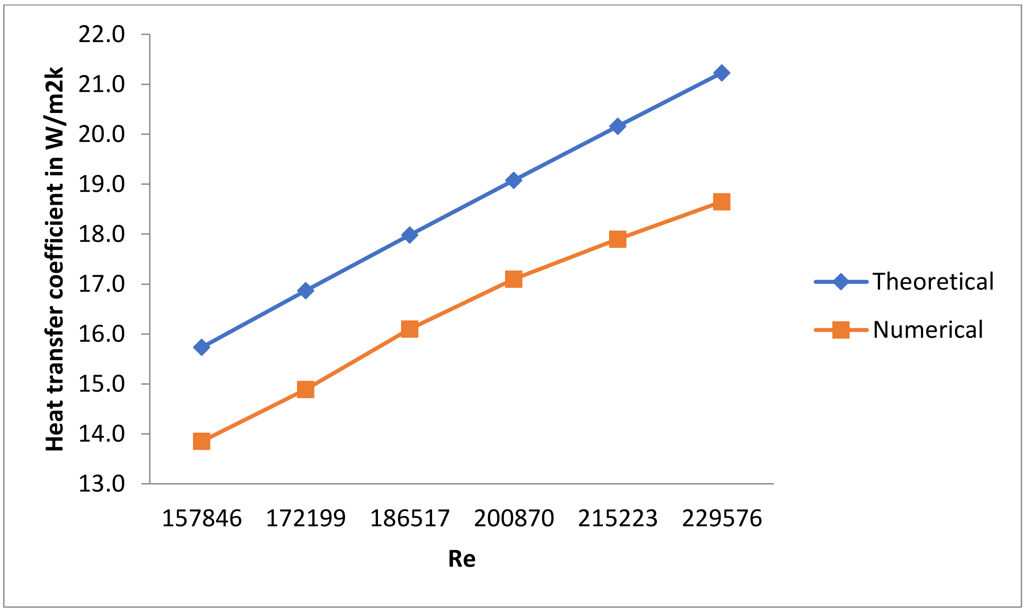

Fig: 4. Heat Transfer Coefficient with Reynolds Number

The heat transfer coefficient is shown as a function of the Reynolds number, with both theoretical and analytical values showing a positive correlation. The analytical values consistently underestimate the theoretical values, with the maximum theoretical coefficient at 229576 and the minimum theoretical value at 157846. The maximum heat transfer coefficient is 18.7 at 229576, while the minimum is 13.9 at 157846. The difference between theoretical and analytical values is approximately 11.7% and 12.8%, respectively.

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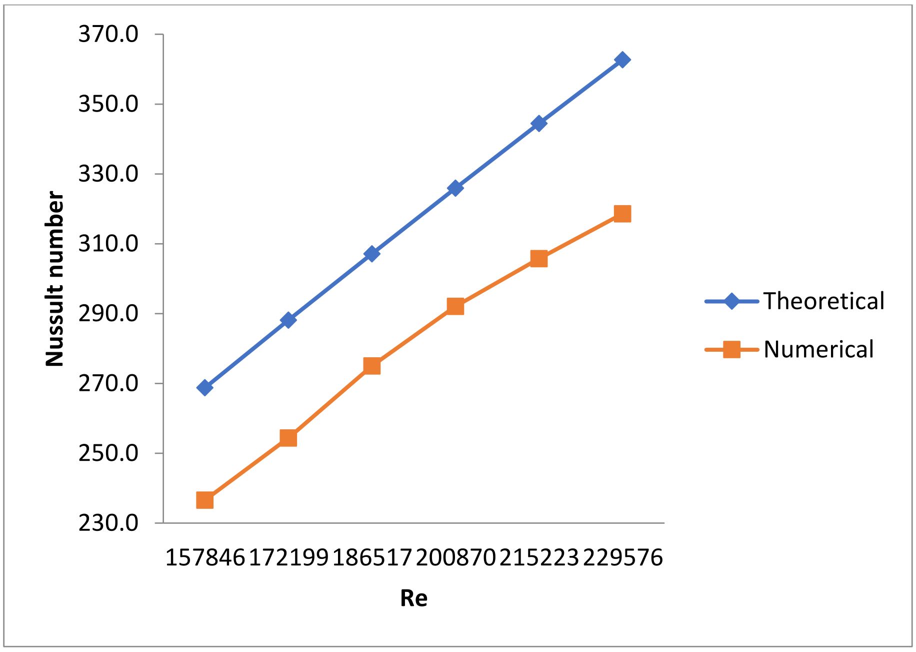

