ANALYZING EFFECT OF VARYING PITCH OF CUT CORRUGATED TWISTED TAPE INSERT ON AUGMENTATION OF HEAT TRANSFER

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Abstract: In the current research work, the results obtained after carrying out experiments dealing with the amplification of turbulent flow heat transfer in a horizontal tube by means of cut corrugated twisted tape inserts with air as the working fluid, are shown. These experiments were conducted for plain tube with and without cut corrugated twisted tape insert at constant wall heat flux and different mass flow rates. The cut corrugated twisted tapes are of same wave-width, but three different twist ratios as 8.33, 9.79 & 10.42. The Reynolds number used varied between 4000 to 9500. Both heat transfer coefficient and pressure drop are calculated and the results are compared with those of bare tube. It was found that the increment of heat transfer with cut corrugated twisted tape inserts as compared to plain tube varied from 18% to 52 % for various inserts. Also the obtained results are compared with the plane twisted tape insert. Further the results obtained are compared with the basic corrugated twisted tape insert.

Index Terms: corrugated twisted tape insert, heat transfer augmentation, wave-width, twist ratio.

I. INTRODUCTION

Number of techniques (both passive and active) are investigated for augmentation of heat transfer rates inside circular tubes and a wide range of inserts have been used, especially when turbulent flow is considered. Few of these methods are applied to increase thermal performance of heat transfer devices such as treated surfaces, coarse surfaces, swirling flow geometries, coiled tubes, and surface tension devices [1]. Twisted tape swirl tabulator is one of the frequently used passive types for heat transfer augmentation as they present advantages of stable performance, simple configurations and ease of installation [2].

Sarma et al. [3] figured out generalized correlations to predict friction factor and convective heat transfer coefficient in a tube fitted with twisted tapes over a wide range of Reynolds numberand the Prandtl number. Design optimization of regularly spaced short-length twisted tapes in a circular tube for turbulent heat transfer was carried out by Wang et al. [4] by using computational fluid dynamics (CFDs) modeling. Eiamsa-ard et al. [5] proposed experimental study of convective heat transfer in a circular tube with short-length twisted tapes inserted under uniform heat flux. Akhavan-Behabadi et al. [6] conducted some experiments to analyze effects of twisted tapes on heat transfer enhancement andpressure drop in horizontal evaporators. The working fluid used was R-134a. Heat transfer and friction factor of CuO/waternanofluid and water were experimentally checked in circular tube equipped with modified twisted tapes was another alternate[7–9].Eiamsa-ardet al. [10] performed research works on heat transfer and friction factor characteristics in a double pipe heat exchanger fitted with twisted tape elements. They performed their analysis for both continuously placed twisted tape and twisted tape placed with various free space inside a circular tube. The heat transfer augmentation and pressure drop during condensation of HFC-134a in a horizontal tube fitted with twisted tapes were experimentally analyzed [11]. Jaisankar et al. [12] experimentally examined the heat transfer, friction factor and thermal behavior caused by twisted tape for solar water heater.

Tape width, twist ratio, space ratio, rod-diameter and phase angle effects on heat transfer and pressure drop we reanalyzed experimentally in a circular tube fitted with regularly spaced twisted tape elements [13].Naphon [14] also made experiments by using conventional twisted tape inserts in horizontal double pipe. Ferroni et al.[15] conducted some experiments in circular tube equipped with physically separated, multiple, short-length twisted tapes. Laminar convective heat transfer enhancement in twisted tape inserted tube was discussed experimentally by Sarma et al.[16]. In some studies, researchers focused the thermal effects of twisted tape inserts in modified tube instead of smooth tube, for example; Thianpong et al. [17] examined heat transfer enhancement in a dimpled tube with a twisted tape swirl generator inserted inside.

They also figured out the empirical correlations based on the experimental outcome of their study for predicting the Nusselt number and friction factor for Reynolds number variation from 12,000 to 44,000. Bharadwaj et al. [18] performed experiments by using conventional type of twisted tapes to determine pressure drop and heat transfer characteristics ofwater in a 75-start spirally grooved tube. Some researchers[19] modified the conventional twisted tape geometries, for example; Murugesan et al. [20] used V-cut twisted tapes to analyze heat transfer and pressure drop in a circular tube.

II. EXPERIMENTAL SET-UP

The schematic diagram of experimental set-up is given in Figure 1.

