



## PERFORMANCE ANALYSIS OF DROPWISE CONDENSATION

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**Abstract**—Drop wise condensation (DWC) is under development as it offers extremely high heat transfer (5-7 times higher) by preventing the buildup of the condensate liquid layer formed in the film wise condensation (FWC). This project work is an experimental based and objective is to break the condensate formed on the cold surface so as to improve the heat transfer rate. Hence an analysis has been done over circular, elliptical and rhombus shaped tubes for the performance DWC and rhombus shaped tube is suggested, as its shape will not permit the buildup of the layer of the condensate.

Considerable attention has been worldwide to promote drop wise condensation on circular tubes using promoter like gold, silver, plasma, organic material etc., In the present work vapour deposition is used for coating the copper tubes is proposed to use Ni-Cr and TEFLON etc., to promote the drop wise condensation. The characteristics of drop wise condensation on various shapes of tube with three different angle tubes were experimentally studied and their results are compared and new Correlations will be derived for DWC for each shape of the be tube with various angles.

**Keywords**—Dropwise condensation, Flimwise condensation, vapour deposition, Ni-Cr, Teflon.

### I. INTRODUCTION

Condensation is a vital process in power generation industries and its result been an area of research for over hundred years. Over this period understanding of condensation process has gradually improved. Theories and models have become more accurate and are known applicable to a wider range of condensation. Dropwise condensation (DWC), when properly promoted, has been known to produce heat transfer coefficients up to 20 times that of filmwise condensation (FWC). Over the past few decades, considerable attention has been paid toward the development of suitable DWC promoters. Gold and silver have been known to consistently show excellent dropwise characteristics (1–7).

Woodruff, D. W and Westwater, J. W [1] have proposed that promotion of DWC on gold-plated surfaces was directly related to the carbon-to-gold ratio on the surface. [2] Studied dropwise condensation of steam on electroplated gold surfaces, and they found that a minimum thickness of approximately 0.2  $\mu\text{m}$  of gold was required to obtain perfect dropwise conditions, [3] the correlation between gold thickness, carbon amount and condensation heat transfer mode appears strong. The present status of the subject is that the purer and the brighter the gold deposit, the better it serves for dropwise condensation. David G. Wilkins et.al [4] pure gold was found to give film-type condensation in the absence of organic contamination. Dropwise condensation was achieved by adding a small amount of promoter. Lauryl mercaptan, n-octadecyl mercaptan, and tetrakis octadecylthiosilane were excellent promoters for dropwise condensation on gold although their life was shorter than on copper. Erb R and Thelen E [5, 6] conducted an extensive investigation of several permanent hydrophobic coatings, including PTFE, sulfide films, parylene-N, and noble metals. However, they concluded that the organic coatings were not as good as a silver coated surface.

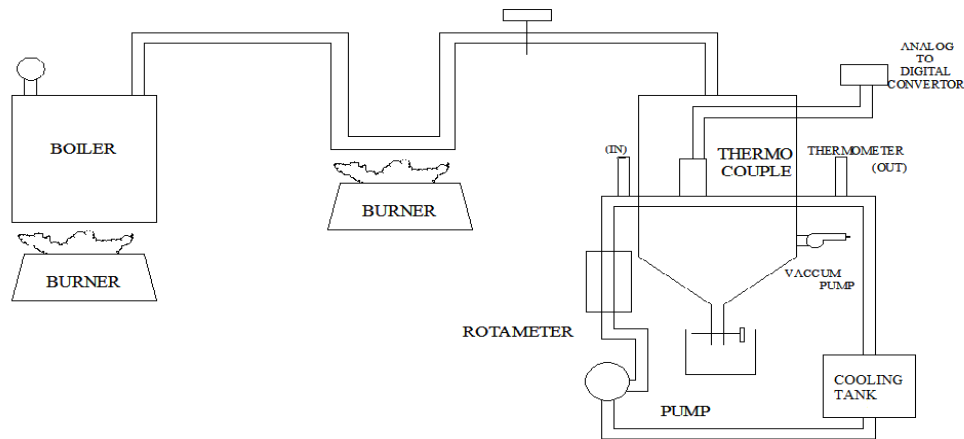
G. A. O'Neill and J. W. Westwater [7] showed very similar results with silver-electroplated copper surfaces they concluded that the lifetime of the silver as a promoter of dropwise condensation depends upon the plating thickness and composition, as well as upon the base metal preparation. Marto P et.al [8] has proposed that The organic coatings were successful in promoting good quality dropwise condensation for prolonged periods of times (> 12,000 h). Dropwise heat transfer coefficients as large as six times the film condensation value were obtained with these coatings and results were not dependent upon the thermal conductivity of the wall. Holden et.al.[9] conducted experiments on 14 polymer-coated surfaces and an electroplated-silver surface, and showed an increase in condensation heat transfer coefficient of about three to eight relative to filmwise at an operating pressure of 11 kPa, with electroplated silver exhibiting the best performance.

