The Effect of Acoustic Vibration on Forced Convective Heat Transfer

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In an effort to study the effect of acoustic vibration on forced convective heat transfer, sound at resonant frequencies of 198, 256, and 322 c./sec., as well as sound at nonresonant frequencies, was imposed on air flowing at Reynolds numbers of 560 to 5,900 in the core of a horizontal, double pipe, steam to air, heat exchanger. Increases in Nusselt number of up to 51% in the nominally laminar region and up to 27% in the nominally turbulent region were obtained. The improvement peaked sharply at resonance and increased with both amplitude and resonant frequency. A qualitative dual mechanism is suggested, and correlations for the experimental results are presented.

In recent years there has been a growing interest in the effect of vibration or pulsation on transport phenomena. This has been evidenced by the increasing number of papers which have appeared on the subject, particularly in liquid-liquid extraction and in heat transfer. For the latter field a variety of situations have been studied. A search of the widely scattered literature reveals that these have included natural convection, external forced convection, aerodynamic or compressional heating, boiling, and other systems, as well as the simple, single phase, axial flow, forced convective heat exchanger.

In this last category Martinelli et al. (10) using pulsations varying from 0.22 to 4.4 c./sec. found little if any improvement in heat transfer to water flowing at N_{Re} ranging from 1,000 to 10,000. Marchant (9) employing pulsations of 0.17, 0.42, and 1.0 c./sec., together with water at N_{Re} of 400 to 100,000, found some improvement in the laminar region but virtually none at higher flow rates. On the other hand West and Taylor (*17*) found, increases in h in excess of 60% by applying 1.7 pulses/sec. to water flowing at N_{Re} of 30,000 to 85,000, but they indicated that no appreciable improvement was obtained for other flow rates. Shirotsuka et al. (*16*) using a pulsation frequency of 1.7 to 8.3 c./sec. for water at N_{Re} of 3,000 to 22,000 found considerable improvement and offered a dimensionless correlation. Linke and Hufschmidt (*7, 8*) reported considerable improvement, particularly in the laminar region, for pulsation applied to oil. Robinson et al. (*15*) reported some increase for ultrasonic vibration of 400,000 c./sec. applied to flowing water.

For air Havemann et al. (2, 3) employing 5 to 40 pulses/sec. and an N_{Re} of 5,000 to 35,000 found an irregular decrease below a certain critical frequency and an irregular increase above it, with the wave form influencing the critical frequency. Mueller (*13*) using a much lower frequency range of 0.038 to 0.25 c./sec. and air flowing at N_{Re} of 53,000 to 77,000 found only a decrease, which was explained in terms of the pulsation being below a somewhat analogous critical level of disturbance and hence in the quasisteady region (for which situation a decrease is usually encountered in accordance with simple exchanger theory). Morrell (*11*) derived a semiempirical relationship which was applied to the extreme conditions in a rocket engine with large amplitude resonance burning.

The disagreement that exists among published results is not surprising in view of the differences among the particular systems, to say nothing of the wide range of vibrational variables involved. Furthermore work has been mostly in the subsonic frequency region where the quasisteady state or proximity to it can negate or complicate any positive vibrational effect. Accordingly the present study was undertaken with frequencies in the acoustic region. A double-pipe, condensing steam to air, exchanger was used. Since vibration can affect both films of a two-

