

# Experimental Investigation of Forced Convection Heat Transfer Enhancement In Dimpled Tube

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## ABSTRACT

The development of high-performance thermal systems has stimulated interest in methods to improve heat transfer. Tubes with artificial roughness obtained by providing dimples on the tube surface are competitive in comparison to performance and cost of other enhanced techniques currently employed in turbulent flow.

In this study forced convection heat transfer characteristics from dimpled tube has been investigated experimentally. The effect of variation in dimple diameter and dimple arrangement (inline and staggered) has been investigated for Reynolds No range 12000 to 26000 dimpled tube. Additionally the effect of variation in dimple tube and diameter on pressure drop across test section has also been investigated. This investigation was carried out to observe if the use of dimples on tube can enhance heat transfer characteristics without severe penalties associated with pressure drops for turbulent flow.

The results show that heat transfer enhancement under forced convection can be achieved by using dimple tubes. The enhancement in convective heat transfer coefficient is observed as 18% and in Nusselt number as 22 % as compared with plain tube for same Reynolds number. The thermal performance factor values for the staggered dimpled array geometry were more than corresponding inline dimpled array geometry in the Reynolds number range studied.

**KEY WORDS:** Dimple tube, Heat transfer enhancement, Forced convection, Thermal performance factor.

## NOMENCLATURE

$Re$	Reynolds number
$Pr$	Prandtl number
$k$	Thermal Conductivity
$C_p$	specific heat at constant pressure
$V$	Velocity of air
$\rho$	Density of air
$\mu$	Kinematic viscosity
$D$	Diameter of tube
$L$	Length of tube
$T$	Temperature
$\Delta p$	Pressure drop
$A$	Area of tube
$h$	Heat transfer coefficient of dimple tube
$h_o$	Heat transfer coefficient of plain tube
$f$	Friction factor of dimple tube
$f_o$	Friction factor of plain tube
$Nu$	Nusselt number of dimple tube
$Nu_o$	Nusselt number of plain tube
$n$	Enhancement efficiency
TPF	Thermal performance factor

## 1.0 INTRODUCTION

The enhancement of heat transfer is an important part of the subject thermal engineering. The heat transfer from surfaces may in general enhanced by increasing the heat transfer coefficient or by increasing the heat transfer area of the equipment.

The need to increase the thermal performance of equipment, thereby effecting energy, material & cost savings have led to development & use of many techniques termed as Heat transfer Augmentation. These techniques are also referred as Heat transfer Enhancement or Intensification. Augmentation techniques increase convective heat transfer by reducing the thermal

resistance in equipment.

Use of Heat transfer enhancement techniques lead to increase in heat transfer coefficient but at the cost of increase in pressure drop. So, while designing a heat exchanger using any of the techniques, analysis of heat transfer rate & pressure drop has to be done. Apart from this, issues like long term performance & detailed economic analysis of heat exchanger has to be studied. To achieve high heat transfer rate in an existing or new heat exchanger while taking care of the increased pumping power, several techniques have been proposed in recent years.

Other studies have introduced shape modifications by cutting some materials from fins to make cavities, holes, slot, grooves or the channels through the fin body to increase heat transfer areas and or the heat transfer coefficient. One popular heat transfer augmentation technique involves the use of rough or interrupted surfaces of different configurations. The concept of heat transfer through dimpled tube is one method of improving the heat transfer characteristic in the forced convection. Continuous research is going on to improve its effectiveness by reducing the thermal boundary layer thickness and increasing the heat transfer surface area.

## 2.0 LITERATURE SURVEY

Over the past couple of years the focus on using concavities or dimples, to provide enhanced heat transfer has been documented by a number of researchers. It is evident from studies that the use of dimples not only provides enhanced heat transfer but it can overcome most of the drawbacks of the other methods employed for augmenting heat transfer. Studies by various researchers have repeatedly yielded heat transfer enhancement comparable to ribs with pressure losses of almost half that experienced under the use of ribs, and even reduced drag coefficient in some cases. In using dimples the extended surface is indented instead of protruding into the flow due to which there is a considerable reduction in the pressure penalty. The heat transfer enhancement provided by dimples is comparable to most rib tabulators but slightly less than some of the complex broken rib configuration. Besides the cooling enhancement and low pressure drop, dimples make manufacturing easier and help reduce the weight of the cooling system which is of critical importance.