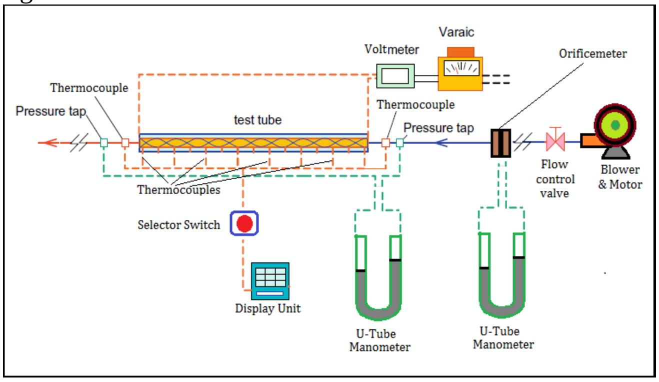

Fig. 1.Experimental setup block diagram

The experimental facility includes a blower, an orifice meter for measurement of the volumetric flow rate and the heat transfer test tube section (700 mm) .The MS test tube of 26 mm inner diameter (D1), 26.4 mm outer diameter (D2), and a thickness (t) of 2 mm. The cut corrugated twisted tapes are testedin this trial, with three various twist ratios as 8.33, 9.79 & 10.42, but they have same wave-width of 13mm. They are fabricatedfrom aluminium. Also one plane twisted tape made up of aluminium is tested. The schematic figure of the test tube with corrugated twistedtape insert is given in Figure 2.

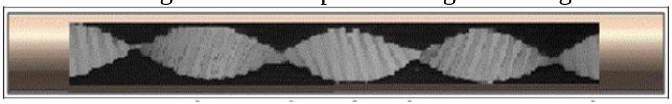

Fig. 2.Schematic of test tube with corrugated tape inserted

The cut corrugated twisted tapes used in thisexperimental study are shown in Fig. 3. A 0.24 hp blower isused to forcefully pass the air through the test tube. Uniform heat flux isapplied to external surface of the test section by means of heating itwith electrical winding andits output power is controlled by avariac transformer to supply stable heat flux along the whole section of the test tube. The outer surface of the test tube is well insulated with glass wool to reduce the convective heat loss to the surroundings. The external surface temperatures of the test tube wall are measured by 6 K-type thermocouples, which are placed on the outer wall of the test tube. Also, the inlet and outlet temperatures of the bulk air are measured by two K-type thermocouples at specific points. An inclined manometers used to measure pressure drop across the test tube. After air passes the test tube, it enters the orifice meter, which is used for finding the volumetric flow rates. For this purposea separate U-tube manometer is placed across orifice meter. The volumetric flow rate of air supplied from the blower is controlled by varying the position of the control valve. The experiments were conducted by varying the flow rate in terms of Reynolds numbers from 4181 to 9466 of the bulk air. The test tubeis heated from the external surface during the experiments,and the data of temperatures, volumetric flow rate, pressuredrop of the bulk air and electrical output are noted afterthe system is approached to the steady state condition. TheNusselt number, Reynolds number, friction factor, heat transfer enhancement are calculated based on the average outer wall temperatures and the inlet and outlet temperatures of air.

Fig. 3.Actual view of cut corrugated twisted tape insert

III. DATA COMPILATION AND ANALYSIS

The data reduction of the obtained results is summarized in following procedures:

A. Heat Transfer Calculations

Avg. Surface Temp., $T_s = (T_2+T_3+T_4+T_5+T_6+T_7)/6$ (1)

Avg. Temp of air, $T_b = (T_1+T_8)/2$ (2)

Air head, $h_a = h_w * \left(\frac{\rho_w}{\rho_a}\right)$ (3)

where,

ρ_w = Density of water = 1000 kg/m^3

Air volume flow rate, $Q_a = C_d * A_o\sqrt{2 * g * h_{air}}$ (4)

where,

A_0= cross sectional area of orifice.

Mass flow rate, $\dot{m} = Q_a * \rho_a$ (5)

Velocity of air, $V = Q_a / A$ (6)

where,

A= cross sectional area of pipe.

Heat carried out, $q = \dot{m} * C_p * (T_8 - T_1)$ (7)

$$h = \frac{Q}{A\,(T_s - T_b)} \quad (8)$$

where,

h = heat transfer coefficient.

