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EFFECT OF VIBRATION ON HEAT TRANSFER FROM A HORIZONTAL CYLINDER TO A NORMAL AIR STREAM

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Abstract—The effect of vibration on heat transfer from a horizontal copper cylinder, 0·344 in. in diameter and 6 in. long, was investigated. The cylinder was placed normal to an air stream and was sinusoidally vibrated in a direction perpendicular to the direction of the air stream. The flow velocity varied from 19 ft/s to 92 ft/s; the double amplitude of vibration from 0·75 cm to 3·2 cm, and the frequency of vibration from 200 to 2800 cycles/min. A transient technique was used to determine the heat transfer coefficients. The experimental data in the absence of vibration is expressed by

$$N_{Nu} = 0{\cdot}226\, N_{Re}{}^{0{\cdot}6} \text{ in the range } 2500 < N_{Re} < 15\,000.$$

By imposing vibrational velocities as high as 20 per cent of the flow velocity, no appreciable change in the heat transfer coefficient was observed. An analysis using the resultant of the vibration and the flow velocity explains the observed phenomenon.

Résumé—L'effet des vibrations sur la transmission de chaleur a été étudié sur un tube de cuivre de 0,344 in. de diamètre et 6 in. de long. Le cylindre, placé normalement au courant d'air était animé d'une vibration sinusoïdale perpendiculaire à l'écoulement. La vitesse de l'écoulement variait de 19 ft/s à 92 ft/s, l'amplitude des vibrations de 0,75 cm à 3,2 cm et la fréquence de 200 à 2800 cycles/mn. Une technique de mesure transitoire a été utilisée pour déterminer les coefficients de transmission de chaleur. En l'absence de vibration, les données expérimentales s'expriment par

$$N_{Nu} = 0{,}226\, N_{Re}{}^{0{,}6} \text{ pour } 2500 < N_{Re} < 15\,000.$$

En imposant des vitesses de vibration atteingnant jusqu'à 20% de la vitesse de l'écoulement aucune modification des coefficients de transmission de chaleur n'a été observée. Une étude utilisant la résultante de la vibration et de la vitesse de l'écoulement explique le phénomène observé.

Zusammenfassung—Der Einfluss von Schwingungen auf den Wärmeübergang wurde an einem waagerechten Kupferzylinder von 0,875 cm Durchmesser und 15,2 cm Länge untersucht. Der Zylinder wurde quer zum Luftstrom angebracht und senkrecht zur Anströmrichtung in sinusförmige Schwingungen versetzt. Die Strömungsgeschwindigkeit reichte von 5,8 m/s bis 28 m/s; die Doppelamplitude der Schwingungen von 0,75 bis 3,2 cm und die Frequenz von 200 bis 2800 Schwingungen je min. Die Wärmeübergangszahlen wurden mit Hilfe eines Kurzzeitverfahrens bestimmt. Werden keine Schwingungen aufgebracht, lassen sich die Versuchsergebnisse im Bereich

$$2500 < Re < 15\,000 \text{ durch } Nu = 0{,}226\, Re^{0{,}6}$$

ausdrücken. Für Schwingungsgeschwindigkeiten bis 20% der Anströmgeschwindigkeit wurde keine merkliche Veränderung der Wärmeübergangszahlen festgestellt. Eine mit Hilfe der Resultierenden der Schwingungs- und Anströmgeschwindigkeit durchgeführte Analyse erklärt das beobachtete Phänomen.

Аннотация—Исследовано влияние вибрации на теплообмен горизонтального медного цилиндра диаметром 0,344 дюйма и длиною 6 дюймов. Цилиндр располагался перпендикулярно к потоку воздуха и вибрировал по синусоидальному закону в направлении, перпендикулярном потоку воздуха. Скорость потока изменялась от 19 до 92 футов в сек; двойная амплитуда вибрации изменялась в интервале от 0,75 до 3,2 см., частота вибрации—от 200 до 2800 колебаний в минуту. Для определения коэффициентов теплообмена использовался нестационарный метод. При отсутствии вибрации экспериментальные данные описываются зависимостью

$N_{Nu} = 0{,}226\ N_{Re}^{0,6}$ в интервале $2500 < N_{Re} < 15\,000$.