**Johann Turnow** *et al.* investigated Vortex structure and heat transfer enhancement mechanisms of turbulent flow over a staggered array of dimples in narrow channel. The vortices on dimpled surfaces are created inside of concave cavities preventing a blockage of the channel and keeping the additional resistance at a minimum. Its formation was in the focus of many studies, but unfortunately, main attention has been paid to time averaged values whereas the flow structures within the cavities and their contribution to the heat transfer mechanism remain still unclear and are not completely understood. Especially, in the turbulent range and at large ratio of dimple depth to dimple diameter  $h/D$  the flow is complicated. Since the form of vortex has a strong impact on heat transfer. The objective of this study is to clarify the role of the vortex formation with respect to the heat transfer on staggered arrangement [1].

**Yu Rao** *et al.* conducted experimental study to investigate the pressure loss and heat transfer in the pin-fin dimple channels with various dimple depths, where dimples are located on the end wall

transversely between the pin fins. Compared to the base line pin fin channel, the pin fin-dimple channel have further improved the performance by up to 19.0%. Also got the results that shallower dimple gets lower friction factor [2].

**C. Biet** *et al.* studied mini channel heat sinks with dimples and cylindrical grooves. Studies on the heat transfer enhancement are mostly focused on the Nusselt number and flow resistance coefficient which do not completely explain the mechanism of heat transfer enhancement related to fluid temperature and its velocity. Good behavior of system can be predicted by velocity field and temperature gradient. Hence it is numerically studied by using synergy principle. The results show that dimple surface represents highest performance of the heat transfer enhancement; performance of cylindrical groove slightly lower and low fin surface presents lowest performance. Performance Evaluation plots are put forth to give most favourable dimple structures for the heat transfer [3].

**S.A. Isaev** *et al.* studied detailed numerical study of heat transfer enhancement by spherical dimple placed in narrow channel. Flow topology studied at different Reynolds no and influence of dimple depth to diameter ratios. Study give the deep insight in flow physics over dimple surfaces and as a benchmark for validation of numerical and experimental methods. It revealed the increase in Reynolds no over the range considered local changes in the flow structure and in the zones of elevation relatively reduced heat flux [4].

**Jonghyeok Lee, Kwan-Soo Lee**, studied the friction factor and Nusselt number in a plate heat exchanger with dimples and protrusions according to geometric and operating conditions. The numerical results of a steady-state laminar model for the laminar region and a steady state  $k-\omega$  turbulence model for the turbulent region were in good agreement with an unsteady analysis. For efficient analysis, a correlation for the critical Reynolds number was derived as a function of the geometric parameters. Friction factor and Nusselt number correlations were also derived as functions of various geometric parameters. In this study, steady-state analysis was conducted using the SST  $k-\omega$  turbulence model to find the friction factor and Nusselt number in a heat exchanger with dimples and protrusions [5].

**Somin Shin** *et al.* Investigated effects of dimple arrangement and channel height on the heat transfer coefficient. The sphere type dimples were fabricated, the diameter ( $D$ ) and the depth of dimple was 16 mm and 4 mm, respectively. Two channel heights of about 0.6 $D$  and 1.2 $D$ , two dimple configurations were tested. The Reynolds number based on the channel hydraulic diameter was varied from 30000 to 50000. As the Reynolds increased, the overall heat transfer coefficients also increased. With the same dimple arrangement, the heat transfer coefficients and the thermal performance factors were higher for the lower channel height. As the distance between the dimples became smaller, the overall heat transfer coefficient and the thermal performance factors increased [6].

**Yu Chen** *et al.* Investigated heat transfer in turbulent channel flow over dimpled surface. Both symmetric (and spherical) and asymmetric dimple with different depth ratios ( $h/D$ ) are considered for Reynolds numbers (based on bulk velocity and full channel height) between 4000 and 6000 while Prandtl number  $Pr$  is fixed at 0.7. It is found that the optimum dimple configuration for enhancing heat transfer measured in terms of the volume goodness factor is obtained for the case of asymmetric dimple

with a depth ratio of  $h/D = 15\%$ . The heat transfer capacity in terms of Nusselt number is significantly increased, while the associated pressure loss is kept almost to the same level as the symmetric dimple with the same depth ratio. The study also suggested that the heat transfer enhancement is closely related to ejection with counter-rotating flow, intensified secondary flow and vortex structures at the downstream rim of asymmetric dimple. All these findings suggest that a carefully designed asymmetric dimpled surface presents a viable means of enhancing heat transfer compared to the symmetric dimple [7].

### 3.0 DESIGN AND DEVELOPMENT OF EXPERIMENTAL SETUP

The schematic arrangement of the experimental set up is as shown in figure 1 and actual photograph of experimental set up as shown in figure 2. The experimental setup consists of the components such as RTD's, Temperature Indicator, Heating element, an arrangement to vary the heat input, pipe, Stand and Hanger, control valve, Orifice plate etc.

