Heat Transfer from Acoustically Resonating Gas Flames in a Cylindrical Burner

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Heat transfer from premixed propane-air flames to the cooled wall of a 5-in. I.D. ramjet type of burner was studied with and without flame-generated, sonic oscillations. Frequency and amplitude measurements revealed that both transverse and longitudinal waves were resonant. The heat flux to the wall was increased significantly by the oscillations, but the gas temperature profile was changed very little. The local heat transfer coefficient was found to be a linear function of the sound pressure amplitude and independent of the mode and frequency of the resonant oscillations.

Heat transfer from vibrating fluids and from flames has recently received considerable attention. The increased rates of heat transfer are a serious problem in the more powerful jet engines but are of possible utility in other applications. The nature, cause, and mechanism of flame-generated oscillations have been studied by a number of investigators (*3, 14, 16, 17*). In ramjet type of burners, properly designed to isolate the fuel and oxidant supplies from instabilities in the combustion chamber, the flame-generated oscillations have been shown to be acoustically resonating waves with the mechanism of generation related to the periodic formation of vortices at the flame holder. Screeching combustion, which has been shown to be due to transverse waves, is particularly interesting and important because it has been reported to have caused burner walls to erode rapidly (*21*).

Heat transfer measurements in the presence of flame-generated oscillations are very limited. In an early study Berman and Cheney (*2*) measured overall rates of heat transfer to a small rocket chamber in the presence of shock and sinusoidal combustion instabilities. More recently Sundstrom and Churchill (*19*) measured local heat flux densities along a combustion tube with and without longitudinal (organ-pipe) oscillations but did not determine heat transfer coefficients.

Some related work has been done with externally imposed sonic oscillations. The effect of oscillations on heat transfer from diffusion flames was studied by Tailby and Berkovitch (*20*), on heat transfer from combustion gases in a closed bomb by Havemann (*7*), on natural convection by Kubanskii (*11*) and others, on natural and forced convection in a pipe by Jackson, Harrison, and Boteler (*8*), and on forced convection in a pipe by Harrje (*6*).

The object of this investigation was to determine the quantitative effects of flame-generated oscillations on the local rates and coefficients of heat transfer. Preliminary studies revealed that (1) intense longitudinal and transverse waves could both be attained over a reasonable range of flows in a cooled tube, 4 in. or more in diameter, with bluff body stabilizers and premixed propane-air flames, (2) that the transverse oscillations could be varied in intensity by the use of different flameholders and propane-to-air ratios, and (3) that the longitudinal oscillations could be damped with a muzzle on the exit of the burner. The equipment was designed accordingly. A vertical cylinder was chosen for reasons of symmetry.

APPARATUS

The equipment included air, propane and water supply, and metering systems. The metered propane and air were joined upstream of a packed bed followed by an open mixing chamber consisting of a 5-in. I.D. by 3 ft. long tube connected directly to the combustion chamber as shown in Figure 1. The rotameters for the propane and air streams were isolated from the sonic oscillations in the combustion chamber by an orifice, ordinarily located between the flanges connecting the mixing and combustion chambers. The combustion chamber was a water-jacketed, ½ in. thick, low-alloy, stainless steel tube,

TABLE 1. FLAME HOLDERS

Designation	Blockage, %	Length, in.
F-1	90.8	1/8
F-2	90.8	5/8
F-3	92.7	1/8
F-4	95.0	1/8
F-5	94.6	5/16
F-6R*	95.3	5/16
S-1	96.5	1/8
S-2	96.5	1-3/8
S-3R*	98.3	1/8
S-4R†	98.3	1/8
S-5R**	97.1	1/4

* A 1/32 in. thick asbestos liner was cemented to the tube wall from the 12½-in. level to the 22½-in. level.

† A 1/32 in. thick asbestos liner was cemented to the tube wall from the 17¼-in. level to the 22½-in. level.

