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HEAT TRANSFER IN OSCILLATING FLOW

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FOREWORD

This progress report covers the first extensive series of experiments in heat transfer with oscillating flow in a duct run with the traversing heater section and revised instrumentation. This experimental approach enables a continuous heat transfer survey to be made between node and antinode locations. The experiments and the data interpretation, using the IBM 7090 as a tool in directly plotting the massive amount of data, represent significant advances over the previous attempts to correlate parametric relationships describing this complex phenomena. The gap variation method, now in the preliminary testing stage, should go one step further in providing the degree of accuracy desired for truly quantitative results.

Within the degree of accuracy associated with the tests presented in this report, a number of parameters have been determined to be important in influencing the oscillating heat transfer contributions. Thus in the more precise tests currently being carried out, the experimental scope can be limited to the most fruitful areas.

ABSTRACT

Pressure oscillations imposed upon a compressible fluid flowing in a duct produce a substantial increase of heat transfer between the heated duct wall and the fluid.

This report records an experimental investigation of the causes of this increase. Resonant conditions in a duct were used to achieve increased oscillating amplitudes. The local heat transfer increase was found to be highly dependent on the amplitude of the local velocity oscillations, and to depend also on the frequency of oscillations and the duct steady-state pressure and Mach number. The presence of flow reversal further increased the observed heat transfer.

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NOMENCLATURE

- A = section area
- a = velocity of sound
- B = hot-wire "cold" balance bridge null dial setting
- C = hot-wire calibration constant
- D = diameter
- E_h = heater voltage
- f = frequency
- f_{ref} = frequency at reference temperature
- h = heat transfer film coefficient
- I = hot-wire current
- i = square wave current
- I_h = heater current
- k = thermal conductivity
- L = duct total length
- l = traverse length
- M = Mach number
- $\dot{m}$ = mass flow
- Nu = Nusselt number
- P = pressure
- R = radius
- Re = Reynolds number
- r = hot-wire resistance ratio
- T = temperature
- T_{ref} = reference temperature at which f_{ref} is defined ($T_{ref}=0^\circ\text{F}$)
- V = gas velocity
- v = traverse speed

w = heater strip width

x = axial coordinate in the duct, from the downstream traverse limit

$X = x/L$, reduced axial coordinate in %

y = distance from the duct wall

$Y = y/R$, reduced radial distance in %

φ = phase lag

λ = wavelength

μ = viscosity of air

ρ = specific mass

τ = time constant

Symbols modifications

$(\bar{})$ = steady parameter level, e. g. $\bar{P}$

$()_0$ = steady-state value, e. g. Nu_0

$()'$ = oscillating parameter, e. g. P_h'

$(\tilde{})$ = time average value of oscillating parameter level

$(\tilde{})_{rms}$ = root mean square intensity, e. g. $\tilde{P}_{rms}$

$(\tilde{})_{pk}$ = peak-to-peak amplitude, e. g. $\tilde{V}_{pk}$

$\langle \rangle$ = average of quantities variations along the duct for a spatial wavelength, e. g. $\langle h \rangle$

$()_h$ = value in the heater section, e. g. V_h

$()_d$ = downstream value

$()_u$ = upstream value

$()^*$ = critical value

$()_t$ = value in the plenum chamber (supply tank)

Symbols definition

Sketch 1

Sketch 2

Sketch 3

Sketch 4

Sketch 5

Sketch 6

Sketch 7

1. Summary

An investigation of the phenomenon of heat transfer in oscillating gas flow was undertaken at the Guggenheim Laboratories, Princeton University. High frequency longitudinal pressure and velocity oscillations were imposed on a steady flow of air passing through a duct under conditions of fully established turbulent pipe flow. Heat transfer was from an electrically heated ribbon of nichrome set in the duct walls to the cold gas.

A general description of the experimental apparatus is given on p.24 . The background information found in the detailed description of the instrumentation (including details on the high response pressure probe and hot-wire anemometer probe, both located in the travelling heater section etc...) is not absolutely necessary for the understanding of the experimental program.

The main part of this program has been an attempt to correlate the local heat transfer increase to the local oscillating conditions. From this point of view, the local oscillating pressure was consistently shown to fail as the basic driving parameter for the heat transfer increase. On the contrary, the local oscillating velocity (and more precisely, the amplitude of this oscillating velocity) provided a reasonable correlation (p. 73).

In this type of experiment, the variation of the oscillating conditions was obtained by the displacement of the heater section with respect to the node-antinode patterns produced in the duct under resonant conditions. The effects of frequency, duct pressure and Mach number have been analysed. The averaging of the quantities along the duct has supplied some preliminary data on the possible engineering aspect of the problem, i. e., how to increase

the overall heat transfer in a duct through utilization of oscillations.