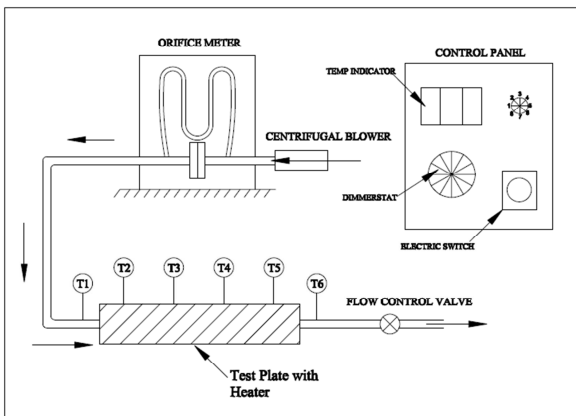


Figure 1: Schematic layout experimental setup.



Figure 2: Photograph of experimental set up at workshop

### 3.1 Test Tube

This stage of work concentrates on developing the dimpled tube with circular shapes with different diameter of perforation. Thermal performance of the dimpled tubes will be compared with the plain tube. The plain tube and dimpled tubes with variation in the diameter of circular dimples are as shown in figure 3. 25 mm (ID) Copper Tubes, 500 mm length, Total Four tubes out of which one with dimple diameter 4 mm, depth 4 mm and inline arrangement and other with staggered arrangement, other two tubes with 8 mm diameter and depth 4 mm with inline arrangement and other with staggered arrangement. Material for plate is copper. These materials are selected considering the thermal properties, manufacturing feasibility.



Figure 3: a) Circular dimples with diameter of 4 mm and depth of 4 mm-Inline arrangement

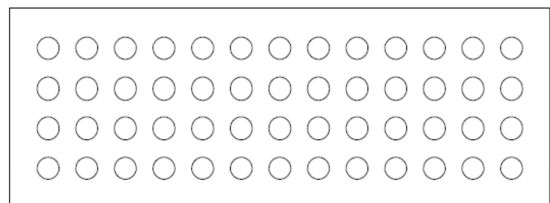


Figure 3: b) Circular dimples with diameter of 4 mm and depth of 4 mm-Inline arrangements

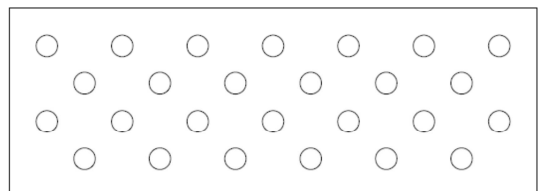


Figure 3 c) Circular dimple with diameter of 4 mm and depth of 4 mm- Staggered arrangement

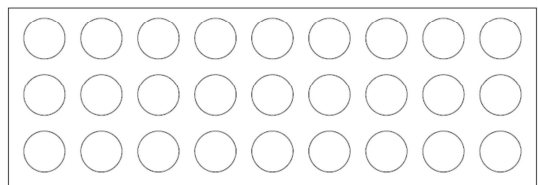


Figure 3: d) Circular dimples with diameter of 8 mm and depth of 4 mm-Inline arrangement

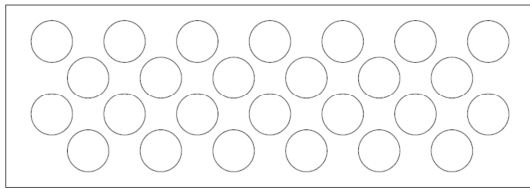


Figure 3: e) Circular dimples with diameter of 8 mm and depth of 4 mm- Staggered arrangement



Figure 3: f) Photograph of the tube with inline dimples



Figure 3: g) Photograph of the tube with staggered dimples

### 3.2 Test Methodology

In order to Experimental Evaluation of heat transfer coefficient through vertical heated roughened surface, it has been decided to vary the mass flow rate of air to for Reynolds number in the range of 12000 to 26000. Readings are taken at steady state. Heater is placed on the dimpled tube to heat the tube. The desired temp is achieved by varying the voltage & current. Whole test set up is mounted horizontally. Voltage supplied is varied with the help of dimmerstat.

Experimentation was carried on the circular tube at various patterns of dimples. All the necessary components were assembled and experimental set was developed. The necessary instruments were attached at correct configuration and the set up is ready for the experimentation.

