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Heat Transfer Enhancement of System with Flow Divider Type Insert in a Circular Pipe

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Abstract

Experimental investigation of circular tube fitted with full length flow divider type insert has been analyzed. The analysis has been carried out with uniform heat flux under turbulent flow condition to investigate heat transfer and friction factor characteristics. The effect of friction factor and pitch length has been presented separately. Enhancement of heat transfer is found to be increased by 15, 43, 48, 61and 61% more than the plain tube for pitch 10,8,6,4 and 2. Pressure drop is increased by 1.1, 2, 2.4, 2.9 and 4.2 times more than the plain tube for inserts having pitch 10,8,6,4 and 2. Nusselt number increases with increase in Reynolds number and friction factor decreases with increase in Reynolds number. Enhancement ratio for all insert is greater than unity, but there is exception for 10cm pitch insert. Hence the flow divider type insert can only be used for heat augmentation only in turbulent flow with less reduction in pumping power.

Keywords: *Pitch, Insert, Nusselt number, Friction Factor, Reynolds Number, Enhancement efficiency.*

1. Introduction

Conventional resources of energy are depleting at an alarming rate, which makes future sustainable development of energy use very difficult. In recent years, high cost of energy and material availability has resulted in an increased effort at producing more efficient heat exchange equipment's. As a result, considerable emphasis has been placed on the development of various augmented heat transfer surfaces and devices [1, 2]. The study of improved heat transfer performance is referred to as heat transfer enhancement, augmentation or intensification. In general, this means an increase in heat transfer coefficient [3]. Energy and materials-saving considerations, as well as economic incentives, have led to efforts to produce more efficient heat exchange equipment. Common thermal-hydraulic goals are to reduce the size of a heat exchanger required for a specified heat duty, to upgrade the capacity of an existing heat exchanger, to reduce the approach temperature difference for the process streams, or to reduce the pumping power [4].

The need to optimize and conserve these expenditures has promoted the development of efficient heat exchangers. Different techniques are employed to enhance the heat transfer rates, which are generally referred to as heat transfer enhancement or heat transfer augmentation techniques.

2. Objectives

- To obtain increased heat transfer rate through higher flow turbulence
- Decrease in heat transfer surface area, size, and hence the weight of a heat exchanger for a given heat duty and pressure drop
- Decrease of exchanger cleaning through reduction of tube side fouling
- Extended tube life due to reduction of fouling

3. Methodology

- Experimental investigations will be performed initially on plain horizontal tube (without using inserts) at a constant heat input and variable mass flow rate. Then Nusselt number (experimental and from correlations), pressure drop and friction factor (experimental and from correlations) across different Reynolds number is calculated.
- Validation of test section by comparing experimental results with correlations
- Experimental investigations will be carried out on plain tube with flow divider type inserts at a constant heat input. Then Nusselt number, pressure drop and friction factor across different Reynolds number is calculated.
- Nusselt numbers and friction factors obtained from plain tube without inserts and with inserts are compared.

3. Literature Survey

Experimental study on friction factor, mean Nusselt number and Thermal Enhancement factor was carried out on circular tube by **Shrirao et al**[1]. The circular tube has different types of internal threads such as buttress, acme and knuckle with pitch of 120 mm. The experiment was conducted under uniform wall heat flux. Data was measured at Reynold's number ranging from 7000 to 14000 with air as working fluid in all three experiments. Test tubes with internal threads such as buttress, acme and knuckle had mean Nusselt number 1.46, 1.30 and 1.19 times better than that of plain tube. The test tubes with buttress, acme and knuckle threads had thermal enhancement in the range of 1.12-1.04, 1.1-1.03 and 1.08-1.02 respectively.

Turbulent flow heat transfer in horizontal tube was experimentally investigated by **Sarada el at**[2]. The study included horizontal tubes with varying width twisted tape insert with air as working fluid. Plain tube

with/without insert were also experimented at constant wall heat flux and different mass flow rate. Three different twist ratio (3,4 and 5), each with five different width (26- full length, 22, 18, 14 and 10mm) were used on twisted tapes. Pressure drop and heat transfer coefficient were calculated and results were compared with those of plain tube. Reynolds number varied from 6000 to 13500. Comparing the results of twisted tube insert to plain tube, it has been found that enhancement of heat transfer varied from 36% to 48% for full width and 33% to 39% for reduced width of 22mm insert. Maximum friction factor rise was 18% for full width of 26 mm and 17.3% for reduced width insert.

