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THE EFFECT OF SOUND ON THE RATE OF HEAT TRANSFER FROM A CYLINDER PLACED NORMAL TO AN AIR STREAM*

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An experimental investigation has been carried out to study the effect of sound waves with a particular frequency on forced convective heat-transfer from a heated circular cylinder over the Reynolds number range from 5000 to 20000.

The results indicate that the convective heat-transfer coefficient can be influenced by a given sound wave, and in this case it causes the Nusselt number to increase appreciably on the back half of the cylinder. On the other hand, the approach of the vortex region formed in the wake to the cylinder surface plays an important role in the mechanism of an increase in the Nusselt number in separated region.

1. Introduction

When the shedding frequency of a vortex formed in the wake of a flow behind individual tubes happens to be near the natural frequency of a gas column perpendicular to the flow and the tube axis, a strong transverse acoustic oscillation of the gas column will be excited, causing vibration and noise in tube-bank heat exchangers. The studies in connection with the prevention of vibration and noise in heat exchangers have been made by many authors⁽¹⁾. However, nothing has been well known about the vibration-and sound-influenced convective heat transfer rates, and until recently, no investigations have been conducted extensively⁽²⁾.

On the other hand, up to this time, much effort has been expended on studies about the influence of free stream turbulence on heat transfer. These studies show that convective heat-transfer rates generally increase with an increase in turbulence. In these studies, many experiments have been conducted with the aid of a grid-induced isotropic turbulence in the free stream. This time, an attempt to understand the details of the mechanism which causes the effect of free stream turbulence to occur must necessarily be speculative and conjectural, and thus the task of explaining the detailed mechanics of the observed phenomena remains unaccomplished. In spite of this, it is suggested that there should be qualitative similarity between the effects of sound and free stream turbulence. Fand and

Cheng⁽³⁾ introduced a sound field normal to the air stream and obtained improvements in heat transfer and observed that for one applied sound frequency the maximum in heat transfer occurs over a range of Reynolds numbers. But they obtained no apparent convection with the frequency of the applied sound field and heat transfer mechanism in the rear of the cylinder and wake oscillations.

In recent years several investigations have been conducted to measure the effects of sound on flow fields around a circular cylinder in crossflow, and particularly, a study of acoustic interferences in vortex-shedding from the cylinder was made and an artificial control of the laminar-turbulent transition in the separated shear layer by external sound was attempted⁽⁴⁾⁽⁵⁾. It is the main purpose of this paper to investigate the influence of sound on the rate of heat transfer from a heated circular cylinder to air in crossflow when the artificial control of the laminar-turbulent transition in the separated shear layer by sound was attempted according to Yamanaka and Adachi's experiments. An experimental investigation has been carried out to study the effect of sound waves with a particular frequency on forced convective heat-transfer from a heated circular cylinder over the Reynolds number range from 5000 to 20000. By changing the frequencies of the applied sound waves and sound pressure levels, the local Nusselt number distributions around the cylinder were measured in detail. Then, external sound caused major changes in the local heat-transfer rate on the back half of the cylinder, and measurements of the velocity profiles and the temperature profiles in the wake region were conducted. Based on the results, discussions were made about an interaction between the wake motion and the induced oscillations caused by the generation of a sound field in the vicinity of the wake and the heat-transfer mechanism in the separated region.

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Nomenclature

C_p	: local pressure coefficient [$=(p-p_\infty)/(\frac{1}{2}\rho U_\infty^2)$]
d	: cylinder diameter
f_a	: frequency of sound wave
f_n	: natural vortex-shedding frequency
f_t	: frequency of the laminar-turbulent transition wave
Nu	: local Nusselt number ($=\alpha d/\lambda$)
p	: static pressure on the cylinder surface
p_∞	: free-stream static pressure
Pr	: Prandtl number
Re	: Reynolds number ($=U_\infty d/\nu$)
SPL	: sound pressure level
St	: Strouhal number ($=f_n d/U_\infty$)
T	: temperature
T_w	: temperature on the surface of the heated circular cylinder
T_∞	: free-stream temperature
U	: stream velocity
U_∞	: free-stream velocity
u, v, w	: velocity fluctuations in x , y and z -directions, respectively
u_{fa}, v_{fa}, w_{fa}	: acoustic particle velocities with frequency of f_a in x , y and z -directions, respectively
x, y, z	: co-ordinates (see Fig.1)
α	: heat-transfer coefficient
θ	: angle around cylinder from front stagnation line
λ	: fluid thermal conductivity
ν	: fluid kinematic viscosity
ρ	: fluid density
-	: mean