However, the difficulty encountered in the separation of pure amplitude effects and second order effects such as wave shape transformations or velocity radial profile deformations (in an axial survey), led to the idea of obtaining the amplitude variation by other means.

The siren gap variation method appears to provide a good way to separate parameters. It consists of staying at a fixed axial location and varying the local oscillating conditions by alteration of the end conditions at the siren.

Local heat transfer increases up to 200% have been obtained in the case of very large amplitude velocity oscillations, normally associated with flow reversal.

2. INTRODUCTION AND BACKGROUND

Under the auspices of the Office of Naval Research, a program to investigate the phenomenon of heat transfer in oscillating gas flow was undertaken as a part of the research program of the Guggenheim Laboratories at Princeton University. The effect of pressure and velocity oscillations on the convective heat transfer coefficient has become of some significance in recent years; two aspects at least are to be pointed out:

First there is evidence of combustion oscillations, which cause heat transfer increases of several hundred per cent (as reported by a number of independent investigators (for example Ref. 2)). These increases are responsible for burnouts or prohibitive erosion of chamber walls. Hence, a better understanding of the phenomenon might well provide a valuable insight into the problem of the reduction of excessive heat transfer in rocket, ramjet, or turbojet combustors.

The second aspect could be to use the oscillations to improve the extraction of energy from heat transfer limited systems such as nuclear power plants.

The problem of unsteady flow, convective heat transfer has been investigated experimentally to some extent in the past by such workers as Layton at Princeton, Romie at UCLA and more recently by Harrje at JPL. The first two investigators worked with non-combustive systems while research at JPL was based upon heat transfer from the products of hydrazine decomposition. In the JPL work, a marked dependence of the heat transfer upon the amplitude of the velocity oscillations in a resonating tube was found. The wave amplitudes attained in that study were never large enough to allow the sinusoidal pulsations induced in the apparatus to degenerate to shock type waves, thus heat transfer increases of less than 10% were observed.

Although that work is probably the nearest approach to date to a controlled duplication of the conditions present in a rocket motor during unsteady heat transfer, the amplitudes, wave shapes and frequencies did not reach the extremes that have been recorded in "screaming" rocket motors. Other experimental investigators dealt with pressure oscillation amplitudes and frequencies which were considerably smaller than those that have been encountered. Thus while previous experimental investigations were able to examine heat transfer phenomena associated with lower amplitude sinusoidal oscillations, much of the area covering heat transfer in the presence of high frequency, high amplitude shock-type waves had yet to be systematically explored.

More recently Condomines (at Princeton) conducted, as an adjunct to the rocket combustion instability program a brief but significant investigation of heat transfer phenomena in a "screaming" rocket motor for both the longitudinal and transverse modes of instability. In the longitudinal case, with a special heat transfer probe mounted in the rocket chamber wall, he found heat flux increases up to 200% accompanying peak-to-peak pressure wave amplitudes of up to 50% of the steady-state pressure. For the transverse mode of instability, heat fluxes greater than 300% of the stable heat transfer level were observed with correspondingly large wave amplitudes.

The investigations of both Harrje and Condomines point to the amplitude of the velocity pulsations in unsteady flow as a highly significant parameter influencing the convective heat transfer. Of particular interest is the case wherein the per cent amplitude of velocity fluctuations is great enough to either stop, or reverse the flow--a condition easily attained when large amplitude shock-type waves are present.

An inherent disadvantage to the use of combustion gases in any research of this type is that the effect of oscillating flow upon convective heat

transfer may be and probably will be superimposed onto heat flux phenomena of a purely combustive nature. It is clear that experimentally a need exists to separate the unsteady flow effects from the combustion effects. The influence of shock waves and large amplitude standing waves must be examined independently of combustion phenomena. This was the goal of the present research carried out at Princeton University.

At the same time, the effect of large amplitude oscillations on heat transfer was studied at Lewis Research Center (Cleveland, Ohio) by Feiler and Yeager (NASA) (Ref. 3). They were interested in the heat transfer from a heated flat plate to an airstream having large-amplitude flow oscillations also using a siren system; however, the siren in that case was used on the inlet flow. The oscillation frequency was varied from 34 to 680 cps. The rms oscillation amplitude was reported to be as great as 65% of the mean flow rate. In the presence of oscillations, the heat transfer coefficient was increased by as much as 65% when compared with the coefficient at the same mean flow rate in the absence of oscillations. The mean flow velocity ranged from about 30 to 100 ft./s. The Nusselt number was correlated by a function of the Reynolds number and the ratio of the rms amplitude of the velocity excursions to the mean flow rate.

$$Nu_x = f_1(\epsilon) Re_x^{f_2(\epsilon)} \quad (\epsilon = (pV)_{rms}/\bar{pV})$$

However, no effect of oscillation frequency was observed that could be attributed solely to frequency and not to the displacement of the node-antinode pattern. Of course these tests covered only a limited frequency range.

