Experimental investigation on effect of helical grooves on condensation heat transfer performance of vertically oriented copper tubes

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Abstract. An experimental investigation was conducted to study the influence of pitch of helical grooves on condensation heat transfer performance on vertically oriented copper tubes. The condensation heat transfer coefficient of bare as well as grooved copper tubes of various pitches ranging from 2.54 mm to 22.4mm were studied. The investigation revealed that copper tube with groove pitch ≤ 6.35 mm has an adverse effect on the heat transfer performance in comparison to bare copper. The helical groove with pitch > 6.35 mm showed improvement in heat transfer coefficient. The helical groove with pitch 8.47 mm showed a maximum enhancement of 68.4% in condensation heat transfer coefficient at $\Delta T = 40^0C$.

1 Nomenclature

ΔT	Degree of sub cooling,(T_3-T_s)	^oC
ΔT_{LMTD}	Log mean temperature difference	^oC
U	Overall heat transfer coefficient	W/m^2K
V	Volumetric flow rate of coolant	LPM
R	Thermal resistance	^oC /W
Subscripts		
3	Condensing Chamber	
i	Inner	
o	Outer	
s	Surface	
w	Wall	
c	Coolant	
f	Fouling factor	
Cu	Copper	
cond	Condensation	
conv	Internal convective	

2 Introduction

Condensation heat transfer process is used in industry for various applications like power generation, electronic thermal management, desalination and air conditioning system, as it is capable of handling large heat flux even in cases with low temperature gradient. During surface condensation as saturated vapour comes in contact with a surface kept below saturation temperature, droplets start to form on the condensation surface, which then grows, coalesce with nearby droplets and finally departs from the surface due to gravity after reaching a critical size.

Lot of research has been done to improve the condensation heat transfer by both active and passive methods. A.A Nicol et.al.[1] has studied the effect of rotational speed of condensing tube on condensation heat transfer coefficient and found out that as the rotational speed increases the condensation heat transfer coefficient also increases. An increase of 360 % in condensation heat transfer coefficient was reported at a rotational speed of 2600 rpm. Experiments were conducted by J.C. Dent et.al. [2] to study the effect of vibration on condensation heat transfer coefficient on copper tube. The vibration was varied from 0 Hz to 140 Hz with a maximum amplitude of 1.5 mm. The maximum enhancement in heat transfer coefficient reported was only 20%. Akhavan-Behabadi et.al. [3] conducted studies to find the effect of twisted tape inserts on condensation of HFC 134a. He reported that augmentation in heat transfer coefficients of 40% was possible at the cost of pressure drop of 240% by introducing twisted tapes. The effect of corrugation and its pitch on condensation heat transfer performance was investigated by Suriyan Laohalertdecha et.al.[4]. They reported that it is possible to improve the condensation heat transfer coefficient by 50% with an increase in frictional loss of 70%. The above studies reveals that both active methods like pulsating steam flow, vibrating condensation surface, rotating condensation surface and passive methods which are comparatively less complicated, low cost and more rugged like insertion of twisted tapes, providing corrugated surfaces can enhance condensation heat transfer but at the cost of significant increase in frictional losses.

An alternative to this which will not incur huge additional pumping cost is to make changes in surface

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morphology at nanometer or micrometer level or a combination of these. Clustered copper hydroxide ribbed-nanoneedles on copper surface were created by Jie Zhu et.al.[5]. He showed such surface can improve condensation heat transfer coefficient by 125%. Most of such nano/ micro level modifications are done user laser beam machining, lithography etc which is usually very costly and difficult to sustain for longer duration of operation

A much more durable and low cost alternative is to introduce millimeter level features like grooves, dimples and fins on the condensation surface which can enhance condensation heat transfer. A. Briggs [6] studied the effect of pin fins on condensation heat transfer. He reported an increase in heat transfer coefficient for R113 in the range 3.6 to 9.9 and 2.4 to 2.9 for steam. Masaaki Izumi et.al [7] studied the effect of round grooves in the enhancement of heat transfer during dropwise condensation. They found that a groove of width 2-3 mm showed a maximum enhancement in heat transfer performance of 80%. Kumar, R et.al [8] compared the performance of circular integral fins, spined integral fins, partially spined circular integral fins with bare tubes. They found partially spined circular integral fins showed better performance with an enhancement of condensation heat transfer coefficient by 20%. The profile of the external features like grooves will lead to changes in the effective condensation area i.e. the area that comes in contact with steam, retentively of condensate on the surface, rate of removal of condensate, turbulence intensity level of the condensate film, condensate film thickness etc. All these factors have a huge impact on the condensation heat transfer rate.