2. Experimental apparatus and instrumentation

2.1 Working section

In order to investigate the influence of intense acoustic vibrations on the rate of heat transfer from a circular cylinder to air in crossflow, we have the working section in wind tunnel (250mm $\times$ 200mm crosssection) as shown in Fig.1. The model used is a copper circular cylinder for measurements of the local heat-transfer rate and the static pressure. The cylinder spans the working section at the center part to maintain two-dimensionality of the flow field.

Sounds superimposed upon the flow field are radiated by a 10W cone-type speaker on both upper and lower sides in working section. The distance between two speakers can be changed from 250mm to 1200 mm on the line perpendicular to free-stream direction to generate standing sound fields with various frequencies. In order to prevent sound reflection from the acrylic resin side walls, and to get an acoustic particle velocity field perpendicular to the uniform flow and the cylinder axis, at first rock wool of 50mm

thickness and then sponge material of 15mm thickness were stuck on the inner surface of both side walls.

2.2 The cylinder for the measurements of local heat-transfer rates

The cylinder used for the measurements of local heat-transfer rates is made of a copper circular tube (18.0mm outer diameter and 16.0mm inner diameter) with such arrangement that the internal space operates as a constant-pressure boiler. Water was boiled at atmospheric pressure so that the vapor condensing on the inside walls would maintain them at a constant temperature of 100°C. A rectangular copper rod, see Fig.2, which is heated electrically by a Kantal wire is set into the cylinder along a line parallel to the axis. It is recessed into the cylinder wall so that its exposed surface is flush with the surface of the cylinder, and is insulated from the cylinder both thermally and electrically by a thin layer of epoxy resin compound and bakelite insulator. Then, the circular cylinder was finally finished by the turning operation of the lathe and sandpaper. The ends of the Kantal wire (0.5mm dia.) were connected to a low voltage AC supply which was independent of the supply to the internal boiler. The temperatures on the rod and the surface of the cylinder were measured by copper-constantan thermocouples (0.1mm dia.). Three thermocouples were embedded inside the rod at 10-mm intervals along its axis, and another three were embedded in the cylinder wall at 120-deg intervals around the cylinder so that their thermocouple junctions were flush with the surface of the rod and the cylinder, and they were fixed with an adhesive. The bakelite supports were installed at both ends of

Fig.1 Working section

the copper tube, and the Pyrex tubes were inserted into the supports to pass the steam through it. The cylinder was mounted over two bearings to make it rotate about its axis. The angle between the local sensor and the front stagnation point was measured using a protractor scale.

3. Experimental procedure

3.1 Sound fields in the working section

In order to study interference in flow field, we had progressive sound fields, and standing sound fields in the working section. The progressive sound fields (the frequency range ; from 102 to 364Hz) consisted of one cone speaker under the cylinder. The standing sound fields (from 1201 to 2976Hz) consisted of two cone speakers of same type located over and under the cylinder with a phase difference of 180 degrees. Plane stationary sound waves were employed and the distance between the two speakers was varied so that the longitudinal axis of the cylinder coincided with pressure nodes of sound pressure levels i.e. displacement anti-nodes of the sound waves. The Reynolds numbers tested were 5000, 11000, 15000 and 20000. In order to compare the influence of sound superimposed upon the flow field at one Reynolds number with the influence of sound on the flow field at the other Reynolds numbers, the intensities of the acoustic particle velocities in the y -direction, $\sqrt{\bar{v}_{ya}^2}}/U_\infty$, were maintained in a range of about 0.25% and 0.30% for all sound fields throughout the experiments.

On the other hand, it is well known that vortices shed periodically behind a circular cylinder in an air flow. Periodic vortices do not shed directly from the cylinder shoulders but at a certain down-

stream position detached from the cylinder. Between the cylinder and this position, flow discontinuity (i.e. the shear layer) is caused by the boundary layers separating from the cylinder surface. In the separated layers, a turbulence, when it develops in it, is preceded by regular two-dimensional transition waves which eventually degenerate to a turbulence⁽⁶⁾. According to Bloor, the frequency of the laminar-turbulent transition wave in the separated shear layer, f_t , is expressed as the function of a Reynolds number:

$$f_t/f_n = 0.093Re^{0.52} \dots\dots\dots(1)$$

where the Reynolds number range is from 10^3 to 10^5 .