При скоростях вибрации даже равных 20 процентам скорости потока, никакого заметного изменения коэффициента теплообмена обнаружено не было. Наблюдаемое явление может быть объяснено, если учесть результирующую скоростей вибрации и потока.

NOMENCLATURE

The following nomenclature is used in this paper:

A	Area of the surface of the cylinder, ft^2;
a	Constant;
b	Constant;
C_p	Specific heat at constant pressure, Btu/lb deg F;
D	Diameter of the cylinder, ft;
d	Amplitude of vibration, ft;
f	Frequency of vibration in cycle/min, 1/min;
h	Heat transfer coefficient, Btu/h ft^2 degF;
k	Thermal conductivity, Btu/h ft degF;
t	Temperature, °F;
t_a	Ambient temperature, °F;
t_0	Temperature at $\tau = 0$, °F;
t_s	Arithmetic mean of the temperature at cylinder axis at beginning and end of the cooling process, °F;
t_f	$(t_s + t_a)/2$, Film temperature, °F;
T	Temperature difference, $(t - t_a)$, degF;
U_v	Root-mean-square velocity in the direction of the air stream, ft/h;
V_F	Flow velocity, ft/h;
V_v	Root-mean-square velocity in the direction perpendicular to the air stream, ft/h;
V_R	Resultant velocity, ft/h;
W	Mass of the cylinder, lb;
X	Amplitude of vibration corresponding to a crank rotation of ψ, ft;
τ	Unit of time, h;
ρ	Density, lb/ft^3;
μ	Viscosity, lb/h ft;
ψ	Crank angle with reference to bottom dead centre. radians;
N_{Nu}	Nusselt Number $= \dfrac{hD}{k_f}$, dimensionless;
N_{Re}	Reynolds Number $= \dfrac{DV_F\,\rho_f}{\mu_f}$, dimensionless;
$\dfrac{V_V}{V_F}$	Ratio of velocities, dimensionless.

INTRODUCTION

Heat convection is the transportation and exchange of heat due to the mixing motion of different parts of a fluid. Heat exchange by convection is promoted by the fluctuating motions in turbulent flow. Consequently heat transfer is higher in turbulent flow than in laminar flow. Vibration of the heat transfer surface may bring about the change from laminar to turbulent flow and hence result in higher heat transfer coefficients.

Studies in this field can be broadly classified into two types—one in which the heat transfer surface is vibrated and the other in which the flow medium is subjected to pulsation, vibrations or agitation.

Van der Hegge Zijnen [1] reported that the heat transfer from a tungsten wire, 0·0005 cm in diameter, to a normal air stream, decreased when the wire was subjected to vibrations in the direction of the air stream. A maximum decrease of 4·3 per cent in the heat transfer coefficient was observed when the ratio of the root-mean-square velocity of vibration to the flow velocity was as high as 45 per cent. Anantanarayanan and Ramachandran [2] observed an increase of 130 per cent in the heat transfer from a vibrating horizontal nichrome wire, 0·018 in. in diameter, to an air stream flowing parallel to the wire. The wire was vibrated in a vertical plane with varying values of vibrational velocity. The ratio of the vibrational velocity to the flow velocity used in their study varied from 0 to 30 per cent. Tessin and Jakob [3] while studying the effects of starting length on heat transfer from a cylinder to a parallel gas stream found that the heat transfer coefficients were not appreciably altered when the cylinder was subjected to a vibrational velocity of about 2 per cent of the flow velocity. While only the above three investigations have been reported on the effects of vibrations on heat transfer by forced convection, studies on the effect of vibration on heat transfer by free convection were reported by Martinelli and Boelter

[4], Lemlich [5], Tsui [6], Shine [7], Maxwell [8] and Richards [9].

In view of the meagre information available on the effect of vibration on heat transfer by forced convection, an investigation was undertaken to study the influence of vibration on heat transfer from a horizontal cylinder to a normal air stream. The basic vibrational parameters affecting the heat transfer are the amplitude and frequency of vibration. Previous investigations indicate that these three can be combined to give the following derived parameters: a vibrational velocity proportional to the product of amplitude and frequency of vibration, the ratio of the vibrational velocity to the flow velocity and the amplitude–diameter ratio. Besides the large accelerations and decelerations to which the heat transfer surface is subjected during the vibration, there appear to be other factors which decide whether the surface carries the boundary layer with it or vibrates in an enclosed fluid film. Indeed, the nature of the phenomenon associated with the vibration of the heat transfer surface is so complicated that a simple correlation or analysis of the mechanism is not possible at this time.