Most of the research that has been done in this area has the condenser tubes oriented horizontally as it has higher heat transfer capability compared to vertically oriented tubes due to lower condensate film thickness. Studies regarding condensation on vertical tubes are very scarce eventhough it has application in areas like in reflux condensers used for distillation. The present work attempts to improve the heat transfer performance of the condenser tubes oriented vertically by providing helical grooves on copper tube which is facile as well as durable.

3 Experimental setup

The schematic of the experimental set up is shown in figure1. The condensation chamber was made of vertical heat resistant borosilicate glass having an outer diameter of 150 mm, thickness 3 mm and height 240 mm. Both the top and bottom of the condensation chamber was closed using 6 mm thick Aluminum plates. The copper tube test specimen was fixed concentrically inside the condensation chamber. Copper tubes used as specimen had a total length 300 mm, effective condensing length 200 mm, outer diameter 19.05 mm and wall thickness 2 mm

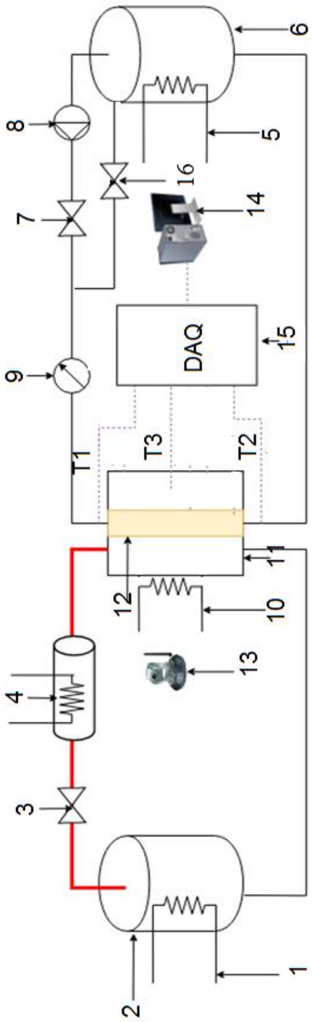
Cooling water was supplied to the copper tube from a 85 L Stainless steel (SS) tank using gear pump (Petec Enviro Engineers, India, 5 LPM, 0.3 MPa) through 19 mm SS pipelines. An industrial grade 3000 W immersion heater along with thermostat was used to maintain the temperature of water in the tank and hence the water entering the specimen. The 19 mm SS hot water supply lines to the test specimen was wrapped with asbestos ropes to prevent heat loss. Two ball valves one on the bypass line and one on the main hot water supply line to specimen were operated simultaneously to maintain a flow rate of 2.76LPM. A variable area flow-meter (Chemcon, India, 1-10LPM, with an accuracy ±1.5%) was provided on the hot water supply line to measure the flow rate. Valves were provided at the exit to the pipe line connected to the test specimen to run the experiment in recirculation mode or to expel the water to drain.

The steam at about 100^0C was admitted through the bottom Al plate of the test chamber. For generating the required steam, a 35 L SS tank with industrial grade 3000W immersion heater along with thermostat was used. The generated steam was transferred to test chamber with minimal heat loss through 19 mm SS tubes wrapped with asbestos ropes. An auxiliary heater of 1000 W was wrapped around the steam line to ensure dry steam in the test chamber. The bottom of the condensation chamber was provided with a valve to drain and collect the condensate.

To study the dynamics of condensate flow on the test specimen a high speed camera (Kron Technologies Canada) was used. As water vapour may condense on the inside of the condensation glass chamber hindering the visualisation process hot air was blown on the outside surface of the condensation test chamber with the help of a heat gun(DeWalt, USA) to ensure visibility as shown in figure2.