Yamanaka et al. studied acoustic interferences in vortex-shedding from a cylinder, and they obtained the following results: there were two acoustic frequency groups which produce powerful effects on vortex-shedding. One has a narrow frequency range centering around the natural vortex-shedding frequency, and the other has a relatively wide frequency range which corresponds to the frequency of the laminar-turbulent transition wave in the separated boundary layer. The latter group has a greater effect on vortex-shedding than the former. Because these acoustic interferences promote shear layer instability, the vortex-shedding position gets nearer to the cylinder, and the strength of the vortices is also increased. But the vortex-shedding frequency is changed only slightly⁽⁴⁾.

Then, considering the results of investigations by Bloor and Yamanaka et al., two groups of sounds superimposed upon the flow fields were used in our test;

(1) the frequency range centering around the natural vortex-shedding frequency (" f_n " group) and

Fig. 2 Test cylinder

(2) the frequency range which corresponds to the frequency of the laminar-turbulent transition wave in the separated shear layer (" f_t " group). The natural vortex-shedding frequency f_n depends upon the Reynolds number as follows:

$$St = f_n d / U_\infty \dots \dots \dots (2)$$

According to Roshko⁽⁷⁾, the Strouhal number is about 0.20 in the Reynolds number range from 10^3 to 10^5 . In our experiments, the Strouhal numbers St were 0.206, 0.202, 0.198 and 0.196 for the Reynolds numbers of 5000, 11000, 15000 and 20000, respectively. Therefore, the values of f_n measured in experiments and the values of f_t calculated with the aid of Eq.(1) were used in our tests as the frequencies of the sound superimposed in the flow fields.

Sound pressure levels in the working section were measured with a sound pressure meter and a condenser microphone. The intensities of three-dimensional velocities of the acoustic particles u_{fa} , v_{fa} and w_{fa} were measured with an X-type hot-wire probe and two constant temperature anemometers.

3.2 Measurements of the local heat-transfer rates

Fig.3 shows the measurements of the local heat-transfer rates. The cylinder used for measurements of the local heat-transfer coefficients⁽²⁾ is heated at a constant wall temperature (about 100°C) throughout the cylinder surface by internal steam circulation from a boiler⁽⁷⁾. At the same time, the heat input to the copper rod is controlled by a transformer⁽⁶⁾ in order to make the copper rod temperature equal to the average surface temperature of the cylinder. Then, the heat input to the copper rod is measured by a voltmeter⁽⁵⁾ and an ammeter⁽³⁾. The surface temperature of the cylinder and the copper rod and the temperature of the air stream are measured by the copper-constantan thermocouples, which are referred to a cold junction bottle⁽¹²⁾, a selector switch⁽¹¹⁾, a potentiometer⁽¹³⁾ and a galvanometer⁽¹⁴⁾.

On the other hand, the velocities of the air stream are adjusted to correspond with the required Reynolds number by a Pitot tube⁽⁹⁾ and a Betz type manometer⁽¹⁰⁾. The velocity profiles in z -direction (cylinder-axis direction) in the vicinity of the test cylinder were found to be essentially uniform, not including the region of the boundary layers in both side walls. Two-dimensionality of the flow field was ascertained by the pressure-distribution measurements around the cylinder. Under steady heat condition on the surface of test cylinder in an air stream, the surface temperature of the copper rod, which was measured by the three thermocouples embedded in the surface of it along its axial direction, was found to be equally entirely within $\pm 0.01^\circ\text{C}$. No heat conduction would then take place in the

copper rod in axial direction, and two-dimensionality of the temperature field was assumed.

During the operation the copper rod was aligned to the desired angular position relative to the stream direction by rotating the cylinder at intervals of 5 deg or 10 deg. In order to compare the heat-transfer coefficients with sound and without sound, both tests at a degree in the protractor scale were conducted as a couple. Throughout the tests, the temperature of the air stream was kept constant within $\pm 0.1^\circ\text{C}$. No heat conduction would then take place between the rod and the circular tube or bakelite insulator, and then all the heat generated in the rod was transferred across the exposed surface to the air and the only heat radiation from the surface of the rod was estimated. This enabled local heat transfer coefficients to be determined.

