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HEAT- AND MASS-TRANSFER CHARACTERISTICS OF A COOLING AND DEHUMIDIFYING COIL AND THE EFFECT OF UPSTREAM TURBULENCE ON THEM

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Abstract—This paper presents an experimental investigation and study of the performance of a cooling and dehumidifying coil and the effect of variation of upstream turbulence (generated by introducing commercially available screens of various mesh sizes in the upstream) on its performance.

The experimental set up consists of a closed circuit air-conditioning tunnel in which velocity, temperature and humidity of air approaching the coil could be varied. The test coil chosen is a commercial evaporator of a 1.5 ton window model air-conditioner.

The experiments are conducted for the following ranges of operating parameters: dry bulb temperature, 20–48°C; wet bulb temperature, 20–36°C; sensible heat to latent heat ratio, 0.4–15; Reynolds number based on evaporator tube diameter and approach air velocity, 650–2400; turbulence level, in percentage, 2.2–6.5; turbulent Reynolds number based on evaporator tube diameter and approach air velocity, 15–150.

The heat-transfer coefficient based on enthalpy potential is presented in dimensionless form as a function of sensible to latent heat ratio, Reynolds number and turbulence (also turbulent Reynolds number). In the ranges of parameters considered the results indicate that the upstream turbulence does not have any significant effect on the heat-transfer coefficient for the evaporator coil. In view of this the following correlation, embracing the entire experimental data, is presented for the heat-transfer coefficient:

$$j_e = 0.0627\,Syl^{-0.02}Re^{-0.498}.$$

NOMENCLATURE

- A, coil external surface area (total) [m^2];
- A_f, coil face area [m^2];
- A_c, coil actual flow area [m^2];
- C_p, specific heat of air $\left[\dfrac{\text{kcal}}{\text{kg}\,°\text{C}}\right]$;
- d, coil equivalent diameter [m];
- d_t, coil tube diameter [m];
- f_e, heat-transfer coefficient $\left[\dfrac{\text{kcal}}{\text{h m}^2\,°\text{C}}\right]$;
- h, enthalpy of air $\left[\dfrac{\text{kcal}}{\text{kg}}\right]$;
- j_e, heat transfer j factor, $St_e Pr^{2/3}$ [dimensionless];
- L, coil thickness [m];
- $\dot{m}$, air mass flow area [kg/h];
- Nu_e, Nusselt number, $\dfrac{f_e d}{K}$ [dimensionless];
- Re, Reynolds number, $U_{ac}d/v\mu$ [dimensionless];
- Re_τ, turbulent Reynolds number, $Rex\tau_f$ [dimensionless];
- Syl, sensible heat to latent heat ratio;
- St_e, heat-transfer Stanton number, $Nu_e/Re \cdot Pr$ [dimensionless];
- U_{ac}, velocity of air through the coil [m/s];
- V, hot wire anemometer voltmeter output with air flow [V];
- V_0, hot wire anemometer voltmeter output with no air flow [V];
- V_{RMS}, hot wire anemometer RMS voltmeter output with air flow [V];
- v, specific volume of humid air [m^3/kg].

Greek symbols

- μ, weighted average viscosity of air–water vapour mixture $\left[\dfrac{\text{kg}}{\text{h m}}\right]$;
- τ_p, turbulence level [%];
- τ_f, turbulence level, fraction.

Subscripts

- 1, 2, c, approach, exit and coil surface conditions;
- m, mean.

Superscript

- *, dimensionless numbers based on coil tube diameter and approach air velocity.

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INTRODUCTION

THE CORRECT evaluation of cooling and dehumidifying coil performance is very important for proper design and selection of evaporators and cooling coils that are used in air-conditioning and refrigeration systems. Use of dry data in the design of wet heat exchangers can result in an overestimated fin area and an under-estimated pressure drop. Guillory and McQuiston [1] have observed, during the course of their experimental investigations of air dehumidification in a parallel plate heat exchanger, that these deviations could be as high as 30%.

The fact that the heat-transfer coefficients for surfaces operating "wet" are higher than those operating "dry" has been observed by Bettanini [2] and Yamakawa *et al.* [3]. However the geometries used by them are simple geometries like flat surfaces.

Bryan [4, 5] has studied the heat, mass and momentum transfer characteristics of a six row bare tube coil [4] and a six row integral fin tube coil [5]. Both the "wet" coil and "dry" coil tests have been conducted. It has been observed that the finned coil yields greater heat transfer in the absence of any dehumidification whereas the bare coil gives higher heat transfer when dehumidification is present. This observation has been attributed to the finned coil effectiveness being reduced below the calculated value when condensate covered the coil surface. It is further stated that this loss in effectiveness could be a function of coil load ratio. It is worthwhile to note that the arrangement of finned tube elements in Bryan's experimental cooling and dehumidifying coil represents a very non-compact type of coil design with a large bypass factor which is likely to increase when the coil operates "wet". This could be the reason why the finned coil heat transfer has been found to be less when dehumidification was present, as observed by Anderson in the discussion given in the paper [5].

Mattaralo and Trapanese [6] have investigated some aspects of performance of cooling and dehumidifying coils. The experiments conducted on a two row finned coil have yielded results which indicate that the air side convection coefficient during simultaneous heat and mass transfer is less than that during heat transfer only. But as observed by Anderson [5], with a good contact factor for a coil of this geometry, it should be expected that the wet coefficient is at least equal to the dry coefficient, if not more.

Bettanini [7] has conducted similar experiments on 4, 6 and 8 row coils built by assembling integral multiples of the basic two-row coil used by Mattaralo and Trapanese [6]. A graphical method of calculating the multirow coil capacities, when approach conditions and velocity of air are known is presented.