A vacuum gauge (manometer, India) and a T type calibrated thermocouple of accuracy ±0.5K was provided to measure the pressure and temperature of the steam in the condensing chamber. One T type calibrated thermocouple of accuracy ±0.5K each was provided at the inlet and outlet of the water flowing thought the test specimen for measuring the inlet and outlet coolant temperature. The coolant water inlet temperature was varied to vary the degree of subcooling and the experiment was conducted for various pitch to develop a correlation for dependence of pitch on condensation heat transfer coefficient

The test specimens are prepared by incorporating millimetre level features on the surface of copper tube of outer diameter 19.1 mm, 2 mm thickness and length 300 mm. In the reference specimen no surface modification was carried out. Helical grooves of various pitch of depth 0.4 mm were machined onto the copper tube using a CNC milling machine (ECOND CNC 26, HMT, India). An alkaline solution of 100 g NaOH,40g Na_2CO_3 and 2000 ml Deionised water (DI) was prepared and the test specimen was fully immersed in the solution and ultrasonicated for 1000 seconds to remove any surface deposits.



1,5 Heater, 2 Steam generator, 3,7,16 Valves, 4 Auxiliary heater , 6 Water tank, 8 Pump, 9 Flowmeter, 10 Heat gun, 11Condensation chamber, 12 Test Specimen , 13 Camera, 14 Computer, 15 Data Acquisition system

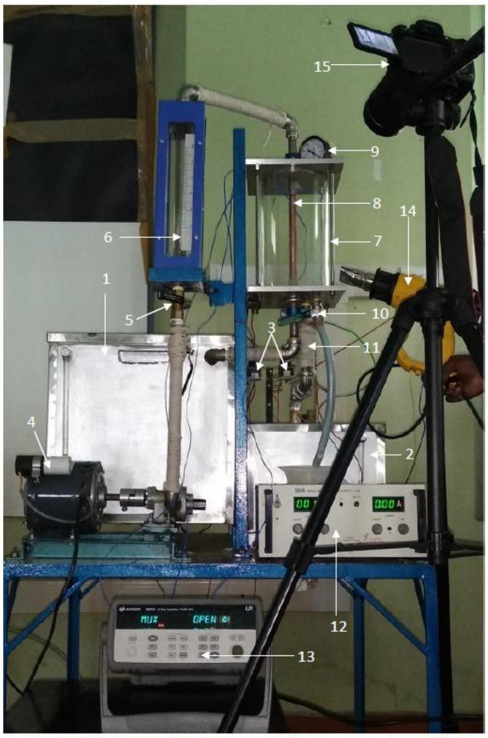
Fig.1. Schematic of experimental setup

4 Data reduction

As the steam condenses on the copper specimen the latent heat of condensation is transferred to the cooling water flowing inside the tube. So the rate at which heat is liberated during the condensation process was calculated using equation 1.

$$Q = mc_p(T_2\text{-}T_1) \tag{1}$$

The overall heat transfer coefficient U was calculated based on equation 2.For all the specimens overall heat transfer coefficient was calculated based on the outside area of bare copper tube A_0, which was found out using solidworks[9]



1- Water tank with immersion heater; 2- Steam generator with immersion heater; 3- Thermostats (Capillary type); 4- Gear Pump; 5- Ball Valve; 6- Rotameter; 7- Condensation chamber with thermocouple; 8- Test section with thermocouple; 9- Pressure Gauge; 10- Drain Valve; 11- Auxiliary Heater; 12- DC Power Supply; 13- Data Acquisition System; 14- Heat Gun; 15- High Resolution Camera

Fig. 2. Photograph of the experimental setup

$$U = \frac{Q}{A_0 \Delta T_{LMTD}} \tag{2}$$

Where $\Delta T_{LMTD} = \dfrac{(T_3\text{-}T_1)\text{-}(T_3\text{-}T_2)}{\ln((T_3\text{-}T_1)/(T_3\text{-}T_2))}$

The total resistance R_{tot} was obtained from the overall heat transfer coefficient and outside surface area of bare tube using equation 3

$$R_{tot}=1/A_0U \tag{3}$$

The various components of total thermal resistance, R_{tot}, are R_{cond}, condensation resistance, R_{fo}, outside fouling resistance, R_{wall}, resistance of the copper wall, R_{fi}, inside fouling resistance, $R_{c.conv}$, convection resistance and their relation is shown in equation 4.