3.3 The measurements of flow fields and temperature fields

The mean velocity $\bar{U}/U_\infty$ and the velocity fluctuation $\sqrt{u'^2}/U_\infty$ in the near-wake of a circular cylinder are measured with the aid of I-type hot-wire probe, constant temperature hot-wire anemometer, linearizer, digital voltmeter (DC component) and RMS meter (AC component). The measurements of flow fields were conducted, with an unheated test cylinder, using a hot-wire probe which was equipped with a traversing mechanism with a vernier scale within $\pm 0.01\text{mm}$ in x and y -directions. Water was boiled at atmospheric pressure so that the vapor condensing on the inside walls would maintain them at a constant temperature of 100°C . Then, temperature fields (T) behind the heated cylinder were measured using a resistance-wire thermometer (wire-diameter 5μ). To compare the ef-

① Wind tunnel ⑧ Condenser
② Test cylinder ⑨ Pitot tube
③ Amperemeter ⑩ Betz type
④ Power supply manometer
 (A.C. 100V) ⑪ Cold junction
⑤ Voltmeter bottle
⑥ Variable ⑫ Selector switch
 resistance ⑬ Potentiometer
⑦ Boiler ⑭ Galvanometer
Section AA: 500×400mm, Section BB: 250×200mm
Fig.3 Setup for measurement of the local heat-transfer rates

fects of sound on the flow and temperature fields, the tests with and without sound at one position of the probe were conducted as a couple.

4. Results and discussions

4.1 Sound fields in the working section

Fig.4 represents the distributions of the sound pressure levels SPL and the intensities of the acoustic particle velocity in the y -direction $\sqrt{v_{fa}^2}/U_\infty$ for the sound fields in the mid-section of the cylinder axis ($x=0$). In addition, the acoustic particle velocities were measured in three directions. However, in both progressive sound fields and standing sound fields, the acoustic particle velocity in the y -direction v_{fa} is much faster than the other components u_{fa} and w_{fa} , in the x and z -directions, respectively. Hence it was assumed that the sounds from the speaker were radiated almost in the y -direction.

It was found that there was no appreciable spanwise variation (in z -direction) in the sound pressure and the acoustic particle velocities except in the neighborhood of the speaker and both side walls. The distribution of the sound pressure levels in Fig.4 was measured in the still air ($U_\infty=0$). But the SPL with the frequency f_a radiating from the speaker in the working section were much greater than the SPL of the background noise in the presence of an air flow in it, and the SPL did not depend on the existence of the air stream. Hence the SPL measured in $U_\infty=0$ would be equal to the SPL with air flow.

4.2 The influence of sound on heat transfer coefficients from a cylinder in crossflow

The local heat transfer coefficient distributions around the cylinder at various Reynolds numbers, both with and without sound waves with a particular frequency, were measured and the local heat transfer coefficient α was expressed in nondimensional form as the Nusselt number $Nu=\alpha d/\lambda$. Here, λ is the thermal conductivity of air and is evaluated at an average film temperature of the cylinder surface temperature T_w and the air stream temperature T_∞ . The test cylinder was rotated at intervals of $\theta=10$ deg and the location at $\theta=360$ deg corresponds to the starting point ($\theta=0$ deg). The readings of meter at a position and the one in the symmetrical position are usually equal within an error. Hence, the measurements were conducted only on the upper surface of the cylinder ($0 \text{ deg} \leq \theta \leq 180 \text{ deg}$) and not on the lower surface ($180 \text{ deg} \leq \theta \leq 360 \text{ deg}$). The effect of the natural convection was substantially neglected for the range of the Reynolds numbers tested.

4.2.1 The heat transfer coefficients around a cylinder without sound

At first the local heat transfer co-

efficients from a cylinder were measured without sound. Fig.5 indicates the results at four Reynolds numbers. For the range of these Reynolds numbers, over the laminar boundary layer in front of the cylinder ($0 \text{ deg} \leq \theta \leq 80 \text{ deg}$), the Nusselt number distribution has a maximum at the stagnation point ($\theta=0 \text{ deg}$), and then falls to a minimum value at around $\theta=80 \text{ deg}$. The Nusselt number at the rear in the separated flow

(a) Progressive sound fields

(b) Standing sound fields

Fig.4 Distributions of sound pressure levels SPL and their particle velocities $\sqrt{v_{fa}^2}/U_\infty$ for $U_\infty=12.5\text{m/s}$

Fig.5 Nusselt number distributions for a cylinder without sound

(80 deg $\leq \theta \leq 180$ deg) begins to rise from $\theta = 90$ deg to another maximum at $\theta = 110$ deg and then increases monotonously to maximum at the rear stagnation point ($\theta = 180$ deg).