Fig: 5. Nusselt Number with Reynolds Number

The Nusselt number is a measure of the friction between two surfaces. It is directly proportional to the Reynolds number, with a positive correlation between the two. The numerical data shows that as the Reynolds number increases, both theoretical and analytical Nusselt numbers also increase, with analytical values consistently lower than theoretical ones. The maximum theoretical Nusselt number is at 229576, while the maximum analytical Nusselt number is at 236.6. The difference between theoretical and analytical values is approximately 12.2%.

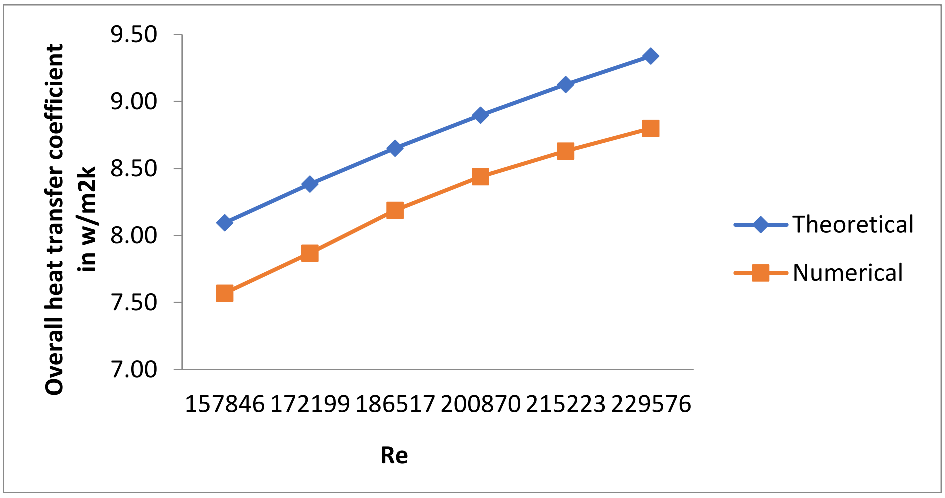

Fig: 6. Overall Heat Transfer Coefficient with Reynolds Number

The Fig illustrates the relationship between the Reynolds number and the overall heat transfer coefficient, showing a positive correlation. As the Reynolds number increases, both theoretical and analytical heat transfer coefficients also increase, indicating a consistent trend. The maximum theoretical heat transfer coefficient is 9.34 W/m²K at a Reynolds number of 229576, while the minimum theoretical value is 8.09 W/m²K at a Reynolds number of 157846. The analytical values show a maximum overall heat transfer coefficient of 8.80 W/m²K at 229576 and a minimum of 7.57 W/m²K at 157846.