Rasi and Bettanini [8] have studied the heat-transfer characteristics of a fan coil unit operating as heating coil and cooling and dehumidifying coil. Their experiments indicate that, up to a certain mass flow rate, the wet coil heat-transfer coefficient is more than the dry coil heat-transfer coefficient and beyond that, the reverse is true. It is pertinent to point out that the relations presented for heat-transfer coefficient are based on results obtained for two airflow rates only. Further mean temperature difference is used in conjunction with the total heat transfer to obtain heat-transfer coefficient, which is not appropriate for simultaneous cooling and dehumidification process.

The experimental results of Buglayev and Kazakov [9] relate to the heat transfer from humid air in cross flow to a vertical tube bundle. It has been observed that increasing the flow velocity of fluid in the space between the tubes and the value of the relative vapour content in approach air augments the heat transfer at the vertical tubes by improving conditions for phase transition and for break off of the condensate film. This is in agreement with the findings of Bryan [4] that bare coil data gave higher heat transfer when dehumidification was present at the coil surface.

All these investigations do not provide a complete insight into the performance of a cooling and dehumidifying coil as there is disagreement amongst various investigations on the performance of such coils. Also, the effect of approaching stream turbulence on the performance of a cooling and dehumidifying coil has not been investigated although Guillory and McQuiston [1] attribute the difference that they noticed between the experimental and theoretical heat transfer coefficients to the free stream turbulence. It has been known for a long time that the second row of tubes in a boiler is more effective than the first row. This has been explained to be due to the influence of the wakes emanating from the first row of tubes on the flow pattern around the second row of tubes. This clearly shows that the approaching stream turbulence could have an effect on the transfer characteristics of the tube bank.

The present experimental investigations pertain to the heat- and mass-transfer characteristics of a finned evaporator over wide ranges of operating parameters and the effect of upstream turbulence on its performance.

TEST SET-UP AND PROCEDURE

The schematic diagram of the experimental set-up is shown in the Fig. 1. It consists of a closed circuit air-conditioning duct of rectangular section 510 × 265 mm, made out of 18 gauge galvanised iron sheet. The entire length of the duct is insulated by resin bonded glass wool to a thickness of 100 mm. Extra thickness of insulation is provided at and around the test section to minimise heat exchange between the system and the surroundings so as to ensure proper evaluation of heat balance on air across the evaporator.

The air from the blower passes first over three banks of electric heaters and then over a steam distributor running across the width at the centre of the duct before passing through the evaporator. The air from the test section is drawn by the blower for recirculation. The temperature of air approaching the test section is varied by controlling the electric input to heaters by means of dimmerstats, the humidity by controlling quantity of steam flowing through the distributor by means of a needle valve and the volume flow rate of air by adjusting a butterfly valve position at the blower discharge end.

The air turbulence is one of the parameters which influences the performance of the test section. The upstream turbulence in air is varied by introducing woven nylon mesh screens of various mesh sizes, at a section about 2.4 m on the upstream side of the test section. The screens are made of 8, 16 and 24 mesh sizes. All the screens are woven out of 0.36 mm diameter nylon wires and held in a framework which

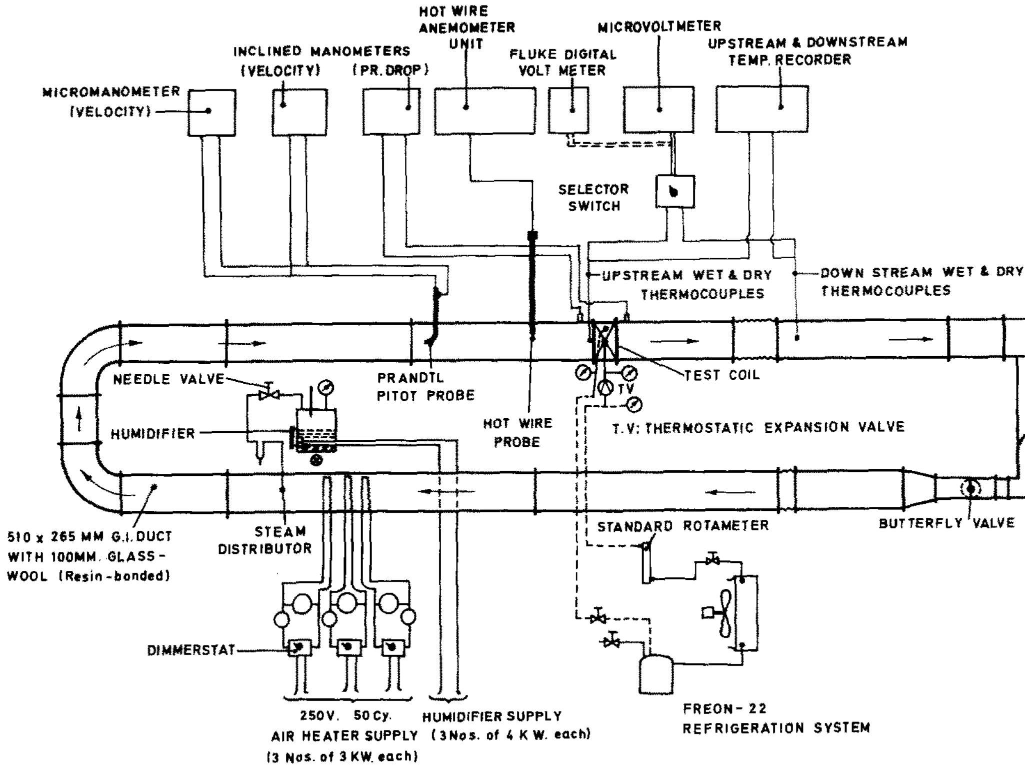

FIG. 1. Schematic diagram of experimental setup.