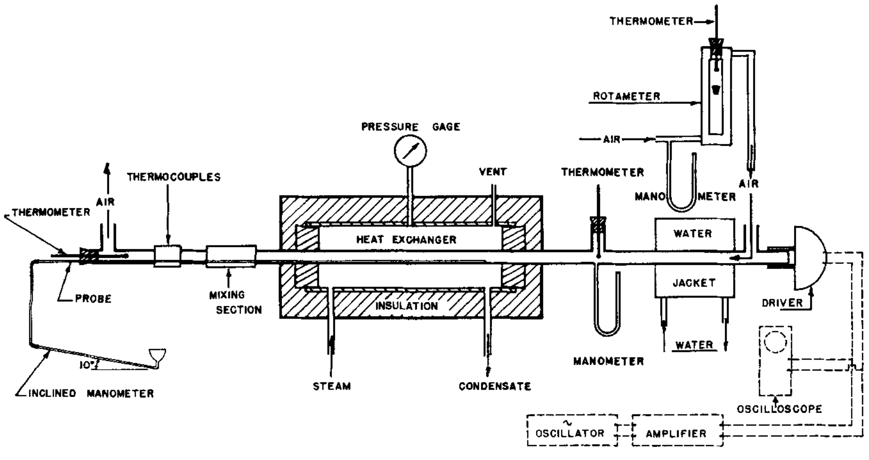

Fig. 1. Apparatus (not to scale).

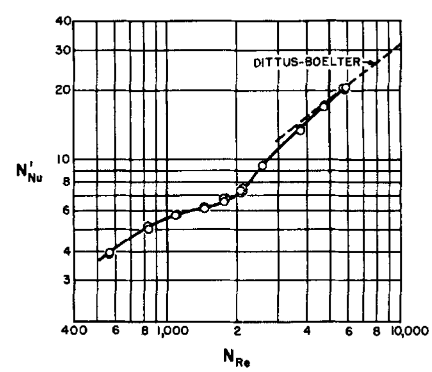

Fig. 2. Control runs without vibration.

fluid exchanger, the choice of this particular system offers an additional simplification in that the thermal resistance of only one film (air) is significant. A recently reported investigation along somewhat similar lines by Jackson et al. (5) is discussed later.

EXPERIMENT

An over-all schematic diagram of the equipment is shown in Figure 1. The heated section consisted of a horizontal 25-in. length of 16-B.w.g. copper tubing with 0.745-in. I.D. concentric with a steam jacket of 2-in. steel pipe. Calming sections of the same core I.D. were added, 65 in. in length upstream and 14 in. in length downstream. They were of Pyrex glass rather than metal in order to minimize longitudinal wall conduction. For a preliminary series of runs the downstream calming section was empty, but for the main series it was packed with a 4-in. length of coarse stainless steel turnings to serve as a mixing section. The entire 104-in. length was insulated with Fiber-glass pads. Air from a service line entered the system through a calibrated rotameter and a large surge tank. Tests showed that the tank effectively stabilized the air pressure and eliminated any spurious effect of vibration on the metering portion of the system.

Temperatures were measured with mercury thermometers as shown. In addition five 24-B.w.g. iron-constantan thermocouple junctions were installed downstream to give some measure of the radial variation in outlet temperature for the preliminary series of runs, as well as to check the uniformity of the mixing temperature in the main series. One junction was placed at the axis, one at a position one-third the way from axis to wall, one two-thirds the way, and two in the wall.

Steam throttled down from a service line was admitted at such a rate that a small purge stream continuously flowed out the vent. Condensate was collected or sewered through a trap. Steam pressure in the annulus was nearly atmospheric.

Vibration was imposed on the air stream at the inlet of the upstream calming section by means of an electromagnetic driver connected to an 8-in. long tubular extension. The driver was actuated through a 20-w. audio amplifier by a variable sinusoidal audio-signal generator. Electrical wave form and relative voltage amplitude were indicated by an oscilloscope connected in parallel with the driver. By always maintaining a sinusoidal wave form on the oscilloscope screen this voltage amplitude was not permitted to exceed the undistorted response of the amplifier. Frequency was simply determined by the setting of the signal generator which in turn had been calibrated against the 60 c./sec. line frequency by means of Lissajous patterns. A water cooling jacket just downstream from the driver took up the waste heat it generated. Comparison of temperatures indicated by the two upstream thermometers showed that this cooler functioned satisfactorily.