The effect of drilling of the cut conical ring turbulators and space ratio on friction factor, heat transfer was investigated by **Jadoaa**[3]. Under Reynolds number ranging from 5000 to 23500. In addition he also investigated correlation Nusselt number, Friction factor and Performance evaluation criterion to assess the real benefit of drilled conical ring turbulator tube. High rates of heat transfer were achieved by the process of drilling of cut conical ring inside the tube as compared to tube without drilling.

Kore et al. [4] carried out experimental investigation to study heat transfer and friction factor by dimpled surface. The aspect ratio of rectangular channel was kept 4:1 and Reynolds number varied from 10000 to 40000. The ratios of dimple depth to dimple print diameter was varied from 0.02 to 0.04 to provide information on the influences of dimple depth. The ratio of channel height to print diameter is 0.5. The heat transfer and friction factor data obtained was compared with the data obtained from smooth plate under similar geometric and flow conditions. It was observed that at all Reynolds number as depth increases from 0.2 to 0.3, the normalized Nusselt number and thermal performance increases and then after when depth increase from 0.3 to 0.4 normalized Nusselt number and thermal performance decreases. These were because of increase in strength and intensity of vortices and associated secondary flows ejected from the dimples. Fig.2.10 shows geometry of dimpled surface.

4. Insert

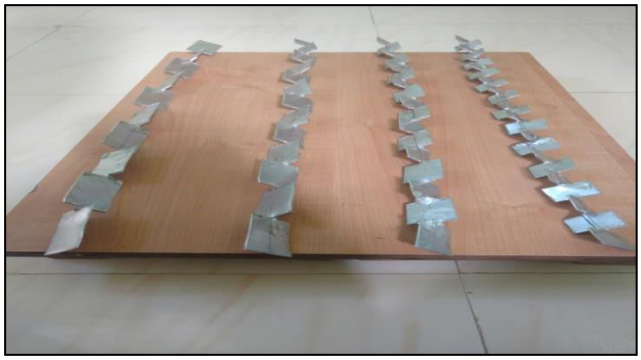

Fig.1 Flow Divider insert

Flow divider type inserts are used for the present experimental analysis. New flow divider type inserts having pitch 2, 4, 6, 8, and 10 cm are made. Material used for making insert is aluminum (al). Physical geometry and dimensions of new flow divider type inserts are shown in fig.1. The dimensions of the new flow divider type insert are chosen such that it forms a tight fit with the horizontal test tube.

4. Heat Transfer Calculation Procedure

4.1 Estimation of Reynolds Number and Prandtl Number

Mean Bulk temperature

$$T_{mb} = \frac{T_1 + T_7}{2} \ (^0\,C)$$

Mean Surface temperature,

$$T_S = \frac{T_2+T_3+T_4+T_5+T_6}{5}, \ (^0\,C)$$

Properties of air ρ_a ,C_P, μ, k and Pr are calculated at bulk mean temperature i.e.at T_{mb}

Equivalent air column height,

$$h_a = \frac{h_w \times \rho_w}{\rho_a}, (m)$$

Volume flow rate,

$$Q_d = \frac{C_d \times A \times A_0 \times \sqrt{2gh_a}}{\sqrt{A^2 - A0^2}}, (m^3/\,s)$$

Mass flow rate,

$$\dot{m} = \rho_a \times Q_d \ , (Kg/S)$$

Mean velocity of air through pipe,

$$U = \frac{Q_d}{A}, (m/s)$$

Reynolds number,

$$Re = \frac{4 \times \dot{m}}{\pi \times D \times \mu}$$

Prandtl Number,

$$Pr = \frac{\mu \times C_p}{k}$$

4.2 Estimation of Experimental Heat Transfer Coefficient

Actual Heat Carried away

$$Q_t = m \times C_p \times (T_7 - T_1), (W)$$

Convective Heat Transfer Coefficient

$$h = \frac{Q_t}{A_s \times (T_s - T_{mb})}, (W/m^2.K)$$

Experimental Nusselt Number

$$Nu_{Ex} = \frac{h \times D_h}{k}$$

Experimental Friction Factor

$$f = \frac{\Delta P_{Ex}}{\left(\frac{L}{D}\right) \times \left(\frac{\rho_a \times U^2}{2}\right)}$$