T_s = surface temperature

The Reynolds number for the fluid is defined by,

$$Re = \frac{VD}{\upsilon} \quad (9)$$

where,

V= velocity of the fluid.

ν = Kinematic viscosity of the fluid.

For internal flow conditions, if Reynolds number (Re) is greater than 4000 then the flow is said to be turbulent. After the flow is decided i.e. laminar or turbulent then the Nusselt number can be calculated. The theoretical Nusselt number is calculated below without considering friction which is theoretical Nusselt number and then calculated by considering friction which will be experimental Nusselt number.

$$Nu_{th} = 0.023*(Re)^{0.8}*(Pr)^{0.4} \quad (10)$$

This equation is called Dittus-Boelter equation.

$$f_s = (1.82 \log_{10}Re-1.64)^{-2} \quad (11)$$

This equation is used to find friction factor and is called as Petukhov equation for smooth surface.
where,
f_s= Friction factor for smooth tube .
Re= Reynolds number.
The actual pressure drop & friction factor is calculated with the help of tappingson both the ends of test pipe connected to U-tube manometer and the friction factor is calculated from the formula given below:

$$f=\frac{\Delta P}{\frac{L}{D}*\frac{\rho_a V^2}{2}} \quad (12)$$

where,
Δ P= pressure difference at both ends of test pipe.
L= length of test pipe.
D= Inner diameter of pipe.
The experimental Nusseltnumber are calculated as given below:

$$Nu = hD/k \quad (13)$$

Nu= Nusselt number
h = heat transfer coefficient
k = thermal conductivity of fluid
D = diameter of test section
The overall enhancement efficiency is expressed as the ratio of the Nusselt number of an enhanced tube with corrugated twisted tape insert to that of a smooth tube, at a constant pumping power.This factor is introduced by Webb[19].

$$PEC= \eta = \frac{Nu\ with\ /Nu\ w/o}{(fwith\ |fw/o)^{1/3}} \quad (14)$$

B. Validation experiments of plain tube
In this study, experimental results of Nusselt number and friction factor for the plain tube are obtained and validated with equations of Dittus Boelter(10) and Petukhov (11) as mentioned above. The comparisons of friction factor and Nusselt number for the present plain tube with existing correlations are shown in Fig. 4 and 5, respectively. These figures shows that validation experiments of heat transfer in terms of Nusselt number and friction factor for the plain tube are in good agreement with the results obtained from Dittus-Boelter and Petukhov equations. The results of present plain tube and previous equations are nearly the same. Thus, this accuracy provides reliable results for heat transfer and friction factor in a tube with twisted tape inserts in this present study. The Reynolds number for validation test were ranged from 4500 to 10500 i.e. the range of Reynolds number used is for turbulent flow. Turbulent flow means Reynolds number greater than 4000. The results of the tests carried for performance checking of present corrugated twisted tape are discussed further in results and discussion.

IV. RESULTS AND DISCUSSION

A. Heat Transfer and Overall Enhancement Characteristics
The variation of Nusselt number with Reynolds number for various cut corrugated inserts is shown in Figure 6. Highest Nusselt number was obtained for tape with twist ratio of10.42. The Nusselt number for cutcorrugated inserts varied from 30% to 146% compared to plain tube. This is due to