** A ⅛ in. thick asbestos ring was cemented to the tube wall from the 17½-in. level to the 18-in. level.

27 in. long, with a honed inside surface 4.920 ± 0.003 in. in diameter. A muzzle consisting of a 10-mesh, stainless steel screen, cooled by a spray of water, extended 3 in. beyond the combustion chamber when attached. All locations in the following description, as in Figure 1, are measured from the exit from the combustion chamber, exclusive of the muzzle. The burning length is defined as the distance from the flame holder to the exit from the combustion chamber, exclusive of the muzzle.

The identification and characteristics of the various flame holders are indicated in Table 1. Three general types were used: (1) ⅛-in. metal disks with several loops of 3/16-in. copper tubing soldered on the back side, (2) the same disks with the addition of a 10-mesh screen of 0.047-in. wire in the annulus between the disk and the tube, and (3) the same disks with a 1 in. high ring of copper sheet metal of the same diameter as the disk soldered to the back of the disk. F designates an open annulus and S a screen type of flame holder. The given blockage includes that of the screen. R indicates the use of a ring of asbestos on the tube wall opposite and upstream from the flame holder.

The flame holders were movable longitudinally and were supported by a ⅝-in. tube containing the water lines to the cooling coil. Sodium bicarbonate powder was injected into the flame for temperature measurement by sending a small stream of air across a layer of the powder, down a small tube inside the ⅝-in. tube, and through a 1/32-in. hole in the center of the flame holder.

The radial temperature gradient in thermally isolated sectors of the combustion chamber wall at distances of 4, 10, 14, 15, and 17 in. from the exit was measured with three thermocouples located on the same radial line. The location of the tips of the thermocouples was determined to the nearest 1/64-in. by cutting up and X-raying the tube wall after completion of the experimental work.

The temperature of the hot gas at the core of the combustion chamber was determined by sodium line reversal (SLR) measurements through pairs of windows ⅝ in. in diameter at 4, 10, and 15 in. from the exit. Thermocouple and pitot-tube traverses of the flame gases were made 4 in. from the exit with a low propane-to-air ratio. The temperatures of the mixed propane and air ahead of the combustion chamber and of the inlet and exit streams of water to the combustion chamber jacket and flame holder coil were also measured. The water flow rates

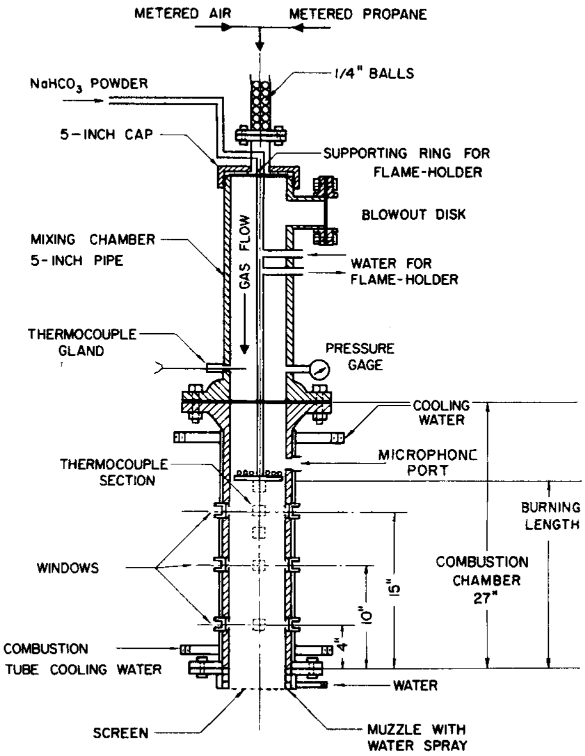

Fig. 1. Detail of mixing chamber, combustion chamber, and muzzle.