In Fig. 6, which shows the plots of $Nu/Re^{1/2}$ determined in this experiment, it can be seen that the general shapes of the curves are the same in front of the cylinder, but that quantitatively there are considerable deviations on the back half. Fig. 6 shows also a comparison of some of the present runs with the results of Schmidt and Wenner⁽⁸⁾, who used an isothermal cylinder surface. An excellent agreement is seen at the front in laminar boundary layer, so that the test cylinder is ascertained to be a cylinder with an isothermal surface.

The overall Nusselt number was found by taking the integral mean of the local Nusselt number distributions from $\theta = 0$ deg to 180 deg ($\bar{Nu} = \int_0^{180} Nu d\theta / \int_0^{180} d\theta$). Fig. 7 shows the overall Nusselt numbers for the cylinder without sound and the comparison of our results with those obtained by the other investigators. This figure shows also the values of $\bar{Nu}$ at the front in the laminar boundary layer and the rear in the separated region as a function of Reynolds number. As shown in Fig. 7, it can be seen that $\bar{Nu}$ is proportional to $Re^{1/2}$ in the laminar boundary layer and to $Re^{1/3}$ in the sep-

Fig. 6 Nusselt number $Nu/\sqrt{Re}$ distributions for a cylinder without sound

Fig. 7 Averaged Nusselt number in the laminar boundary layer, in the separated region and over all region from a cylinder without sound

arated region. Thus in the laminar boundary layer and the separated region, respectively, the values are:

$$\bar{Nu} = 0.99Re^{0.48}, \bar{Nu} = 0.09Re^{0.67} \dots (3)$$

The overall Nusselt numbers (0 deg $\leq \theta \leq 180$ deg) are given for both the laminar boundary layer and the separated region, using Eq. (3), as follows.

$$\begin{aligned} \bar{Nu} &= (80/180)(0.99Re^{0.48}) \\ &\quad + (100/180)(0.09Re^{0.67}) \\ &= 0.44Re^{0.48} + 0.05Re^{0.67} \dots (4) \end{aligned}$$

Equation (4) is in good agreement with the data of Richardson⁽⁹⁾, Hilpert⁽¹⁰⁾ and v. Meel⁽¹¹⁾.

4.2.2 The heat transfer coefficients around a cylinder with sound

Fig. 8 shows the local Nusselt number distributions around the cylinder with sound. In this figure Nu value is plotted as a function of θ at constant values of the sound pressure levels SPL for different values of applied acoustic frequencies f_a over the Reynolds number range from 5000 to 20000. From Fig. 8, it can be seen that the applied sound waves have a large effect on the local Nusselt numbers, and they cause the Nusselt numbers to increase appreciably on the back half of the cylinder, but to increase only slightly in the laminar boundary layer. As the result of these experiments, it is found that the Nusselt numbers are increased as the SPL becomes large, and the sounds of the f_t group have a greater effect on the increase of the Nusselt number than the sounds of the f_n group. For example, it can be seen from Fig. 8(b) that the Nusselt numbers are increased much more in the presence of sound waves with the frequencies of the f_t group ($f_a = 1201$ Hz) than with the f_n group ($f_a = 102$ Hz) in the separated region.

Fig. 9 shows the overall Nusselt numbers (0 deg $\leq \theta \leq 180$ deg) around the cylinder with and without sound as a function of Reynolds number. Use of the sound waves increases the overall Nusselt number, the increase being greatest at the lowest Reynolds number tested (20 percent increase) and least at the highest Reynolds number.

4.3 The pressure distributions around the cylinder

The pressure distribution measurements around the cylinder ($Re = 15000$) are plotted in Fig. 10. From this figure, it can be seen that the applied sound waves have a large effect on the pressure distributions around the cylinder and that the sound waves cause the base pressure to reduce. Hence, the sound waves used would cause an appreciable increase in drag coefficient of circular cylinder at the Reynolds numbers investigated in these tests. A higher sound pressure level has a greater effect on C_p , and the sounds of the f_t group have a greater effect on C_p than ones of the f_n group.

