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Heat Transfer Analysis Between Two Aluminium Parallel Plates with Natural Convection

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Abstract

As a type of energy, heat can be transferred from heated systems using the natural convection heat transfer method, which has several uses in different industries. Natural convection method of heat conveyance is suitable in many cases due to absent of any external sources like fan. The transfer of this kind of energy can be listed as electrical and electronics equipment, nuclear reactors, domestic convection, dry cooling towers, thermo siphons, installed in ground and many more. On the account of functional continuity and longevity aspect the heat generates due to work execution must be extracted from the machineries. There are various ways, such as provision of vertical tubes and parallel plates to transfer heat from heated system by natural convection method. The improvement of heat transport through the use of natural convection method is probable by facility of internal impediments inside the vertical tube, by the geometrical position of the test section and also depending on material selection. In this work heat transfer properties through two parallel aluminum composite parallel plates have been presented. The magnitude of plate with thickness, breadth and length are 5 mm, 150 mm and 500 mm correspondingly. The both plates are heated by provision of heating coil. The exterior region of both plates are maintained insulation, the shifting of heat energy is admitted to regulate from interior region towards the top of plates. The constant wall heat flux maintained inside the plate has a magnitude of 2188 W/m^2. The basis aim of this work to find the wall temperature along the plates and air temperature at exit from the plate. The result from experimental set up is compared with theoretical by using ANSYS software.

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INTRODUCTION

Natural conduction is a cost-effective cooling technique that is extensively employed in the electronics industry to remove heat energy from electronic devices, which are firmly established on limited platforms to increase their lifespan and efficiency. The exact shifting of heat energy implies that the transmission of heat through the heated plate to air due to temperature difference between them. Hot bodies can be cooled faster on behalf of forced heat convection than other means. Natural convection occurs because of the difference in mass compactness of fluid resulting out of the temperature dissimilarity. The ambient air in close proximity of the hot bodies is less than the air away from it. Due to difference in density of fluid, a buoyant force is created which enables the lighter hot air to flow in upward direction. Natural convection observed in diverse domains such as nuclear reactors, power plants adhere pipe carrying steams, the very cooling towers, convectors in domestic means, siphons of thermal means etc. The exact aim behind this investigation implies findings of natural convection via heated upright ducts in both theoretically and experimentally. The two parallel plates that make up the test portion have their outer surfaces electrically heated, which releases heat into the air from the inner surface. The observational part excited via electrical means keeping fixed heat energy surge on the boundary. The temperature of air in the vicinity of heated plate increases which results in decrease of density of air near the heated plate. Due to variation of density of air a buoyant force is created which causes the hot air through the two plates to move in upward direction. Natural convection heat flow experiment is carried out with continual wall heat flux conditions for plates of dimensions 500 mm length, 150 mm width and 3 mm thickness. Dynamics of fluid flow and heat flow in vertical pipes and parallel flat plates have been investigated by many researchers. Nevertheless, the existing literature on heat conveyance in the presence of rings is unsatisfactory. Naturally occurring convective heat flow across staggered discrete plates was experimentally investigated by Mallik and Sastri [1], who found that it is