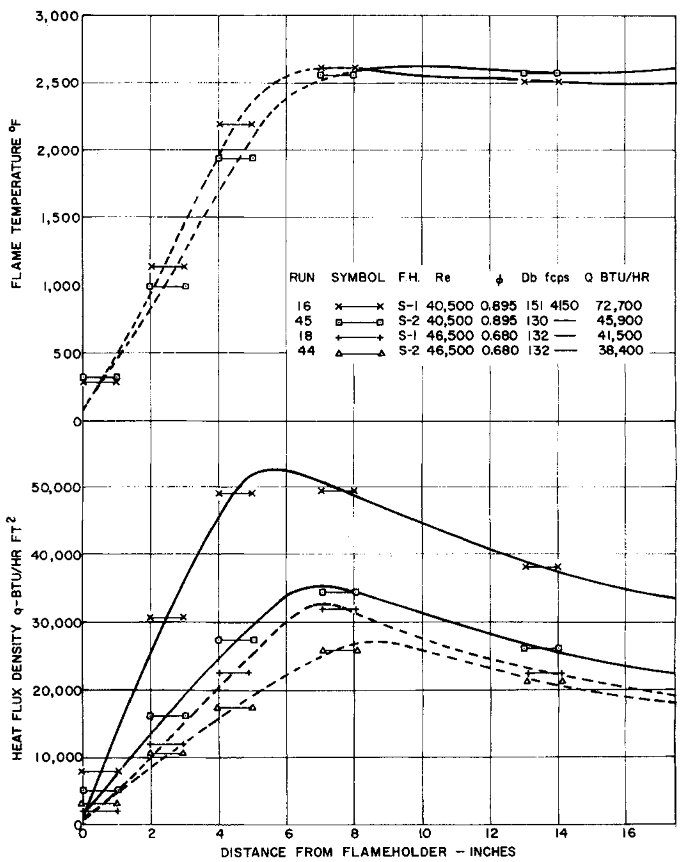

Fig. 2. Effect of transverse oscillations on heat flux density and gas temperature.

were measured with rotameters. The pressure drop between taps 25 and 2 in. from the exit was measured within 0.02 in. water with a set of manometers.

The flame-generated oscillations were measured with a microphone mounted flush with the inside tube surface, 22 in. from the exit. A sound level meter and sound analyzer were connected to the microphone. Selected events were recorded with a tape recorder.

The commercial grade propane used in the experiments was found by analysis to be 98 propane and 2% propylene.

EXPERIMENTAL CONDITIONS

Seventy-four runs were made over a range of flow rates corresponding to N_{Re} from 35,000 to 48,000 and propane-to-air ratios corresponding to ϕ from 0.57 to 1.03. The flame holders were located at 17½ in. from the exit and in some cases at 13½ and 8 in. The holders with a free annulus were used to study longitudinal oscillations. The intensity of the oscillations was increased by using holders with greater blockage and hence greater reflectivity. The muzzle was used to damp out the longitudinal oscillations completely, the water droplets absorbing the acoustical energy. The flame holders with a screen in the annulus were used to study transverse oscillations. The screen stabilizes the flame closer to the wall and thus results in a greater release of energy near the circumference of the tube where the amplitude of the transverse waves is greatest and the energy is utilized most effectively to drive the oscillation. The transverse oscillations were damped by using the type 3 (extended) flame holders which cause the unburned gases to jet through the annulus and sweep the region near the tube wall free of active combustion. The transverse oscillations were kept free of interaction with longitudinal oscillations by the use of the muzzle.

Sodium bicarbonate was injected only during the SLR measurement. Satisfactory SLR measurements were possible at the 4 and 10-in. levels with a 17½ in. burning length but only at the 4-in. level with a 13½ in. burning length and not at all with an 8 in. burning length. The powder had no observable effect on the oscillations or the heat transfer to the walls.

In order to determine the drag of the flame holder and tube wall the pressure drop was measured with cold air flowing at the same inlet N_{Re} as that of each run. The radial temperature gradients in the tube wall and the gas temperature were also measured with air entering the combustion chamber at 170°F. and flow rates corresponding to the average N_{Re} of the burned gases.