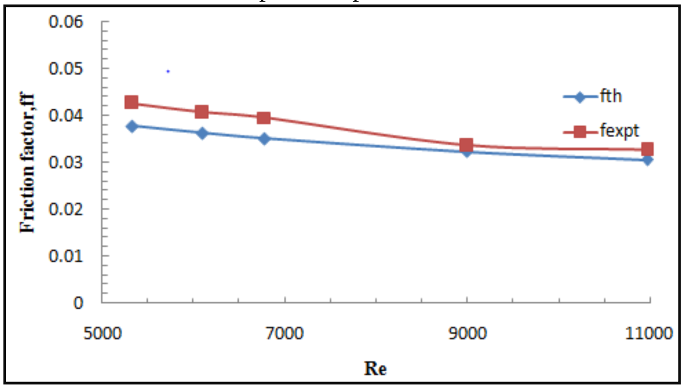

Fig. 4. Validation results for friction factor

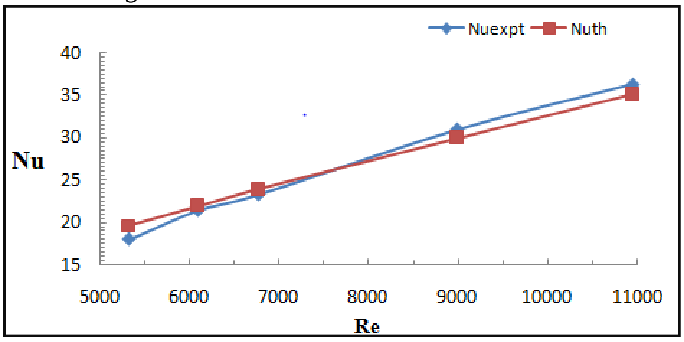

Fig. 5. Validation results for Nusselt number

strong turbulence intensity generated by corrugations& cuts on inserts leading to rapid mixing of the flow causing heat transfer enhancement. The variations of friction factor with Reynolds number for corrugated tape inserts are presented in Figure 7. It is observed that the friction factor gradually reduced with rise in Reynolds number. It is observed to be maximum, for insert having twist ratio of 10.42. It is clear from Figures 6, 7 and 8 that when a corrugated twisted tape is inserted into a plain tube there is a significant improvement in Nusselt number because of secondary flow, with greater augmentation being realized at lower Reynolds numbers and higher twist ratio keeping wave-width same. This enhancement is mainly due to the centrifugal forces resulting from the spiral motion of the fluid and partly due to the tape acting as fin. It is observed that the rise in twist ratio causes increment in Nusselt numbers as well as rise in pressure drop. From Figure 6, the percentage rise in Nusselt numbers for corrugated twisted tapes compared to plain tube are about 30-89%, 33-100% and 58-146% respectively for tape with twist ratio 8.33,9.79 & 10.42 respectively keeping wave-widths same as 13mm.The overall enhancement ratio is useful entity to evaluate the quality of heat transfer enhancement obtained over plain tube at constant pumping power. It is found to be more than unity for all the corrugated twisted tape inserts under test. Variations of overall enhancement ratio η against Reynolds number for various tapes are shown in figure 8. It is observed that overall enhancement tended to decrease gradually with the rise of Reynolds number for all twist ratios. The highest value of overall enhancement is 1.70 for corrugated twisted tape insert having wave-width of 13mm with twist ratio 10.42. It is seen in figure 8 that, for tapes of twist ratios 8.33, 9.79 & 10.42 curves are of decreasing order for a given wave-width in the range of Re from 4100 to 9400.