$$R_{tot}= R_{tcond}+ R_{fo} +R_w + R_{fi} + R_{c.conv} \tag{4}$$

As the scales are removed by ultrsonification of the specimens in alkaline bath which was detailed in section experimental setup, the inside and outside fouling resistances was neglected and equation 4 is rearranged as equation 5.

$$R_{cond}= R_{tot} – R_w – R_{c.conv} \quad (5)$$

Sieder-Tate correlation [10] was used for the calculation of the coolant Nusselt number (Equation.6)

The constants C and m in the above correlation was found out experimentally using modified Wilson plot which has been detailed below. From the above found value of Nusselts number, convective heat transfer coefficient was found using on Equation.7, which was used to find the convective resistance based on equation 8.

$$Nu_c= C\, Re_c^{\,m} Pr^{0.4} \quad (6)$$

$$h_c = Nu_c k_c/D_i \quad (7)$$

$$R_{c.conv}= 1/h_c A_i \quad (8)$$

The wall resistance was found out using equation 9

$$R_w=ln(D_0/D_i)/2\pi k_{Cu}) \quad (9)$$

As the total resistance and all other thermal resistances except condensation resistance was found as per the procedure stated above condensation resistance was found out using Equation.5.Once condensation resistance, (R_{cond}), was found out, the condensation heat transfer coefficient was calculated based on Equation.6.

$$h_{cond}= 1/A_0 R_{cond} \quad (10)$$

The detailed account of the above procedure has been outlined by Alwazzanet al. [11].

4.1 Determination of constants C and m

Ferdeznán-Seara et.al.[12] conducted a series of experiments to find the values of constants C and m for smooth pipes used in equation 6. Using modified Wilson plot they found the values of C and m as 0.0213 and 0.8067 respectively. The geometry of the specimen used for the present work is different hence the above values cannot be used directly. Modified Wilson plot done on the present specimen was made to find the constants C and m.

The iterative procedure used for finding C and m is summarized in the following steps

1) Assume a value for m preferably around 0.8
2) Make a plot of R_{tot} vs Re^{-m}.
3) Find out the intercept,I and slope S of the plot
4) Make a plot of Ln(R_{tot}-I) vs Ln Re.
5) Find out the slope m^* of the plot
6) If m^* = m then stop else update the value of m = $m^*$and go to step 2

Once the above iterative procedure converges, constant C was found out from the slope of R_{tot} vs $Re^{-0.853}$ and the slope Ln(R_{tot}-I) vs Ln(Re) was used for finding out m . The plot of R_{tot} vs $Re^{-0.853}$ and Ln(R_{tot} -I) vs Ln(Re) used for finding the constants for bare tube tubes is depicted in figure 3 and figure 4 respectively.

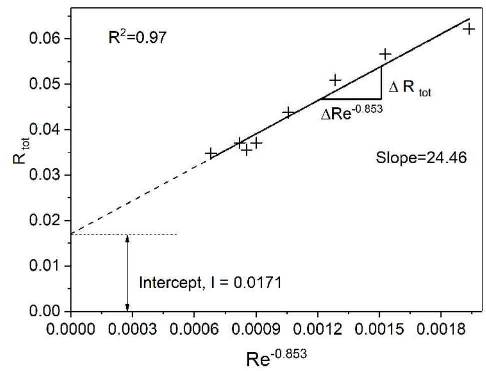


Fig. 3. Variation of total thermal resistance with Reynolds number for bare copper specimen

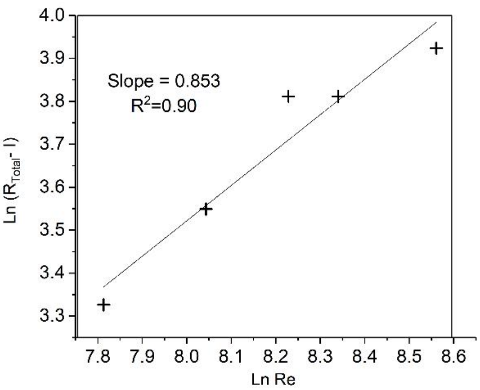


Fig. 4. Variation of condensation thermal resistance with Reynolds number for bare copper specimen

Due to space constrains the detailed procedure of the modified Wilson plot outlined by Ferdeznán-Seara et.al. [13] which is an iterative procedure is not presented in this paper. For all the tubes the value of C and m in Equation 6 was approximately 0.054 and 0.853 respectively.

