Measurement of Steam Film Condensation Coefficient in Laminar and Turbulent Flow

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Abstract:

Steam and other vapour condensation processes occur widely in power stations, refineries, and in most chemical plants. Hence the importance of arriving at the actual values of the heat transfer coefficient in such equipment.

Experimental work was carried out for the determination of film-wise condensation coefficients for steam in laminar and turbulent flow on tubes of various geometries similar to those found in actual heat exchangers.

The results were correlated and comparison with the theoretical Nusselt type expressions was conducted. The result of comparison shows that the heat transfer coefficients are larger than those theoretically predicted. The reason for this is believed to be due to the effect of high vapour velocity. Local and average values of the coefficients are also discussed.

The suggested correlation is:

$$h_o = 1.89\left(\frac{K^3 \rho^2 g \lambda}{\mu \chi \Delta T_f}\right)^{0.25}$$

Keywords:

Steam condensation, heat transfer coefficient, laminar and turbulent flow.

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قياس معامل تكثف بخار الماء الغشائي عند الجريان الطباقي والاضطرابي

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الخلاصة:

تحدث عمليات تكثف بخار الماء والابخرة الأخرى في محطات توليد الطاقة وفي المصافي النفطية وأغلبية المصانع الكيمياوية، وعليه تتولد الحاجة الى الوصول الى القيم الحقيقية لمعامل أنتقال الحرارة في هذه المعدات .

تم القيام بتجارب عملية لقياس معاملات التكثف الغشائي لبخار الماء في حالة الجريان الطباقي والاضطرابي على أنابيب ذات أبعاد مختلفة مشابهة لتلك في المبادلات الحرارية الحقيقية، وقد تم أيجاد علاقة للنقاط التجريبية وأجريت مقارنة مع التعابير النظرية التي تعطيها معادلات نسلت، وكانت نتائج المقارنة أن معاملات أنتقال الحرارة أكبر من تلك المحسوبة نظرياً، ويعتقد السبب هو سرعة البخار العالية، كما تم مناقشة قيم معامل أنتقال الحرارة الموقعية وتلك التي تمثل المتوسط.

والمعادلة المقترحة هي:

$$h_o = 1.89\left(\frac{K^3 \rho^2 g \lambda}{\mu \chi \Delta T_f}\right)^{0.25}$$

الكلمات الدالة

تكثف بخار الماء، معامل أنتقال الحرارة، الجريان الطباقي والاضطرابي.

Nomenclature:

A	Heat transfer area	m^2
Cp	Specific heat	J/kg K
D	Diameter	mm
G	Acceleration due to gravity	m/s^2
h	Heat transfer coefficient	W/m^2 K
K	Thermal conductivity	W/m k
q	Heat transfer rate	W
R	Universal gas constant	J/g mole K
Re	Reynolds number	-
T	Temperature	K
U	Overall heat transfer coefficient	W/m^2 K
V	Velocity	m/s
χ	Distance along condensing surface	M

Subscripts

o	outside
i	inside
b	bulk
f	film

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G	gas or vapour	
w	wall	

Greeks

Δ	Difference	
ρ	Density	Kg/m^3
μ	Viscosity	Kg/ms
λ	Latent heat	J/kg
ν	Specific volume	m^3/kg
Γ	Mass flow rate per unit periphery	Kg/ms

1. Introduction:

Condensation mode offer high rates of heat transfer. This makes the equipment smaller in size than single-phase equipment. Hence, it is important to arrive at the values of the heat transfer coefficient.

Mathematically, Nusselt's analysis [(6)] of film condensation on vertical surfaces (tubes or plates) gives the condensation coefficient as:

$$h_o = 0.943\left(\frac{K^3\,\rho^2\,g\,\lambda}{\mu\,\chi\,\Delta T_f}\right)^{0.25} \qquad \ldots (1)$$

With the physical properties evaluated at the mean temperature of the condensation film.

The equation is arrived at using many assumptions, the most serious ones are:

i- The film is in laminar flow, with the viscous force balanced by the gravitational force.

ii- Film surface is smooth i.e. no ripples or waves.

iii- The temperature of the film surface is that of the saturated vapour.

iv- Zero shear on the surface in contact with the vapour. i.e. the vapour stagnant.

In actual heat equipment, the steam moves at high velocities, hence sweeping the condensate off the surface and causing waves and ripples at the condensate surface. In addition, there exists an interfacial resistance between the steam and condensate film making the temperature of the steam different from that of the film surface.