greater over discrete vertical plates than over flat plates. Natural convection through staggered isolated plates has been studied by Sparrow and Prakash [2], who also contrasted the outcomes of parallel flat plates and staggered isolated vertical plates while taking constant wall temperature into account. They noticed that with increase in spacing, decrease in height of channels results in improvement of heat transfer. Hung and Shiau [3] studied natural convection flow through vertical parallel plates with rectangular ribs theoretically as well as experimentally. They observed that the flow in turbulent flow circumstances is similar to the heat exchange phenomena in the area across the ribs. In order to calculate heat transfer characteristics; they suggested two correlations for forecasting the Nusselt number downstream of the rib. The heat exchange and hydrodynamics in vertical channels were experimentally studied by Gortysov *et al.* [4]. They observed that provision of discrete rings in the internal surface results in increase in heat transmission from vertical walls to atmosphere. Experimental investigation by Sparrow and Bahrami [5] showed imposing three variants of boundary states on the horizontal wall. Dixit *et al.* [6–8] employed numerous optimization strategies for example bacterial colony optimization (BCO), cuckoo search (CS) along with group search optimization (GSO) so as to determine the most favorable composition of the structure. Levy *et al.* [9] pointed out the issue concern to most favorable plate gap on behalf of enhancement of streamlined usual heat energy surge among the specimens. Through heated vertical tubes, Roul and Nayak [10] demonstrated the properties of heat flow. Nayak *et al.* [11–15] investigated the improvement of heat energy shift in upright tube owing to the accessibility of the internal rings. These were made with rectangular segments as impediments. Churchill and Chu [16] developed general correlation equations applying the experimental and theoretical data for the laminar natural convection flow through vertical channels which is driven by buoyancy. Dey *et al.* [17] observed that the flow dynamics in the close proximity of a fin can significantly improve the heat transmission through a heat sink owes to a forced convection. The rise in heat flow is owing to an enhancement in temperature gradient, as the depth of the thermal boundary barrier decreases due to presence of fins. The typical Nusselt Number on the fin surface rises as a result of the temperature shift being strengthened. Fluid dynamics and pressure drop predictions through sudden expansions and contractions have been studied using CFD modeling. Pradhan *et al.* [18] examined the passage of heat characteristics through 180 degrees bent pipe having various cross segment utilizing nano-enhanced ionic liquids. Heat transmission through natural convection through an upward micro channel under continuous heat flow was investigated by Buonomo and Manca [19]. To assess their impact on mass flow rate, velocity profile, wall temperature, and Nusselt number, they computed Rayleigh number as an indicator of wall heat flux. They perceived that there is spike in the wall operating temperature with enhancement in Knudsen number for higher range of the Rayleigh number. However, they noticed that the wall temperature is minimum for lower values of Rayleigh number for $Kn = 0.05$. They also noticed that Nusselt number declines when Kn increases and mass flow rate upsurges when Kn increases. El-Morshedy *et al.* [20] experimentally explored the natural convection-induced heat transmission in thin vertical channels having rectangular heated walls. For forced and natural convection flows, respectively, they suggested a pair of correlations on the local Nusselt number. When the correlations' prediction and the experimental data were compared, it was found that the two sets of data suited each other rather well, around 5.8% standard deviations. Nayak et al. [21–27] presented some innovative works with natural convection method heat transfer. The investigation of fluid flow characteristics inside parallel composite plates is the focus of this work. The observations from experiments are compared with theoretical.