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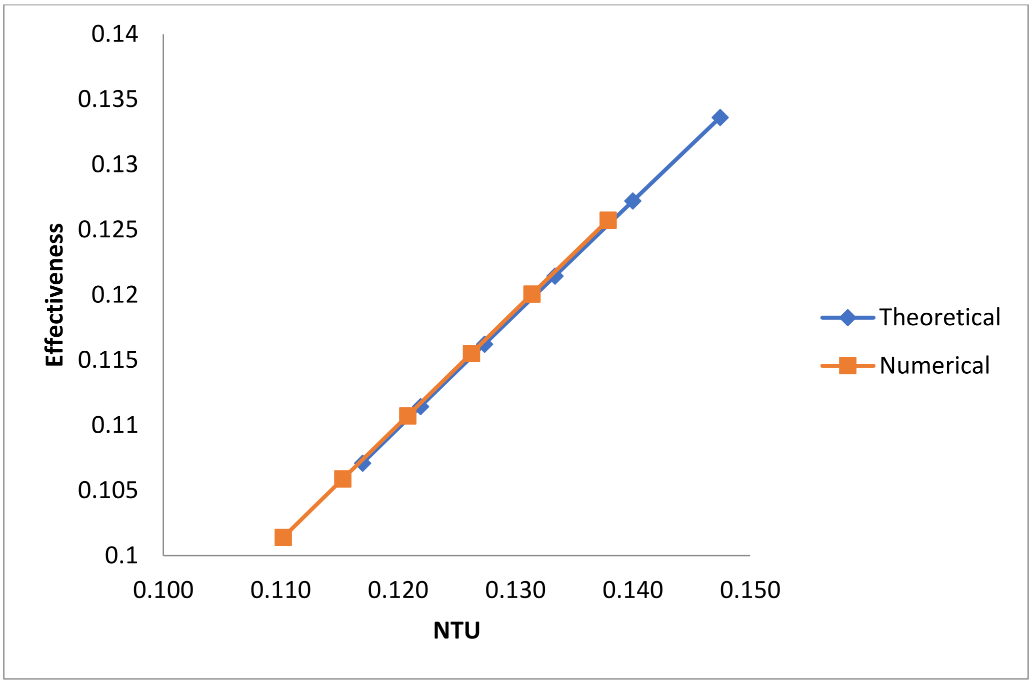

Fig: 7. NTU with Effectiveness

The Fig shows an inverse relationship between NTU and effectiveness, with a maximum theoretical effectiveness at 0.147 and a minimum at 0.110. The analytical effectiveness is consistently lower than the theoretical values, with a maximum at 0.126 and a minimum at 0.101. The maximum difference between theoretical and analytical effectiveness is approximately 7.6%, while the minimum difference is 8.2%. The trend remains consistent across both theoretical and analytical values.

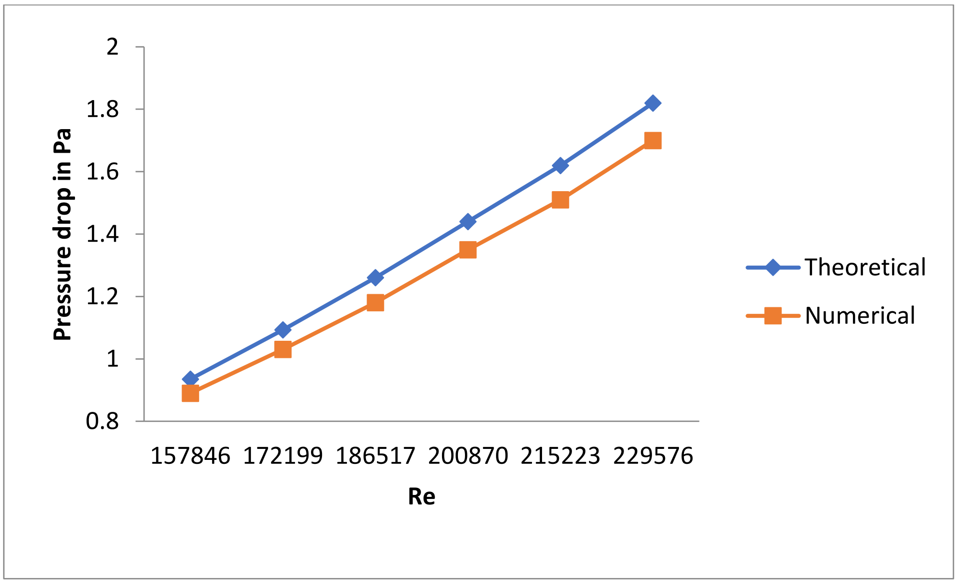

Fig: 8. Pressure Drop with Reynolds Number

The Fig shows the pressure drop as a function of the Reynolds number, with both theoretical and analytical values showing a direct relationship. The maximum theoretical pressure drop is 1.82 Pa at a Reynolds number of 229576, while the minimum is 0.935 Pa at a Reynolds number of 157846. The analytical values show a slightly lower pressure drop, with a maximum of 1.7 Pa at 229576 and a minimum of 0.89 Pa at 157846.