DESCRIPTION OF APPARATUS

The test apparatus consisted of a thermal "capacitor" with a high thermal capacity, an arrangement for vibrating the cylinder at various amplitudes and frequencies, provision for a controlled flow of air, and finally instrumentation to determine the temperature–time history of the capacitor during cooling. In Fig. 1 is shown the arrangement of the test apparatus. The honey-comb and the straightener in the 8 in. square duct straightened out the discharge from the blower. The duct discharged into atmosphere and the test cylinder was placed 1 ft from the duct. A pitot traverse indicated a uniform velocity profile at this section. The air velocity was measured with an impact tube made of standard dimensions. The velocity head was measured with a Meriam manometer, model A-750. The impact tube was swung into the plane of the test cylinder after each cooling run. This prevented the impact tube from disturbing the flow field around the cylinder during cooling. As the duct was discharging into the atmosphere, the impact tube directly gave the velocity head. In Fig. 2 is shown the solid copper rod, 6 in. long and 0·344 in. diameter, which was used as the thermal "capacitor" and which also provided the necessary heat transfer surface. The test cylinder constructed as shown in Fig. 2 satisfied the following conditions:

(a) The test cylinder made of copper was essentially isothermal at every instant of cooling.
(b) The end caps reduced the end leakage of heat to a negligible value.
(c) The radiation effects were reduced by the highly polished electroplated surface of the test cylinder.
(d) The test cylinder and its "transite" supports were heated simultaneously, thus reducing any parasitic heat transfer from the supports to the cylinder or vice versa, to a negligible value.

A 40 B and S gauge iron–constantan thermocouple was inserted inside the test cylinder. The reference junction of the thermocouple was placed upstream of the cylinder. The temperature at that station was measured using a mercury-in-glass thermometer reading accurately to 0·05°F. The copper cylinder with its transite supports was held in a yoke support fixed to the reciprocating member of the connecting rod and crank mechanism. A heater of a special design was used to heat the cylinder when fixed in position.

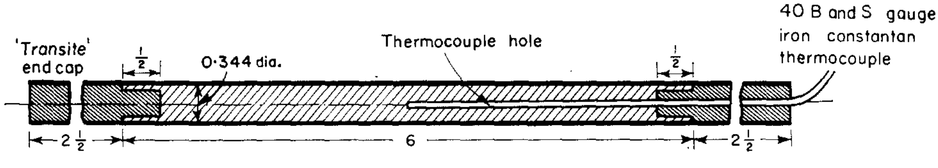

FIG. 2. Thermal capacitor.

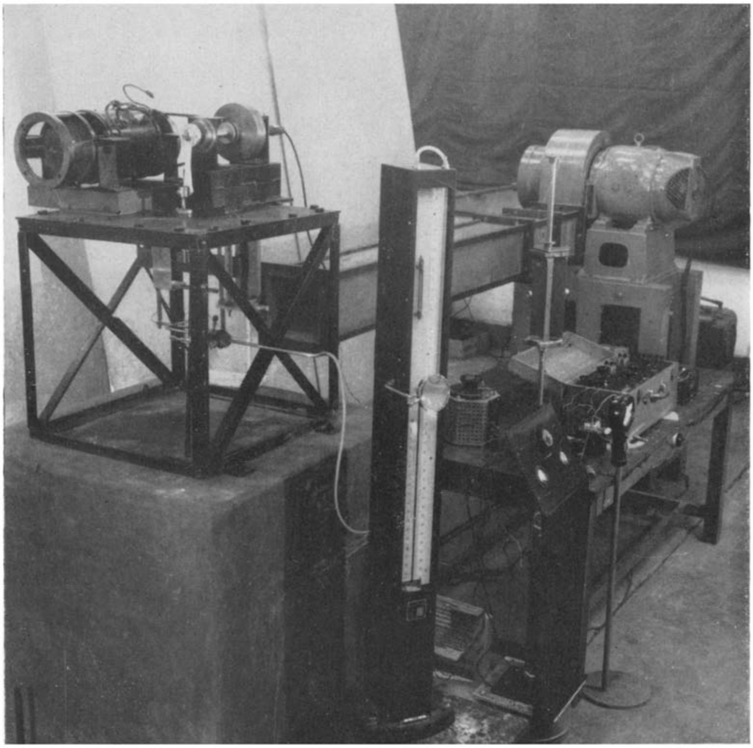

FIG. 1. General arrangement of the test apparatus.