EXPERIMENTAL SETUP

Figure 1 demonstrations the investigational set up. It involves of electrical panel board, thermocouple, test section (aluminum composite parallel plates), heating coil, and insulating layer. The electrical panel board consists of temperature indicator, selector switch, voltage regulator.

Figure 2 shows the structure of the aluminum composite plate, it consists of three layers, PVDF panel layer at top, epoxy adhesive at middle and polyester back layer, such composite panel is strong in nature due to its three layers are metal.

The test section (plate) is attached with the heating coil, which is heated by supply of voltage from electrical panel board. There is provision of an insulating layer at external part of the test section, which helps to insulate the test section. To gauge the surface temperature while air flows through the plate, a number of thermocouples are affixed to the wall. The temperature of the fluid is measured towards

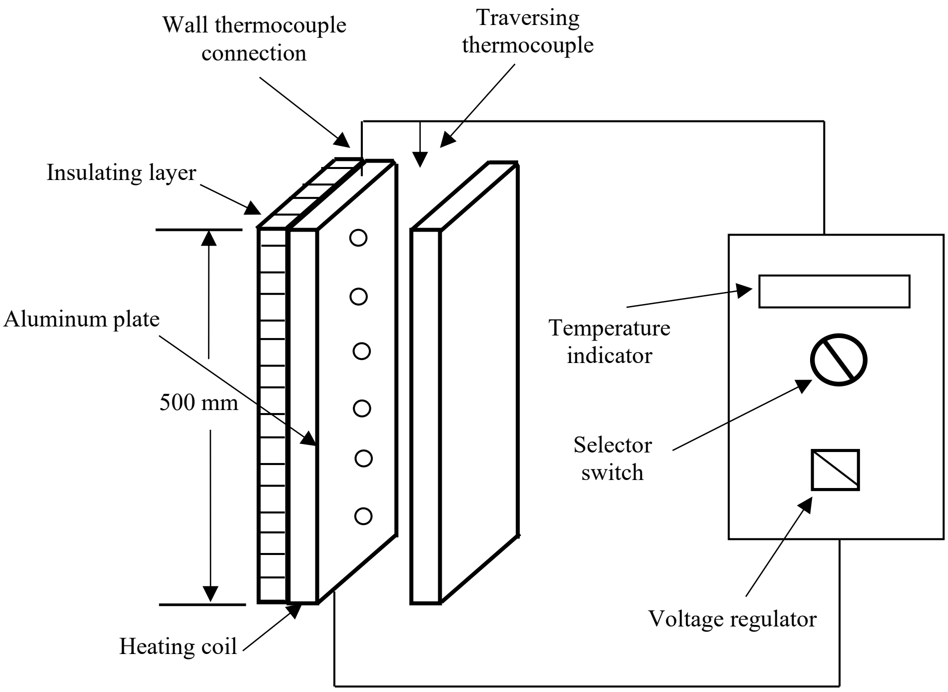

Figure 1. Heated vertical parallel plates.

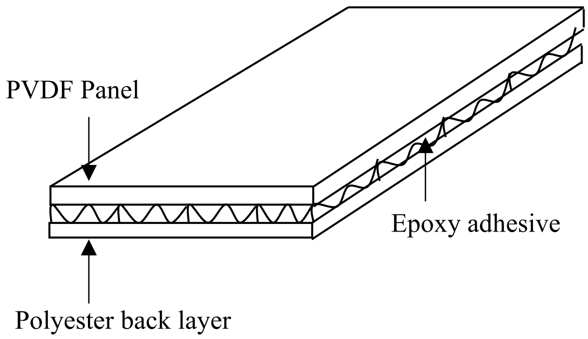

Figure 2. Structure of aluminum composite plate.

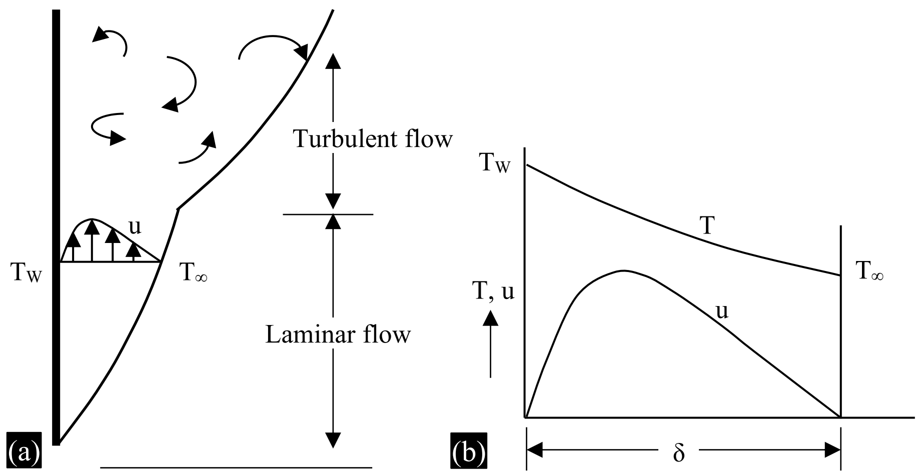

Figure 3. (a) Boundary layer (b) velocity and temperature profile on a vertical plate.

exit of the tube by attaching a traversing thermocouple. The magnitude of plate with thickness, breadth and length are 5 mm, 150 mm and 500 mm correspondingly. Inside the plate, a continuous wall heat flow with a magnitude of 2188 W/m^2 is maintained. Figure 3 shows the boundary layer and profile for velocity and temperature during air flow inside the plate.