### 4.0 FORMULAE USED FOR CALCULATION

In the present work air is used as test fluid in flowed through a uniform heat flux and insulated tube. The steady state of the heat transfer rate is assumed to be equal to the heat loss in the test section which express as Formulas used for calculation are as under.

$$Q_{air} = Q_{conv} \quad (1)$$

$$Q_{air} = mC_p\Delta T \quad (2)$$

$$Q_{conv} = hAT_{mean} \quad (3)$$

$$\text{Heat Supplied (Q)} = \text{Voltage} \times \text{current} \quad (4)$$

$$\text{Reynolds No (Re)} = VD/\mu \quad (5)$$

$$\text{Pressure Drop } (\Delta p) = \frac{f\rho V^2}{2D} \quad (6)$$

$$\text{Nusselt Number (Nu)} = hD/k \quad (7)$$

$$\text{Nusselt Number (Nu}_0) = 0.023 \text{ Re}^{0.8} \text{ Pr}^{0.4} \quad (8)$$

$$\text{Friction factor (f)} = 2\Delta pD / L\rho V^2 \quad (9)$$

$$\text{Prandtl no (Pr)} = \frac{\mu C_p}{k} \quad (10)$$

$$\text{Thermal Performance Factor} = \frac{Nu}{\left(\frac{f}{f_0}\right)^{1/3}} \quad (11)$$

$$\text{Enhancement efficiency} = \frac{h}{h_0} \quad (12)$$

Friction factor  $f_0$  is as per Moody's chart

## 5.0 RESULTS AND DISCUSSION

The effect of Inline and staggered arrangement of dimples on the heat transfer characteristics are presented in terms of both local and relative mean Nusselt number and heat transfer coefficient for plane dimpled tubes to the corresponding ones for dimpled tubes at the same Reynolds numbers. The local heat transfer coefficient and Nusselt number is calculated by varying surface of tube by considering location of thermocouples.

### 5.1 Variations in Nusselt number with Reynolds Number for Plain and dimpled tubes

Table 1: Nusselt number for various dimple tube

Reynold number	Nusselt number				
	Plane Tube	Inline Ø4	Stagge red Ø4	Inline Ø8	Stagge red Ø8
12912	27.52	29.92	31.76	33.92	35.37
18174	31.24	32.84	34.90	37.28	39.38
22215	34.82	35.80	38.36	40.70	43.36
25649	35.74	36.58	38.94	41.66	44.60

From the table 1 we observe that the enhancement of Nusselt number in the dimple tube Staggered Ø 8 arrangement is more when compared with plain tube, the same is plotted in figure for different Reynolds number as shown in Figure4.

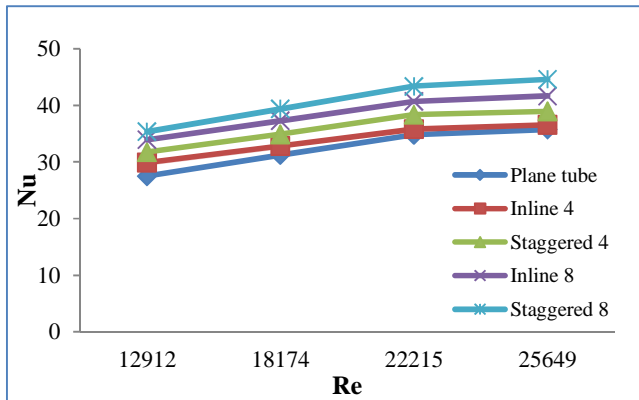


Figure 4: Variations in Nusselt number with Reynolds number for different dimpled tubes

### 5.2 Variations in heat transfer coefficient with Reynolds Number for Plain and dimpled tubes

Table 2 and figure 5 shows the heat transfer coefficient for different Reynolds number for plane and dimple tube.

Table 2: Heat transfer coefficient for various dimple tube

Reynold number	Heat Transfer coefficient				
	Plane Tube	Inline Ø4	Staggered Ø4	Inline Ø8	Staggered Ø8
12912	30.49	33.15	35.19	37.59	39.197
18174	34.91	36.68	38.99	41.65	43.99
22215	39.32	40.43	43.33	45.96	48.97
25649	40.69	41.65	44.33	47.43	50.77

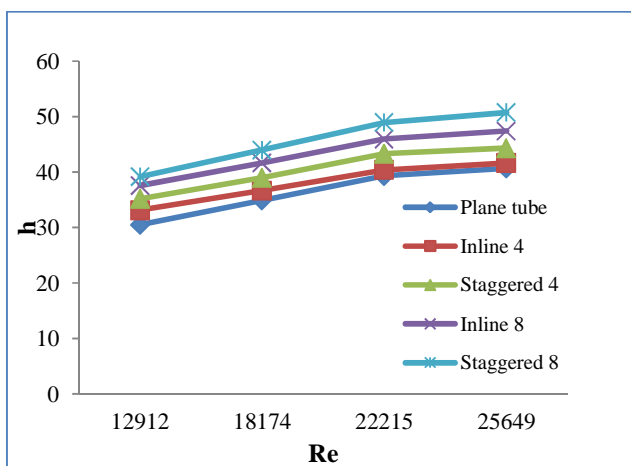


Figure 5: Variations in Heat transfer coefficient with Reynolds number for different dimpled tubes