could be introduced at the section indicated above. The blockage coefficient, which is defined as the ratio of the area occupied by the wires of the screen to the total cross section of the screen (which is equal to the area of cross section of the duct) is for each one of the screens as follows:

8 Mesh screen 0.23
16 Mesh screen 0.46
24 Mesh screen 0.69.

The test section is an evaporator of a window model commercial airconditioner of 1.5 ton capacity. The physical data relating to this evaporator are as follows:

Number of rows of tubes:	3
Number of tubes in each row:	8
Tube arrangement:	Staggered
Tube material:	Copper
Tube diameter (outside), d_t:	12.5 mm
Number of equispaced fins:	224
Fin thickness:	0.28 mm
Fin material:	Aluminium
Coil face area, A_f.	$0.135\,m^2$
Coil actual flow area A_0:	$0.0718\,m^2$
Coil thickness, L:	84 mm
Coil external surface area (total), A:	$9.19\,m^2$
Coil equivalent diameter, d:	2.62 mm.

The refrigeration system uses Freon-22 as the working fluid. Calibrated pressure gauges and thermocouples located at various points in the refrigeration circuit and a calibrated rotameter incorporated in the liquid refrigerant circuit, after the condenser and before the thermostatic expansion valve facilitates the evaluation of heat transfer between the evaporating refrigerant within the evaporator tubes and air flowing across them on the outer surface.

The velocity profiles upstream and temperature profiles upstream and downstream of the evaporator were taken at sections shown in Fig. 1 for various conditions of air approaching the test section. The velocity profiles were fairly flat, with average velocity being 0.92 times the central velocity. The temperature profiles were also flat on both sides of the test section. The maximum variation in temperature across the duct section was ± 0.3°C at the highest temperature. At moderate temperatures, the variations were of the order of ± 0.1°C.

The distribution of turbulence level across the section of the duct before the evaporator was also measured. It was found that the turbulence level, in general, varied across the depth of the duct, being a minimum at the centre and maximum towards the walls. However, there seemed to be some symmetry in the distribution about the horizontal axis of the duct and a core of a fairly uniform turbulence level was obtainable when screens were introduced. It is probably not possible to obtain a homogeneous distribution of turbulence level in such systems. In view of this, in the correlations which are presented later in this investigation turbulence levels as measured at the centre of the duct are used.

Static pressure tappings are provided at the centre of each of the four sides of the duct on either side of the evaporator. Interconnecting these four tappings enables averaging of the pressure transmitted to the manometer column.

The velocity of air approaching the evaporator was measured using a Prandtl Pitot static probe along with a Hero micromanometer, the dry bulb and wet bulb temperatures before and after the evaporator by 30 gauge, previously calibrated, Leeds and Northrup copper–constantan thermocouples along with a Philips DC microvoltmeter/Fluke digital voltmeter and the turbulence level by DISA hot wire anemometer along with a DC microvoltmeter and RMS voltmeter. The pressure drop across the evaporator was measured by an inclined manometer.

The experimental procedure consists in taking the steady-state outputs of micromanometer, DC microvoltmeter, anemometer voltmeter and RMS voltmeter and the pressure drop across evaporator for the air side, the steady state values of temperature and pressure of liquid refrigerant before the expansion valve, the liquid refrigerant flow rate in the circuit, the average pressure of refrigerant in the evaporator and the temperature of refrigerant in the evaporator and the temperature of refrigerant after the evaporator, for the refrigerant circuit side. This procedure is repeated for different values of velocity, temperature, humidity and turbulence of air.

ANALYSIS OF EXPERIMENTAL DATA

The air at the coil surface is assumed to be saturated at a temperature corresponding to the saturation temperature of the refrigerant at the average evaporator pressure.

The properties of humid air—humidity, specific volume, specific heat, enthalpy before and after the evaporator and at the coil surface are found as per the procedure outlined in [14] when wet and dry bulb temperatures are known.

The coefficient of heat transfer is calculated using enthalpy potential. Thus

$$f_e := \frac{\dot{m}(h_1 - h_2)}{A(h_1 - h_c)} \times C_{pm}\,\mathrm{kcal/(h\,m^2\,C)} \qquad (1)$$

where h_1 and h_2 are the upstream and downstream enthalpy of air, h_c is the enthalpy of air at coil surface, and C_{pm} is mean specific heat of air

$$[\sim (C_{p1} + C_{p2})/2].$$

Some authors have evaluated the heat-transfer coefficient based on "temperature difference" and total heat transfer [8] and some authors based on "temperature potential" and total heat transfer [10]. However use of enthalpy potential seems more logical in the case of a cooling and dehumidifying coil where the energy transfer is both by sensible heat and latent heat transfer between air and the coil surface.

In the evaluation of dimensionless numbers the properties of air used are the weighted average properties. The characteristic length is the coil equivalent diameter (d) and velocity of air is the actual velocity through the coil. Some times coil tube diameter (d_t) and approach air velocity are used, in which case the dimensionless numbers are distinguished by a superscript (*). The diffusivity of air water vapour system is calculated using the correlation suggested by Fuller, Schettler and Gidding in [11]. All the properties of air are evaluated at the upstream conditions of the evaporator.