4.3 Correlation for Plain Tube

Nusselt number (Dittus-Boelter Correlation)

$$Nu_{Dittuse-Boelter} = 0.023 \times Re^{0.8} \times Pr^{0.4}$$

Friction Factor (Blasius Correlation)

$$f = 0.316 \times Re^{-0.25}$$

4.3 Heat Transfer Calculation with Insert

Velocity of air Flow,

$$U_i = \frac{Q_t}{A_{fr}}, \left(\frac{m}{s}\right)$$

Free area for airflow,

$$A_{fr} = \left(\frac{\pi}{4} \times D^2\right) - (w \times t), (m)$$

Wetted perimeter,

$$P = (\pi \times D) + 2(w + t), (m)$$

Reynolds number,

$$Re = \frac{\rho_a \times U_i \times D_h}{\mu}$$

Nusselt Number

$$Nu = \frac{h \times D_h}{k}$$

Friction Factor

$$f = \frac{\Delta P_{Ex}}{\left(\frac{L}{D}\right) \times \left(\frac{\rho_a \times U_i^2}{2}\right)}$$

Overall Enhancement Ratio

$$\eta = \frac{\left(Nu_i/Nu\right)}{\left(f_i/f\right)^{(1/3)}}$$

Enhancement Efficiency,

$$\Psi = \frac{h_i}{h}$$

5. Experimental Setup

Fig.2 Experimental Setup

The schematic diagram of the open loop experimental setup is shown in Fig.2. The loop consists of a blower unit fitted with a tube in horizontal orientation. The blower fan runs at a constant speed. Outlet of blower is connected to a GI pipe having inside diameter 27.5 mm and outside diameter 33.9 mm. The U-tube manometer is connected across the orifice meter to measure pressure drop. Nichrome bend heater of resistance 135 Ω encloses the test section to a length of 500 mm to cause electric heating. Supply to the heater is given from variable transformer. Power input to the test tube heater is varied using a variable transformer, which is used to vary the voltage of the AC current passing through the heater and by keeping the current less than 2A. Two thermocouples are placed one at the entrance and the other at the exit of the test section at a distance of 50 mm from test section to measure air inlet and outlet temperatures (T_1 and T_7) respectively. Other five thermocouples i.e. T_2, T_3, T_4, T_5 and T_6 measure the temperatures at various points along the test tube surface, which are located at distance of 50 mm, 150 mm 250 mm, 350 mm and 450 mm from inlet end of test tube respectively. The outer surface of the test section was well insulated to minimize convective heat loss to the surrounding. The control valve at the exit section controls the airflow rate into the test section. Necessary precautions were taken to prevent leakages from the system. Pressure tapings at each end of the test section connects to digital pressure sensor to measure the pressure drop across the test section.

6. Experimental Procedure

- Supply is given to the blower motor and the valve is opened slightly.
- The blower was started and the airflow rate was adjusted by operating the control valve, so that the desired difference in manometer level (40 mm, 80 mm, 120 mm or 160 mm) was obtained.
- Start the Nichrome bend heater wound on the test section by giving desired heat input by adjusting the dimmer stat.
- Thermocouples 2 to 6 are fixed on the test surface to measure surface temperature and thermocouples 8 and 7 are fixed inside the pipe to measure inlet air temperature and outlet air temperature respectively
- The readings of the thermocouples are observed every 5 minutes until they show constant values.
- The variation of tube wall temperatures (T_2, T_3, T_4, T_5 and T_6) were observed until the constant value was attained and then the outlet air temperature (T_7) was observed until the temperature didn't deviate over fifteen to twenty minutes. Steady state temperatures from T_1 to T_7 are tabulated.
- Under steady state condition, the readings of all the seven thermocouples are recorded. The experiment is repeated for different openings of the valve, thus varying the airflow rate.
- For each test run, temperatures, volumetric flow rate and pressure drop of bulk air at steady state conditions were noted down.
- The fluid properties were calculated as the average between the inlet and the outlet temperature.
- It took 90 minutes to reach steady state conditions.
- Experiment was carried out at constant heat flux conditions and constant heat input at different mass flow rates.
- The Reynolds number of the air was varied from 6000 to 17000 during the experimentation. The various characteristics of the flow, Nusselt number and the Reynolds number on the average tube wall temperature and the inlet and outlet air temperature were plotted,
- Same procedure is repeated with insert in flow.

