

Performance Analysis of Annular Fins in Cylindrical Pipes: A Study on Heat Transfer and Fin Efficiency under Various Convection Modes

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ABSTRACT

This research examined how annular fins fabricated in a cylindrical pipe enhanced heat transfer on the air side. It evaluated the heat transfer coefficient, efficiency of the fins, and velocity of air in both natural and forced convection scenarios. Experimental measurements are used to examine heat transfer properties, and the results are compared with simulations performed using ANSYS software. The geometric models for these studies were created using the SolidWorks software. The experimental setup incorporated thermocouples and a heater regulated by a voltage controller to ensure precise measurement. Using a stainless-steel cylinder of 103.4 mm in length with six annular fins (spaced 23.90 mm) and a radius of 63.40 mm (outer) and 25.6 mm (inner). Studies have demonstrated that, in free convection, the heat transfer coefficient increases as the temperature difference increases. This improves the fin efficiency at higher temperature differentials owing to stronger convective currents. In contrast, forced convection demonstrated an inverse correlation between the heat transfer coefficient and the temperature differential. Elevated air velocities enhance the heat transfer coefficients; however, they simultaneously decrease the fin efficiency owing to the formation of narrower boundary layers near the fin surface.

Keywords: Heat transfer enhancement, Annular fins, Forced convection, Heat transfer coefficient, Fin efficiency.



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1. Introduction

The structure of a heat exchanger, which is widely utilized in many thermal systems, has a considerable impact on the flow and heat transfer properties.

Heat can be transferred via three distinct mechanisms: radiation, convection, and conduction. This phenomenon occurred naturally. Convection is the most basic and extensively utilized method for cooling electrical and mechanical parts [1]. Systems for cooling air via natural convection are dependable and easy to use [2]. In addition, their design was simple, affordable, and noiseless. Newton's law of cooling states that the cooling process can be enhanced by increasing either the surface area A or the heat transfer coefficient h in the equation $Q = hA$. Convective heat transfer from a surface can be accelerated by modifying the area and temperature while maintaining a constant power input and staying below a specified maximum temperature. Applying a better fluid to improve the convective heat transfer coefficient is expensive. For instance, the use of a liquid as a coolant necessitates the installation of a heat exchanger, pump, and related facilities [3].

The most typical application for improving the heat transfer is the use of fins to expand the heat transfer area. Fin arrays can be used in a wide variety of other contexts, from nuclear fuel models to air-land-space vehicle power sources, conventional furnaces, and gas turbines for waste-heat boilers [4]. Fin design is the only variable that can be changed to enhance convective heat transfer [5].

Using reliable parts allows the designer to use more advanced tactics to boost performance, which is a key idea in engineering design. Because effective cooling increases dependability [6], it has become increasingly important as electronic technology has advanced, as has the dissipation of

power and component density, both of which have significantly increased. Natural convection is the most basic and extensively utilized approach for cooling electronic components [7]. Systems for cooling air via natural convection are dependable and easy to use. In addition, their design is uncomplicated, economical, and noiseless. To maintain a steady-state condition, systems must continuously release the heat that travels through solid materials, walls, or boundaries into the surrounding environment. Many technical applications require the removal of substantial amounts of heat from the confined spaces. The implementation of fins enhances the surface area, thereby facilitating improved convective heat transfer.[8].Extensive studies have been conducted on the free convection of rectangular fins. However, limited research has addressed the interaction between free and forced convection in cylindrical pipes equipped with annular fins. This study aims to experimentally investigate the enhancement of air-side heat transfer in such configurations, considering horizontal fin arrangement. Consequently, this study aims to experimentally investigate the air-side heat transfer enhancement in cylindrical pipes with annular fins, concentrating on forced and free convection while excluding other convection modes, such as mixed convection.

2. Design of Fin Assembly

Design of fin-assembly dimensions

Length of the cylinder, = 406.4 mm

Thickness, $t_1 = 1.4$ mm

Outer radius of the cylinder, $R_2 = 25.6$ mm

Inner radius of the cylinder, $R_1 = 23.7$ mm

Fin outer radius, $r_2 = 63.40$ mm

Fin inner radius, $r_1 = 25.6$ mm

Fin thickness, $t_2 = 1.7$ mm

3. Experimental Setup

The power supply was controlled by a voltage regulator for free convection. A multimeter checks the present supply current and the heater heats the fin assembly. Temperature-sensing elements located at specific intervals on the surface of each fin measured the wall and fin temperatures, which were used to determine the fin parameters. For forced convection, a table fan was used to supply the air velocities, and an anemometer was used to measure the airflow velocity for the horizontal fin. For vertical mounting, wood pieces and a nail support the fin, whereas a fan placed approximately one foot away supplies the air. Fig.1, Air was provided during forced convection by a fan that held the foot distance, as shown in Fig.1, and in vertical mountings of the fins.