Amplitude of vibration within the exchanger along its entire length was measured in terms of the variation in local pressures. For this purpose a traveling 8-ft. long pressure probe of 1/8-in. O.D. stainless steel tubing was used. It was connected to an inclined, open ended, water manometer. The probe lay longitudinally along the bottom of the core with its inlet facing upstream. Tests with a similar but more slowly responding 1/16-in. O.D. probe gave substantially the same results, indicating the 1/8-in. probe was of sufficiently small diameter not to affect the measurement or system appreciably.

Steady state was generally achieved within 30 min. to 2 hr.

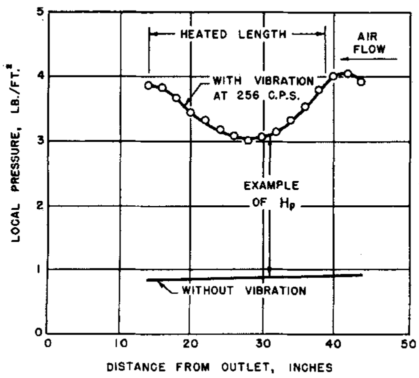

Fig. 3. Example of variation in local pressure along the heated length of the exchanger. Reynolds number equals 1,440.

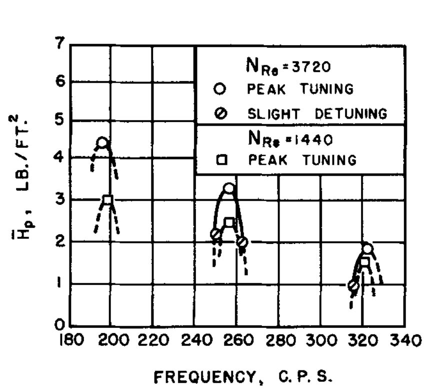

Fig. 4. Typical amplitudes at maximum sinusoidal voltage input to the driver.

RESULTS

Control Runs

Control runs without vibration were first carried out. After the relatively large heat losses to the surroundings (measured with runs of zero air flow) were subtracted, heat balances closed with an average difference of about 10%. Heat transfer coefficients were calculated by Equation (1):

$$h = \frac{w\, c_p\, \Delta t}{A\, \Delta t_m} \qquad (1)$$

The very small condensing steam film and wall resistances were neglected. Results of these control runs are shown in Figure 2, together with the well known Dittus-Boelter equation ($N_{Pr} = 0.7$) for turbulent flow.

Acoustic Properties

Despite the somewhat complicating attachments, such as the branch tubes and mixing section which absorbed some acoustical energy, and despite the superposed steady state air flow, the system behaved essentially as a simple vibrating air column. This was demonstrated by the traveling pressure probe from which curves, such as that of Figure 3, were developed. Except at or near resonance no appreciable sound pressure could be measured. Peaks of the resonance pressure wave correspond to quiescent regions, while troughs correspond to regions of maximum vibrating air motion, either positive or negative in direction. Thus the pressure wave length is half the sound wave length.

The wave length and frequency of sound are related by Equation (2). The condition for resonance in an open ended air column is given by Equation (3):

$$c = \lambda F \qquad (2)$$

$$\frac{L}{\lambda} = \frac{n}{2} \qquad (3)$$

Now, resonance was obtained at or very close to 198, 256, and 322 c./sec. Taking the velocity of sound as 1,200 ft./sec. at average experimental conditions, and substituting in Equations (2) and (3), one can find an effective open ended vibrating length very nearly equal to the 112 in. of exchanger plus calming sections plus driver extension, with n very close to 3, 4, and 5. Accordingly the three resonant frequencies correspond to the third, fourth, and fifth harmonics respectively.