CFD Analysis

Computational grid generation

For the study of the numerical performance of experimental setup STAR CCM+ package is used. Test section geometry is developed in 3D CAD modeling software CATIA V5 and thereafter geometry is imported in STAR CCM+ for simulation. Polyhedral meshing model is used for along with Embedded thin mesher and prism layer thickness. Cell count for plain tube with 20 mm insert is 162571. The grid generation for 20 mm insert is shown below in Fig.3

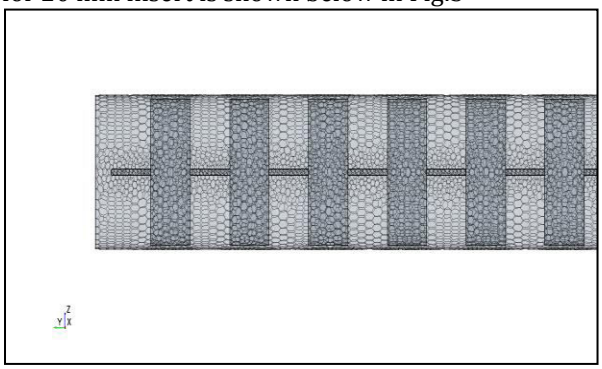

Fig.3 Computational grid for tube with 20 mm insert

For solving this problem in software steady-state, three-dimensional, ideal gas and turbulent model is used. Furthermore k-ε turbulence model is used to solve the problem. k-ε turbulence model is widely used in for engineering application. As the working fluid is air, Segregated flow model is choose as air shows variation in flow with change in density. Velocity input is given at the inlet while the heat flux boundary condition is given to the tube surface.

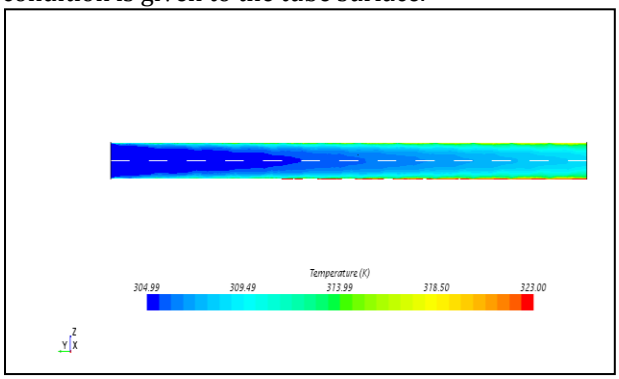

Fig.4 Temperature variation along the length of tube for insert with 20 mm pitch at Re=7500

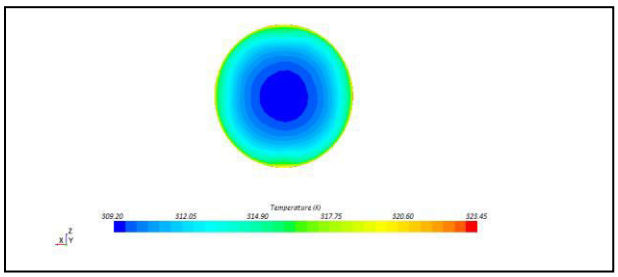

Fig.5 Temperature variation at the outlet surface for insert with 20 mm pitch at Re=7500

7. Result and Discussion

Fig.6 shows the comparison of experimentally obtained Nusselt numbers for plain tube and inserts. It was observed that 2 cm pitch insert yielded the highest value of Nusselt number. This may be due to free flow area of the flow decreases, leading to more turbulence and rapid mixing of flow. It was observed that the Nusselt number increased with increase in Reynolds number and decrease in pitch. This can be expected, as the pitch decreased from 10 cm to 2 cm, the obstruction to airflow increases, that increases the turbulence created within the test section, leading to rapid mixing of the flow. As a result, more heat is carried away by air, which results in increase of Nusselt number. Nusselt number increased by a maximum of 61% more than the plain tube in the presence of 2 cm pitch inserts.