Fig.1 Experimental setup.

3. Experimental Data

Table 1 presents the changes in temperature at different voltages and currents supplied by free convection. Table 2 presents the change in temperature at different voltages and currents supplied in forced convection with the variation in power. Table 3 shows the change in temperature at different air speeds in forced convection, where the voltage is constant.

Table 1 Free-convection data

Exp No	Volt V	Q watt	tH °C	tA °C	tFav °C	tH-tA °C	tFav-tA °C
1	80	82.56	115	29	39.5	86	10.5
2	105	134.4	158	29	49	129	20
3	120	181.4	190	29	59.33	161	30.3
4	150	288	240	29	75.5	211	46.5

Table 2 Data for forced convection (air velocity 4.2 m/s).

Exp. No	Volt V	Q watt	tH °C	tA °C	tFav °C	tH-tA °C	tFav-tA °C
1	80	82.56	50	29	30.13	21	1.13
2	105	141.96	62	29	31.26	33	2.26
3	145	196.04	70	29	32.25	41	3.25
4	150	288	82	29	33.67	53	4.67

Table 3 Forced convection (for voltage 120 V).

Exp. No	Q watt	tH °C	tA °C	tFav °C	tH-tA °C	tFav-tA °C	UA m/s
1	183.36	140	29	37.25	111	8.25	2.5
2	183.36	135	29	33.41	106	4.41	3.0
3	183.36	115	29	31.91	86	2.91	3.5
4	183.36	110	29	30.08	81	1.08	4.0

4. Numerical Computation

4.1 Models Preparation

The models were first developed using SolidWorks before conducting simulations with empirical measurements obtained under controlled experimental settings. The physical dimensions exhibited reduced precision attributable to coarser surface finishes, whereas the simulation models featured smoother surfaces, facilitating superior accuracy. Consequently, simulations produced accurate model parameters. Fig.2, depicts the three-dimensional model constructed in SolidWorks.

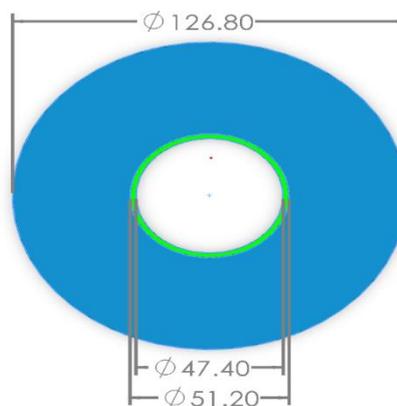
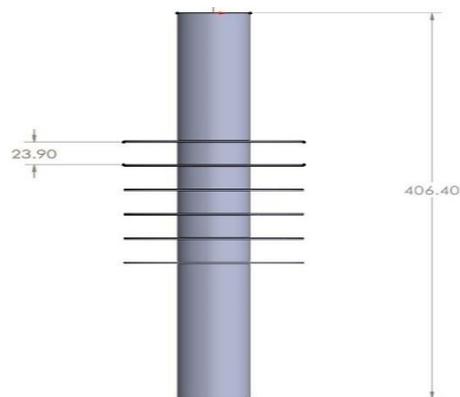


Fig.2 Computational model

Table 4 emphasizes the principal physical and inertial characteristics of the model, encompassing the volume, mass,

centroid coordinates, and principal moments of inertia, which are essential for evaluating thermal and structural performance in a steady-state context.

Table 4 Properties of the model.

Properties	Values
Volume	$2.274 \times 10^{-004} \text{ m}^3$
Mass	1.8089 kg
Centroid X	$2.7986 \times 10^{-018} \text{ m}$
Centroid Y	0.20569 m
Centroid Z	$2.462 \times 10^{-018} \text{ m}$
Moment of Inertia I_{p1}	$1.6159 \times 10^{-002} \text{ kg} \cdot \text{m}^2$
Moment of Inertia I_{p2}	$2.5586 \times 10^{-003} \text{ kg} \cdot \text{m}^2$
Moment of Inertia I_{p3}	$1.6159 \times 10^{-002} \text{ kg} \cdot \text{m}^2$

4.2 Material Properties

The use of 316 stainless steel closely resembled the thermal characteristics of the physical model. The thermal variables, which range from -0.15°C to 2726.9°C and are essential for accurate thermal modeling and performance analysis, are detailed in Table 5 based on ANSYS data.

