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Sonically enhanced heat transfer from a cylinder in cross flow and its impact on process power consumption

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Abstract—Tests were performed to compare the power requirements of various combinations of low frequency sound (infrasound) levels and steady flow rates which achieve the same heat transfer rates. The experiments used steady flow rates up to $Re = 50\,000$. The sound levels ranged from 153 to 170 dB at a frequency of approximately 18 Hz. For the range of parameters tested, the results indicated that the power requirements for cooling a cylinder with an infrasound system could be 12 to 75% less than a fan system, also, the enhancement in heat transfer provided by the maximum sound level ranged from 5 to 700% depending on the steady flow rate. The applied sound was also found to provide more uniformity in the heat transfer distribution around the cylinder.

INTRODUCTION

IN A NUMBER of electrical utilities, there is a large impetus to minimize the growth of electricity consumption. This has led to investigations of existing and potential electricity conservation options in the industrial, commercial, and residential sectors. The end use of electricity is composed of many different loads, particularly in the industrial sector. There are, however, a few prominent loads, one of which is fans. It is estimated (from ref. [1]) that the fan load for industrial convection processes (heating, drying and cooling) in the province of Ontario is approximately 400 MW. Techniques that enhance convection could therefore provide significant load reduction.

The focus of this study was on the use of sound for enhancing heat transfer, and the resulting impact on the process power consumption. The sonic frequencies used in the experiments were approximately 18 Hz, which is in the infrasound, or sub-audible, range of frequencies. Infrasound was used for this study instead of higher frequencies for the following reasons.

(1) Industrial infrasound systems are commercially available.
(2) The wavelength of infrasound is long enough for most materials to be within the useful portion of the standing wave (i.e. the section at the pressure anti-node, where the velocity is greatest).
(3) Infrasound is not audible, and therefore at similar pressure levels, infrasound is less of a nuisance in plant environments than audible sound.

The objective of this study was to assess whether sonically enhanced heat transfer from a cylinder uses more or less power than a fan system achieving the same heat transfer rate. It was not the intention of this paper to present any new theories on the mechanism of sonically enhanced heat transfer, but the experimental results are presented primarily to provide input for the development of theories and applications. The range of sound levels used in this study are higher than the levels used in most of the previous literature and therefore provides a different perspective on at least the following two conclusions reached in earlier studies.

Fand and Cheng [2] measured heat transfer from a cylinder in cross-flow at sound levels up to 150 dB and frequencies of 1100 and 1500 Hz. They observed that the enhancement in heat transfer had a minimum at Reynolds numbers between 4500 and 6000, depending on the frequency. Since the Reynolds number corresponding to the minimum enhancement shifted in approximate proportion to the frequency, Fand and Cheng concluded that the phenomenon was due to a resonance excited by the sound.

Anantanarayanan and Ramachandran [3] measured the heat transfer from a vibrating wire to an air stream. The air velocities ranged from 10 to 19 m s^{-1}, and the amplitude of the wire vibration ranged from 3 to 20 mm, at frequencies between 75 and 120 Hz. They observed that heat transfer increased gradually as the oscillating velocity was increased and concluded that a vibrational Reynolds number based on the mean oscillating velocity could be used to predict the effect on heat transfer.

APPARATUS AND PROCEDURE

A sketch of the apparatus which was used for measuring the effect of infrasound on heat transfer is shown in Fig. 1. The infrasound was generated using a two

NOMENCLATURE

c	speed of sound
COP	coefficient of performance
d	cylinder diameter
e	d.c. voltage on heater
h	heat transfer coefficient
i	d.c. current through heater
k	thermal conductivity
Nu	Nusselt number (hd/k)
p_{rms}	root-mean-square air pressure
Pr	Prandtl number
Q_c	total convection heat transfer
Q_d	total heat dissipated
Q_r	total heat radiated to environment
Re_{sf}	steady flow Reynolds number, ud/ν_k
Re_{res}	Reynolds number based on resultant velocity, $V_{res}d/\nu_k$
S	skewness in oscillating velocity
T	period of oscillation
T_s	surface temperature
T_a	free stream air temperature
u, v	perpendicular velocity components
V_{res}	resultant of two perpendicular velocity components.

Greek symbols

ε	emissivity of surface
ρ_0	air density
σ	Stefan–Boltzmann constant
ν_k	kinematic viscosity.

Subscripts

p	peak value (of oscillating variable)
rms,i	indicated root-mean-square value
rms,a	actual root-mean-square value.

cylinder air compressor, and two quarter wavelength resonators made from 7.6 cm diameter ABS (acrylonitrile–butadiene–styrene) pipe. The air compressor operated such that the pistons were 180° out of phase, and the cylinder head was replaced with a plate incorporating the resonator pipes. A variable speed d.c. motor was used to control frequency, and thus establish resonance in the system. The oscillating velocities in the test section were controlled by changing the motor speed, which brought the system on, or slightly off resonance. This system resulted in relative velocity and pressure distributions in the resonators that were approximately as shown in Fig. 2.