Runs were made with the sonic orifice at various locations in the mixing chamber, but this did not prove to be a significant variable. The combustion chamber wall temperature was raised for a few runs by reducing the water rate to the cooling jacket.

INTERPRETATION OF DATA

The heat flux through the five thermally isolated sectors of the combustion chamber wall and the surface temperature were calculated from the measured radial temperatures with the following equation:

$$Q = \frac{kA_s}{r_s} \frac{T_i - T_j}{\ln \dfrac{r_j}{r_i}} \qquad (1)$$

The maximum error in the computed heat flux due to distortion of the heat flux by the thermocouples was estimated from the analyses of Churchill (*4*) and Beck and Hurwicz (*1*) to be less than 4%. The effect of the possible mislocation of the thermocouple tips was negligible. The area under an equal area plot of the incremental heat-flux densities vs. wall area gives the total heat flux to the combustion chamber wall. This value checked within 4% with the energy picked up by the cooling water streams.

The accuracy of the SLR measurements depends on the lamp calibration, the accuracy of the current reading, the scatter of light by the lamp-side lens, the sensitivity of the reversal point, and absorption by cool gas next to the tube wall. The total net limits of uncertainty are estimated to be −50° to +90°F., which is better than can be achieved for a burning gas at 3,300°F. by most other methods (*12*).

Conduction and radiation for the corrections for the 16-gauge, bare thermocouples used for the gas temperature measurement were made with the charts of Scadron and Warshawsky (*18*). The velocity and temperature distributions were found to agree closely with the predicted distributions for fully developed thermal and momentum boundary layers, giving support to the following relationship obtained between the bulk and *SLR* temperatures:

$$T_b = 0.82\, T_{SLR} + 0.18\, T_s \qquad (2)$$

The combustion efficiencies were computed from the bulk temperature and the heat flux through the wall. The gas temperature and the combustion efficiency were also calculated from the pressure-drop measurements for the hot and cold flows, with one-dimensional mass, energy, and momentum balances. The details of the derivations, assumptions, and calculations are given in reference *23*. For quiet combustion the average difference between the computed efficiencies was less than 2% with a 17½ in. burning length. With a 13½ in. burning length the efficiencies obtained from the pressure-drop measurements averaged about 6% higher than from the temperature measurements. With reasonating oscillations the efficiencies obtained from the pressure-drop measurements were meaningless, presumably owing to the invalidity of some of the assumptions in the above balances and to the incorrect response of the manometers under oscillating conditions.

The temperature profile between the flame holder and the measured temperature at 10 in. from the exit was computed from the heat flux by assuming the ratio of the heat transfer coefficients for the burning gases and cold air was the same as the measured ratio at the 4- and 10-in. levels. This method is inaccurate but gives some indication of the gas temperature in the precombustion and initial-combustion zones where it cannot be measured by thermocouples owing to stabilization of the flame on the thermocouple or by the SLR method because of insufficient radiation.

The local heat transfer coefficient was calculated only at the stations where the heat fiux and the gas temperature were measured. The radiant transfer from the gas was estimated by the method described in McAdams (*13*) and subtracted from the measured heat flux to obtain the convective coefficient. The computed contribution of radiation to the local heat flux density had a maximum value of 15% and a mean value of 10%.