A.K.Das et.al[10] have proposed that the SAM coating increased the condensation heat transfer coefficient by factors of 4 for gold-coated aluminum, and by about 5 for copper and copper-nickel tubes, under vacuum operation (10 kPa). The respective enhancements under atmospheric conditions were about 9 and 14. Comparatively, the heat transfer coefficient obtained with a bare gold surface (with no organic coating) was 2.5 times that of the filmwise condensation heat transfer coefficient under vacuum, and 3.4 at atmospheric conditions.

During this performance analysis study, copper condenser tubes to study their condensation characteristics. The copper tube could be directly coated with Ni-Cr and TEFLON etc., and eliminated the expensive gold and silver. The steps were exactly the experiments were conducted in a horizontal single-tube condensation apparatus. A brief description of the experimental program and the apparatus follows. The heat transfer results were in good agreement with existing theory and with data from other studies.

## II. EXPERIMENTAL SETUP

The experimental setup has two major units and they are 1.Steam Generation Unit.  
 2. Condensing Unit.



**Figure. 2.1 Experimental Setup**

## III. CALCULATION PROCEDURE (FORMULAE USED)

### Theoretical Heat Transfer Rate:

$$h = Q/A \cdot (T_{\text{sat}} - T_{\text{sur}}) \text{ (kW/m}^2\text{K)}$$

### Theoretical Heat Transfer co-efficient:

$$Q = M_s \cdot X \cdot h_{\text{fg}} \text{ (kW)}$$

### Actual Heat Transfer Rate:

$$Q_{\text{actual}} = h_{\text{actual}} \cdot A \cdot (T_{\text{sat}} - T_{\text{sur}}) \text{ (kW)}$$

### Overall Heat Transfer co-efficient:

$$U = Q_{\text{actual}} / A \cdot (\text{LMTD}) \text{ (kW/m}^2\text{K)}$$

$$LMTD = (T_{sat} - T_{wi}) - (T_{sat} - T_{wo}) / \ln [(T_{sat} - T_{wi}) / (T_{sat} - T_{wo})] \quad (^{\circ}C)$$

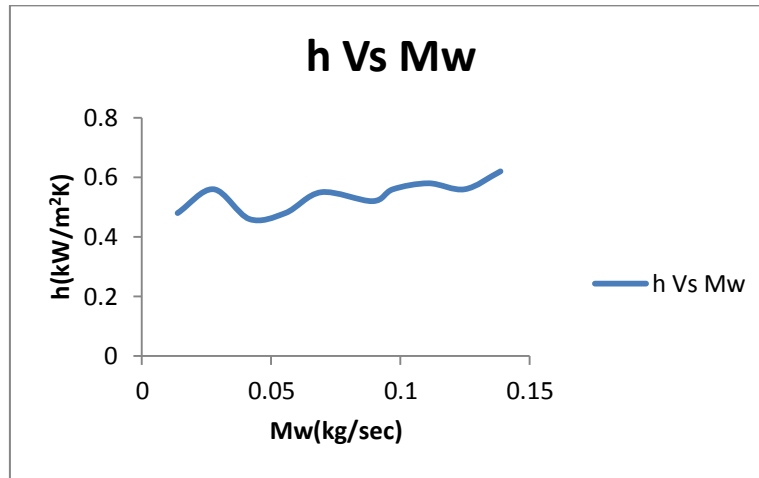


Figure. 3.1 Heat transfer coefficient (h) Vs Mass flow rate of Water (Mw)

The graphs are drawn between heat transfer co-efficient h (Kw/m<sup>2</sup>k) and mass flow rate of cooling water M<sub>w</sub> (Kg/sec) by taking M<sub>w</sub> on X-axis and h on Y-axis.

#### Relations For Calculating h<sub>actual</sub> (Circular Dwc)

- Linear →  $h = 0.9043M_w + 0.468$
- Logarithmic →  $h = 0.0451\ln(M_w) + 0.6617$
- Polynomial →  $h = 7.9833M_w^2 - 0.3101M_w + 0.5016$
- Power →  $h = 0.6761M_w^{0.0847}$
- Exponential →  $h = 0.4701e^{1.6906M_w}$

Model calculation of h<sub>actual</sub> for 1<sup>st</sup> reading is shown below by taking the linear relation

$$h = (0.9043 \times 0.0138) + 0.468$$

$$= 0.480479 \text{ (kW/m}^2\text{K)}$$

$$\text{Error percentage} = [(h_{\text{theoretical}} - h_{\text{actual}}) / h_{\text{theoretical}}] \times 100$$

$$\text{Error percentage} = (0.480 - 0.468533) / 0.0138 \times 100 = 2.389002\%$$

Similarly error percentages for all the relations are calculated.