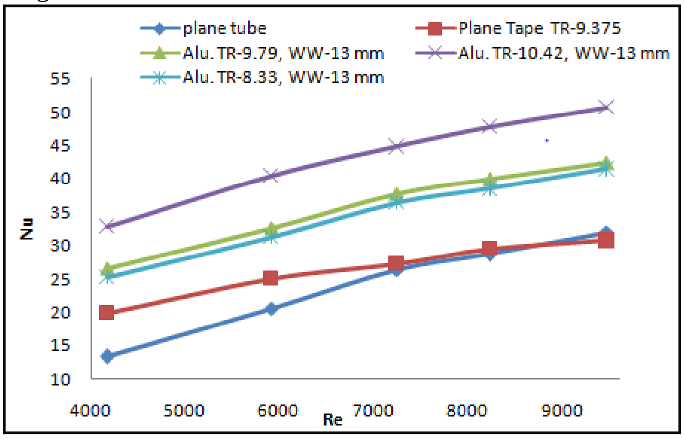

Fig. 6. Variation of Nusselt number for different insert configurations

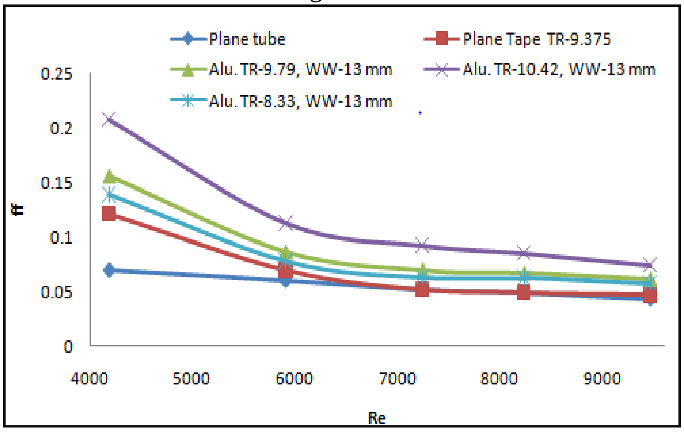

Fig. 7.Variation of friction factor for different insert configurations

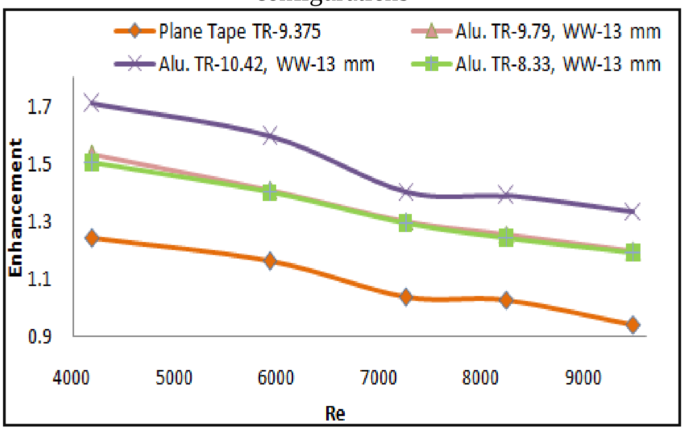

Fig. 8. Variation of Enhancement for different insert configurations

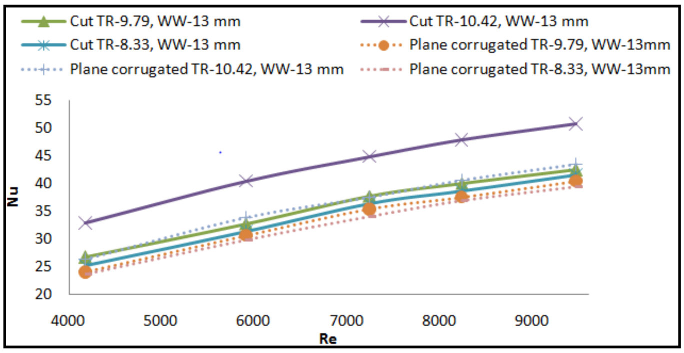

Fig.9.Comparison of Nusselt number with plane corrugated insert

The comparison of Nusselt number obtained in case of cut corrugated tape insert with plane corrugated tape insert i.e. without cut is shown in Fig. 9. It is seen that the rise in Nusselt number ranges from 5-11 % , 16-25% & 5-6% for the inserts of 9.79, 10.42 & 8.33 twist ratios respectively.

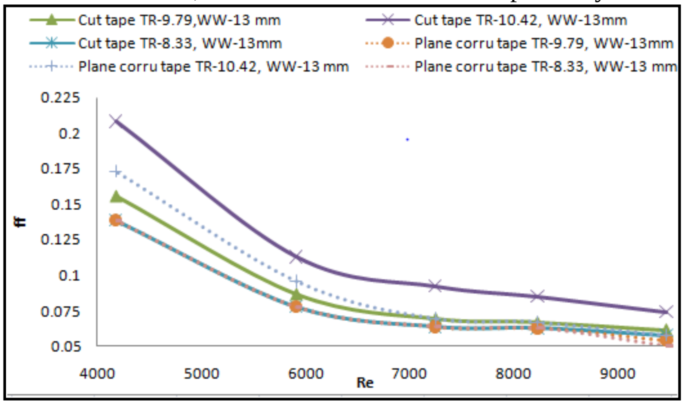

Fig.10.Comparison of Friction Factor with plane corrugated insert

The comparison of friction factor obtained in case of cut corrugated tape insert with plane corrugated tape insert i.e. without cut is shown in Fig. 10. It is seen that the rise in friction factor ranges from 11-12% ,20-29% &1-13% for the inserts of 9.79, 10.42 & 8.33 twist ratios respectively.