The first and second harmonics could not be obtained owing to driver cutoff at low frequencies. Attempts to force the driver only produced a fuzzy tone

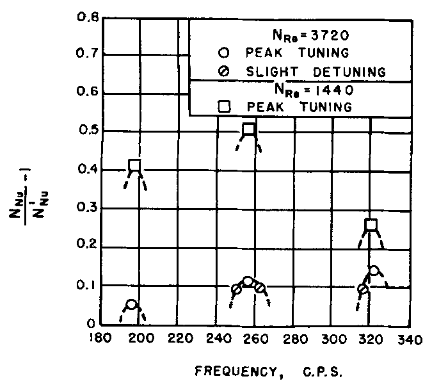

Fig. 5. Typical improvements in heat transfer at maximum sinusoidal voltage input to the driver.

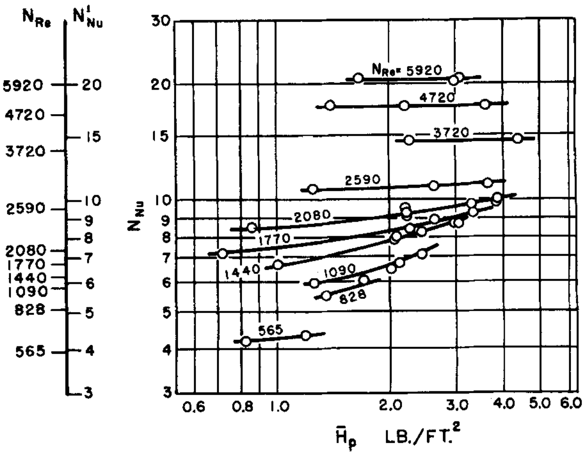

Fig. 6. Heat transfer with vibration at 198 c./sec. Control runs without vibration are indicated for comparison.

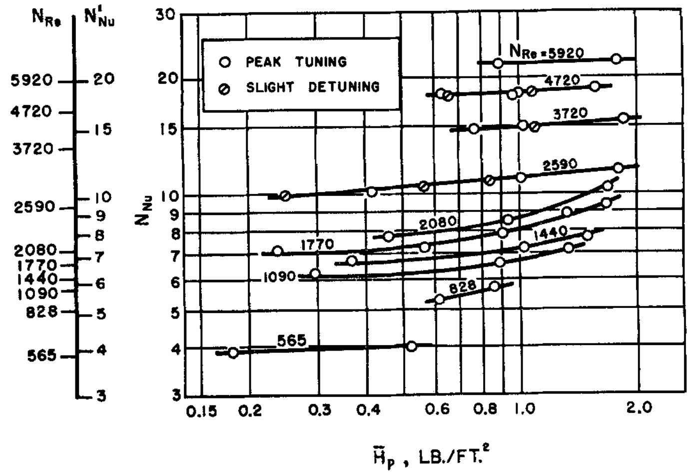

Fig. 8. Heat transfer with vibration at and near 322 c./sec. Control runs without vibration are indicated for comparison.

with little vibrational pressure rise in the probe. For the third, fourth, and fifth harmonics at a constant voltage input to the driver the net mean-pressure amplitude decreased as resonant frequency increased. ($\overline{H}_p$ is an indirect measure of the disturbance due to vibration at a given frequency and N_{Re}.)

Figure 4 illustrates this amplitude decrease for the case where the voltage input was at the maximum sinusoidal level. This limiting voltage level was independent of frequency. The effect of slight detuning from resonance by ±6 c./sec. is also illustrated. Raising the resonant frequency above the fifth harmonic decreased $\overline{H}_p$ still further. This came about chiefly from the increase in loss of sonic energy from the far end of a vibrating air column that accompanies an increase in resonant frequency (*12*).

Some blank runs were carried out at resonance with high amplitude but with the steam shut off. For this case the vibration caused only a small rise in air temperature across the exchanger. This indicates that any conversion of sonic energy to heat within the exchanger is small compared with the heat transferred from steam during the regular runs.