The cylinder used to measure heat transfer consisted of a wooden dowel (3.2 cm diameter) covered by a film heater and thin RTDs (resistance temperature detectors) as shown in Fig. 1(b). The film heater had a constantan element, therefore its resistance did not change with temperature. The constant resistance allows the local heat dissipation to be constant around the cylinder, regardless of the surface temperature distribution. The emissivity of the cylinder was measured with an Agema infrared camera. The convection heat transfer rate (Q_c) from the cylinder was the difference between the total heat dissipated (Q_d) and the heat radiated (Q_r). The total heat dissipated was simply the product of the d.c. voltage (e) and current (i), and the total heat radiated from the surface was found as

$$Q_r = \varepsilon\sigma A_s[(T_s+273)^4-(T_a+273)^4]. \quad (1)$$

The heat transfer coefficient (h) was then found as

$$h = \frac{Q_c}{A_s(T_s-T_a)} = \frac{ei-\varepsilon\sigma A_s[(T_s+273)^4-(T_a+273)^4]}{A_s(T_s-T_a)}. \quad (2)$$

The steady air flow was provided by a blower through a chamber, and into the test section through a 7.6 cm diameter ABS pipe. The chamber was necessary to provide a uniform velocity profile. The steady air velocity was monitored during the tests using a pitot–static tube. The uniformity of the flow was measured with the pitot–static tube and found to be constant through approximately 90% of the pipe diameter.

The sound pressure level in the system was measured near the pistons in the resonators (as shown in Fig. 1(a)), using a model 800B Larson Davis sound meter with a 6.3 mm microphone. Since the sound pressure levels approached 170 dB, an attenuator was constructed so that the microphone would not be damaged while taking the measurement. The attenuator consisted of plastic tubing connected to a brass coupling filled with tissue. The attenuation provided was between 31 and 34 dB depending on the frequency. Before each test, the attenuator was calibrated throughout the range of infrasound frequencies used.

The oscillating air velocity was measured with a hot wire anemometer (Dantec model 56C01) in place of the cylinder, and was recorded along with the corresponding sound pressure level measured at the pressure node. The velocity was then established during the heat transfer tests using the corresponding sound pressure level. With no steady flow applied, the velocity distribution of the oscillating flow was checked with the anemometer and found to be uniform across the test section of the cylinder.

Since the anemometer can only measure the magnitude of the velocity (i.e. it does not detect direction), an oscillating velocity with no steady component will appear rectified. The indicated mean and r.m.s. velocities are therefore not the actual mean and r.m.s.

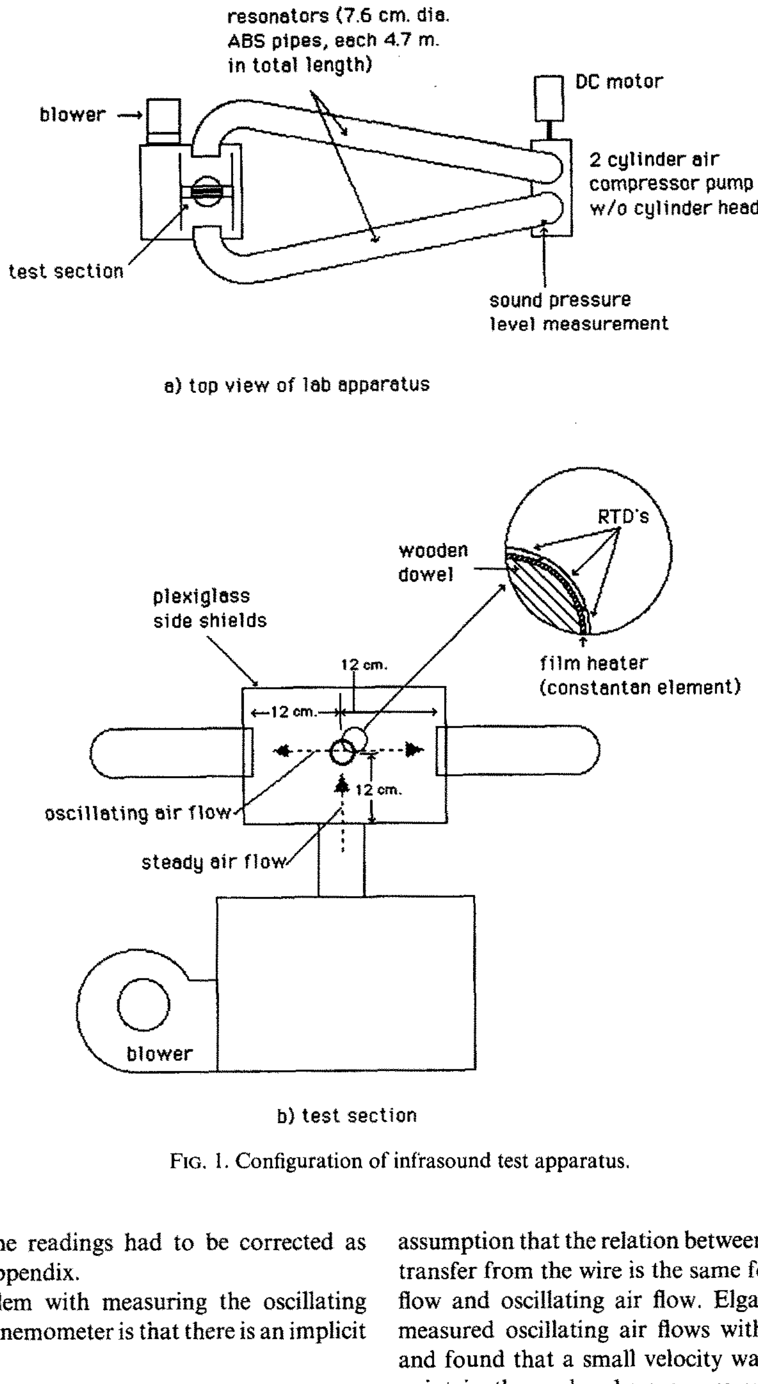


FIG. 1. Configuration of infrasound test apparatus.

velocities, and the readings had to be corrected as outlined in the appendix.