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In the present work, an experimental program was conducted to measure the actual steam condensation coefficients in situations similar to those occurring in practical heat equipment, where the vapour is not stagnant and compare the result with those predicted by analytical relations such as the Nusselt equation.

2. Theory:

The steam condensation coefficient is given by the basic heat equation:

$$q = h_O \, A \, \Delta T_f$$

where :

$$\Delta T_f = (T_G - T_W) \qquad \ldots (2)$$

There exists a practical difficulty in measuring metal surface temperature, hence one method of getting around this difficulty is to measure the overall coefficient between the condensing steam and cooling water and subtracting the resistance of the separating metal wall which, is fairly constant, and the resistance on the cooling water side which is given by well established and accurate single phase heat equations like the Dittus – Boelter eqn. to arrive at the resistance on the steam side and hence the steam condensation coefficient.

This idea was first exploited by Wilson [2] in what then became known as the 'Wilson Plot Method' to estimate the steam side coefficient.

the theoretical basis for this method could be explained as follows:

$$\frac{1}{U_o} \quad \frac{1}{h_o} = \frac{1}{h_i} + \frac{D_o}{D_i} \quad \frac{Do \ln (D_o/D_i)}{2 k} \qquad \ldots (3)$$

The term [$\frac{D_o \ln (D_o/D_i)}{2 k}$] is fairly constant, since the dependence of the thermal conductivity (k) on temperature is small (for steel as an example, 100% change on temperature causes about 10% change in K).

The Dittus-Boelter eqn. [6] gives a fair estimate of (h_i) as follows:

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$$h_i = 0.023 \left(\frac{\rho V D}{\mu} \right)^{0.8} \left(\frac{\mu\, Cp}{k} \right)^{0.4} \left(\frac{k}{D} \right) \qquad \ldots (4)$$

for Re range of 10,000 - 120,000

Hence the term $\left(\frac{D_o}{h_i\, D_i} \right)$ could be represented by $(1/a\, V^{0.8})$

i.e. $hi = a\, V^{0.8}$

Where $a = 0.023 \left(\frac{\rho D}{\mu} \right)^{0.8} \left(\frac{\mu\, Cp}{k} \right)^{0.4} \left(\frac{k}{D} \right)$

$\therefore$ Eqn. (2) becomes: $\frac{1}{U_o} \quad \frac{1}{a\, V^{0.8}} \quad \frac{1}{h_o} \quad \frac{Do\, ln\, (Do/Di)}{2\, k}$

$$\frac{1}{U_o} = \frac{1}{aV^{0.8}} + \frac{1}{h_o} + \frac{D_o \ln (D_o / D_i)}{2k} \qquad \ldots (5)$$

$\frac{1}{a\, V^{0.8}} \quad \frac{1}{U_o} \quad \frac{1}{a}$

letting $\frac{1}{V^{0.8}} = x$, $\frac{1}{U_o} = y$ and $\frac{1}{a} = m$

$\therefore$ $y = m\, x + c$ $\quad \frac{1}{h_o} \quad \frac{Do\, ln\, (Do/Di)}{2\, k}$

where $c = \left[\frac{1}{h_o} + \frac{D_o \ln (D_o / D_i)}{2k} \right]$

Eqn. $(5)_1$ when plotted gives a straight line with a slope $= \frac{1}{a}$ and an intercept on the y – axis when x = 0 of

$$\left[\frac{1}{h_o} + \frac{D_o \ln (D_o/D_i)}{2\, k} \right]$$

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note that at x = 0 , resistance on the inside is zero since velocity is infinite.

If

($\frac{D_o \ln (D_o/D_i)}{2\,k}$) is subtracted from the value of the intercept, then a value for (h_o) is readily found.

There are, however, two serious assumptions in above analysis:

i- (h_o) is constant with respect to (V) and hence the rate of heat transfer (q).

ii- $h_i = a\, v^{0.8}$ i.e. hi is a function of (V) only.

The second assumption could be avoided by rewriting Eqn. (4) as:

$$h_i = a\acute{o}\,(1 + b\acute{o}\,T_b)\frac{V^{0.8}}{D^{0.2}} \quad \ldots (6)$$

To allow for the variation of the physical properties of water with bulk temperature (T_b).