Runs with Vibration

Runs with vibration covered an N_{Re} range of 560 to 5,900 and an $\overline{H}_p$ of 0.18 to 4.4 lb./sq. ft. Frequencies were mostly those of resonance discussed above and their side bands of ±6 c./sec. However a few runs at nonresonant frequencies were also carried out. These yielded no significant $\overline{H}_p$, change in outlet radial temperature profile (preliminary series), or change in *h*. This was the case even at the maximum sinusoidal voltage input to the driver. Thus without resonance insufficient sonic energy is built up within the exchanger to disturb the fluid appreciably and affect the rate of heat transfer.

At resonance the vibration causes significant disturbance. This is believed to occur by two mechanisms. First the vibratory motion is a disturbance in and of itself; Second the vibration acts as a turbulence trigger in the flowing air.

For $N_{Re} < 2{,}100$, $\overline{H}_p$ and the minimum in H_p increase with N_{Re} as well as with voltage input. Thus in the nominally laminar region increasing N_{Re} apparently increases the influence of vibration in causing disturbance. This accords with the second mechanism. On the other hand for $N_{Re} > 2{,}100$ increasing N_{Re} generally decreases $\overline{H}_p$ somewhat and diminishes the crest to trough values of H_p. Thus in the nominally turbulent region increasing the natural turbulence appears to scatter the standing wave and diminish its effect.

The effect on heat transfer is illustrated in Figure 5 for the same typical runs that are presented in Figure 4. The similarity in many respects with Figure 4 itself is evident. Resonant vibration increases N_{Nu} by as much as 51% for the region below N_{Re} of 2,100 and by as much as 27% (not shown) above N_{Re} of 2,100. Detuning decreases improvement. Further comparison of the

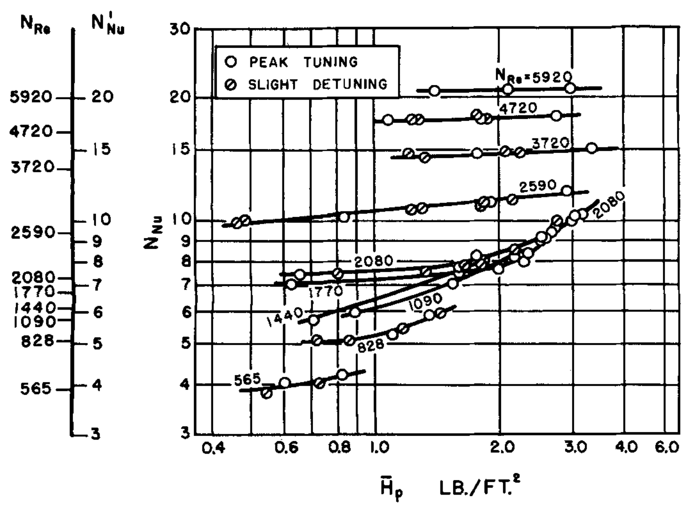

Fig. 7. Heat transfer with vibration at and near 256 c./sec. Control runs without vibration are indicated for comparison.

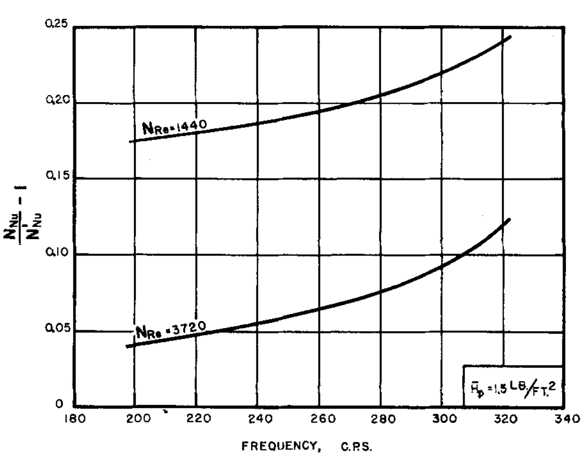

Fig. 9. Typical cross plot illustrating the effect of resonant frequency on the improvement in heat transfer.