Another problem with measuring the oscillating velocity with an anemometer is that there is an implicit assumption that the relation between air flow and heat transfer from the wire is the same for both steady air flow and oscillating air flow. Elgar and Adams [4] measured oscillating air flows with an anemometer and found that a small velocity was indicated at the point in the cycle where a zero velocity should be indicated. They therefore concluded that an oscillating flow had to be calibrated separately from steady flow. Bremhorst and Gilmore [5] found, however, that the static and dynamic calibrations were identical within 2%. In the current lab tests, an approximate check of the anemometer was provided by converting the sound pressure level at the pressure node to a velocity at the anti-node using the following formula (from ref. [6], p. 42)

$$v_{\mathrm{rms}} = \frac{p_{\mathrm{rms}}}{(\rho_0 c)} \tag{3}$$

where p_{rms} is the r.m.s. pressure at the pressure node,

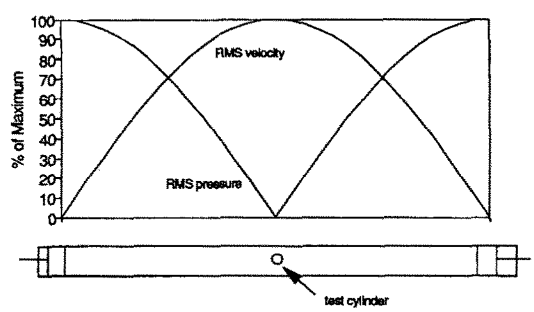


FIG. 2. Distribution of air pressure and velocity along the length of the resonator.

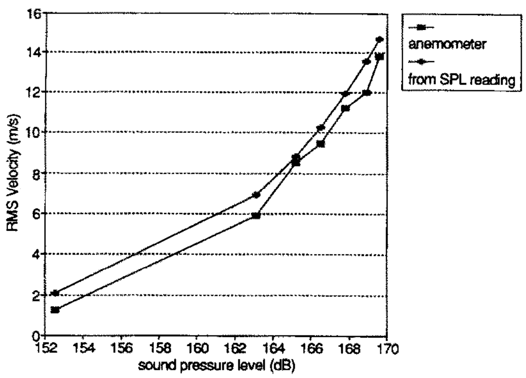


FIG. 3. Independent indications of the oscillating velocity using an anemometer and the sound pressure level.

v_{rms} is the r.m.s. velocity at the pressure anti-node, ρ_0 is the air density, and c is the speed of sound in the air. The results of these tests are shown in Fig. 3. As expected, the velocity indicated by the sound pressure level is a little bit higher than the velocity measured with the anemometer. This difference is to be expected because the small expansion of the air (as it passes through the test section—see Fig. 1) is not accounted for in the sound level. At the lower velocities, the relative difference is larger than at the higher velocities, probably since the air will disperse to a higher extent at low velocities.

In order to assess whether the increase in heat transfer can be accounted for by the increase in velocity or kinetic energy, a total air velocity was defined which is a simple vector addition of the steady and oscillating components (see Fig. 4). The total velocity is not intended to describe the pattern of air flow around the cylinder. It will allow, however, an assessment of the conclusion of Anantanarayanan and Ramachandran [3] that the enhancement of heat transfer could be accounted for by the mean velocity. The resultant of the two velocities was found as

$$\overline{V_{res}} = \left(\frac{1}{T}\int_0^T [(v_p \sin \omega t)^2 + u^2]\,\mathrm{d}t\right)^{1/2} = (v_{rms}^2 + u^2)^{1/2} \qquad (4)$$

where T is the period of the oscillating flow, u is

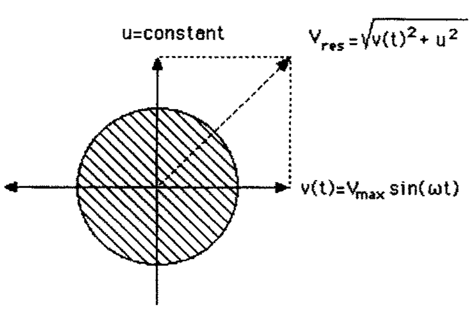


FIG. 4. Transverse velocity components in the air stream.

the steady velocity, and v represents the oscillating velocity.

ASSESSMENT OF MEASUREMENT UNCERTAINTY

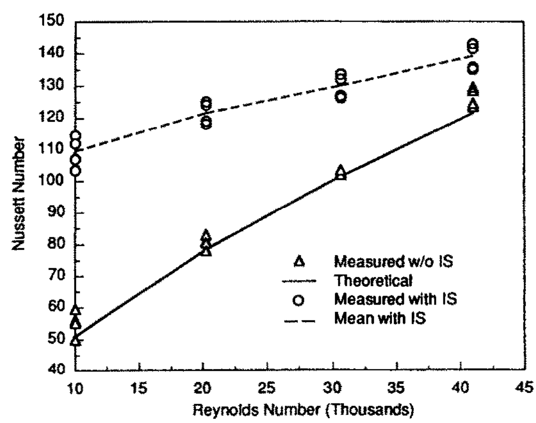
The measurement uncertainty in the lab tests was determined by measuring the heat transfer at four different steady flow velocities, both with and without infrasound. This set of tests was repeated on four separate occasions.