The average coefficients concern to heat energy shift may expressed in different environments as per the details given below:

$$\overline{Nu_f} = C\left(Gr_f Pr_f\right)^m \tag{1}$$

In above notation '*f*' specifies unit-less category of fluid properties where the factor "temperature" is evaluated via:

$$T_f = \frac{T_W + T_\infty}{2} \tag{2}$$

The Rayleigh number is the assessment experienced via the multiplication of Prandtl number and Grashof number:

$$\mathrm{Ra} = \mathrm{Gr}*\mathrm{Pr} \tag{3}$$

The typical length taken inside determination of Grashof as well as Nusselt number depends upon the very physical configuration considered within issue. The specimen laminate height (L) is considered as the characteristic dimension for flat vertical plates and the diameter (D) are considered as the distinctive dimension for horizontal cylinders. Hence equations used to determine the heat transfer for vertical flat plates can also are considered for vertical cylinders when width concern with outside film not being very comparison with cylinder diameter. As a matter of fact, an upright cylinder is generally considered as one upright flat plate under the following conditions.

$$\frac{D}{L} \geq \frac{35}{GR_L^{1/4}} \tag{4}$$

In equation 4, D is taken as the cylinder diameter. For determination of Nusselt number the following equation is considered.

$$\mathrm{Nu_f} = 0.10(\mathrm{Nu_f Pr_f})^{1/3} \tag{5}$$

More relevant equations which are appropriate over extensive varieties of Rayleigh number

$$\overline{\mathrm{Nu_f}} = 0.68 + \frac{0.670\mathrm{Ra}^{1/4}}{\left[1+(0.492/\mathrm{Pr})^{9/16}\right]^{4/9}} \text{ for } \mathrm{Ra_L} < 10^9 \tag{6}$$

$$\overline{Nu_f}^{1/2} = 0.825 + \frac{0.387Ra^{1/6}}{\left[1+(0.492/Pr)^{9/16}\right]^{8/27}} \text{ for } 10^{-1} < Ra_L < 10^{12} \quad (7)$$

For boundary conditions with continuous heat transition, equation 7 can also be taken into account. The fluid properties are taken corresponding to the fluid temperature in the above equations. Numerous experimental investigations are being documented in the literature for natural convection heat flow from flat upright as well as tilted plates to fluid for fixed heats urge boundary values. In these experimentations, outcomes have been expressed through modified Grashof number, Gr^*.

$$Nu_x^* = Gr_x Nu_x = \frac{g\beta q_w x^4}{kv^2} \quad (8)$$

Here, the unit of wall heat flow, q_w as watt/m^2. In case of laminar flow conditions, the local Nusselt number is estimated out of the function.

$$Nu_{xf} = \frac{hx}{k_f} = 0.60(Gr_x^* Pr_f)^{1/5} \text{ for } 10^5 < G_x^* < 10^{11}; \quad (9)$$

Similarly, in case of turbulent flow conditions, regional Nusselt number is calculated using the relation given below.

$$Nu_x = 0.17(Gr_x^* Pr)^{1/4} \text{ for } 2 \times 10^3 < Gr_x^* Pr < 10^{16}; \quad (10)$$

The local Nusselt number is calculated using the relation given below.

$$Nu_x = C(Gr_x Pr)^m \quad (11)$$

Substituting the value of Grashof number in equation 11, we have,

$$Nu_x = C^{1/(1+m)}(Gr_x^* Pr)^{1/(1+m)} \quad (12)$$

For laminar and turbulent surge conditions, ‘m’ is considered as 1/4 and 1/3 correspondingly.

Churchill and Chu [19] have proposed that equation12 can be customized as per the details given below so that it can be applied for the case of constant heat flux conditions.

$$\overline{Nu_L}^{1/4}[\overline{Nu_L} - 0.68] = \frac{0.67(Gr_L^*)}{\left[1+(0.492/Pr)^{9/16}\right]^{4/9}} \quad (13)$$

i.e.,

$$\overline{Nu_L} = q_w L/(k\overline{\Delta T}) \text{ and } \overline{\Delta T} = T_w(\text{at } L/2) - T_\infty \quad (14)$$

In equation 14, is the expression for the mean Nusselt number with conjunction to the boundary heat energy surge conditions. Temperature distinction is taken at the mid-point of a plate namely from x = L/2.

THEORETICAL ASPECTS

A Theoretical study was carried out by using ANSYS Fluent 15.0 software. Following steps has been followed for theoretical analysis

1. *Geometry creation:* ANSYS Fluent 15.0 Design Modeller is used to construct the computational domain's geometry.
2. *Mesh generation:* Create the mesh when the domain has been generated, maintaining that the number of nodes and elements is constant. In areas where considerable fluid property fluctuations are anticipated, use finer mesh refinement.
3. *Solution setup:* Establish general parameters, choose suitable models, allocate resources, and specify boundary conditions to configure the solution.
4. *Solution initialization and execution:* Set up the solution and work out the governing equations to model heat transfer and fluid flow.