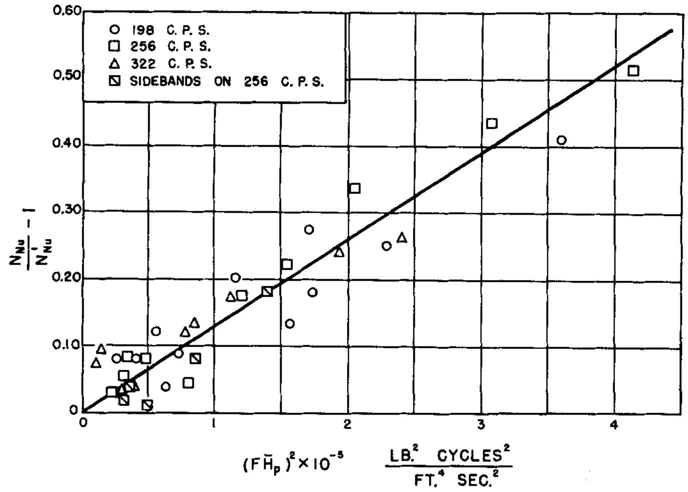

Fig. 10. Correlation of results for Reynolds numbers below 1,500.

two regions shows that for $N_{Re} < 2{,}100$ the improvement at resonance is generally the greater and for fixed frequency and voltage input it generally increases with N_{Re}, particularly at low N_{Re}. For $N_{Re} > 2{,}100$ the improvement is generally the lesser and for the same parameters decreases with N_{Re}. These observations also accord with the two mechanisms proposed.

The use of $\overline{H}_p$ as a measure of amplitude reduces the effect of N_{Re} particularly in the nominally laminar region. This is shown in Figures 6, 7, and 8 which represent N_{Nu} vs. $\overline{H}_p$ for the entire main series of runs. Tabular presentation of data is also available (4).

For fixed frequency and N_{Re}, N_{Nu} increases with $\overline{H}_p$. For $N_{Re} < 2{,}100$ the effect becomes very pronounced at higher values of $\overline{H}_p$. This is also in accord with the notion of turbulence or disturbance promotion. At low $\overline{H}_p$ the disturbance contributed by the vibration is small compared with that otherwise present. Only at higher $\overline{H}_p$ and particularly in the nominally laminar region does the vibratory disturbance become relatively important.

Figure 5 shows that increasing frequency at $N_{Re} > 2{,}100$ increases the improvement in N_{Nu}. At $N_{Re} < 2{,}100$ this is not the case. However if comparison is made at the same $\overline{H}_p$ from Figures 6 to 8 (rather than at the same voltage input from Figure 5), the improvement generally increases with frequency in both regions. A representative cross plot is shown in Figure 9. These observations demonstrate the utility of $\overline{H}_p$ as a measure of amplitude. They also accord with the notion of disturbance promotion. Increasing the number of times per second that a momentary disturbance of given amplitude occurs is likely to increase the over-all effect and hence increase the improvement in heat transfer.

Finally the few runs at the side-band frequencies cluster reasonably well about their respective parent curves in Figures 7 and 8, in contrast with the more marked drops in N_{Nu} shown for these side bands in Figure 5. This further demonstrates the usefulness of $\overline{H}_p$, while it shows the relative insensitivity of N_{Nu} to small changes in frequency at constant $\overline{H}_p$.

No heat transfer effect due to longitudinal shift, with frequency, of standing wave location could be isolated, despite the fact the heated length was only of the same order as the length of the pressure wave. Apparently a still smaller heated length would be required to show any such effect within the precision of this experiment.