5 Uncertainty analysis

Kline and McKlintock [14] method was used to determine the uncertainty in condensation heat transfer coefficient from the measured parameters. Evaluation of uncertainties of the independent parameters was done with confidence interval of 68.27% [15]. At low degree of sub cooling the difference in coolant inlet and outlet temperature is very small and is of the order of the uncertainty of the thermocouple. Therefore the uncertainty associated with determination heat transfered using equation 1 and hence uncertainty in finding out overall and condensation heat transfer coefficient is significantly high. However at high degree of sub-cooling the temperature rise of the coolant is significant and hence uncertainty in determination of condensation heat transfer coefficient is less..This has been clearly explained in our earlier work [16]. As the uncertainty bar will hinder the readability of the plot, it has not been included in figure 5. The uncertainties associated with the measured quantities and dependant parameters is summarized in Table 1

Table 1. Uncertainty of determination of condensation heat transfer coefficient

Parameter	D (mm)	L (mm)	V (l/min)	T (°C)	A_0 (%)	Degree of sub-cooling	h_{cond} (%)
Uncertainty	0.02	0.02	0.01	0.5	0.11	ΔT= 38.86°C	5.02
						ΔT= 7.09°C	80.27

6 Results and discussion

The condensation heat transfer coefficient for grooves of various pitches for various degrees of sub cooling ranging from 7^0C to 40^0C is depicted in figure 5. Figure 5 shows that specimen with pitch 8.47 mm outperforms all other tubes for the entire range of subcooling. Also it was seen that specimens with groove of pitch ≤ 6.35 mm had adverse effect on condensation heat transfer coefficient.

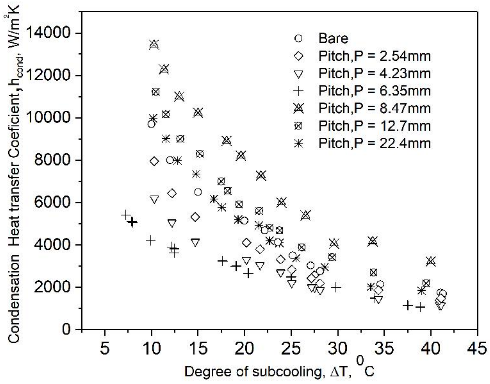


Fig. 5. Condensation heat transfer coefficient for various degrees of subcooling

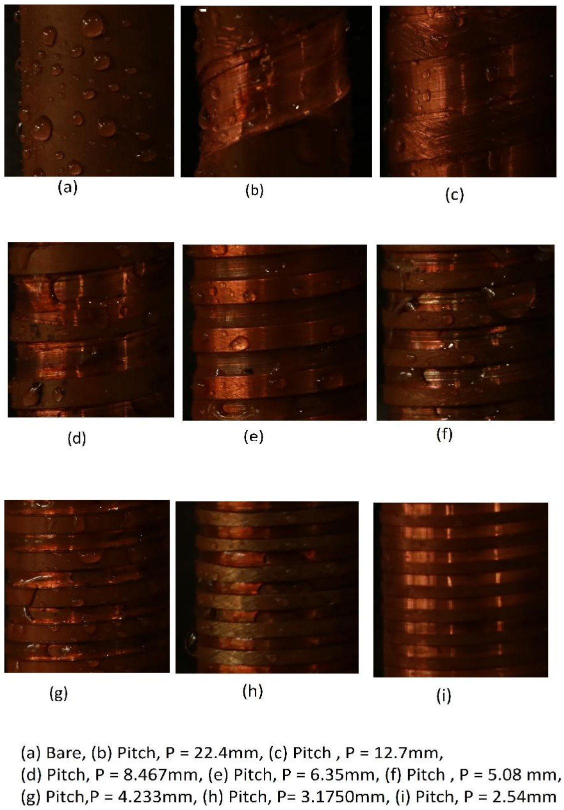


Fig. 6. Photograph of retention of condensate by various grooves

The drastic drop in performance of grooved tubes with pitch ≤ 6.35 mm over grooved tube with pitch 8.47 mm and more is due to the difference in the condensate retention capability of grooved tubes. It is seen from figure 6 (e-i) that grooves of tubes with pitch ≤ 6.35mm were flooded with condensate at all degrees of subcooling . This condensate retained in the grooves reduced the effective condensing area in contact with steam thereby reducing the condensation heat transfer coefficient.