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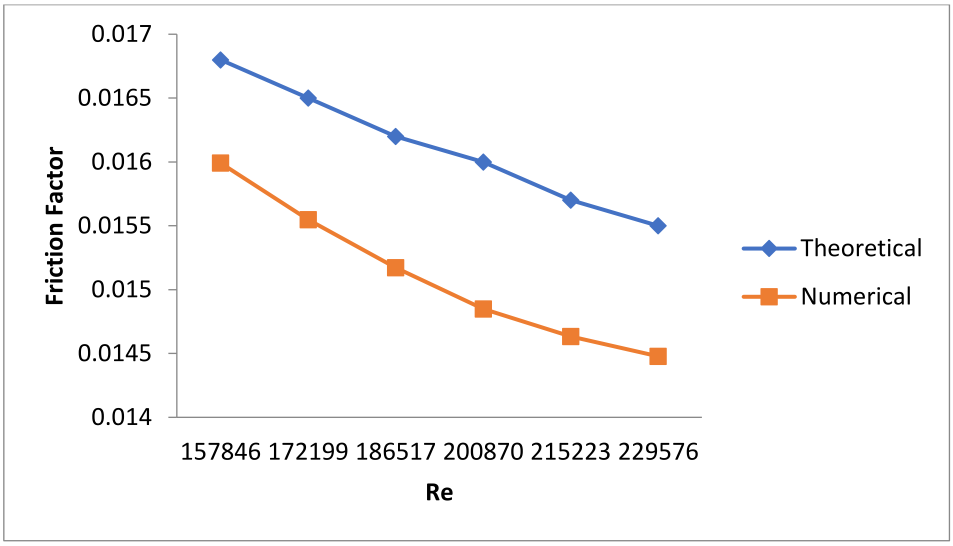

Fig: 9. Friction Factor with Reynolds Number

The Fig shows the friction factor as a function of Reynolds number, with theoretical and analytical values. As Reynolds number increases, friction factor decreases, indicating an inverse relationship. Analytical values are consistently lower than theoretical values, but the overall trend remains consistent. The maximum theoretical friction factor is at 157846, while the minimum is at 229576. The maximum difference between theoretical and analytical friction factors is approximately 4.8%, while the minimum difference is about 5.8%.

The study highlights the potential of waste heat recovery for energy-intensive processes like food drying. The counterflow configuration of the heat exchanger maximized temperature gradients, ensuring effective heat transfer. While initial costs for system installation may deter small-scale applications, the long-term energy savings and reduced emissions justify the investment. Future work could explore advanced heat exchanger designs or integrate real-time monitoring to enhance system adaptability.

V CONCLUSION

The study explores that food drying system uses waste heat for preheating of air. The system uses waste heat from exhaust gases to enhance performance. The study used both theoretical and analytical methods, using CATIA software for modelling and ANSYS Fluent for computational fluid dynamics simulations. The analysis evaluated performance metrics like heat transfer coefficients, Nusselt number, pressure drop, and effectiveness. The mesh convergence study ensured the accuracy and reliability of the results. The study found good agreement between theoretical and analytical values for heat transfer coefficient and Nusselt number, and consistent trends across both theoretical and analytical data. The results suggest that the heat recovery system, modelled in CATIA and analysed with ANSYS, is an efficient way to utilize waste heat for drying processes. However, small discrepancies in results highlight the challenges in achieving perfect agreement between theoretical and numerical methods.

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