The turbulence level is calculated using King's formula which is quite accurate for the levels encountered during the investigations. The formula is:

$$\text{Turbulence level, } \tau = \frac{4VV_{\mathrm{RMS}}}{V^2 - V_0^2}.$$

All calculations are based on the heat balance made on the air side of the system on the assumption that the heat loss through the insulation around the test section and the sensible heat carried away by the condensate draining from the coil surface are negligible. In order to check the validity of this assumption heat balances were made on the refrigerant side, by the refrigeration cycle analysis at various running conditions. Good agreement was noticed between the heat lost by air flowing across the evaporator and heat gained by the refrigerant flowing through the tubes of the evaporator.

The data analysed included four cases: no screen, 8 mesh screen, 16 mesh screen, 24 mesh screen, introduced in turn in the upstream of the evaporator, with approximately the following ranges of operating parameters covered for each case:

Drybulb temperature:	20–48°C
Wetbulb temperature:	20–36°C
Sensible to latent heat ratio:	0.4–15
Reynolds number based on coil tube diameter:	650–2400
Turbulence level, percentage:	2.2–6.5.

The above Reynolds number range covers the approach air velocity range between 0.8 m/s and 3.2 m/s normally encountered in airconditioning practice.

RESULTS AND DISCUSSIONS

The correlations relating the dimensionless heat-transfer coefficient, the sensible heat to latent heat ratio, Syl, the Reynolds number Re and the turbulence level, τ for each of the four cases studied and final correlation embracing all the experimental data are obtained in the form $Y = Syl^m Re^n(a + b\tau)$ by multivariable linear regression technique using the method of least squares. In the above expression a and b are constants. This form of the equation enables the effects of screens to be included so that the effects of flattening the velocity profile and the effects of altering overall turbulence level can be distinguished. The results obtained are tabulated in Table 1.

Prandtl number, Pr and Schmidt number, Sc, do not appear in the correlations presented because variations of these quantities are quite negligible in the ranges of operating parameters covered. In fact it was observed that the ratio of two dimensionless numbers was sensibly constant within $\pm 1.5\%$.

Some of the correlations presented in Table 1 are represented in Figs. 2–6. In these figures the experimental values of dimensionless heat-transfer coefficients are plotted against the correlated values. It is clear from the figures that all the correlations are excellent fits with a high degree of correlation and low standard errors of estimate. That the effect of upstream turbulence on the performance of the evaporator coil, in the range of operating parameters considered, is not significant is clearly indicated in the individual correlations presented in Figs. 2–5 and the overall correlations presented in the Fig. 6 embracing the data of all the four cases studied. The value of the coefficient "b" in the overall correlation is -1.43×10^{-4} and hence the contribution of the term $b\tau_p$, where τ_p is the percentage turbulence level, to the total heat-transfer coefficient is very insignificant.

Table 1. Heat- and mass-transfer correlations
Correlation of the form:
$Y = Syl^m Re^n (a + b\tau_f)$
where
$Y = Nu, St, j$ factor; a and b are constants
CC = indicates the correlation coefficient

Case		Nu_c	St_e	j_e	Number of experimental data sets
No screen	m	−0.050	−0.047	−0.049	
	n	0.558	−0.443	−0.442	
	a	0.0146	0.0204	0.0163	108
	b	0.4332	0.6199	0.4872	
	CC	0.878	0.974	0.976	
8 mesh	m	−0.015	−0.011	−0.014	
screen	n	0.636	−0.367	−0.365	
	a	0.0137	0.0197	0.0154	96
	b	0.2467	0.3537	0.2783	
	CC	0.961	0.953	0.951	
16 mesh	m	−0.010	−0.006	−0.009	
screen	n	0.415	−0.587	−0.586	
	a	0.0919	0.1313	0.1039	98
	b	−0.0175	−0.0275	−0.0238	
	CC	0.896	0.987	0.988	
24 mesh	m	−0.033	−0.029	−0.032	
screen	n	0.517	−0.485	−0.484	
	a	0.0463	0.0661	0.0523	89
	b	0.1646	0.2360	0.1850	
	CC	0.972	0.987	0.987	
Overall	m	−0.02	−0.017	−0.019	
(all cases	n	0.503	−0.500	−0.498	
together)	a	0.0556	0.0800	0.0627	391
	b	−0.0123	−0.0193	−0.0143	
	CC	0.886	0.963	0.961	

In order to emphasise the above fact, the experimental values of j_e divided by $Syl^m Re^n$ are plotted against the percentage turbulence level, τ_p for the overall correlation, in the Fig. 7. The correlation line is a straight line with a slope of -1.43×10^{-4} and an intercept of 0.0627 on Y-axis. It is important to note that the correlation line is almost parallel to the x-axis (or τ_p axis). Further, it is significant that almost all of the 391 experimental points lie within $\pm 10\%$ of the average value lines.

The deduction that the upstream turbulence does not have a significant effect on the performance of evaporator coil seems reasonable in view of the similar findings of Cornings *et al.* [12] during the course of their experiments related to the study of the effect of air turbulence on heat and mass transfer from single cylinders in cross flow of air. They have found that at low Reynolds numbers, changes in turbulence levels over a wide range did not appreciably affect the heat-transfer rate, whereas at high Reynolds numbers, the effect of a change in turbulence level becomes increasingly significant especially at low turbulence levels. Increasing turbulence levels at high Reynolds numbers beyond certain values did not bring about any significant change in heat-transfer rate. The heat-transfer rate was thus independent of turbulence level at low Reynolds numbers. It may be noted that in the present investigations, the turbulence levels (between 2.5% and 6.5%) and Reynolds numbers range (between 650 and 2500) involved are low.

Thus the observation of Guillory and McQuiston [1] that the deviation in j factor (based on enthalpy potential) between the experimental values and the analytical values could be explained by the presence of free stream turbulence induced in flattening the velocity profile in the duct prior to heat exchanger does not seem to be correct.