5. *Post-processing:* Examine the findings by creating contour plots of the distributions of pressure, temperature, and velocity.
6. *Data visualization:* Retrieve several charts, such as the velocity and temperature characteristics, to comprehend the simulation's results.

Natural convection heat flow happens owing to the movement of fluid because of density differences produced by the heating processes. The force of buoyancy causes the movement of fluid in the upward direction as the density of fluid in the proximity of the heated plate is decreased due to heat transmission from the hotter surface to the atmosphere. The buoyant force has an important role to play in natural convection flow. Figure 2 shows the boundary layer and profiles of velocity and temperature over a heated vertical flat plate. In this boundary layer, the inertia force as well as the buoyant and viscous forces is predominant. It is observed from Figure 2 that the velocity happens to be zero at the wallowing to zero slip boundary state and it enhances to attain highest value and there after it decline reaches null value towards part of the edging film as the air is at rest away from the wall. After a specific distance from the leading edge, the boundary layer begins to transit from a lamina to a turbulence one, depending on the fluid's characteristics and the wall's temperature. In transitional region turbulent eddies are formed. Farther up the vertical plate, fully turbulent boundary layer could be developed.

Compared to the imposed convection flow velocity, the actual convective flow velocity is substantially lower. As a result, the coefficient of convective heat transfer is often one order factor lower. Within free convection circumstances, the Grashof number, the ratio comparing buoyancy force associated with the viscous force, acts similarly to the Reynolds number in forced convection. The Grashof number is the determining element in the transition through laminar to turbulent flow. A differential equation for motion for the boundary region must be obtained before we can begin to analyze the heat-transfer problem. A natural convection flow is created by uniform wall heat flow, where the flow enters the heated flat plate along the lower side and exits at the upper side. Therefore, difference in the density of fluid which is caused because of the temperature gradients produces the buoyant forces which are responsible for causing the motion of fluid. The following assumptions have been considered as the computational analyses of natural convection flow of heat through the heated upward flat plate.

1. Fluid properties except density are considered to be constant.
2. Density variations are significant only in the gravity term which is responsible for producing the buoyant force.
3. Therefore, flow is characterized as laminar, inviscid, steady andincompressible.

Thus governing equations for 2-D, steady, inviscid flow are given below.

Continuity:

$$\left(\mathrm{u}\frac{\mathrm{d}\rho}{\mathrm{dx}}+\mathrm{v}\frac{\mathrm{d}\rho}{\mathrm{dy}}\right)+\left(\rho\frac{\mathrm{du}}{\mathrm{dx}}+\rho\frac{\mathrm{dv}}{\mathrm{dy}}\right)=0 \tag{15}$$

Momentum:

$$\frac{\partial \mathrm{p}}{\partial \mathrm{x}}=-\left(\rho \mathrm{u}\frac{\partial \mathrm{u}}{\partial \mathrm{x}}+\rho \mathrm{v}\frac{\partial \mathrm{u}}{\partial \mathrm{y}}\right) \tag{16}$$

$$\frac{\partial \mathrm{p}}{\partial \mathrm{y}}=-\left(\rho \mathrm{u}\frac{\partial \mathrm{v}}{\partial \mathrm{x}}+\rho \mathrm{v}\frac{\partial \mathrm{v}}{\partial \mathrm{y}}\right) \tag{17}$$

Energy:

$$\rho c_v\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)=-\left(\frac{\partial q_x}{\partial x}+\frac{\partial q_y}{\partial y}\right)-T\left(\frac{\partial \rho}{\partial T}\right)_p\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}\right) \tag{18}$$