The first assumption comes from the fact that (h_o) depends on the rate of heat transfer (q). This dependence is clear in film type condensation where the rate of heat transfer depends on the thickness of the condensate film. The larger this thickness, the slower the rate of heat transfer and hence the smaller the heat transfer coefficient. This would be the case as long as the condensate film is in laminar motion, but once the film becomes turbulent, then the heat transfer coefficient begins to increase due to this turbulence. Mathematically, and is accordance with Nusselt type expressions, this dependence could be written as:

$$h_o = \frac{C_1}{\Delta T_f^{\,0.25}} \quad \ldots (7)$$

Eqn.(7) when coupled with the basic heat transfer Eqn.(2):

Then we have:

$$h_o = c_2\,(q)^{-\frac{1}{3}} \quad \ldots (8)$$

3. Experimental Apparatus and Procedure:

Two test sections, one of (1219 mm) and inside diameter of (12 mm) and the other of (203 mm) heated length, having inside diameter of (25 mm) were used for the inside tubes. These tubes were surrounded by outer tubes of (24 mm)and (50 mm) respectively.

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Water, flowing inside the inner tube was supplied from mains through a valve (see Fig.1) passing through a rotameter and its inlet and outlet temperature are measured before passing to drain.

Steam is supplied through a steam valve through the annulus at a given pressure and temperature as indicated by a pressure gauge and a thermometer and the condensate is spent to drain via a steam trap.

Sixty experimental runs were carried out on each tube. In each run inlet and outlet water temperatures, water flow rate, steam pressure and temperature, condensate flow rate were recorded.

The rotameter was calibrated by weighing the flow accumulating in a measured period of time and compared to that given by the rotameter chart. The calibration is shown in Fig. (7).

The geometry of the test sections used reflects the geometry of tube bundles in industrial equipment ,where the steam moves and condenses in narrow confined spaces between tubes, instead of being stagnant as assumed by analytical equations.

4. Results and Discussion:

The experimental results, when plotted as (h_o) vs. water flow rate, Fig.(2), show a clear dependence on steam pressure. The heat transfer coefficient is expected to depend on the steam pressure in two ways; for a given tube an increases in pressure lowers the volumetric flow rate and hence the heat transfer coefficient decreases in view of the effect of the vapour velocity in draining the condensate film quicker.

On the other hand an increase in pressure, results in a larger number of molecules striking the surface of the condensate film and hence more would stand a chance of being absorbed into the surface after giving up their latent heats. In view of the present results, the second phenomenon seems to dominate. Silver [(1)] points out that the resistance on the steam side is really made up of two parts:

i- An interfacial resistance.

ii- Resistance due to the condensate film.

and derived an expression for the interfacial heat transfer coefficient (hió) from the kinetic theory:

$$hi\acute{o} = f\left(\frac{g}{2\pi RT}\right)\frac{\lambda^2}{TV} \qquad \ldots (9)$$

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Where f = the molecular exchange condensation fraction (non – dimensional).

There seems to be uncertainty about the exact value of (f). However, silver takes it to be (0.036).

Using eqn. (9), the following values are reached:

Condenser Temp.(^oC)	Condenser Temp.(^{o} k)	Condenser Pres. (Pa) Absolute	hió (W/m^2 ^ok)
26.7	**299.7**	**3500**	**19,494**
35.6	**308.6**	**5800**	**29,526**

Recognizing this dependence on steam pressure, our experimental data, although giving a combined coefficient i.e. the reciprocal of the resistance across the condensate film plus the interfacial resistance, show a similar trend, see Fig.(2).

Testing the experimental data of this work against the Colburn correlation [(5)], Fig. (3), shows that in general the values are higher than those predicted by this correlation for (Pr No. = 1) for the condensate, which is nearest to the Prandtl No. for our condensate film (being 0.8). This is due to the high steam velocities reached in the annulus, where condensation takes place (being about 10 m/s).

The steam velocity helps to drain the condensate and hence giving thinner film resulting in a higher heat transfer coefficient. On average, the data in the laminar and in the turbulent regions are twice the values predicted by the analytical Nusselt eqn. (1), see Fig. (4). Mc Adams [(4)] states that limit for laminar flow is ($Re = 4\Gamma/\mu_f = 2100$).

The experimental data for steam condensation confirms the work of Chu et.al.[(3)] condensing organic vapours, that the heat transfer coefficient is dependent on water flow rate and hence on (q), see Fig.(5). Also from Fig. (5), the results show that (h_o) decreases as (q) increases. This is in accordance with eqn. (8), where (h_o) varies with (q) to a power of (-1/3).

The resultant (h_o) for the whole experimental runs of this work, could be obtained by plotting the data in accordance with eqn. (5), the results of this plot is shown in Fig.(6). From the intercept, the value of (h_o) is obtained.