Another important observation can be made with reference to the Table 1. It is seen that the values of the exponent of the ratio of sensible heat to latent heat, m in the correlations are quite small implying that this ratio also does not have significant effect on the heat-transfer coefficient based on enthalpy potential and total heat transfer. This is in agreement with the observation of Trapanese *et al.* [13] that the external convection coefficient and the total efficiency of the finned coil do not depend on the sensible heat to total heat ratio.

CONCLUSIONS

Heat- and mass-transfer characteristics of a cooling and dehumidifying coil have been tested. The effect of approaching stream turbulence has been investigated. From the results obtained the following conclusions are drawn:

(i) The upstream turbulence does not have any significant effect on the coil performance.

(ii) The external convection coefficient based on the total heat transfer and enthalpy potential is not influenced much by the ratio of sensible heat to latent heat or the coil load ratio.

(iii) Within the ranges of the parameters tested the following correlation could be employed to evaluate the air side heat-transfer coefficient.

$$j_e = 0.0627 Syl^{-0.02} Re^{-0.498}.$$

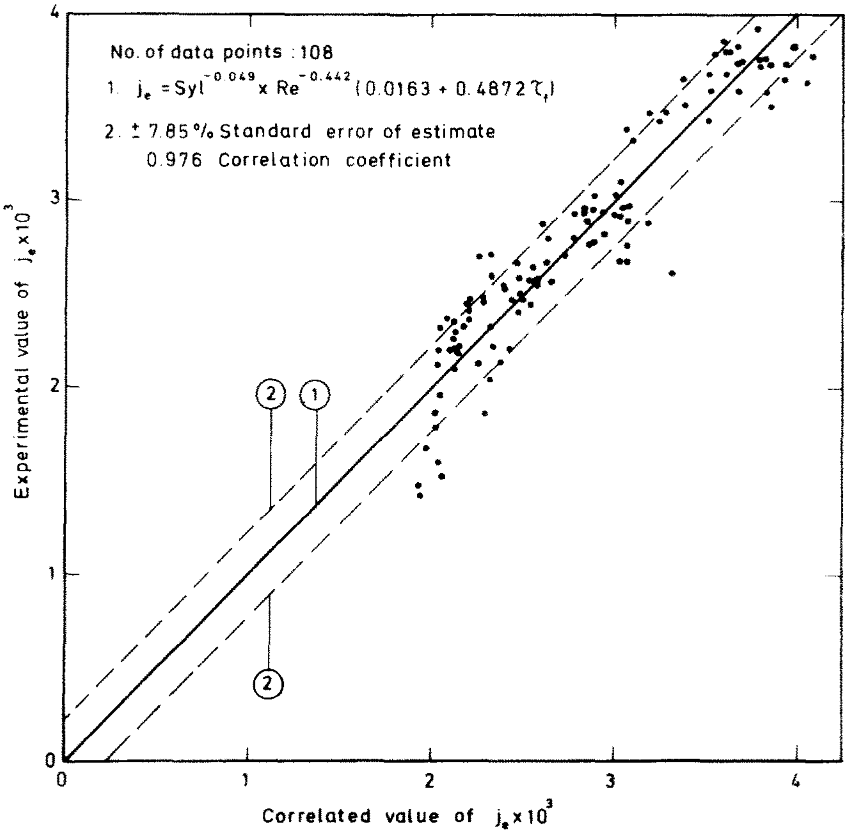

FIG. 2. Correlation of total heat-transfer data—no screen case.

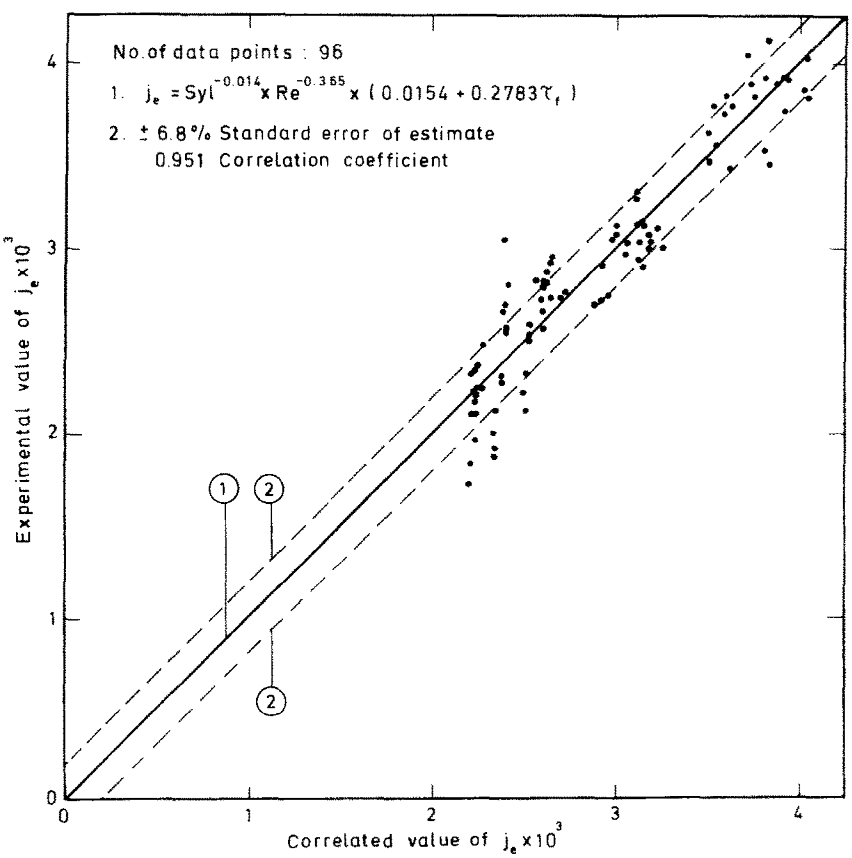

FIG. 3. Correlation—heat transfer j factor, j_e—8 mesh screen case.