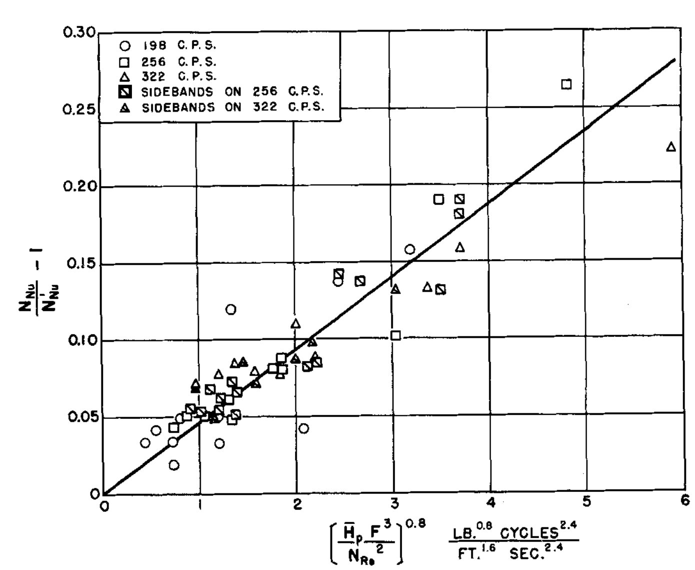

Fig. 11. Correlation of results for Reynolds numbers above 2,500.

Comparison with Literature

The work of Jackson et al. mentioned previously is sufficiently similar to the present investigation to warrant some direct comparison. In that study the effect of acoustic vibration on combined forced and free convection was measured in a vertical, double pipe, steam to air, exchanger. Sound, at frequencies ranging from 125 to 2,400 c./sec. and measured upstream intensities of up to 138.6 decibels, was impressed on air flowing at N_{Re} of 1,170 to 4,750 (but with most runs in the approximate N_{Re} neighborhood of 2,300). No significant effect was found at nonresonant frequencies. At resonance improvements in h of up to 120% were reported. These improvements increased with sound intensity; above N_{Re} of 2,300 they decreased abruptly with N_{Re}. These observations are qualitatively in accord with those of the present study.

Jackson et al. also report that for a given N_{Re} and upstream intensity changing the resonant frequency apparently has little effect on the improvement in h, or at most a slightly inverse effect. This would appear to contradict the findings of the present investigation. However the discrepancy may be at least partially explained in terms of the very different means employed to define and measure amplitude, which of course result in different criteria for constancy of amplitude. Jackson et al. measured amplitude as the sound intensity at only one fixed upstream point. On the other hand in the present study the amplitude is taken from the complete pressure wave within and along the entire heated portion of the exchanger. This is believed to be a more representative measure.

As indicated earlier the use of acoustic frequencies in the present study precludes detailed comparison with earlier heat transfer work in the subsonic region. However to view the present results in the light of certain investigations in acoustics itself (1, 14) is of interest. Such investigations have shown that resonance within a tube sets up stable vortices. Next to the wall there is a boundary layer, the thickness of which decreases as frequency increases. Now, it is reasonable to believe that under conditions of a superimposed steady flow something of this pattern remains if N_{Re} is not too great, while at high N_{Re} the natural turbulence destroys the pattern. In any case the results of the present investigation do not contradict such a physical picture; rather they lend some support. For example the increase in heat transfer improvement with frequency accords with a decrease in thickness of the frictional layer adjacent to the wall.

Correlation

Attempts to develop a simple over-all correlation applicable to the results at all N_{Re} proved unsuccessful. The differences between laminar and turbulent flow, as well as the complicated nature of the interaction among variables in the neighborhood of N_{Re} of 2,100, are too involved. Nevertheless a pair of correlations have been devised, which together cover the major portion of the nominally laminar and turbulent regions.