#### IV. Result and discussion

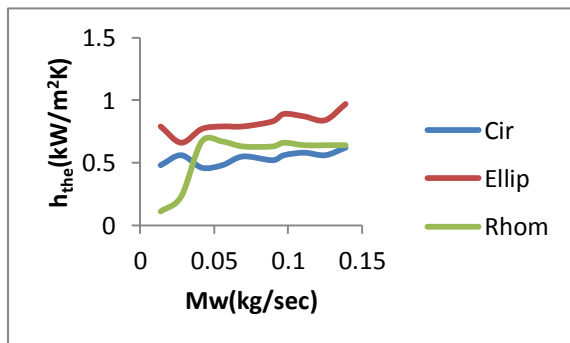


Figure. 4.1 Comparison of hthe Vs Mw

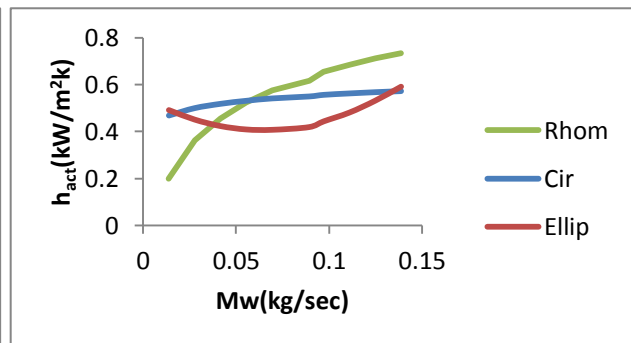


Figure. 4.2 Comparison of hact Vs Mw

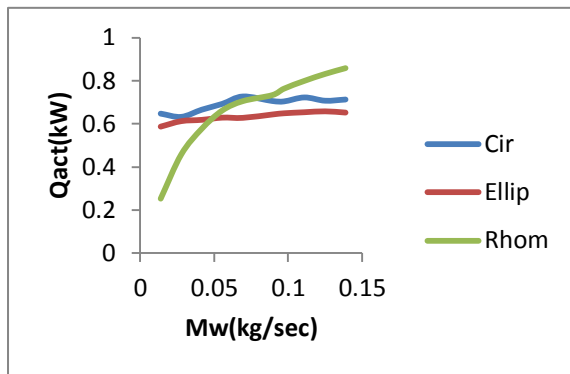


Figure. 4.3 Comparison of Qact Vs Mw

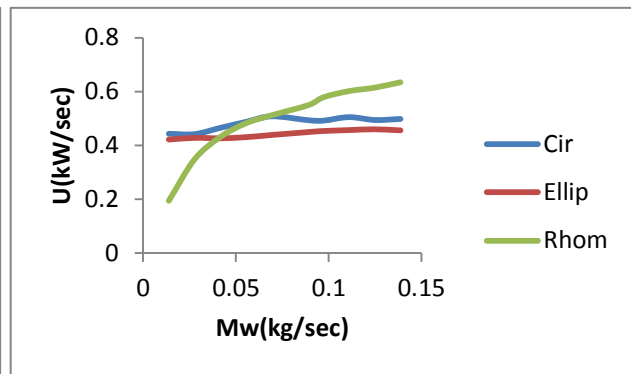


Figure. 4.4 Comparison of U Vs Mw

**Figure 4.1:** The Rhombus shaped tube is found to have higher heat transfer co-efficient when compared with Elliptical and circular shaped tubes. However Circular shaped tube has higher heat transfer co-efficient than Elliptical shaped tube.

**Figure 4.2:** The actual heat transfer co-efficient is found to be nearly same For Circular and Elliptical shaped tubes and found to increasing for Rhombus shaped tube. However Rhombus tube is found to have higher heat transfer co-efficient when compared to other shapes.

**Figure 4.3:** The actual heat transfer rate is found to be increasing in all the three shapes. In that the Rhombus shaped tube is found to have higher heat transfer rate than Elliptical tube. But Circular shaped tube is having slightly lesser heat transfer rate at the final stage than Elliptical shaped tube.

**Figure 4.4:** The graph clearly shows the Rhombus shaped tube is found to have higher overall heat transfer co-efficient than Circular and Elliptical shaped tubes. Similarly calculation has done for various inclination (0°, 15°, 30°, 45) using Teflon coating and (Ni-Cr) Coatings.