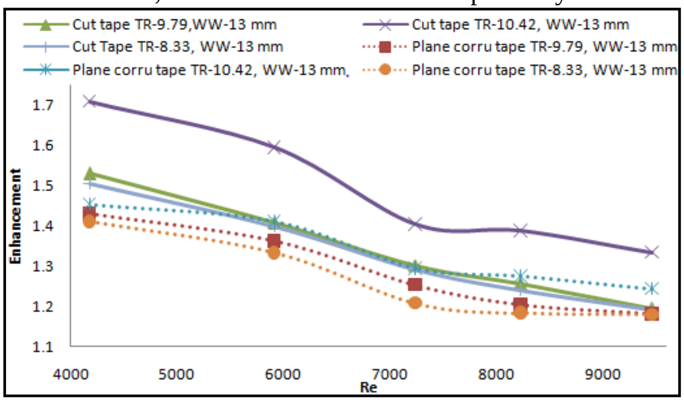

Fig. 11. Comparison of Enhancement with plane corrugated insert

The comparison of enhancement obtained in case of cut corrugated tape insert with plane corrugated tape insert i.e. without cut is shown in Fig. 11. It is seen that the rise in enhancement ranges from 1-7 % , 7-18% & 0-7% for the inserts of 9.79, 10.42 & 8.33 twist ratios respectively.

V. CONCLUSIONS

The study presents an experimental investigation of the potential of corrugated twisted tape inserts to enhance the rate of heat transfer in ahorizontal circular tube with inside diameter 26 mm with air as working fluid. The Reynolds number varied from 4100 to 9400. Theeffects of parameters such as modified twist ratio, Reynolds number on the heat transfer and overall enhancement ratio are studied.

The following conclusions can be drawn.

- The enhancement of heat transfer with cut corrugated twisted tape inserts as compared to plain tube varied from 19 to 52% for twist ratio 8.33 and from 33 to 70% for twist ratio of 10.42. This enhancement is mainly due to the centrifugal forces resulting from the spiral motion of the fluid. Rise in twist ratio causes rise in Nusselt numbers as well as friction factors.
- The maximum friction factor rise was about 88% for twist ratio 8.33 and 189% for twist ratio 10.42 for corrugated twisted tapeinserts compared to plain tube.
- The overall enhancement for the tubes with corrugated twisted tape inserts is 1.50 for twist ratio 8.33 and 1.70 for twist ratio 10.42 for corrugated twisted tape insert.
- The rise in enhancement in case of cut corrugated twisted tape insert as compared to plane corrugated tape insert was 1 to 18%. Maximum rise observed with tape of twist ratio 10.42 having its wave-width 13 mm.
- Thus the improved performance can be achieved using cut corrugated twisted tapes as compared to plane twisted tape. Thus, from the considerations of enhanced heat transfer and savings in pumping power, corrugated cut width tape inserts are seen to be attractive for enhancing turbulent flow heat transfer in a horizontal circular tube.

Future work may be extended as below:

A. Chang tape material from Aluminium to Copper.

B. Compound enhancement techniques maybe applied i.e., the tape inserts can be coupled with coil wire inserts for better enhancement.

NOMENCLATURE

A_0area of orifice, (m^2)
Atest section inner tube area, ($\pi/4\ D^2$) (m^2)
Cp specific heat of air, (J/kg K)
$Q_a$air discharge through test section (m^3/sec)
DInner diameter of test section, (m)
Hpitch, (mm)
w width of corrugated tape insert,(mm)
H/D twist ratio
f_{th}friction factor(theoretical) for plain tube
f friction factor(experimental) for plain tube
$f_i$friction factor obtained using tape inserts
$h$experimental convective heat transfer coefficient, W/m^2K)
$h_w$manometer level difference,(m)
h_{air}equivalent height of air column, (m)
$k$thermal conductivity, (W/mK)
L length of test section, (m)
$\dot{m}$mass flow rate of air, (Kg/sec)
Nu_iNusselt number (experimental) with tape inserts, (hD/k)
NuNusselt number (experimental) for plain tube
Nu_{th}Nusselt number for plain tube (theoretical)
PrPrandtl number
p pitch, (m)
ΔPpressure drop across the test section, (Pa)
Qtotal heat transferred to air (W)
Re Reynolds number, ($\rho\ V D/\mu$)
T_1-T_8 - air temperature at inlet and outlet, (°K)
T_2, T_3, T_4, T_5, T_6 - tube wall temperatures, (°K)
Ts average Surface temperature of the working fluid, (°K)
$T_b$bulk temperature, (°K)
V air velocity through test section, (m/sec)

Greek symbols

ν Kinematic viscosity of air, (m^2/sec)
μ dynamic viscosity, (kg/m s)
η Over all enhancement
$\rho_w$density of water, (kg/m^3)
$\rho_a$density of air (kg/m^3)

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