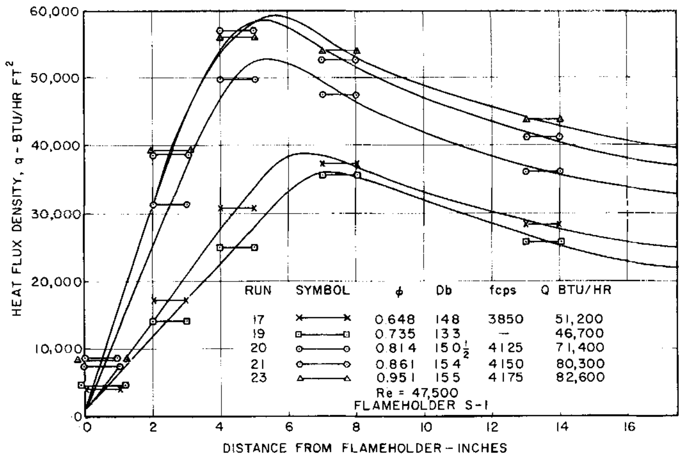

Fig. 3. Effect of propane-to-air ratio on heat flux density.

The acoustically resonant frequencies in the combustion chamber were predicted from the solution to the wave equation for a cylinder with one open and one closed end:

$$P = C \cos m\theta \cos \left(\frac{n_x + 1/2}{L} \right) \pi x \; J_m\left(\frac{\alpha_{mn}\, \pi r}{R} \right) e^{2\pi i f t} \qquad (3)$$

where n, n_x, and $m = 1,2,3 \ldots$ are the wave numbers. Values of α_{mn} which satisfy the condition that $dJ_m/dr = 0$ at $r = R$ are given by Morse (*15*). The frequency f is

$$f = \frac{a}{2} \sqrt{\left(\frac{n_x + 1/2}{L} \right)^2 + \left(\frac{\alpha_{mn}}{R} \right)^2} \qquad (4)$$

The measured frequency for the organ-pipe oscillations of about 350 cycles/sec. agreed closely with the computed frequency obtained from Equation (4) for a primary mode of longitudinal oscillation. The frequency calculated for resonance between the exit and the sonic plate rather than the flame holder, with the method of Jost (*9*) for the columns of hot and cold gas in series, did not agree with the measured frequency, clearly indicating that the flame holder acted as an acoustically closed end.

The measured frequency for the screeching combustion of 4,000 ± 250 cycles/sec. agreed well with the computed frequency for a primary mode of transverse oscillation and excluded the possibility of radial oscillation. The frequency difference between a pure, primary, transverse mode and a primary, transverse mode resonating also in the longitudinal direction is less than the change observed with propane-to-air ratio, and the latter mode cannot be excluded as a possibility. However the detailed probings of screeching flames by previous investigators (*5*, *22*) and the fact that the frequency did not change significantly as the burning length was shortened from 17½ to 8 in. is persuasive evidence against this possibility.

RESULTS

Typical profiles of the heat flux density along the combustion chamber for a screeching and a quiet flame are illustrated by the upper two solid curves in the lower portion of Figure 2. The screeching flame produced a sound level of 151 decibels with a dominant frequency of 4,150 cycles/sec. as compared to 130 decibels with no dominant frequency for the quiet flame. The

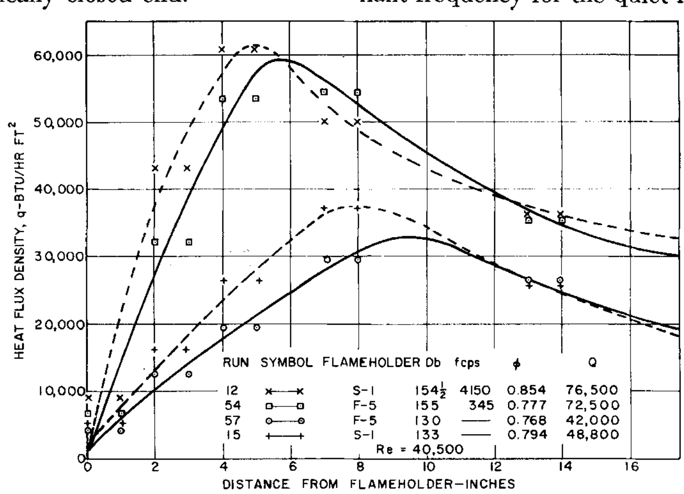

Fig. 4. Effect of longitudinal oscillations on heat flux density.