Table 5: 316 Stainless Steel Field variables

Variable	Default	Lower	Upper
Name	Unit	Data	Limit
Temperature	C	22	-0.15 2726.9

316 stainless steel isotropic thermal conductivity as per ANSYS data has been plotted in Fig.3, having temperature-dependent properties increasing with temperature until around 1300°C is suitable for calculation of heat transfer in high-temperature engineering applications.

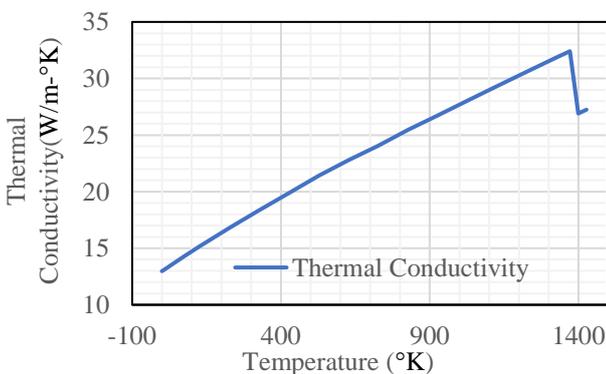


Fig.3 Isotropic thermal conductivity.

4.3 Computational grid (Mesh)

Fig.4 presents the mesh settings from the ANSYS Steady-State Thermal analysis, featuring a mechanical-physics configuration. The element size was 5 mm with

adaptive sizing, and a precise fine span angle. The mesh quality metrics showed minimum and average qualities of 0.129 and 0.720, respectively. The model includes 196,186 nodes and 96,863 elements, facilitating an accurate thermal analysis, as illustrated in Fig. 4

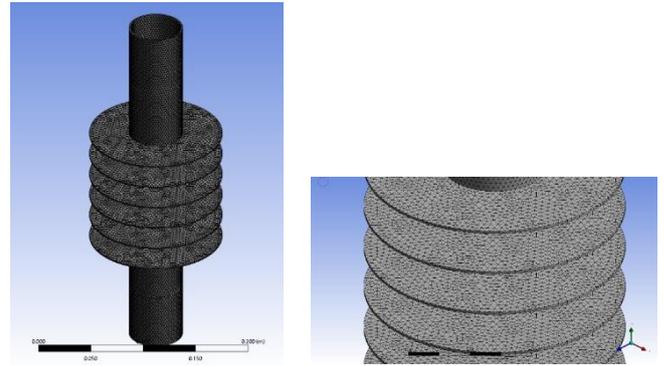


Fig.4 The meshed model

5. Results and Discussion

5.1 Code Validation

A validation study was performed by simulating three annular fins attached to a horizontal pipe geometry investigated by Nagarani and Mayilsamy [9], using the same conditions as those used in this work. Fig.5 presents a comparison between the present simulation and the simulation results of Nagarani and Mayilsamy [9]. There was 3% of error.

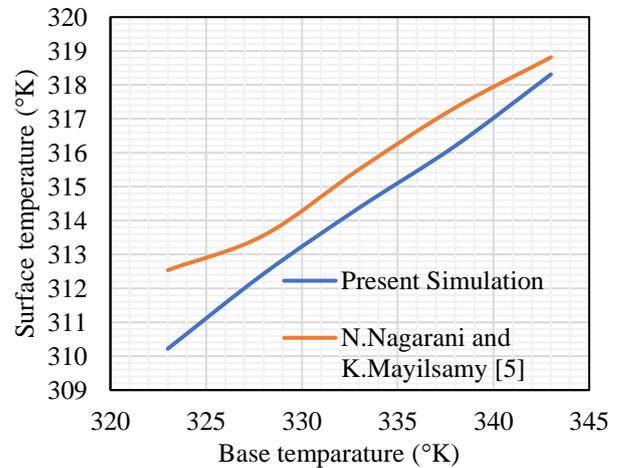


Fig.5 Variation in surface temperature with base temperature validation.

5.2 Temperature Distribution

In the thermal analysis conducted using ANSYS Steady-State Thermal, the temperature distribution of an annular fin attached to a steel pipe was studied to evaluate its heat transfer characteristics. The model depicted in Fig.6 illustrates the temperature distribution across the fin, with color contours representing the temperature gradients. The pipe was subjected to heating to simulate the operational conditions, allowing for a detailed examination of how the fin dissipates heat.