5. Conclusions and Recommendations:

i- The actual values of the condensation coefficient as in industrial heat equipment are much higher (up to 100%) than those predicted by analytical equations such as Nusselt type expressions, because of the high vapour velocities prevailing in such equipment. Hence, for design purposes, the

coefficient calculated from such expressions should be multiplied by a factor of (2) rather than (1.2) usually recommended in literature.

ii- Higher steam pressures give higher heat transfer coefficients.

iii- This work avoids the necessity to measure the tedious metal surface temperatures and utilizes bulk temperatures only to measure vapour condensation coefficient, while maintaining accuracy.

iv- The same analysis and experimental program of this work could be employed on orientations (horizontal and inclined surfaces) to measure vapour condensation coefficients in spite of the different hydrodynamics existing. In addition, it could be applied for any type of vapour and not only steam.

v- The results from the shorter test section (representing local) gives better values for the coefficient than the longer test section (representing average).

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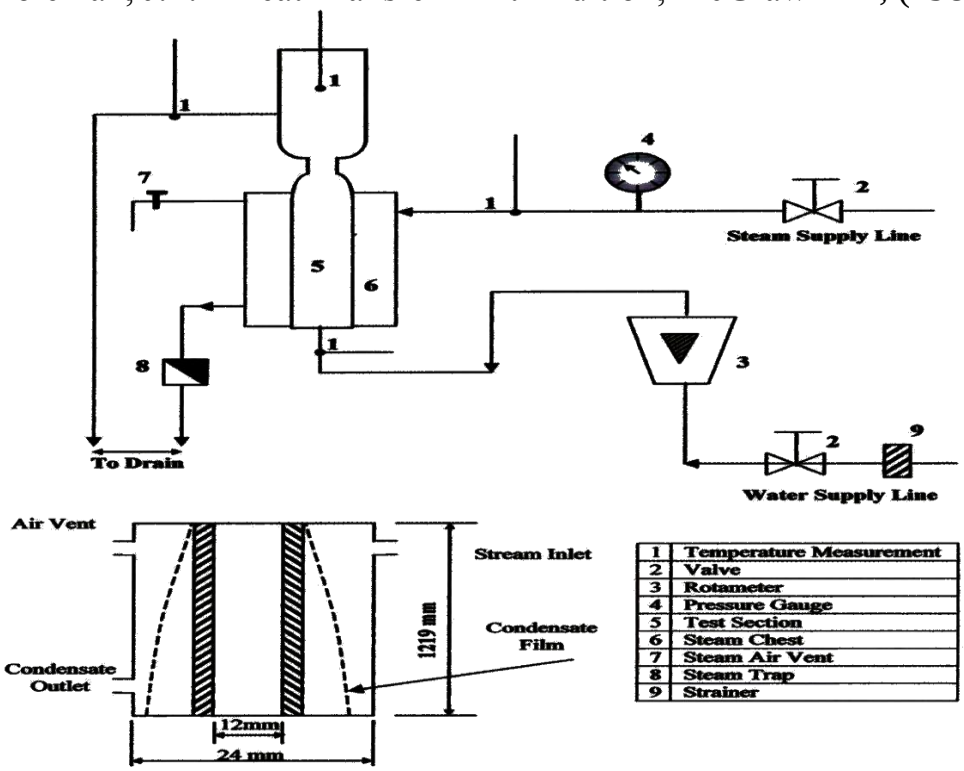

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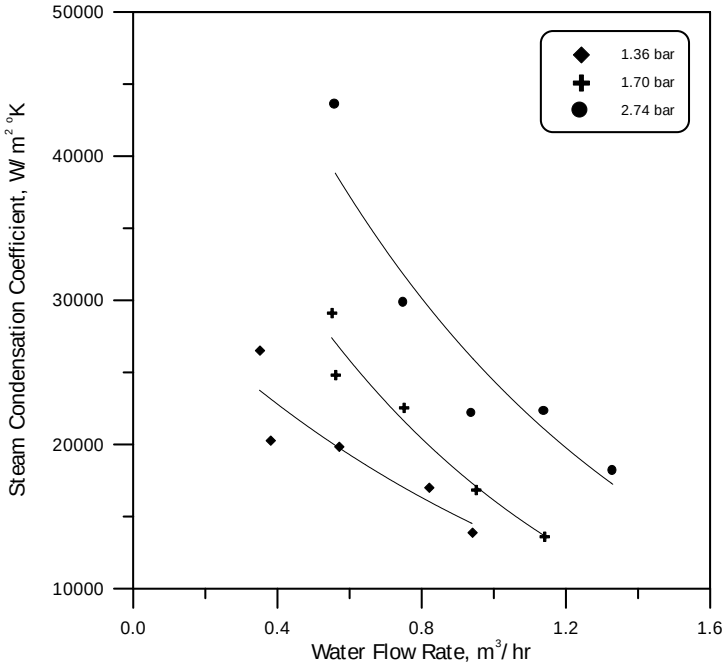

Fig.(2) Variation of steam condensation coefficient with water flow rate.