The correlations give the fractional improvements in N_{Nu} rather than absolute N_{Nu}. As has been pointed out (6) this permits some cancellation of experimental error and makes for somewhat greater generality. They are shown in Figures 10 and 11. The equations are as follows:

For $N_{Re} < 1500$

$$\frac{N_{Nu}}{N'_{Nu}} - 1 = 1.3 \times 10^{-6} (\overline{H}_p F)^2 \quad (4)$$

For $N_{Re} > 2500$

$$\frac{N_{Nu}}{N'_{Nu}} - 1 = 0.047 \left(\frac{\overline{H}_p F^3}{N^2_{Re}} \right)^{0.8} \quad (5)$$

The term $\overline{H}_p F^3/N^2_{Re}$ is a measure of the ratio of vibrational disturbance to inherent turbulence. It shows the decrease in vibrational effect as N_{Re} increases above 2,500. Of course care should be taken in extending either of these correlations much beyond the experimental range of this investigation.

If the improvement in N_{Nu} depends only on vibrational variables and flow variables, the term $\overline{H}_p F$ in Equation (4) can be generalized by dimensional analysis to the dimensionless group $D^4 \overline{H}_p \omega \rho^2/\mu^3$ with appropriate change in the multiplying constant. The relatively large positive exponent on D as well as the sign of the exponents on ρ and μ would seem to accord qualitatively with the physical picture mentioned above. Similarly the term $\overline{H}_p F^3/N^2_{Re}$ in Equation (5) can be generalized dimensionlessly to $D^6 \overline{H}_p \omega^3 \rho^2/V^2 \mu^3$ which can also be partly written in terms of the first dimensionless group, viz. $(D^4 \overline{H}_p \omega \rho^2/\mu^3) \times (D\omega/V)^2$. However these generalizations are largely speculative. The restriction in the present work to a single diameter and a very narrow range of physical properties precludes checking the effect of these variables with the data at hand.

SUMMARY OF CONCLUSIONS

1. Acoustic vibration can substantially improve the rate of forced convective heat transfer, increases of up to 51% in N_{Nu} having been obtained.
2. The improvement peaks sharply at resonance and generally increases with both amplitude and resonant frequency.
3. The improvement peaks asymmetrically at a flow rate below the nominally critical N_{Re} of 2,100 and is generally greater below N_{Re} of 2,100 than above.
4. Amplitude is better measured along the entire length of the exchanger rather than at one fixed point.
5. The improvement in heat transfer is believed to result from a dual mechanism of contributory disturbance, the first contribution being the vibration itself and the second its effect as a turbulence trigger.
6. Equations (4) and (5) correlate the present results over most of the N_{Re} range.

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NOTATION

A = inner heated surface of the core, sq. ft.
c = velocity of sound, ft./sec.
c_p = heat capacity at constant pressure, BTU/lb. mass °F.
D = inner diameter of the core, ft.
F = frequency, c./sec.
h = individual heat transfer coefficient, BTU/hr. sq. ft. °F.
H_p = local pressure rise resulting from vibration, lb. force/sq. ft.
$\overline{H}_p$ = root-mean-square (taken along the heated length) of H_p, lb. force/sq. ft.
k = thermal conductivity, BTU/hr. ft. °F.
L = effective length of a vibrating air column, ft.
n = positive integer, dimensionless
N_{Nu} = Nusselt number, hD/k, dimensionless
N'_{Nu} = Nusselt number without vibration but at the same N_{Re} as that with vibration, dimensionless
N_{Pr} = Prandtl number, $c_p\mu/k$, dimensionless
N_{Re} = Reynolds number, $DV\rho/\mu$, dimensionless
Δt = temperature rise upon heating, °F.
Δt_m = logarithmic mean temperature difference between steam and air, °F.
V = average linear velocity, ft./hr.
w = air flow rate, lb. mass/hr.

Greek Letters

μ = dynamic viscosity, lb. mass/ft. hr.
π = 3.14159. . ., dimensionless
ρ = density, lb. mass/cu. ft.
ω = angular frequency, $2\pi F$, radians/sec.
λ = wave length of sound, ft.

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