RESULTS AND DISCUSSION

Utilized theoretically to explore via excited plates maintained upright and parallel, in exact two upright parallel plates, excited by means of heating coils using electric energy accomplished at the outer region to ensure steady heat surge status at the boundary. The dimensions considered for the plates are 500 mm, 150 mm, 5 mm for the designation of length, breadth and thickness respectively. Again here the outside boundary is kept conduction restrict and consequently permitting the surge of heat energy commencing at inside region is the ambient and heat flux equal heat energy surge. At the wall, the heat movement is maintained at $q''= 2188\ W/m^2$. The temperature of the wall is measured experimentally using thermocouples at different locations on the inner face of the vertical plate. The hypothetical consequence intended for partition temperature in addition to investigational data for heat flux of 2188 W/m^2 are compared with each other. Figure 3 indicates the ambient temperature at the wall surface which remains elevated from the bottom to the top of the plate when there is uniform heat flow at the wall. It is also evident from Figure 4 so as to hypothetical consequence goes with differ methodically through the investigational consequence. The divergence concern to hypothetical consequence out of the investigational consequence found practical within ± 10 %.

Similarly, the temperatures of air at the top of the vertical plates are measured experimentally and theoretically. The temperature of air at the middle of the two parallel plates is found to be minimum and it increases towards the wall in the lateral direction. The fluid's temperatures appear to be at its peak close to the wall. A hypothetical consequence intended for flowing calculated fundamentally near out of plate is assessed by means of experimental consequence for the heat energy surge of value 2188 W/m^2 as illustrated in the figurev5. The hypothetical consequence nearly confirms to the observational fact which has established by the graphical presentation. Ultimately the divergence of both the magnitudes i.e. hypothetical as well as practical remains within $\pm 10\%$.

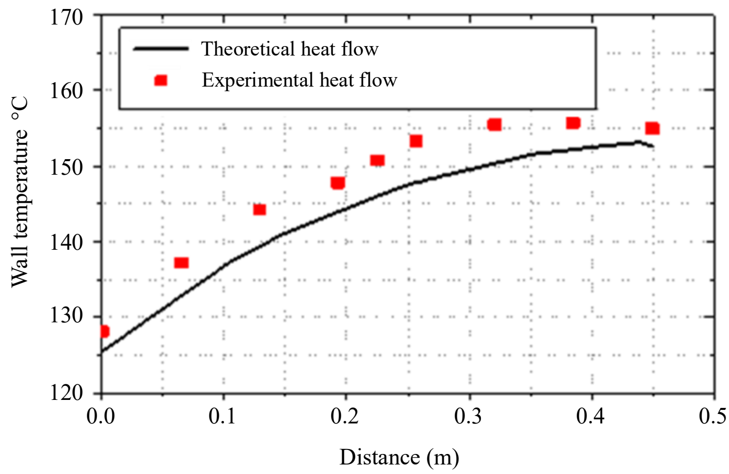

Figure 4. Theoretical and experimental wall temperature.

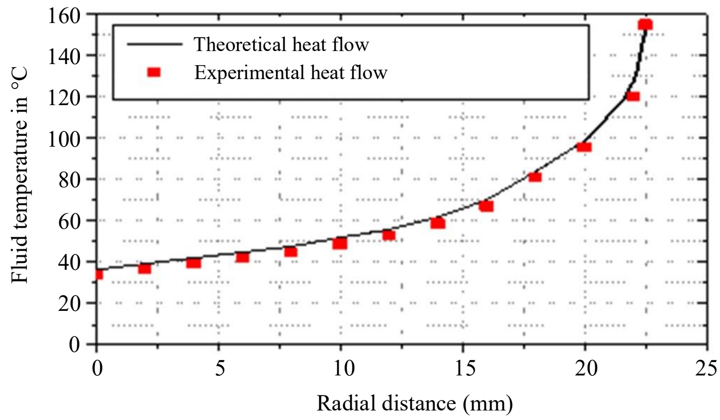

Figure 5. Theoretical and experimental fluid temperature.

Air temperature at the middle of the two parallel plates was observed and it found minimum at the top and it increases gradually towards the walls it also found the air temperature near the plates is vary and it indicate maximum.

CONCLUSIONS

Natural convection flow of heat through two heated upward flat plates is being investigated theoretically as well as experimentally. The major conclusions drawn from the investigation can be enumerated as follows.

1. The temperature increases through the bottom with the highest part of the plates in order to maintain constant heat flux values on the wall.
2. The air temperature is found to be lowest at the center of both parallel plates and to rise radially through the direction surface of wall and the fluid temperature reaches its maximum.
3. With an inaccuracy of ±10%, the theoretical findings appear to be quite similar to the experiment's findings.

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