The research which has been carried out at Princeton University since July 1st, 1958 under sponsorship of the Power Branch, Office of Naval Research, has been conducted in a slightly different way: The pressure waves of a known

frequency and amplitude (comparable in many respects to those encountered in combustion instability) are induced axially in a duct of such a length that the waves will steepen into shock waves. The pressure waves are produced by varying periodically the area of a sonic orifice at the end of a resonating duct. The area variation is produced by the high speed rotation of a special disc which has slots cut in its periphery to correspond in size to that of the sonic orifice (siren arrangement). The heat transfer takes place from the duct walls to the flowing gas and is accomplished by electrically heating a strip of nichrome metal set flush into the walls of the duct. The temperature of the metal strip provided a means of measuring the heat transfer if the electrical power is kept constant.

In the early experiments conducted by E. Croke (Ref. 5), the heater section could not travel along the duct. However, through the use of additional duct sections it was possible to vary the resonant duct length between three and twenty-four feet; thus a wide range of resonant frequencies was available. Furthermore, this arrangement permitted the heater section to be placed at a number of different stations along the length of the duct so that a discrete survey of the heat transfer as a function of location in the resonant system was possible.

The fact that most of the experiments had been conducted with a turbulence grid consisting of a fine wire mesh supported on either side by a coarse mesh screen, explained the apparent absence of resonance phenomena in the duct, the upstream extremity of the duct behaved neither like a "closed" end nor like an "open" end, but absorbed the waves with no reflection whatsoever. The waves thus generated had an amplitude of only a few per cent of the steady-state pressure.

Croke fixed the heater section at a certain X location in the duct and

tried to correlate that local heat transfer increase to the maximum pressure oscillation intensity ($\tilde{P}_{drms}$) at different frequencies.

Because of the absence of strong resonant conditions in the duct, the axial location of the heater section was quite unimportant and such a correlation was actually of an "overall" nature (see p. 64 in the present report). The heat transfer variations observed by Croke may be attributed to frequency and wave shape effects rather than to the relative location of the heater section with regard to the node-antinode patterns associated with resonant conditions. The general conclusions from Croke's results are that the higher frequencies are more effective in improving heat transfer, since, although the intensity of oscillations decreases as f increases, the heat transfer ratio h/h_0 is either constant (high Mach number) or an increasing function (low Mach number).

A second set of experiments has been made by D. T. Harrje after removal of the turbulence grid, thereby obtaining resonant conditions in the duct (peak-to-peak amplitudes as high as 0.05 of steady-state pressure). Such oscillations produced heat transfer increases as high as three times the steady-state values and, by using three locations of the heater section, the increases were shown to be dependent on the proximity to the velocity antinode locations in the duct, with minimum increases at velocity nodes.

Hence, the most interesting conclusion of the report is that the heat transfer increase was found to reach its highest values in the vicinity of velocity antinodes. However, with only 3 data points along X , only tentative conclusions could be drawn as to the heat transfer profile along the duct. Hence, a great number of additional locations were necessary which led to the idea of the traversing heater section. Furthermore, as the velocity oscillations were pointed out to be the interesting parameter, it was necessary to be able

to measure the oscillating velocity (rather than to use a standard oscillating velocity-oscillating pressure relationship and pressure measurements).

A third set of experiments was conducted by R. C. Saunders (Ref. 7). A major portion of his efforts was involved in extensive modification of the apparatus necessary for the development of an adequate experimental tool to perform the desired research (traversing heater section, including a high frequency pressure instrumentation and a hot-wire anemometer probe capable of automatic radial traverse).

In his investigations, Saunders focused his interest on "combined" and linked modes of oscillation, longitudinal and transverse (tangential, standing and spinning, and radial).

Strong first radial modes have been observed in the case where the frequencies of these modes were almost exact multiples of the longitudinal mode.

Triggering of the spinning tangential oscillations by a wave deflector, to promote transverse modes, showed important increases of heat transfer (80% instead of 25%) and this occurred when the transverse mode was almost an integer multiple of the longitudinal mode.

Saunders, using the automatic traverse equipment, showed that the heat transfer sinusoidal-like pattern along and the oscillating pressure pattern were displaced by one-quarter wavelength with respect to each other. He drew the conclusion that the oscillating velocity amplitude pattern was in phase with the heat transfer pattern.

Based on the results of previous investigations, only a few of which were mentioned here, the aims of the present study may be better outlined. Foremost is the objective of a more precise determination of the various important parameters and their relationships to the heat transfer effects

observed. The experimental approach is dictated because of the difficulty involved in theoretically predicting heat transfer in turbulent, unsteady flow without limiting the number of variables involved. Even with such restrictions, a suitable theoretical treatment may still be unattainable. Thus, it is quite possible that the engineering solution based on dimensional analysis and experimental values will constitute the ultimate result of this research.