In Fig. 3 is shown the connecting rod and crank mechanism which vibrated the cylinder in a vertical direction. The large length of the connecting rod gave a sensibly sinusoidal motion during vibration. A General Electric hand tachometer was used to measure the speed and the double amplitude of vibration was measured with a travelling telescope.

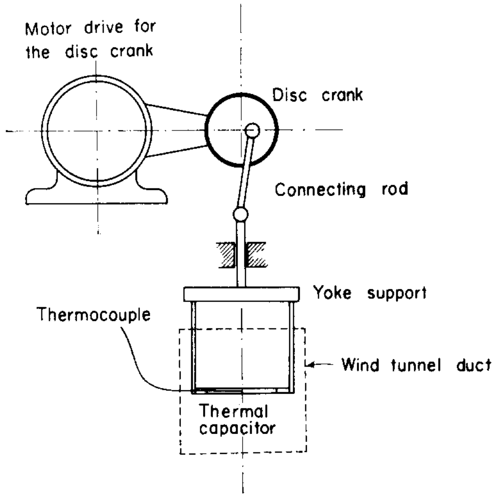

FIG. 3. Connecting rod and crank mechanism.

The size of the cylinder was so chosen that the cooling curve at the maximum velocity could be easily determined using a potentiometer and a stop watch. A Leeds and Northrup portable precision potentiometer was used along with an Anglo-Swiss stop watch for this purpose.

The transient technique adopted to determine the heat transfer coefficients was similar to the one suggested first by London *et al.* [10].

EXPERIMENTAL PROCEDURE

The initial runs were conducted to establish the heat transfer data in the absence of vibration. The test cylinder was heated by placing the heater near it. The blower was started. When the test cylinder had reached a temperature greater than a pre-set value, the heater was removed and the cylinder allowed to cool. One of the stop watches was started when the potentiometer balance was obtained at the pre-set value as indicated by the galvanometer. Immediately afterwards, the potentiometer reading was reduced by a known amount. When the poteniometer balance was reached at this value, the first stop watch was stopped and the second started. This procedure was repeated to obtain the complete cooling data. The temperature at the beginning of the cooling run was in the range 140° to 220°F. The temperature–time history was recorded till the surface reached a temperature of 90°F. The potentiometer current was obtained from a battery of six cells so as to minimize the de-standardization of the calibration during the cooling run. Further, the calibration was checked and rectified as and when the cooling rate permitted. At the conclusion of the test, all the non-transient data were recorded. This was repeated at various values of the flow velocity. The next set of runs was conducted with the cylinder vibrating. Keeping the amplitude of vibration and the flow velocity constant, the vibrational frequency was varied from 200 to 2800 cycle/min. This was repeated at the same amplitude for different values of flow velocity ranging from 19 ft/s to 92 ft/s. Further test data were obtained for different amplitudes of vibration. The range of parameters covered was

Flow velocity	19 ft/s to 92 ft/s
Double amplitude of vibration	0·75 cm to 3·2 cm
Frequency of vibration	200 cycle/min to 2800 cycle/min.

ANALYSIS OF DATA

The theory of the transient test technique [10] has been well established and will be briefly given here.

Writing the heat balance of the test cylinder at any instant of cooling,

$$W\,C_p\left(\frac{dt}{d\tau}\right) = -\,hA\,(t - t_a). \tag{1}$$

Putting $T = t - t_a$

$$W\,C_p\left(\frac{dT}{d\tau}\right) = -\,hAT. \tag{1a}$$

Equation (1) holds good only when the following assumptions are satisfied (a) the test cylinder loses all the heat through its cylindrical surface only, i.e. no end conduction and radiation take place, (b) the test cylinder remains isothermal at every instant of cooling. Schneider [11] has pointed out that for the value of the modulus $K/h\ \delta_1 > 10$, where δ_1 is a surface radius, the temperature is nearly uniform through the cylinder. Under maximum heat flow conditions in this investigation, the value of the modulus is far in excess of 10, thus justifying the assumption that the cylinder is isothermal.