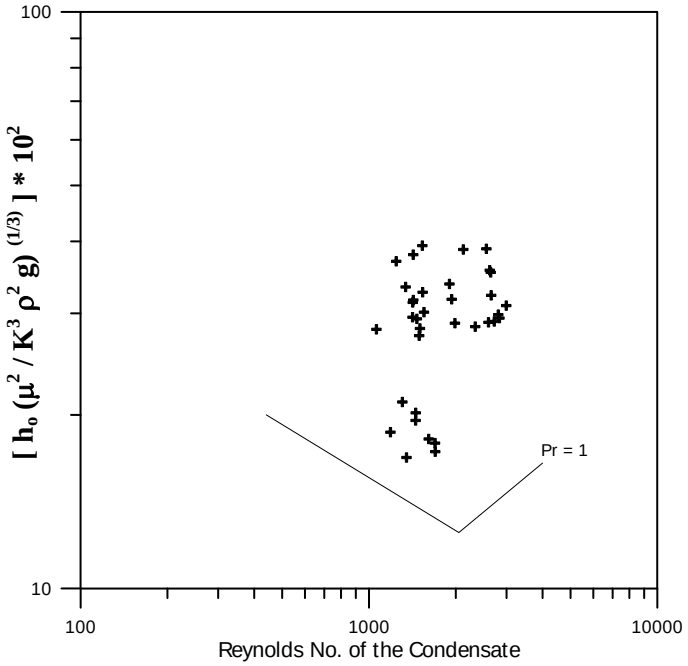

Fig.(3) Testing the experimental results against the Colburn correlation.

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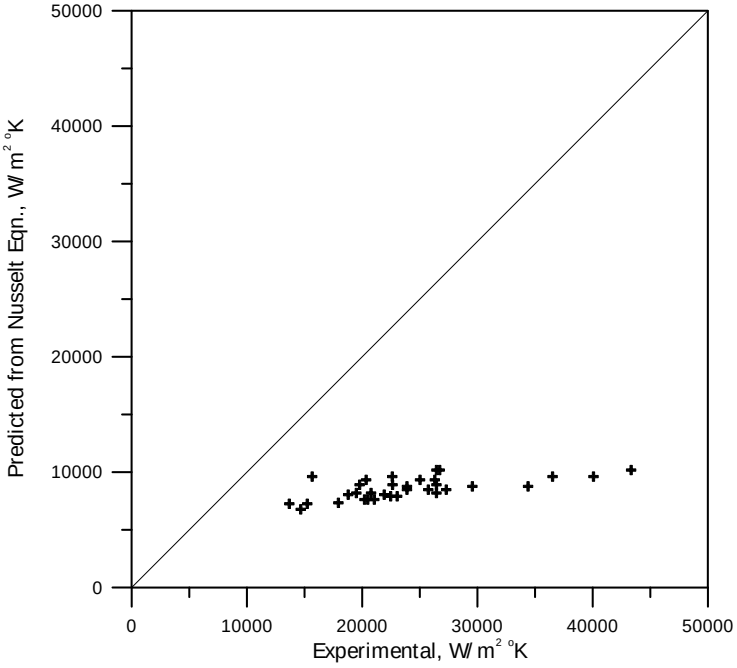

Fig.(4) Predicted against experimental heat transfer coefficient together with the 45° line.