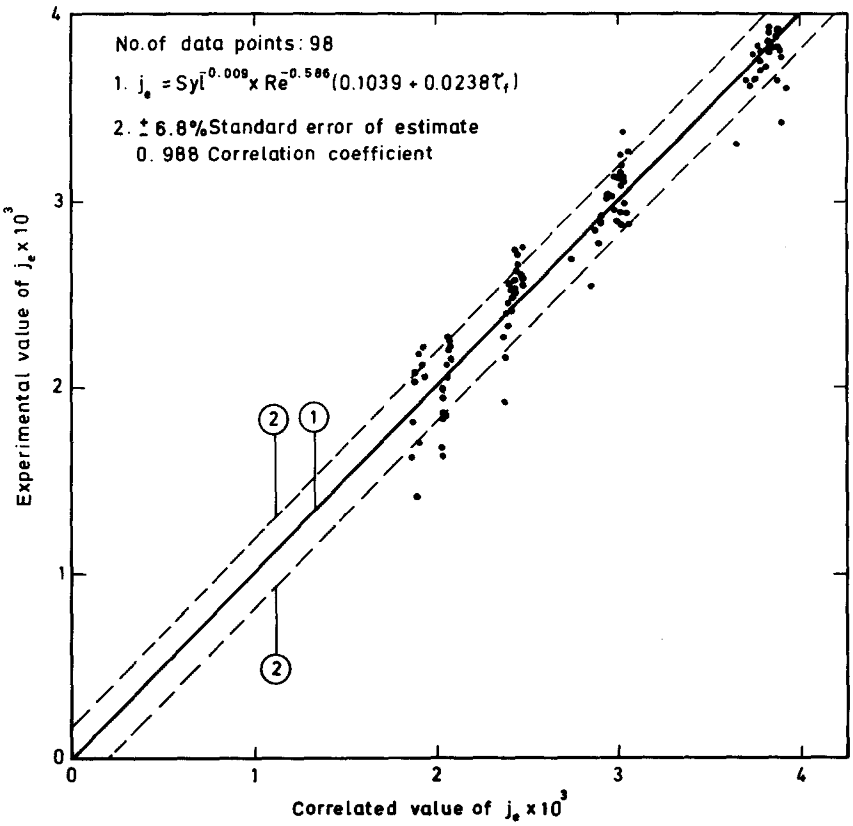

FIG. 4. Correlation—heat transfer j factor, j_e—16 mesh screen case.

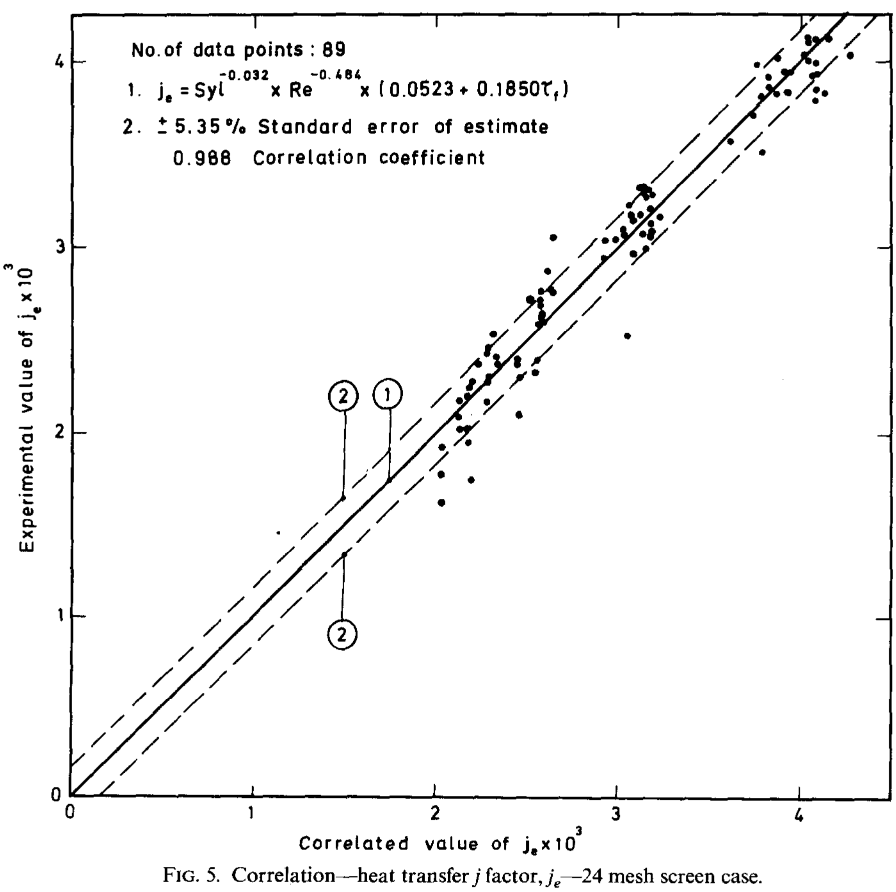

FIG. 5. Correlation—heat transfer j factor, j_e—24 mesh screen case.