## V. CORRELATION FOR TEFLON AND NI-CR COATING

| S.NO | SHAPE      | DEG | TEFLON                                                      | Ni-CR                                                     |
|------|------------|-----|-------------------------------------------------------------|-----------------------------------------------------------|
| 1    | CIRCULAR   | 0   | Logarithmic $\rightarrow h = 0.0451 \ln(Mw) + 0.6617$       | Polynomial $\rightarrow h = 21.54Mw^2 - 0.805Mw + 1.035$  |
| 2    | CIRCULAR   | 15  | Polynomial $\rightarrow h = 31.102Mw^2 + 2.5181Mw + 1.1044$ | Exponential $\rightarrow h = 1.060e^{1.998Mw}$            |
| 3    | CIRCULAR   | 30  | Polynomial $\rightarrow h = -70.83Mw^2 + 13.90Mw + 0.64$    | Linear $\rightarrow h = 2.028Mw + 1.028$                  |
| 4    | CIRCULAR   | 45  | Exponential $\rightarrow h = 0.701e^{1.989Mw}$              | Exponential $\rightarrow h = 1.088e^{-0.30Mw}$            |
| 5    | ELLIPTICAL | 0   | Polynomial $\rightarrow h = 33.75Mw^2 - 4.321Mw + 0.544$    | Power $\rightarrow h = 0.529Mw^{0.049}$                   |
| 6    | ELLIPTICAL | 15  | Logarithmic $\rightarrow h = 0.019 \ln(Mw) + 0.611$         | Polynomial $\rightarrow h = 27.92Mw^2 - 3.041Mw + 0.879$  |
| 7    | ELLIPTICAL | 30  | Logarithmic $\rightarrow h = 0.011 \ln(Mw) + 0.791$         | Polynomial $\rightarrow h = -0.289Mw^2 + 0.763Mw + 0.654$ |
| 8    | ELLIPTICAL | 45  | Polynomial $\rightarrow h = 47.67Mw^2 - 6.451Mw + 1.237$    | Linear $\rightarrow h = 0.014Mw + 0.950$                  |
| 9    | RHOMBUS    | 0   | Logarithmic $\rightarrow h = 0.232 \ln(Mw) + 1.195$         | Polynomial $\rightarrow h = 6.802Mw^2 + 0.456Mw + 0.966$  |
| 10   | RHOMBUS    | 15  | Logarithmic $\rightarrow h = 0.118 \ln(Mw) + 1.276$         | Logarithmic $\rightarrow h = 0.031 \ln(Mw) + 0.866$       |
| 11   | RHOMBUS    | 30  | Polynomial $\rightarrow -4.750Mw^2 + 1.970Mw + 0.536$       | Linear $\rightarrow h = 0.634Mw + 0.679$                  |
| 12   | RHOMBUS    | 45  | Power $\rightarrow h = 1.387Mw^{0.108}$                     | Logarithmic $\rightarrow h = 0.0902 \ln(Mw) + 1.0718$     |

Table.5.1 correlation for Teflon and ni-cr coating

## VI. CONCLUSION

From the above graphs it has been proved that the Overall heat transfer co-efficient of Rhombus shaped tube is comparatively higher than Circular and Elliptical shaped tubes for Teflon Coating. Hence the Rhombus shaped tube is suggested for better DWC from the above observations for Teflon Coating.

Similarly from the above graphs it has been proved that the Overall heat transfer co-efficient of Elliptical and Circular shaped tubes are almost same and comparatively higher than Rhombus shaped tubes for Ni-Cr Coating. Hence the Elliptical and Circular shaped tubes are suggested for better DWC from the above observations for Ni-Cr Coating.

TEFLON and Ni-Cr has negligible heat transfer resistance and poses no contamination threat to the system. However the Durability of the TEFLON and Ni-Cr is to be determined before it can be commercialized.

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## NOMENCLATURE

|                                                                             |                                                                    |
|-----------------------------------------------------------------------------|--------------------------------------------------------------------|
| $h$ = heat transfer coefficient( $\text{kW/m}^2\text{K}$ )                  | $U$ = Overall heat transfer coefficient( $\text{kW/m}^2\text{K}$ ) |
| $Q$ = Heat Transfer rate(watts)                                             | $h_{fg}$ = Latent Heat of Evaporation( $\text{kJ/kg}$ )            |
| LMTD=Log Mean Temperature Difference( $^{\circ}\text{C}$ )                  | $T_{\text{sur}}$ =Surface Temperature( $^{\circ}\text{C}$ )        |
| $T_{\text{wi}}$ =Coolant Inlet temperature( $^{\circ}\text{C}$ )            | $T_{\text{wo}}$ =Coolant out let Temperature( $^{\circ}\text{C}$ ) |
| $T_{\text{sat}}$ =Saturation temperature of the steam( $^{\circ}\text{C}$ ) | $M_w$ =mass flow rate of coolant( $\text{kg/sec}$ )                |
| $M_s$ =mass of the condensate( $\text{kg/sec}$ )                            | $A$ = Surface area of the condenser tube( $\text{m}^2$ )           |
| $D$ = Outer diameter of the Condenser tube( $\text{m}$ )                    | $L$ = Length of the condenser tube( $\text{m}$ )                   |