The uncertainty in the measurements consisted of two components; a random error and a fixed error. The random error is indicated by the standard deviation of the results at a certain condition. The fixed error is a systematic error which remains constant during a test. The fixed error can be estimated by comparing the mean results with theoretical or standard published results.

The results from the four repeatability tests are shown in Fig. 5. The random error is indicated in these figures as the standard deviation at each steady flow rate (Re). The standard deviation is expressed as a percentage of the mean value. With the exception of the lowest steady flow rate, the random error corresponding to a 95% confidence level was, therefore, less than 6%. In the worst case, the random error corresponding to the 95% confidence level was 15%.

The fixed error can be seen in Fig. 5 by comparing the measured heat transfer in steady flow, with the theoretical results. The theoretical values were calculated using the following formula from [7]

$$Nu = C Re^n Pr^{1/3} \qquad (5)$$



Re	Standard Dev % with IS	Standard Dev % w/o IS
10000	4.5	7.3
20250	2.8	2.5
30600	2.8	0.6
41000	2.9	2.2

FIG. 5. Repeatability of heat transfer measurements with and without infrasound (IS), and comparison of steady flow heat transfer measurements with theoretical values.

Table 1. Constants for use with equation (5)

Re	C	n
40–4000	0.683	0.466
4000–40 000	0.193	0.618
40 000–400 000	0.0266	0.805

where n and C are constants (listed in Table 1) which are dependent on the Reynolds number.

RESULTS AND DISCUSSION

The main results from the heat transfer measurements are shown in Fig. 6. Figure 6 shows the Nusselt number in different steady flow velocities (indicated as the steady flow Reynolds number) with different levels of infrasound applied. The infrasound level is indicated as the r.m.s. velocity of the oscillating air flow. The data points in Fig. 6 represent the average heat transfer measured in two separate tests at the same conditions. Averaging the data did not change any trends in the data.

One trend that is apparent in Fig. 6, is that as the steady flow Reynolds number increases with constant sound levels, the heat transfer reaches a minimum near a Reynolds number of 5000. This seems to be consistent with the results of Fand and Cheng [2] in which the heat transfer enhancement reached a minimum in the range from $Re = 4500$ to $Re = 6000$ at frequencies of 1100 and 1500 Hz, respectively. Fand and Cheng concluded that since the heat transfer minimum seemed to depend on the frequency, the effect must be due to a resonance in the flow. The current results, achieved at 18 Hz, suggest, however, that the steady flow Reynolds number corresponding to the minimum heat transfer, may be independent of frequency.

In general, it is not believed that resonance occurred in the flow during any of the tests. To check for resonance in the flow, the anemometer was used to look at the frequency spectrums in the flow at various locations around the cylinder. The frequency spectrum was looked at for a number of steady flow rates and infrasound intensities (note that the infrasound intensity was adjusted by slightly varying the frequency). None of the frequency spectrums observed had any detectable differences.

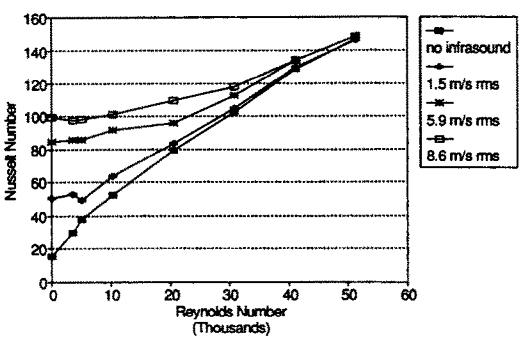
The main objective of this study was to estimate the impact on power consumption from using infrasound to enhance heat transfer. Simple measurements of the electricity consumption would, however, be misleading since the efficiencies of the lab equipment used will not be typical of full scale equipment. The assessment of the power requirement was, therefore, divided into two components: the effect of steady and oscillating velocities on heat transfer, and the electrical power required by the driver (e.g. fan or infrasound source) to produce the velocities.

For a fan, the input power is roughly proportional to the velocity cubed. The instantaneous friction loss in an oscillating air flow will also be approximately proportional to the instantaneous velocity cubed. A dimensionless coefficient of performance (COP) was, therefore, defined to relate the input power and the air velocities (or fluid power requirement), such that

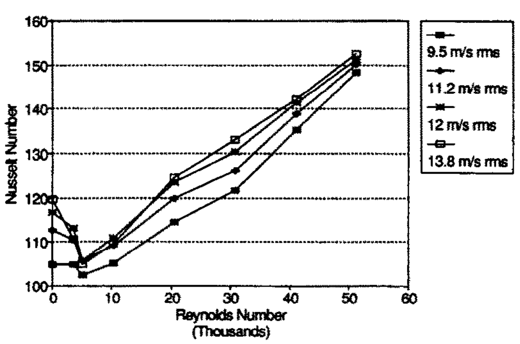
$$\mathrm{COP} = \frac{\frac{1}{2}\rho A \bar{V}^3}{P_{\mathrm{inp}}} \tag{6}$$

where $\bar{V}^3$ is the mean magnitude of the velocity (oscillating or steady) cubed, A the cross sectional area of the flow, ρ the density of air and P_{inp} the electrical input to the fan or infrasound source. For this study, the numerator in equation (6) is used to represent the power requirements of the oscillating flow and the steady flow. Admittedly, the accuracy of this assumption is uncertain, however the main purpose of defining the COP is to relate the input power and velocity level achieved with a certain driver (i.e. fan or infrasound source). This will ultimately link the input power to the heat transfer rate and will also allow the analysis to be updated as the expected driver performance is changed.