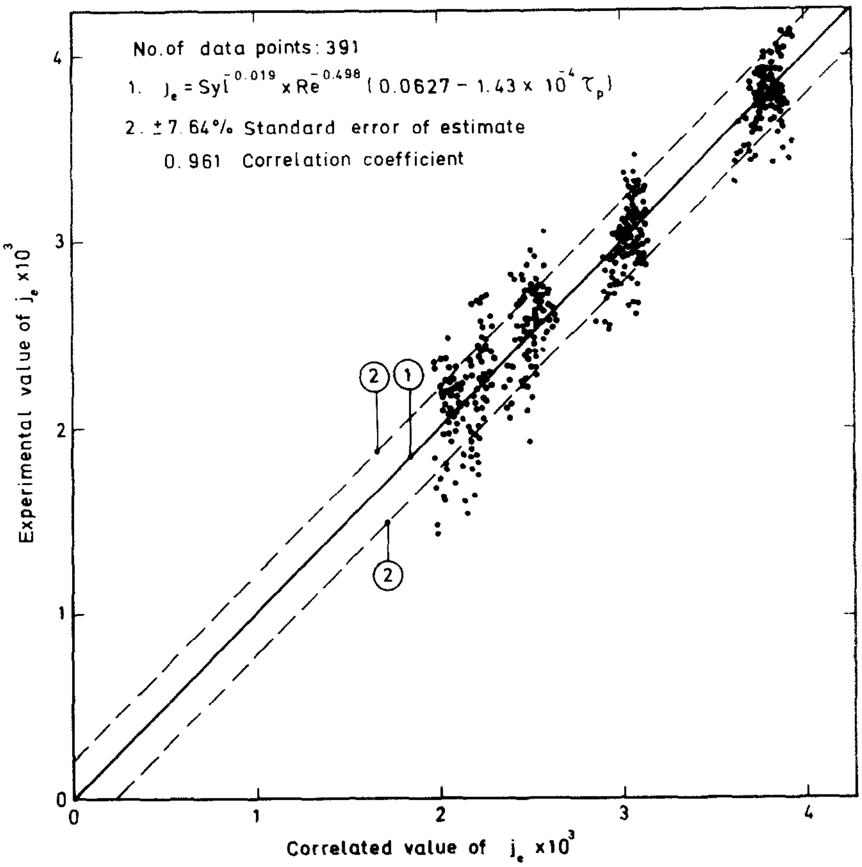

FIG. 6. Correlation of total heat-transfer data—effect of upstream turbulence.

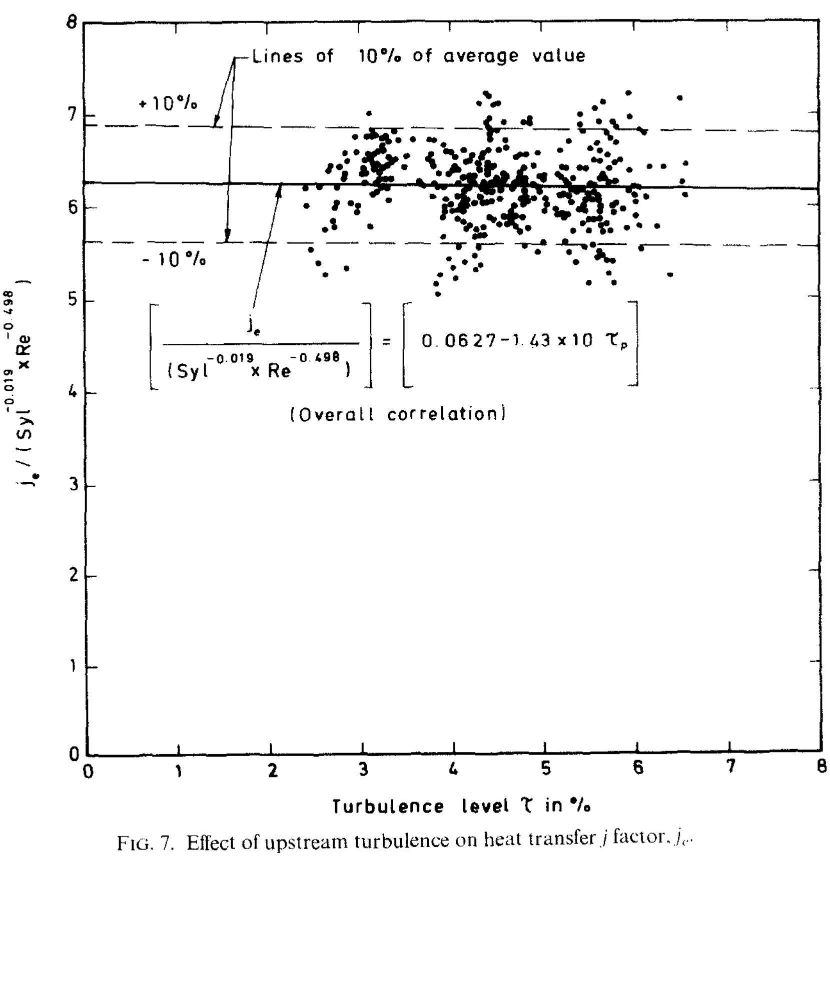

FIG. 7. Effect of upstream turbulence on heat transfer j factor, j_e.

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LES CARACTERISTIQUES DE TRANSFERT THERMIQUE ET MASSIQUE D'UN SERPENTIN DE REFRIGERATION ET DE DESHUMIDIFICATION ET L'EFFET DE LA TURBULENCE EN AMONT

Résumé—On présente des mesures expérimentales et une étude sur un serpentin de réfrigération et deshumidification et sur l'effet de la variation de la turbulence en amont (générée en introduisant des écrans disponibles dans le commerce et possédant plusieurs tailles de maille). Le montage expérimental comprand un circuit fermé d'air conditionné dont on peut faire varier la vitesse, la température et l'humidité de l'air qui attaque le serpentin. Celui-ci est un évaporateur commercial de conditionneur d'air.