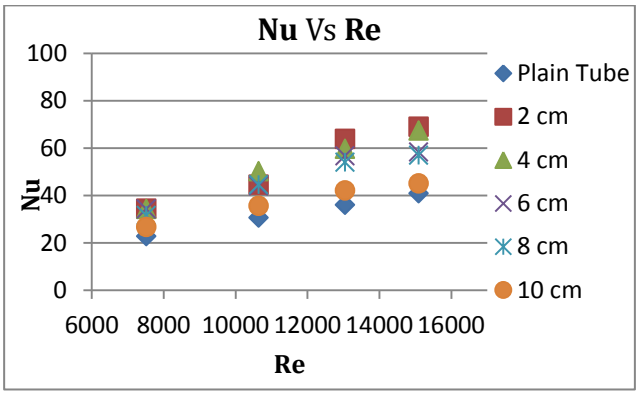

Fig.6 Variation of Nusselt number with Reynolds number

Fig.7 shows Pressure drop increases with increase in Reynolds number. Pressure drop (i.e. head loss) is a function of velocity squared. The increase in pressure drop with Reynolds Number is due to increased velocity associated with increase in mass flow rate. Also, pressure drop increases with decrease in pitch of insert. This is due to the fact that, as pitch decreases, obstruction to the flow increases. Maximum pressure drop is observed to be 4.2 times more than that of plain tube for insert with pitch 2 cm.

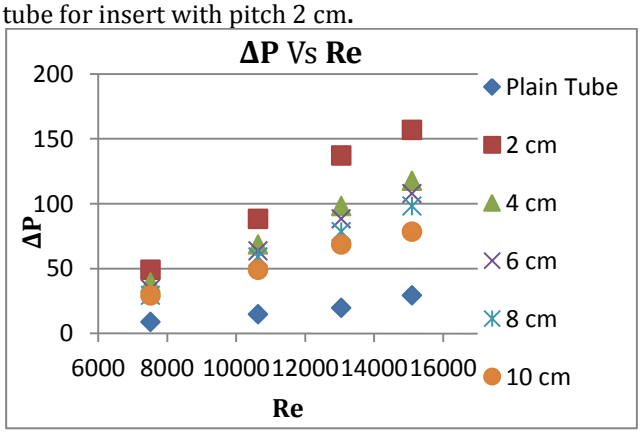

Fig.7 Variation of pressure drop with Reynolds number

Fig.8 indicates the variation of friction factor with Reynolds number. The friction factor, which is high at lower Reynolds number, tends to reduce with the increase of Reynolds number. This is due to the fact that, *f* is inversely proportional to square of velocity for a given pipe. So, friction factor is decreases due to increased velocity associated with increase in mass

flow rate. Also, friction factor increases with decrease in pitch of insert. This is due to the fact that, as pitch decreases, obstruction to the flow increases.
Friction factor for inserts having pitch values 10, 8, 6, 4, and 2 cm is increased to 74, 100, 123, 146, and 225% respectively compared to plain tube. This is due to the obstruction caused to airflow leading to higher friction factors

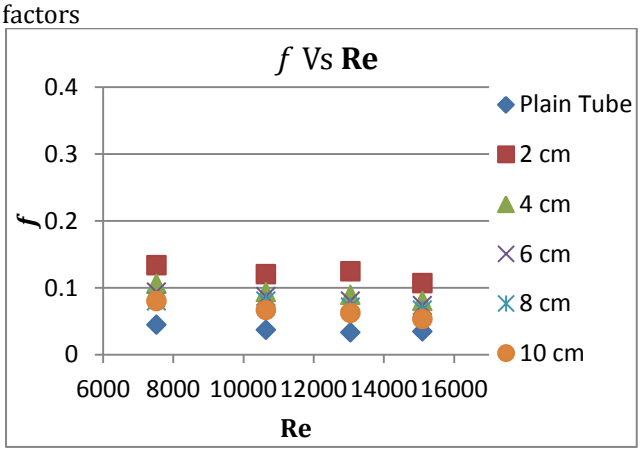

Fig.8 Variation of friction factor with Reynolds number

Fig.9 shows the variation of enhancement ratio with Reynolds number. Enhancement ratio for all insert is greater than unity, but there is exception for 10 cm pitch insert. This is due to the fact that friction factor ratio dominates Nusselt number ratio for this insert. Also, greater increase of pressure drop and friction factor for insert of pitch 10 cm enhancement ration is less than one.
Above experimental results are validated with the results obtained from analysis. Both pressure drop and Nusselt number are compare with the simulation results and the graph is plotted. It is Shown in Fig.10 and Fig.11