4.4 Flow fields behind the cylinder

In order to examine the mechanism of interaction between the sound waves and the wake motion, and to establish at least a preliminary understanding of the flow in the vicinity of the wall and effects of sound waves on the heat transfer, we measured mean velocity and fluctuating velocity in the flow fields of the near-wake of an unheated circular cylinder. Fig.11 indicates the profiles of mean velocity and velocity fluctuations in y -direction at the reference positions x/d for a Reynolds number of 15000. Fig.12 shows the profiles of mean velocity and velocity fluctuations in x -direction at $y/d=0$. The frequency of the sound used is 1918Hz (the f_t group), and SPL in the working section

is 110dB. These profiles without sound are shown by the solid lines and the profiles with sound are shown by the dotted lines in Figs.11 and 12.

In order to express the flow fields from the separated shear layers to the rolling-up major vortices, those profiles in Figs.11 and 12 are used to generate the mean velocity isopleths shown in Fig.13 and the RMS fluctuation velocity isopleths shown in Fig.14. In both Figs.13 and 14, the isopleths without sound are shown by the solid lines and the isopleths with sound are shown by the dotted lines. Judging from Figs.13 and 14, it is likely that the flow fields with sound are, in general, similar to the flow fields without sound, but that these acoustic interferences cause the shear layer to diverge

Fig.8 Nusselt number distributions for a cylinder with sound

Fig.11 Distributions of mean velocities and fluctuating velocities in y -direction in the wake of an unheated cylinder.
 $Re=15000$, $f_a=1918\text{Hz}$, $SPL=110\text{dB}$

Fig.12 Distributions of mean velocities and fluctuating velocities on the wake axis of an unheated cylinder.
 $Re=15000$

and the vortex-shedding position gets nearer to the cylinder.

4.5 Temperature fields behind the heated cylinder

The profiles of mean temperature in y -direction at the reference x -positions were measured with the aid of a resistance-thermometer probe. Those profiles are used to generate the mean temperature $(\bar{T}-T_\infty)/(\bar{T}_w-T_\infty)$ isopleths behind the heated cylinder shown in Fig.15. The isopleths without sound are shown by the solid lines and the isopleths with sound are shown by the dotted lines in Fig.15. From this figure it can be seen that the temperature fields with sound are, in general, similar to the temperature fields without sound, but that these acoustic interferences cause each isotherm to come nearer to the surface at the rear of the cylinder, so that the temperature gradients in the thermal boundary layer on it become large, and consequently the heat transfer coefficients in the range are increased.

Fig.13 Mean velocity $\bar{U}/U_\infty$ at $Re=15000$: isopleths

Fig.14 Fluctuating velocity $\sqrt{u'}/U_\infty$ at $Re=15000$: isopleths

Fig.15 Mean temperature $(\bar{T}-T_\infty)/(\bar{T}_w-T_\infty)$ at $Re=15000$: isopleths

Thus, it is concluded that because the laminar-turbulent transition in the separated shear layer is promoted by the superposition of sound field, the shear layer becomes thick and the vortex-shedding position gets nearer to the cylinder, the velocity in the vicinity at the rear of the cylinder is increased, the thermal boundary layer becomes thin and the heat transfer at the rear in the separated flow is increased.

5. Conclusions

An experimental investigation has been carried out to study the effect of sound waves with a particular frequency on forced convective heat-transfer from a heated circular cylinder over the Reynolds number range from 5000 to 20000, and the following conclusions are drawn.

(1) The convective heat-transfer coefficient can be influenced by a given sound wave, and for the range of the SPL tested, the sound waves with the frequency of laminar-turbulent transition wave in the separated shear layer (the f_t group) have a greater effect on $\bar{Nu}$ than the sound waves with the natural vortex-shedding frequency (the f_n group).

(2) The sound waves with the frequency of the f_t group cause the Nusselt number to increase appreciably on the back half of the cylinder (in the separated region),

and Nu is increased as the SPL becomes large.

(3) The approach of the vortex region formed in the wake to the cylinder surface plays an important role in the mechanism of an increase of the Nusselt number in separated region.

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