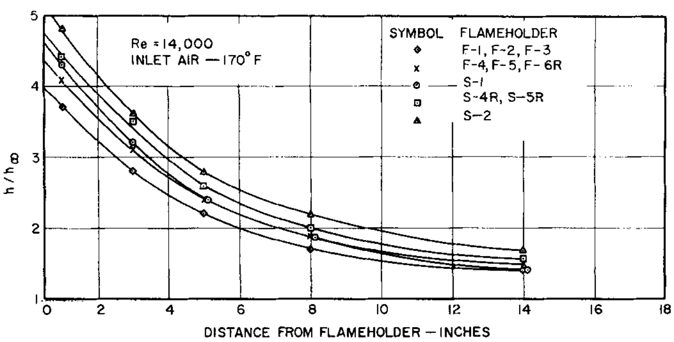

Fig. 5. Normalized local heat transfer coefficients for air.

heat flux density is observed to be about 60% greater in the presence of the resonant oscillation. This change was produced by the use of different flame holders. However in the two dashed curves the heat flux profile is illustrated for the same two flame holders at a lower propane-to-air ratio such that screeching did not occur for either, and it is observed that the difference in the heat transfer rate which can be attributed to the flame holders themselves is less than 8%.

The temperature profile is shown in the upper portion of Figure 4 for two of the same runs. The curves are dashed from 0 to 4 in. to indicate a greater uncertainty than in the heat flux density. Since the effect of the resonant oscillation on the gas temperature profile is apparently very small, the previously observed increase in the heat flux density must be almost wholly attributable to an increase in the heat transfer coefficient.

The effect of propane-to-air ratio on the heat flux density profile is illustrated in Figure 3. As the equivalence ratio was increased from 0.648 to 0.735, the resonant frequency of 3,850 cycles/sec. disappeared and the sound level dropped from 148 to 133 decibels. This was accompanied by a drop of about 9% in the heat flux. As the equivalence ratio was further increased to 0.814, a dominant frequency of 4,125 appeared with an accompanying increase in decibels to 150½ and an increase in the heat flux as shown. With further increases in the equivalence ratio the resonant frequency shifted slightly upward and the intensity and heat flux increased. The change in the resonant frequency from 3,850 to 4,175 cycles/sec. as the propane-to-air ratio was increased can be accounted for by the increase in the gas temperature and hence in the sonic velocity.

The effect of longitudinal oscillations on the heat flux density is illustrated by the solid curve in Figure 4. The increase in sound level from 130 to 155 decibels and the 73% increase in the heat flux resulted from the removal of the screen in the muzzle only and is wholly attributable to the resonance. The peak in the heat flux density is observed to be shifted toward the flame holder by the resonance.

Heat flux profiles for transverse oscillations and quiet combustion with the same pair of flame holders at the same rate of flow and intensity of sound are represented by the dashed curves in Figure 4. The similarity between the curves for transverse and longitudinal resonance is striking. The heat transfer rate is obviously more a function of the intensity of sound than of frequency. The different intensities and modes of oscillation were obtained by using different flame holders and fuel/air ratios. The effects of the flame holders and fuel/air ratio, apart from their effect on the oscillations, was shown in previous figures to be relatively small.

The local heat transfer coefficients measured with heated air at N_{Re} = 14,000 are shown in Figure 5. These coefficients are normalized by dividing by the coefficient for fully developed flow in a pipe at the same N_{Re} and N_{Pr} as obtained from the Dittus-Boelter equation (*13*). The curves thus indicate the effect of the flame holders themselves on the heat transfer rate.