The relation in equation (6) was defined as a coefficient of performance instead of an efficiency,



a)



b)

FIG. 6. Heat transfer measurements at different steady velocities and different infrasound intensities (stated as r.m.s. velocity).

because an infrasound system could have a COP ratio greater than one. The COP could be greater than one because a standing wave has no inherent power (ref. [6], p. 42); in a closed infrasound system, all the power is used to overcome the losses, because the kinetic energy stays within the system. If, therefore, the losses could approach zero, the COP would approach infinity.

To assess the power requirements of an infrasound system compared to a fan system, the different combinations of steady and oscillating velocity which achieve similar rates of heat transfer were estimated. The estimation of the power requirement was done by interpolating from the results in Fig. 6. Using Fig. 6, a line of constant Nusselt number could be drawn and the points of intersection indicate the different combinations of steady flow and infrasound which achieve the same heat transfer rate. The total power requirement could then be calculated for each combination of steady and oscillating flow. It was discovered that the savings in the total required power varied with the level of infrasound used. As illustrated in Fig. 7, the relative fluid power savings reached maximum levels from 20 to 80%, depending on the Nusselt number.

This study did not include an assessment of the COPs for infrasound and fan systems. The COP for a fan system will, however, be less than 0.7 (a good efficiency for a fan and motor combination), and it is estimated, using data from ref. [8], that existing infrasound units can have a COP of 0.5. Using these COPs to adjust the data plotted in Fig. 7, the power requirement of the infrasound system would be 12 to 75% less than the fan system.

Another way to analyse the power requirements is to plot the amount of displaced fan power against the infrasound power used to maintain the same amount of heat transfer. These results are shown in Fig. 8. If the fan COP and the infrasound COP are equal, then the breakeven line (where infrasound displaces the same amount of power as it uses) can be drawn along

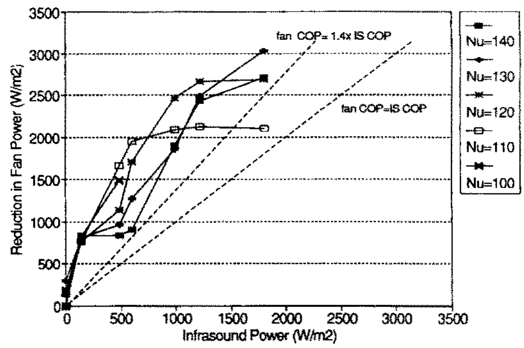


FIG. 8. Reduction in fan power as the infrasound intensity is increased, while maintaining the same net heat transfer. For points above the breakeven (dashed) lines, infrasound displaces a larger amount of fan power.

the diagonal as shown in Fig. 8. For points above the line, the infrasound power displaces a greater amount of fan power. It is also easy to show the breakeven line for any COP ratio (also shown in Fig. 8).

To determine whether the enhanced heat transfer rate can be simply explained by the increased velocity in the system, the resultant velocity was calculated for each test as the mean vector sum of the steady and oscillating velocity (as shown in Fig. 4). As mentioned earlier, the resultant velocity is not intended to describe the pattern of the air flow. Figure 9 shows the heat transfer at various resultant Reynolds numbers (Re_{res}), and each curve represents an increasing sound level with a constant steady flow velocity. The results in Fig. 9 indicate that the increase in heat transfer can not be accounted for by the resultant velocity, or total kinetic energy, around the cylinder. Figure 9 also shows that for $Re_{res} < 40\,000$ the marginal increase in heat transfer is greater with infrasound than steady flow. While for $Re_{res} > 40\,000$ the marginal increase in heat transfer is greater with steady flow than with infrasound.

The fact that heat transfer was increased beyond the level that would be expected using the resultant velocity, again brings into question the validity of using an anemometer to measure the oscillating vel-

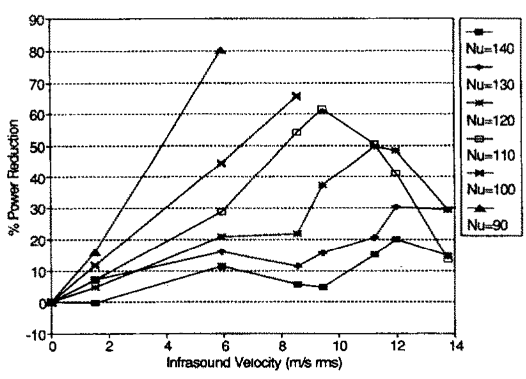


FIG. 7. Reduction in total power consumption (fan + infrasound power) as the infrasound level is increased and displaces fan power, while maintaining the same heat transfer rate.

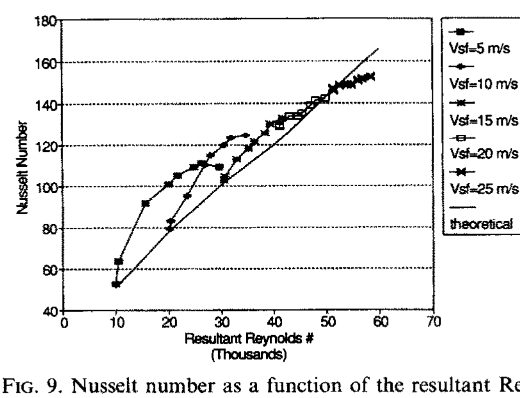


FIG. 9. Nusselt number as a function of the resultant Reynolds number (based on the resultant of the perpendicular steady and oscillating velocities).

ocity. There are, however, two distinctions between the above heat transfer results and the velocity measurements.