In order to quantify retention of condensate due to variation in pitch, fine droplets of water was sprayed onto the specimen, which can simulate the condensate retention during condensation process as outlined by Ali et.al. [17] .The amount of water retained was found out by measuring the difference in weight of the specimen before and after spraying and the results are tabulated in table 2.

As no flooding was observed ,figure 6 (b-d) ,on grooved tube with pitch ≥ 8.47 mm there was no considerable decrease in effective condensation area and hence they showed better condensation heat transfer performance than bare copper.

The decreasing trend in condensation heat transfer coefficient for grooved tube with pitch ≥ 8.47 mm as pitch increases can be explained based on the increase in surface area and length of helix of grooved specimen.

Table 2. Condensate retained on the specimen

Groove Pitch (mm)	Quantity of condensate retained (mg)
Bare	856
2.54	1013
3.18	1331
4.23	1534
5.08	1694
6.35	2697
8.47	913
12.7	898
22.4	879

Table 3 depicts the increase in surface area and length of helix of grooved specimen. The surface area as well as the length of helix of grooved copper tube was found out by modeling in CADD software Solidworks [11].It is seen that as the pitch increases the surface area available for condensation of steam decreases. In all cases the overall heat transfer coefficient was calculated using equation 2 where the surface area was kept as constant (area of bare copper tube, A_0). Hence any increase in area will be reflected as increase in overall as well as condensation heat transfer coefficient.

Table3. Enhancement in surface area and length of helix of copper tube of various pitches

Pitch (mm)	Outside surface Area,A_0 (mm^2)	Increase in outer surface area, (%)	Length of helix where condensate was acummlated (mm)	Angle of Helix (in degree)
Bare	11963.4	-	-	
2.54	15488.13	29.46	9753.41	2.46
3.18	14666.43	22.59	7638.36	3.05
4.23	13824.96	15.56	5490.54	4.06
5.08	13522.04	13.03	4716.84	4.86
6.35	13152.16	9.94	3781.10	6.05
8.47	12945.8	8.21	3251.12	8.08
12.7	12476.77	4.29	2109.01	12.04
22.4	12228.27	2.21	1123.12	20.55

The surface tension of condensate retained on the grooves also have significant effect on condensation heat transfer coefficient. When the groove was not fully flooded, the condensate in the groove will draw the water on the flank as well as the tip of the extended surface as outlined by Mori et.al.[18]. It is seen from table 3 that as the pitch decreases the length of helix along which condensate was retained on the grooved tube increases and hence the amount of condensate drawn towards the groove increase. Due to this effect, for grooved tubes with smaller pitches, fresh surface was exposed more rapidly for condensation and hence showed higher condensation heat transfer coefficient. In case of flooded grooves also the surface tension of condensate in the groove will draw water from the tip of the condensate surface hence flooded grooved tubes with lower pitch will show better performance compared to flooded grooves with higher pitch

The angle of helix was calculated using solidworks [11]. It did not have a major impact on condensation heat transfer coefficient as it was not large enough to make the condensate drain through the grooves. However for groove with pitch > 12.7 mm, it was seen that the groove acted as a patch for easy removal of condensate contributing to a small increase in condensation heat transfer coefficient.

7 Conclusions

The presence of rectangular grooves on the condensation surface of the specimen can improve or deteriorate the heat transfer performance. The flooding characteristics of the groove is the major factor that will determine whether the groove will increase or decrease heat transfer coefficient. By providing grooves of pitch 8.45 mm a maximum enhancement in condensation heat transfer coefficient of 68.4% over bare tube at $\Delta T = 40^0C$ was obtained. While providing rectangular grooves of pitch 6.35 mm resulted in a drastic drop of heat transfer coefficient of 39.01% at $\Delta T = 40^0C$ due to flooding of grooves.

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