The local heat transfer coefficients for the flame were normalized by dividing by the coefficients measured for air at the same N_{Re} and position but corrected for physical properties and temperature level as indicated by Zellnik and Churchill (*24*). The normalized coefficients would thus be unity for a nonreacting gas stream in the combustion tube. Coefficients for 4 in. from the exit are plotted in Figure 6 vs. the sound pressure amplitude divided by the sound pressure amplitude for a quiet flame in the same equipment, that is 130 decibels. A linear correlation is evident. The equation of the line sketched through the data is

$$\frac{h}{h_1} = 0.044 \frac{P}{P_o} + 1.08 \quad (5)$$

The variation in the normalized heat transfer coefficient for quiet combustion is undoubtedly due to changes in the burning gas stream which cannot be accounted for by the cold flow coefficient and the bulk temperature alone. This variation occurs for resonant combustion as well and contributes to the scatter in the correlation but is overshadowed by the effect of the oscillations.

Increasing the wall temperature 200°F. to about 450°F. increased the intensity of the transverse oscillations from 153 to 157 decibels and the heat transfer coefficient correspondingly. An asbestos liner extending approximately 5 in. into the combustion chamber beyond the flame holder was found to dampen screeching. An asbestos ring in the flame holder annulus had little effect on the acoustical phenomena.

All of the above results are for a 17½ in. burning length. Similar results were obtained with other flame holder positions. The principal effect of shortening the burning length is an increase in the intensity of the transverse oscillations with a corresponding increase in the heat transfer coefficients. These additional data and other details concerning the work are available in reference *23*.

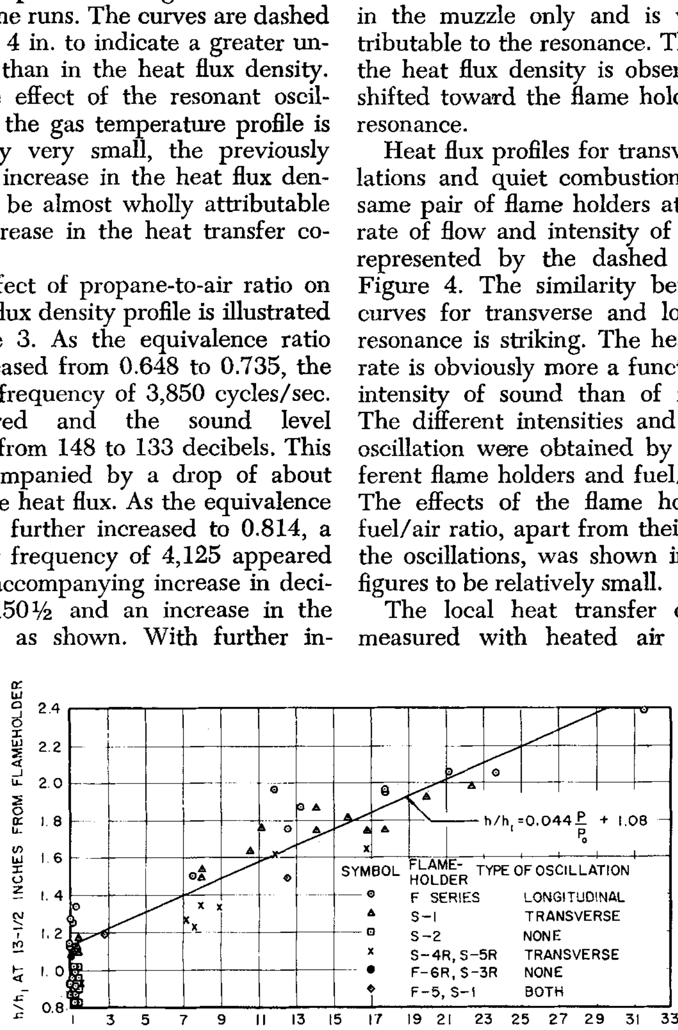

Fig. 6. Normalized local heat transfer coefficients for combustion with resonant oscillations. Burning length = 17½ in.