(1) The probe wire on the anemometer has a diameter of 5 μm and the diameter of the test cylinder was 3.5 cm.
(2) The experimental heat transfer results were obtained with both steady flow and oscillating flow, while the anemometer measured the oscillating velocity without a steady flow component.

Currently, it is not known how much either of these two factors would influence the results.

In much of the previous literature on the effect of sound on natural convection heat transfer, the results are plotted as the ratio of heat transfer with sound to that without sound, vs the sound pressure level. In Fig. 10 the current results are plotted in these coordinates to allow comparison with results published by Fand [9] using sound at approximately 1500 Hz. Although the sound pressure levels in the current study did not overlap with the levels in Fand's study, the results line up well.

The cylinder used for measuring heat transfer had eight RTDs around the circumference which provided a rough indication of the local heat transfer around the cylinder. The variation of the Nusselt number around the cylinder's circumference is shown in Fig. 11 for various steady flow rates and infrasound levels. As shown in Fig. 11(b), the heat transfer at 0° (the stagnation point for steady flow) stays approximately constant as the infrasound level is increased. At a certain infrasound level (which varies with the steady flow velocity) the heat transfer at 0° decreases slightly as the sound level increases (shown in Fig. 11(c)). Beyond this infrasound level, the heat transfer around

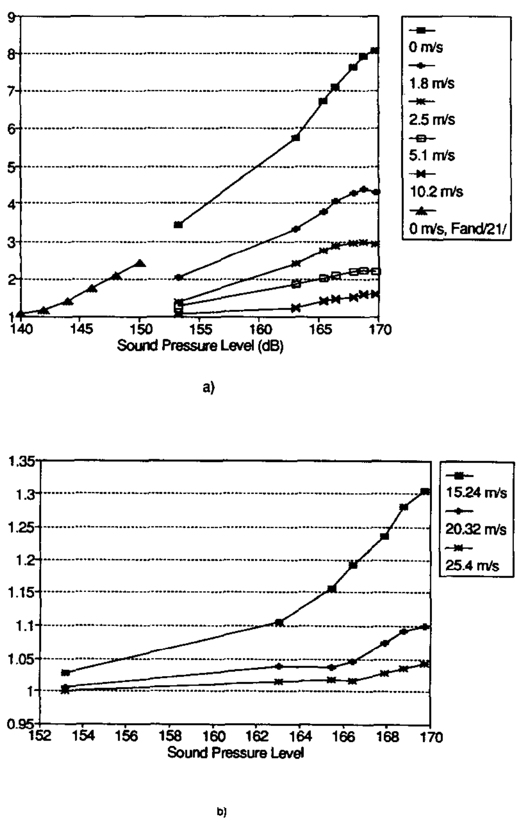


FIG. 10. Enhancement ratio (Nu_{IS}/Nu_0) at different steady flow rates as the sound pressure level is increased.

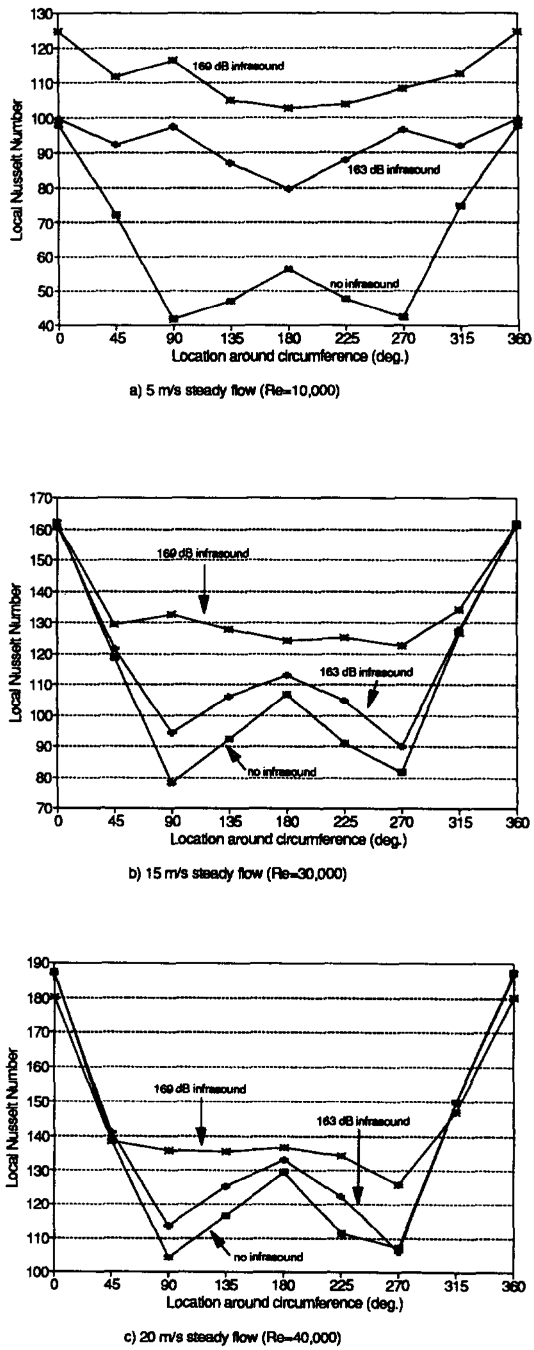


FIG. 11. Distribution of the Nusselt number around the circumference of the cylinder.

the circumference increases more uniformly (shown in Fig. 11(a)). These trends were consistent though all the results with the exception that at higher steady flow velocities, greater sound pressure levels are needed to progress the change in the heat transfer distribution.