Equation (1a) can be integrated only when (i) h is independent of t and hence τ; (ii) C_p and t_a are assumed constants.

Equation (1a) can be written as

$$h = -\frac{W\ C_p}{A}\frac{1}{T}\frac{\mathrm{d}T}{\mathrm{d}\tau}. \tag{2}$$

Assuming that the chord has the same slope as the tangent at the mid-point of the chord, the value of $(1/T)\ (\mathrm{d}T/\mathrm{d}\tau)$ can be calculated from the temperature–time history.

Integration of equation (1) with the boundary conditions $\tau = 0$, $T = t_0 - t_a$; and $\tau = \tau$, $T = t - t_a$, yields

$$\frac{t - t_a}{t_0 - t_a} = \exp\left(-\frac{hA}{W\ C_p}\tau\right). \tag{3}$$

This relation should yield a straight line when plotted on a semi-log sheet. Fig. 4 is typical plot of a cooling run. Departure from linearity would reflect combinations of end leakage, variation of h and possibly instrument lag effects.

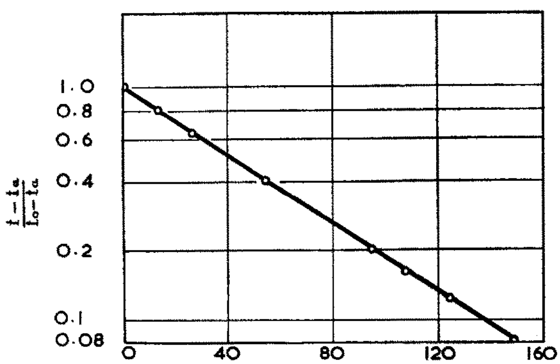

FIG. 4. Cooling curve in terms of $(t - t_a)/(t_0 - t_a)$ versus time.

After obtaining h, the Nusselt and Reynolds numbers were calculated using the values of the physical properties at the film temperature $t_f = (t_s + t_a)/2$.

As the length of the connecting rod was large the vibration was sensibly sinusoidal. The motion of the test cylinder under conditions of maximum amplitude is expressed by

$$\frac{X}{d} = \cos\psi + 0{\cdot}026\cos 2\psi. \tag{4}$$

The root-mean-square velocity of vibration was calculated from the equation

$$V_v = 60\ (\sqrt{2}\ \pi f d). \tag{5}$$

A similar equation was used by Martinelli and Boelter [4].

The ratio of the vibrational velocity to the flow velocity, V_v/V_F, was then calculated in each case.

ACCURACY OF INSTRUMENTATION

The following summarizes the accuracies of the various measurements:

Physical quantity	*Measured accurate to*
Temperature difference	$\pm$ 0·2 degF
Time interval	$\pm$ 0·1 s
Flow velocity	$\pm$ 0·6 per cent
Frequency of vibration	$\pm$ 10 cycle/min
Double amplitude of vibration	$\pm$ 0·002 cm.

In some runs, a fluctuation of about 1·0 per cent in the flow velocity was observed. The cumulative error resulting from these inaccuracies was estimated to be 4 per cent. On this basis, the maximum error associated with the Nusselt and Reynolds numbers was 4 per cent under the most adverse conditions. A similar estimation indicated an error of 5 per cent in the vibrational velocity.

DISCUSSION OF RESULTS

The results in the absence of vibration are presented in Fig. 5. These can be expressed in the form

$$N_{Nu} = 0{\cdot}226\ N_{Re}^{\ 0{\cdot}6} \tag{6}$$

in the range $2500 < N_{Re} < 15\,000$.