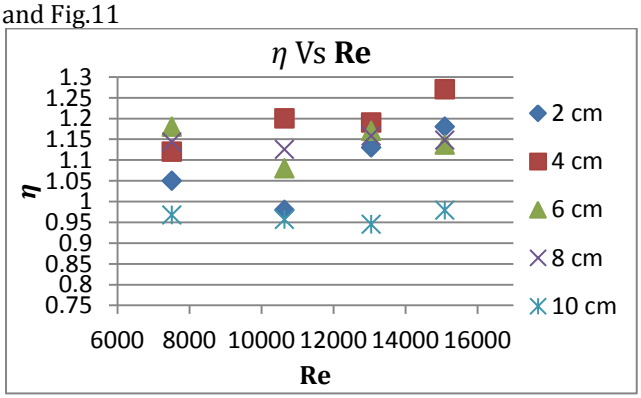

Fig.9 Variation of Enhancement ratio with Reynolds number

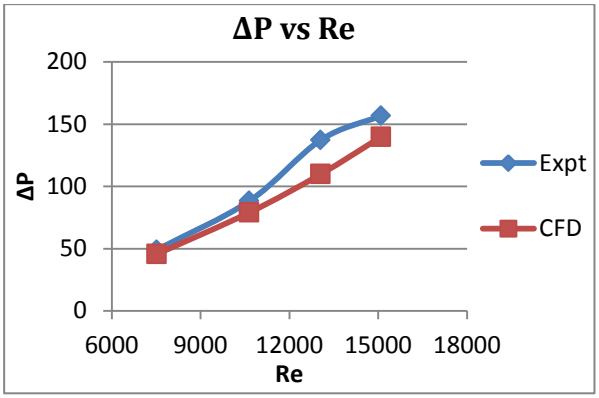

Fig.10 Comparison of numerical and experimental pressure drop for 20 mm insert

Fig.10 Shown above display the variation of experimental pressure drop and numerical pressure drop with Reynolds number. It shows that experimental results shows 10% to 15% variation as compare to numerical results.

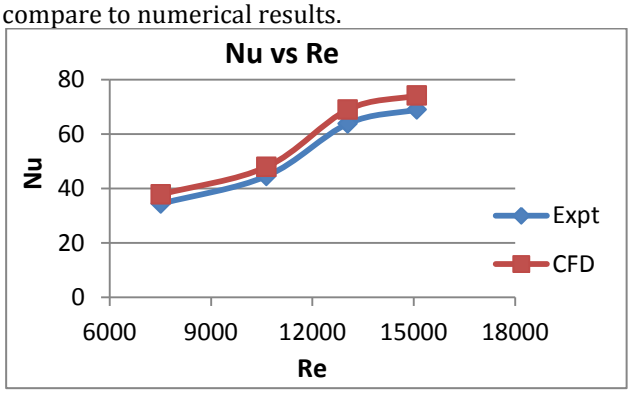

Fig.11 Comparison of numerical and experimental Nusselt number for 20 mm insert

From the above Fig.11 it is clearly seen that the numerical and experimental Nusselt numbers are shows 5% to 10% difference. It is shown that the Nusselt number and friction factor both increases with increses in Reynolds number.

Conclusions

In case of Plain Tube it is observed that, the experimental data are in good agreement with the correlation values, i.e., Dittus–Boelter Correlation and Blasius Correlation. Experimental data of the present work is reasonably agreed within 10% for Nusselt number from Dittus-Boelter correlation and 13.7% for friction factor from Blasius correlation
For the different type of Insert, it is observed that the enhancement of heat transfer is found to be increased by 15, 43, 48, 60 and 61% more than the plain tube for pitch 10,8,6,4 and 2 cm respectively and Pressure drop is increased by 1.1, 2, 2.4, 2.9, and 4.2 times more than the plain tube for inserts having pitch 10,8,6,4 and 2 cm respectively. So the enhancement ratio for all insert is greater than unity, but there is exception for 10cm cm pitch insert.

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