Also evident in Fig. 11 is an asymmetry in the heat transfer when the maximum sound level is applied. The asymmetry may have been due to different levels of flow losses in the two resonators, which would result in slightly different velocities on the two sides of the test cylinder. The non-symmetric oscillating velocity could not, however, be confirmed with the hot wire anemometer; it would be seen as a higher peak velocity in one half of the cycle. Since an asymmetry did not appear in the oscillating flow, the asymmetry in the heat transfer may be due to the combination of the steady and oscillating flow. This was also supported by the observation that the asymmetries were most noticeable with the highest infrasound level.

CONCLUSIONS

The current study looked at the effect of using intense infrasound to enhance heat transfer from a cylinder. The main conclusions from the experimental results are as follows.

(1) Infrasound provided significant increases in the heat transfer from a cylinder in a steady flow. The most significant increases in heat transfer occur with steady flow Reynolds numbers less than 40 000. The increase in heat transfer could not be accounted for by just the increase in kinetic energy in the air as suggested by Anantanarayanan and Ramachandran [3].
(2) Combinations of infrasound and steady air flow can reduce the power required by 12 to 75% depending on the heat transfer rate required.
(3) A minimum in the enhanced heat transfer near a steady flow Reynolds number of 5000 does not appear to be due to resonance, as was suggested in previous work by Fand and Cheng [2] at 1100 and 1500 Hz.
(4) The use of infrasound to enhance heat transfer from a cylinder resulted in a more uniform distribution of heat transfer around the cylinder.
(5) The heat transfer distribution around the cylinder becomes slightly non-symmetric as the intensity of the infrasound was increased. The asymmetry may be due to the interaction of the transverse oscillating and steady air flow.

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APPENDIX. CORRECTION OF THE OSCILLATING VELOCITY MEASUREMENTS

When a single wire anemometer is used to measure a flow velocity, only the magnitude of the flow ($|v|$) is indicated and not the direction. In an oscillating flow where the flow reverses, the anemometer signal will appear to rectify the flow velocity reading. This is illustrated in Fig. A1. If the actual flow velocity is represented by $v(t) = v_{max} \sin \omega t$, with no steady flow component, then instead of indicating the actual mean velocity of zero, the anemometer software indicates a mean velocity defined by

$$\bar{v}_i = \frac{1}{T}\int_0^T |v(t)|\,dt = \frac{2}{T}\int_0^{T/2} v_p \sin \omega t\,dt = \frac{2}{\pi} v_p \quad \text{(A1)}$$

where v_i is the indicated mean velocity, v_p the peak magnitude of the oscillating velocity, and T the period of oscillation in the flow.

The anemometer software will then find the r.m.s. velocity using the relation

$$v_{rms,i} = \left(\frac{1}{T}\int_0^T [|v(t)| - \bar{v}_i]^2\,dt\right)^{1/2} \quad \text{(A2)}$$

where $v_{rms,i}$ is the r.m.s. velocity indicated by the anemometer software. However, the actual r.m.s. velocity ($v_{rms,a}$) is

$$v_{rms,a} = \left(\frac{1}{T}\int_0^T v(t)^2\,dt\right)^{1/2}. \quad \text{(A3)}$$

Expanding equation (A2), and using equations (A1) and (A3) gives

$$v_{rms,i} = \left(\frac{1}{T}\int_0^T [|v(t)|^2 - 2\bar{v}_i|v(t)| + \bar{v}_i^2]\,dt\right)^{1/2}$$
$$= (v_{rms,a}^2 - 2\bar{v}_i^2 + \bar{v}_i^2)^{1/2} \quad \text{(A4)}$$

or

$$v_{rms,a} = (v_{rms,i}^2 + \bar{v}_i^2)^{1/2}. \quad \text{(A5)}$$

In order to find the mean magnitude of the velocity cubed ($|v^3|$), the 'skewness' calculated by the anemometer software is utilized. The skewness (S) is calculated by the software as

$$S = \frac{\bar{v}_i^3}{v_{rms,i}^3} \quad \text{(A6)}$$

where the indicated mean cubed velocity is found as

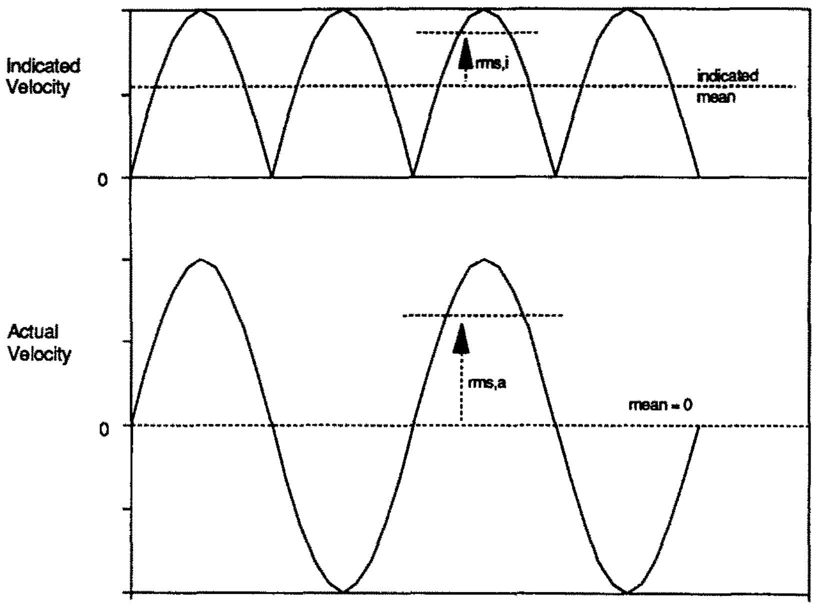

FIG. A1. Anemometer signal in an oscillating flow with no steady flow component.