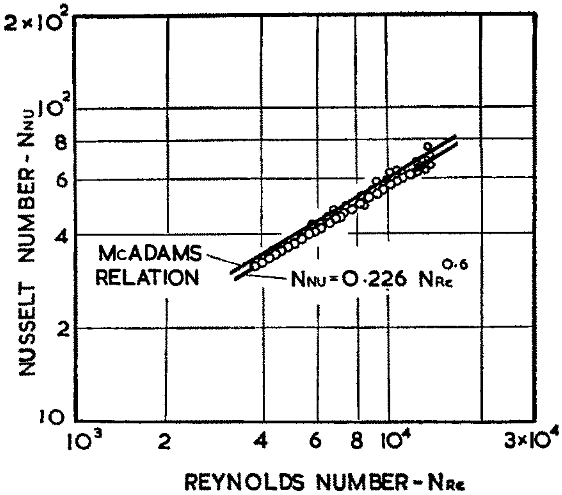

FIG. 5. Test results without vibration.

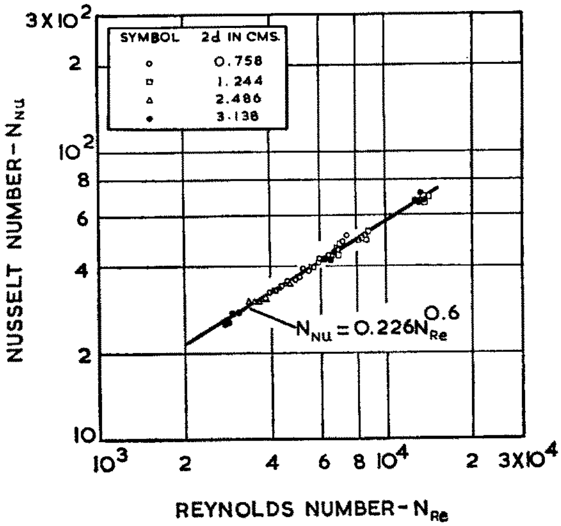

FIG. 7. Test results for a vibrational frequency of 990 cycle/min.

McAdams [12] recommends a relationship of the form

$$N_{Nu} = 0{\cdot}24\ N_{Re}{}^{0{\cdot}6} \tag{7}$$

in the range $1000 < N_{Re} < 50{,}000$.

The heat transfer coefficients obtained in this investigation are 6 per cent lower than those given by equation 7.

In Figs. 6–8 are shown the results with vibration. In Fig. 6 is shown the test data for a constant amplitude with varying vibrational frequency and flow velocities. In Fig. 7 is presented test data at constant frequency with varying amplitude and flow velocities. In Fig. 8 is presented the data of all the sixty-four runs conducted with different combinations of flow velocity, amplitude and frequency of vibration. In each figure the forced convection correlation without vibration is also shown to facilitate comparison. They indicate that the heat transfer

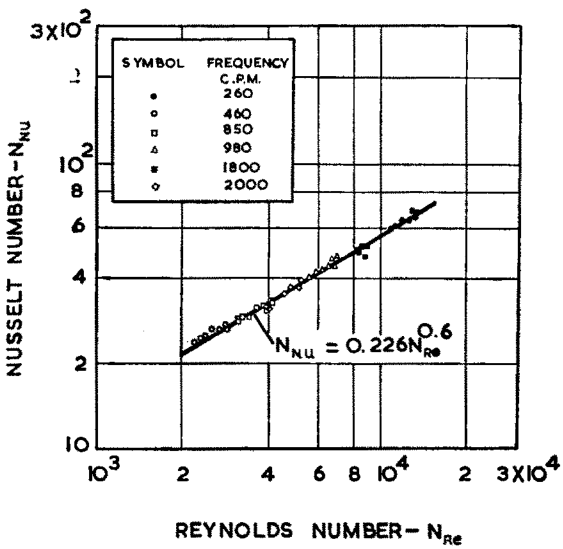

FIG. 6. Test results for a vibrational double amplitude of 1·244 cm.

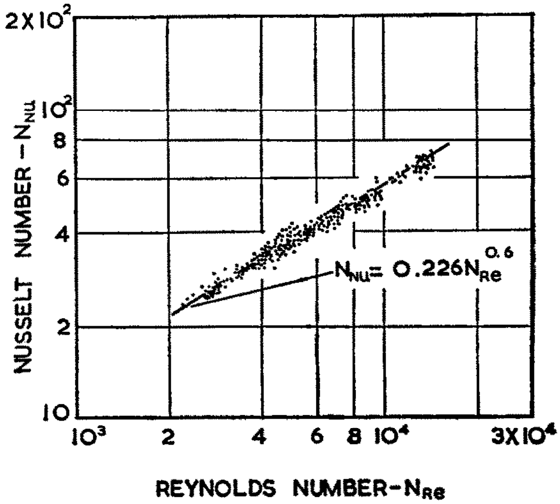

FIG. 8. Test results with vibration.