DISCUSSION

Several significant conclusions can be drawn from the results of this investigation. The coefficient for forced convective heat transfer from a premixed flame to the wall of the combustion chamber is shown to be a linear function of the sound pressure level above the sound pressure level of a quiet flame with no resonant frequencies. The coefficient increased as much as 100% under the conditions of this investigation, and an even greater increase appears to be attainable with other conditions and burners. The coefficient appears to be independent of the resonant frequency; that is longitudinal (organ-pipe) oscillations are just as effective as transverse (screeching) oscillations. It is further shown that the mode and hence the frequency of the resonant oscillations can be controlled by minor variations in the flame holder as well as by changing the dimensions of the combustion chamber or adding acoustical dampers, and that the sound pressure level of resonant oscillations can be controlled by minor variations in the flame holder, the fuel-to-air ratio, the wall temperature, the dimensions of the combustion chamber, and acoustical dampers.

The resonant oscillations may produce the observed changes in the heat transfer coefficient by either or both of two mechanisms: changes in the combustion process, for example in flame shape, flame speed, and combustion efficiency; changes in the dynamics of the hot gas stream downstream from the flame, for example in boundary-layer thickness and intensity of turbulence. The similarity in the temperature profiles and the relative invariance of the combustion efficiency suggest that fluid-dynamic effects rather than combustion and thermal effects are primarily responsible for the changes in heat flux.

The results should be useful in the design and operation of jet engines and industrial heating devices. For example the work indicates that a small increase in the wall temperature of combustion chamber due to any change in operating conditions may result in more intense transverse oscillations, leading to a higher rate of heat transfer to the wall, an even higher wall temperature, etc. This may explain the observed failures of burner walls in jet engines under conditions of screeching combustion. It should be noted that the investigation was limited to a single tube diameter, and hence the results cannot be generalized quantitatively for combustion in chambers of other dimensions.

The results are generally consistent with those of the previously mentioned investigators with imposed oscillations. Harrje (*6*) also observed a linear increase in forced convective heat transfer with the amplitude of the oscillations, although the maximum increase he encountered was only 10%. Jackson, Harrison, and Boteler (*8*) and Havemann (*7*) also found that frequency effects were minor compared with the sound pressure level. The exception is Tailby (*20*) who found a 600 cycle/sec. note more effective than a 1,700 cycle/sec. note in promoting mixing and heat transfer. The critical level of the sound pressure for the promotion of heat transfer was found by Jackson, et al. to be 118 decibels as contrasted with 130 decibels in this work, suggesting that the critical level is the normal level for any process and apparatus in the absence of resonance.

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NOTATION

a = speed of sound, ft./sec.
A = area, sq. ft.
C = proportionality constant
Db = intensity of sound, decibels
f = frequency, cycles/sec.
h = local coefficient for convective heat transfer, B.t.u./(hr.)(sq. ft.) (°F)
i = $\sqrt{-1}$
J_m = cylindrical Bessel function
k = average thermal conductivity of tube wall, B.t.u./(hr.) (ft.) (°F)
L = length of cylinder of oscillating gas, ft.
m = wave numbers for transverse oscillations
n = wave numbers for radial oscillations
n_x = wave numbers for longitudinal oscillations
N_{Re} = Reynolds number based on tube diameter and cold, inlet gas
N_{Pr} = Prandtl number
P = pressure, lb.$_F$/sq. ft.
Q = heat flux, B.t.u./hr.
q = heat flux density, (B.t.u./hr.) (sq. ft.)
R = radius of cylinder of oscillating gas, ft.
r = radial distance, ft.
T = temperature, °F.
t = time, sec.
x = longitudinal distance, ft.
α_{mn} = eigenvalues
ϕ = equivalence ratio, propane-to-air ratio in inlet mixture divided by the stoichiometric propane-to-air ratio

Subscripts

b = bulk
i,j = any positions
s = tube surface
l = nonreacting gas
o = reference
∞ = for fully developed flow
SLR = sodium line reversal

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