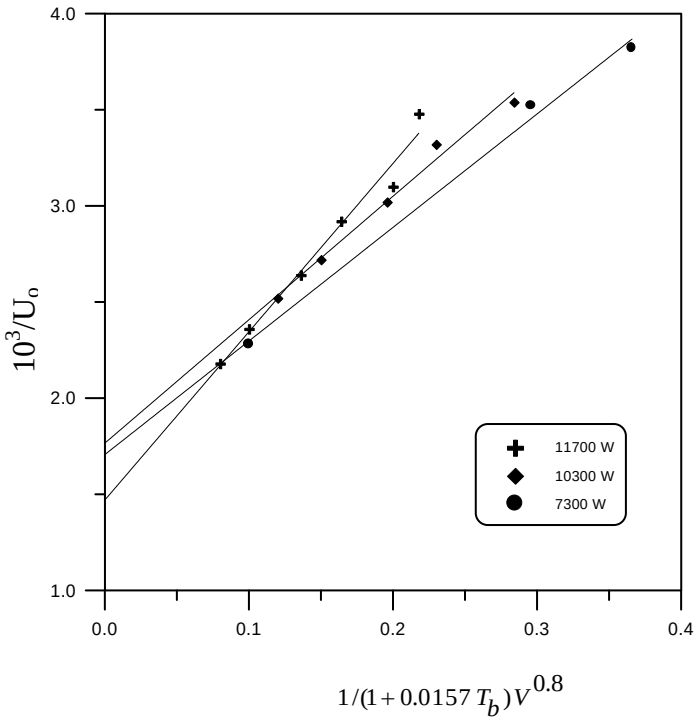

Fig.(5) Plotting the results in accordance with Chu et. al. method.

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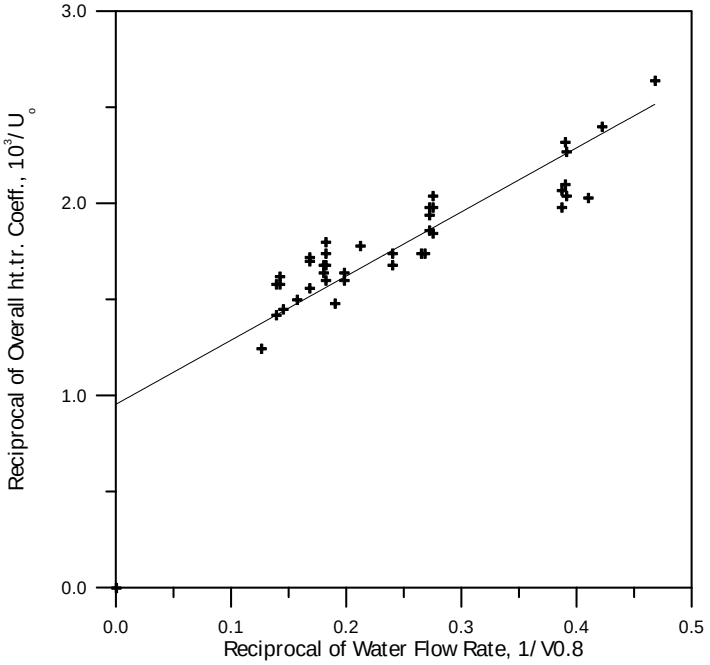

Fig.(6) Reciprocal of the overall heat transfer coefficient vs. reciprocal of water flow rate to power (0.8).

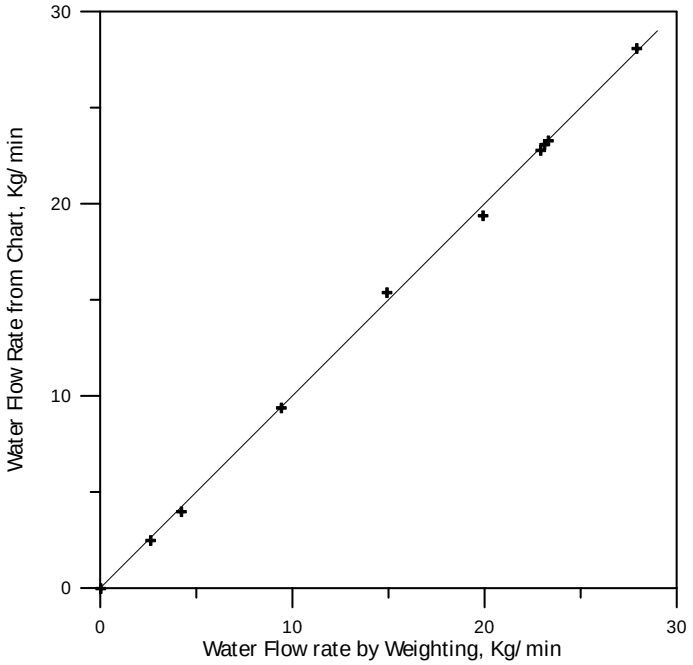

Fig.(7) Calibration of the water flow rotameter.

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