$$\bar{v}_i^3 = \frac{1}{T}\int_0^T [|v(t)| - \bar{v}_i]^3 \, dt \tag{A7}$$

therefore, expanding equation (A7) and using equation (A6) gives

$$\begin{aligned} Sv_{\mathrm{rms,i}}^3 &= \frac{1}{T}\int_0^T [|v(t)|^3 - 3|v(t)|^2\bar{v}_i + 3|v(t)|\bar{v}_i^2 - \bar{v}_i^3] \, dt \\ &= \overline{|v_a^3|} - 3\bar{v}_i v_{\mathrm{rms,a}}^2 + 2\bar{v}_i^3 \end{aligned} \tag{A8}$$

and using equation (A5) in equation (A8) gives

$$\overline{|v_a^3|} = Sv_{\mathrm{rms,i}}^3 + \bar{v}_i^3 + 3\bar{v}_i v_{\mathrm{rms,i}}^2 \tag{A9}$$

From the indicated mean velocity, indicated r.m.s. velocity, and skewness, it is, therefore possible to find the actual r.m.s. velocity (equation (A5)) and the actual mean cubed velocity (equation (A9)).

ACCROISSEMENT ACOUSTIQUE DU TRANSFERT THERMIQUE AUTOUR D'UN CYLINDRE EN ATTAQUE FRONTALE ET PUISSANCE DEPENSEE

Résumé—Des essais sont conduits pour comparer les puissances requises dans différentes combinaisons de niveaux sonores à basse fréquence (infrason) et de vitesses d'écoulement, pour obtenir le même taux de transfert thermique. Les expériences concernent des débits allant jusqu'à $Re = 50\,000$. Les niveaux sonores varient depuis 153 dB jusqu'à 170 dB à des fréquences proches de 18 Hz. Les résultats montrent que la puissance requise pour refroidir un cylindre par ce procédé peut être inférieure de 12 à 75% au cas de ventilation. L'accroissement du transfert thermique produit par le niveau sonore optimal varie de 5 à 700% selon le débit de l'écoulement. Le son appliqué donne une distribution plus uniforme du transfert thermique autour du cylindre.

VERBESSERUNG DES WÄRMEÜBERGANGS AN EINEM QUER ANGESTRÖMTEN ZYLINDER DURCH SCHALL UND DIE AUSWIRKUNG AUF DEN ENERGIEVERBRAUCH DES VORGANGES

Zusammenfassung—Der Leistungsbedarf verschiedener Kombinationen von Schall geringer Frequenz und stationärer Strömung, die jeweils zu denselben Wärmeübergangskoeffizienten führen, wird experimentell untersucht. Die Reynolds-Zahl der Strömung beträgt maximal $Re = 50\,000$. Der Schallpegel liegt im Bereich zwischen 153 und 170 dB bei einer Frequenz von Näherungsweise 18 Hz. Im untersuchten Parameterbereich zeigt sich, daß der Leistungsbedarf zur Kühlung eines Zylinders mit Hilfe eines derartigen Schallsystems gegenüber einer zwangsweisen Anströmung um 12 bis 75% reduziert werden könnte. Die entsprechende Erhöhung des Wärmeübergangs bei maximalem Schallpegel liegt zwischen 5 und 700%, je nach Strömungsgeschwindigkeit. Es ergab sich ferner, daß die Verteilung des Wärmeübergangs am Zylinder infolge der Beschallung gleichförmiger wird.

ЗВУКОВАЯ ИНТЕНСИФИКАЦИЯ ТЕПЛОПЕРЕНОСА ОТ ОБТЕКАЕМОГО ПОПЕРЕЧНЫМ ПОТОКОМ ЦИЛИНДРА И ЕЕ ВЛИЯНИЕ НА ПРОЦЕСС ПОТРЕБЛЕНИЯ ЭНЕРГИИ

Аннотация—Проводились эксперименты для сравнения энергетических потребностей различных комбинаций низкочастотных (инфразвуковых) уровней звукового давления и стационарных расходов, при которых достигаются одинаковые скорости теплопереноса. В экспериментах использовались стационарные расходы вплоть до $Re = 50\,000$. Уровни звукового давления изменялись от 153 до 170 дб при частоте, равной приблизительно 18 Гц. Показано, что энергетические потребности для охлаждения цилиндра в случае инфразвуковой системы на 12–75% ниже, чем для вентиляционной системы. Кроме того, интенсификация теплопереноса, обеспечиваемая максимальным уровнем звукового давления, варьировалась от 5 до 700% в зависимости от стационарного расхода. Найдено, что применение звука способствуют также достижению большей однородности в распределении теплопереноса вокруг цилиндра.