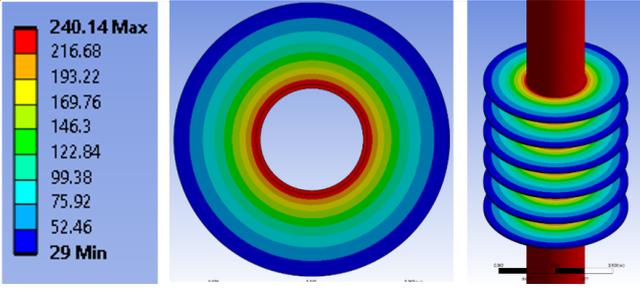


Fig.6 Temperature distribution.

5.3 Comparison Between Experimental Result and Simulation Result

The experimental and simulated results for horizontal positions under free convection conditions were compared to validate the accuracy of the thermal modeling approach. Fig. 7, present the surface temperature profiles as a function of base temperature for horizontal orientations, respectively, all conducted in a free convection environment. There is 6% error between simulation result and experimental result.

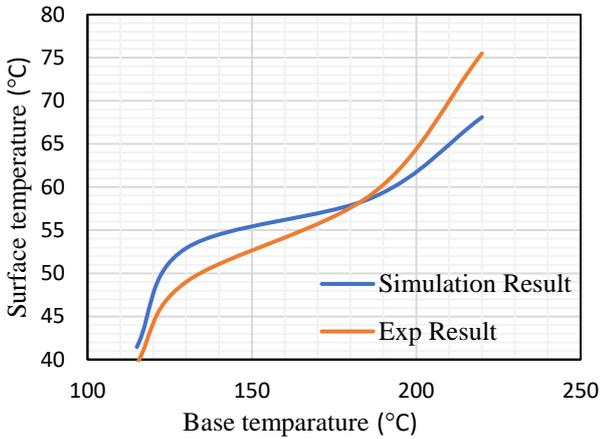


Fig.7 Comparison between the simulated and experimental results for horizontal position free convection.

5.4 Heat transfer coefficient

Fig.8 shows that when the temperature difference between the wall and ambient environment increased, the heat transfer coefficient h also increased. The increase in the heat transfer coefficient is caused by increased convection, allowing faster heat transfer from the fin surface to the ambient air.

5.5 Fin efficiency

Fig. 9 shows that in natural convection, the fin efficiency increases as the heat transfer coefficient increases. Conversely, in forced convection, the graph indicates that fin efficiency declines with higher heat transfer coefficients. This decrease occurs because mechanically driven air diminishes the interaction with the surrounding environment, resulting in reduced heat-transfer coefficients and overall fin efficiency.

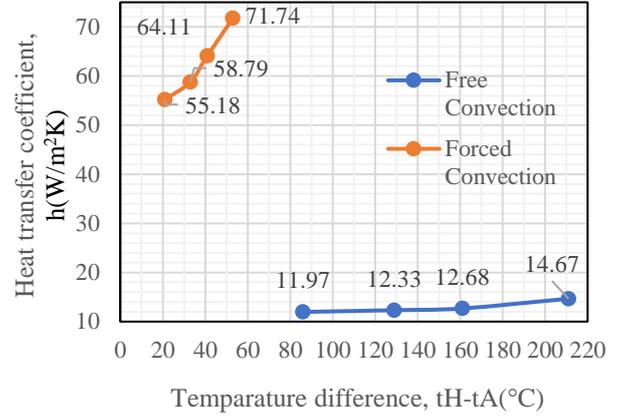


Fig.8 Heat transfer coefficient against temperature difference

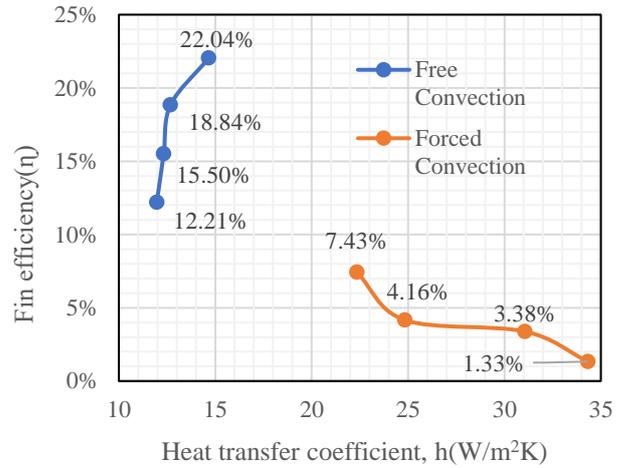


Fig.9 Fin efficiency against Heat transfer coefficient

5.6 Fin efficiency

As illustrated in Fig.10, the efficiency of the fins increases as the temperature difference between the wall and the surrounding environment increases. This is due to the fact that greater temperature gradients enhance the convective currents, thereby improving the transfer of heat from the fin to the surrounding air.