coefficients are not affected under the influence of vibrations in the range of parameters investigated. The root-mean-square velocity of vibration was varied from 0·5 ft/s to 4·2 ft/s. Further increase was prevented by mechanical limitations of the connecting rod and crank mechanism. The ratio V_v/V_F varied from 0·04 to 0·20. The trend of the results was the same in every run and no other characteristic behaviour was observed at the highest amplitude or at the highest flow velocity. A few runs were conducted wherein the frequency of vibration was varying throughout the cooling of the cylinder. Even these variations did not have any effect on the heat transfer coefficients.

Van der Hegge Zijnen [1] using the resultant of the flow velocity and the vibrational velocity found that the heat transfer from a wire subjected to velocity fluctuations perpendicular and parallel to the direction of flow could be expressed in the form

$$N_{Nu} = a + b\sqrt{(V_F)}\left[1 - \left(\frac{U_v}{V_F}\right)^2 + 0{\cdot}25\left(\frac{V_v}{V_F}\right)^2\right]. \tag{8}$$

This was obtained by substituting the resultant velocity for the flow velocity in the equation

$$N_{Nu} = a + b\sqrt{(V_F)}. \tag{9}$$

Higher power terms were neglected in the binomial expansion given in equation (8). His experimental results were in good agreement with equation (8).

A similar analysis is made below using equation (6) which gives the heat transfer data in the absence of vibration.

$$N_{Nu} = 0{\cdot}226\, {N_{Re}}^{0\cdot6} \tag{6}$$

$$N_{Nu} = K\, V_F^{0\cdot6} \tag{10}$$

where

$$K = 0{\cdot}226\left[\frac{D\rho}{\mu}\right]^{0\cdot6}.$$

On imposing a vibrational velocity V_v, the resultant velocity becomes

$$V_R = \sqrt{(V_F^2 + V_v^2)}. \tag{11}$$

On substituting this value, equation (10) transforms to

$$N_{Nu} = K(V_F^2 + V_v^2)^{0\cdot3}$$

$$= K\, V_F^{0\cdot6}\left[1 + \left(\frac{V_v}{V_F}\right)^2\right]^{0\cdot3}. \tag{12}$$

Writing the binomial expansion of the terms in the parentheses, we get

$$N_{Nu} = K\, V_F^{0\cdot6}\left[1 + 0{\cdot}3\left(\frac{V_v}{V_F}\right)^2 - 0{\cdot}105\left(\frac{V_v}{V_F}\right)^4 + 0{\cdot}06\left(\frac{V_v}{V_F}\right)^6 - + \ldots\right] \tag{13}$$

As the highest value of V_v/V_F is 0·2, the third and the succeeding terms can be neglected. Then equation (13) reduces to

$$N_{Nu} = K\, V_F^{0\cdot6}\left[1 + 0{\cdot}3\left(\frac{V_v}{V_F}\right)^2\right]. \tag{14}$$

Substituting $(V_v/V_F) = 0{\cdot}2$ which is the maximum used in this investigation, we get

$$N_{Nu} = 1{\cdot}012\, K\, V_F^{0\cdot6} \tag{15}$$

thus indicating the improvement in heat transfer to be only 1·2 per cent. An increase of 1·2 per cent could not be observed with the instrumentation used in this investigation.

CONCLUSIONS

(1) The forced convection heat transfer from a horizontal cylinder to a normal air stream can be expressed in the form

$$N_{Nu} = 0{\cdot}226\, {N_{Re}}^{0\cdot6}$$

in the range $2500 < N_{Re} < 15\,000$.

(2) In the above range of Reynolds number, the heat transfer from the cylinder subjected to vibration in a direction perpendicular to the direction of the air stream remains unaffected when the ratio of the root-mean-square velocity of vibration to the flow velocity is varied from 4 to 20 per cent.

(3) An analysis using the resultant of the vibrational and the flow velocity indicated that there will be no appreciable change in the heat transfer coefficients.

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