Figure 5 shows the heat transfer coefficient increases with increase of the Reynolds number, also the heat transfer coefficient of the tubes with dimples is more than the plain tubes.

### 5.3 Variations in Thermal performance factor with Reynolds Number

Table 3 shows the comparison for Nusselt number and friction factor for different dimpled tube as compared plain tube. It shows that there is increase of Nusselt number as well as friction factor.

Table 3: Table for comparison of Heat transfer coefficient & Nusselt number with plain tube for different Reynolds number

Re	Nu/Nu0 & f/f0							
	Inline Ø4		Staggered Ø4		Inline Ø8		Staggered Ø8	
	Nu/Nu0	f/f0	Nu/Nu0	f/f0	Nu/Nu0	f/f0	Nu/Nu0	f/f0
12912	1.08	1.04	1.15	1.09	1.23	1.15	1.28	1.18
18174	1.05	1.02	1.11	1.04	1.19	1.07	1.26	0.97
22215	1.02	1.01	1.10	1.04	1.16	1.06	1.24	0.85
25649	1.02	1.02	1.08	1.02	1.16	1.04	1.24	0.82

Table 4: Table for comparison of Thermal performance factor for different dimpled tubes for Reynolds number

Reynolds number	Thermal performance factor (TPF)			
	Inline Ø4	Staggered Ø4	Inline Ø8	Staggered Ø8
12912	1.016	1.119	1.175	1.217
18174	1.022	1.100	1.165	1.270
22215	1.045	1.087	1.146	1.313
25649	1.075	1.080	1.150	1.332

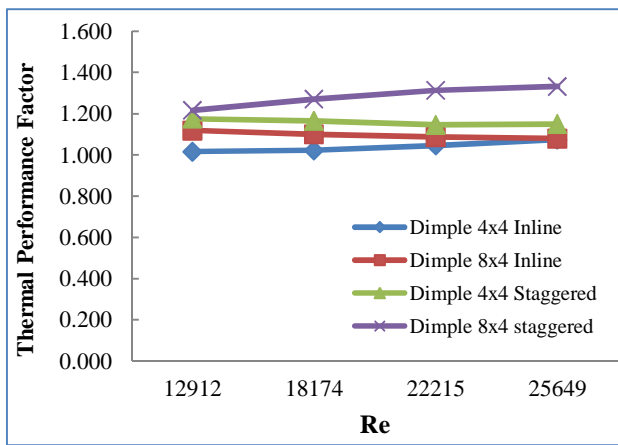


Figure 6: Thermal performance factor with Reynolds number for different dimpled tubes

Figure 6 and Table 4 shows the variation of thermal performance factor with Reynolds number. The thermal performance factors for the dimple tube tend to increase with increasing Reynolds number. Also the tube with staggered array & large dimple diameter the thermal performance factor is more.

## 6.0 CONCLUSION

In this study forced convection heat transfer characteristics from dimpled tube has been investigated experimentally. The effect of variation in dimple diameter and dimple arrangement (inline and staggered) has been investigated for Reynolds number in the range of 12000 to 26000. The whole experimentation has been carried out under turbulent flow condition. This investigation was carried out to observe if the use of dimples on tube can enhance heat transfer characteristics without severe penalties associated with pressure drops for turbulent flow.

The following conclusions were drawn from this study.

1. Heat transfer enhancement under forced convection can be achieved by using dimple tubes. It is also observed that staggered array facilitates higher heat transfer augmentation when compared to the inline array.
2. The enhancement in convective heat transfer coefficient is observed as 18% and in Nusselt number as 22 % as compared with plain tube for Reynolds number 12000 to 26000.
3. Larger dimple diameter shows higher enhancement.
4. The thermal performance factor values for the staggered dimpled array geometry were more than corresponding inline dimpled array geometry for Reynolds number 12000 to 26000.

Thus, the dimpled surface on the tube was found to enhance heat transfer over a plain tube surface for turbulent airflows. The staggered dimpled array geometry proved to give a better thermal performance than the inline dimpled array.

## 7.0 REFERENCE

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