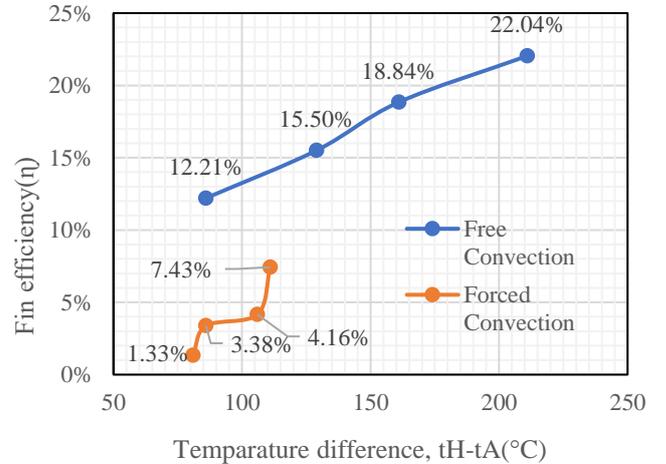


Fig.10 Fin efficiency against temperature difference

5.6 Effect of airspeed

As shown in Fig. 11, increasing the airspeed increased the heat transfer coefficients because more air contact enhanced the fin surface temperature differences to transfer heat.

At higher airspeeds, the airflow adjacent to the wing may transition from laminar to turbulent flow. For turbulent flow, increasing the boundary layer thickness to the core is comparatively thinner, and the interactions of layers within the flow improves heat transfer. This change tightened the thermal boundary layer, leading to a substantial reduction in thermal resistance. However, near the fin surface, the reduced boundary layer thickness can lead to a decrease in fin efficiency. Despite this the overall heat transfer may improve due to the lower thermal resistance

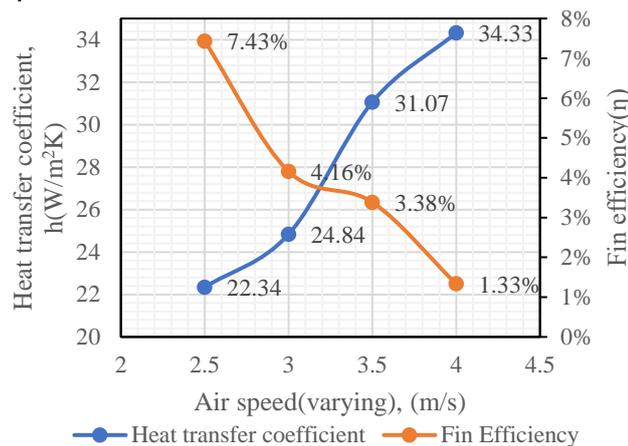


Fig.11 Heat transfer coefficient and Fin efficiency with respect to Air speed (Varying)

6. Conclusion

The present study aims to investigate the effects of convection type on heat transfer in a horizontal stainless-steel cylindrical pipe with annular fins.

Free Convection:

- The coefficient of heat transfer increases as a result of higher air current speeds, which are promoted by more pronounced temperature gradients under steady-state conditions.
- The efficiency of the fins is enhanced by the increased thermal gradient between the pipe wall and its environment.

Forced Convection:

- Larger temperature differences enhance the heat-transfer coefficient, resulting in decreased turbulence between the air and fins.
- The fin efficiency increases with temperature gradients but diminishes with elevated heat transfer coefficients, likely because the enhanced airflow adversely affects the thermal efficacy at the fin interface.

Effect of Mass Flow Rate in Forced Convection:

- The heat transfer coefficient was augmented by increasing the overall mass flow rates with a larger contact area, as available by the air-fin matrix.
- Fin effectiveness declines with the rate of mass flow because the boundary layer of the fins is reduced; thus, the thermal resistance of the finned surfaces is reduced to allow for maximum heat transfer.

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NOMENCLATURE

Q	: Rate of heat transfer, $J.s^{-1}$
h	: heat transfer coefficient, w/m^2k
A	: surface area A, m^2
T	: temperature, K
t	: Celsius temperature, $^{\circ}C$
tH	: Cylinder base temperature, $^{\circ}C$
tA	: Ambient temperature, $^{\circ}C$
$tFav$: Average temperature of the fins, $^{\circ}C$
$tH - tA$: Temperature difference between the cylinder base and ambient ($^{\circ}C$)
$tFav - tA$: Temperature difference between the average fin temperature and ambient ($^{\circ}C$)
UA	: Airspeed, ms^{-1}
m/s	: Meter per second (unit of airspeed)
V	: voltage, Volt