Les expériences concernent les domaines suivants des paramètres:

Température sèche	20–48°C
Température humide	20–36°C
Rapport de la chaleur sensible à la chaleur latente	0,4–15
Nombre de Reynolds basé sur le diamètre du tube de l'évaporateur et sur la vitesse amont de l'air	650–2400
Niveau de turbulence, en pour cent	2,2–6,5
Nombre de Reynolds turbulent basé sur le diamètre du tube de l'évaporateur et la vitesse amont de l'air	15–150.

On présente sous forme adimensionnelle le coefficient de transfert thermique, basé sur le potentiel enthalpique, en fonction du rapport de la chaleur sensible à la chaleur latente, du nombre de Reynolds et de la turbulence (aussi du nombre de Reynolds turbulent). Dans les conditions considérées, les résultats indiquent que la turbulence en amont n'a pas d'effet sensible sur le coefficient de transfert pour le serpentin évaporateur. Par suite la formule suivante, englobant tous les résultats expérimentaux, est présentée pour le coefficient de transfert de chaleur:

$$j_e = 0{,}0627\, Syl^{-0{,}02}\, Re^{-0{,}498}.$$

WÄRME- UND STOFFAUSTAUSCH-CHARAKTERISTIKA EINES WÄRMEAUSTAUSCHERS ZUM KÜHLEN UND ENTFEUCHTEN UND DIE WIRKUNG VON VORLAUFTURBULENZ

Zusammenfassung—In dieser Veröffentlichung wird über eine experimentelle Untersuchung und Studie über das Verhalten eines Wärmeaustauschers zum Kühlen und Entfeuchten berichtet und über die Wirkung einer Aenderung der Vorlaufturbulenz (hervorgerufen durch Einsetzen von im Handel erhältlichen Gittern verschiedener Maschengröße im Vorlauf) auf sein Verhalten. Die experimentelle Anordnung besteht aus einem geschlossenen Klimatisierungskreislauf in dem Geschwindigkeit, Temperatur und Luftfeuchtigkeit in der Nähe des Wärmeaustauschers verändert werden können. Der verwendete Test-Wärmeaustauscher ist ein handelsüblicher Verdampfer aus einem Fensterklimagerät mit einer Leistung von 4500 kcal/h. Die Experimente wurden für folgende Bereiche der Betriebsparameter durchgeführt:

Trockentemperatur	20°C–48°C
Feuchttemperatur	20°C–36°C
Verhältnis der fühlbaren zur latenten Wärme	0,4–15
Reynolds-Zahl, gebildet mit Verdampherrohrdurchmesser und Luftanströmgeschwindigkeit	650–2400
Turbulenzgrad in %	2,2–6,5
Turbulente Reynolds-Zahl, gebildet mit Verdampferrofrdurchmesser und Luftanströmgeschwindigkeit	15–150

Der auf Enthalpiepotential basierende Wärmeübertragungskoeffizient wird in dimensionsloser Form als Funktion des Verhältnisses aus fühlbarer und latenter Wärme gebildet sowie aus Reynolds-Zahl und Turbulenz (auch turbulente Reynolds-Zahl). Die Ergebnisse zeigen im Bereich der betrachteten Parameter, daß die Vorlaufturbulenz für den Verdampfer keine entscheidende Wirkung auf den Wärmeübertragungskoeffizienten hat. Angesichts dessen wird folgende Beziehung–sie umfaßt sämtliche experimentellen Daten–für den Wärmeübertragungskoeffizienten aufgestellt:

$$j_c = 0.0627\ \mathrm{Syl}^{-0.02}\ Re^{-0.498}.$$

ХАРАКТЕРИСТИКИ ТЕПЛО- И МАССООБМЕНА ОХЛАЖДАЮЩЕГО И ОСУШАЮЩЕГО ЗМЕЕВИКА И ИХ ЗАВИСИМОСТЬ ОТ ТУРБУЛЕНТНОСТИ НАБЕГАЮЩЕГО ПОТОКА

Аннотация — Проведено экспериментальное исследование режима работы охлаждающего змеевика и его зависимости от турбулентности набегающего потока, генерируемой стандартными решетками с различным шагом. Экспериментальная установка состоит из замкнутого воздушного кондиционера, в котором можно было изменять скорость, температуру и влажность поступающего в змеевик воздуха. В качестве змеевика использовался стандартный испаритель 1,5-тонного воздушного кондиционера. Эксперименты проводились в следующих диапазонах рабочих параметров: температура сухого термометра: 20–48°C; температура мокрого термометра: 20–36°C; отношение теплосодержания к скрытой теплоте: 0,4–15; число Рейнольдса, отнесенное к диаметру трубы испарителя и скорости подачи воздуха: 650–2400; степень турбулентности: 2,2–6,5%; турбулентное число Рейнольдса, отнесенное к диаметру трубы испарителя и скорости подачи воздуха: 15–150. Рассчитанный по потенциалу энтальпии коэффициент теплообмена представлен в виде безразмерной функции отношения теплосодержания к скрытой теплоте, числа Рейнольдса и турбулентности, а также турбулентного числа Рейнольдса. Результаты, полученные в рассмотренных диапазонах параметров, показывают, что турбулентность набегающего потока не оказывает существенного влияния на коэффициент теплообмена змеевика. Исходя из вышеизложенного, предложено следующее корреляционное соотношение для коэффициента теплообмена $j_c = 0.0627\ Syl^{-0.02}\ Re^{-0.498}$.