3. EXPERIMENTAL APPARATUS

31. General Description

Pressure oscillations are imposed on flowing air, in a high length to diameter ratio duct, by interrupting the air through an orifice at the exit. (Fig. 1) A traversing heater section is located about midway down the duct; the heat transfer is measured from an electrically-heated, flush-mounted, metal strip covering half the inner circumference of the cross section. Across from the heater strip on the duct wall are located a pressure transducer and a hot wire anemometer probe. The probe position can be continuously varied in the radial direction from the wall to the center of the duct.

32. Wave Generator

The resonance duct, made of brass tubing, is 9 feet long and has an inner diameter of 1.5 inches. It is fed from a 4.5 cu. ft. stagnation chamber connected to a controlled high pressure air supply. The flow is controlled by a large dome-loaded regulator which reduces the feed pressure from a 2700 psi supply source to any desired level. (Fig. 1)

The duct is terminated by a rectangular sonic orifice. Through the choice of various orifice sizes the Mach number of the flow in the duct may be controlled. At present this investigation is being carried out for low Mach number flows by the use of three orifices which permit three different Mach numbers: about 0.03, 0.07, and 0.1, respectively; for the small, medium and large orifice.

The sonic orifices are of a rectangular shape in accordance with the results of a design analysis which indicated that, for a given frequency, the steepest waves (i. e., those which will most readily degenerate to shocks) are formed by the rapid closing of a rectangular hole. (Ref. 5)

The "siren" arrangement consists of an 11.3" steel disc around the periphery of which are cut a series of equidistant rectangular orifices to correspond with the sonic orifice. (Fig.2) This disc is positioned such that a gap of a few thousandths of an inch separates it from the sonic orifice. As it rotates, it opens and (virtually) closes the sonic orifice in a periodic fashion. A degree of control of the wave amplitudes induced in the duct is obtained by remote control of the intra-orifice gap. (Fig.3) It is not possible to completely close the sonic orifice (i. e., requiring a reduction of the gap to zero), however, the spacing may be made small enough so that the leakage area with the orifice "closed" is a very small fraction of the total orifice area.

The revolving orifice is powered by a 15 hp Varidrive motor which drives the orifice shaft by a belt and pulley arrangement. There are three ways to control the magnitude of the frequencies induced in the duct:

First, the rpm of the Varidrive is subject to remote control via a servomotor: the induced frequency may, in this manner, be varied continuously over a range of several tens, hundreds or thousands of cycles depending on the disc used and the pulley arrangement.

Second, there are available three revolving orifices of 28, 4 and 2 holes, respectively. (Fig.2)

A third method of frequency range selection is made possible by the choice of several belt and pulley combinations in the Varidrive rotating orifice shaft linkage. At the present time the Varidrive output may be slowed down by either one-to-one, 3.5-to-one, or

(3.5)² 12-to-one pulley ratios. The combination of these three methods of frequency control makes available an extremely broad operating range, which spans from 5 to 5700 cps covering an interesting spectrum comparable to that associated with combustion instability oscillations. Note that the overlapping of some of the partial ranges often permits a choice of the most convenient disc and pulley arrangement for the purpose of the considered experiment (Fig. 5). Arrangements No. 5, 6, and 9 have been used in the present experiments.

33. Heat Transfer Unit

The heart of this unit is a 0.5 inch wide, 0.0125 inch thick ribbon of high resistance nichrome metal, rolled into a 1.5 inch diameter half-cylindrical tube to conform with the walls of the resonance duct. (Fig. 6) The width of the heater strip was 1.5 inch in the previous systems. (Ref. 5) It has been reduced for greater discrimination of the local heat transfer at the higher frequencies (p. 62). This ribbon is set into the walls of a cylindrical sleeve of machinable boron nitride ceramic having an internal diameter of 1.5 inch and a wall thickness of 0.5 inch. The boron nitride sleeve forms a part of the resonance duct itself when the heater unit is installed in the duct. While this material exhibits excellent high temperature capabilities, unfortunately it has a rather high heat capacity and a high thermal conductivity. In order to prevent excessive heat loss from the nichrome element to the boron nitride sleeve, it was necessary to provide a 1/16 inch layer of mica and asbestos between the nichrome and the ceramic. The nichrome element, the asbestos and the boron nitride are bonded together with high-temperature cement.

Electric power to heat the nichrome ribbon is delivered through two 3/8 inch diam. silver electrodes. This power is supplied from a 440 volts 2.5 kW, 60 cycles AC source. It is passed through a step-down transformer which delivers low voltage, high current power to the heater unit. Present operation has not required more than about 100 W of the available power.

Regulation of the energy delivered to the heater is accomplished through a servo-driven Variac transformer rated at 4.8 kVA. The servomotor is powered by a Brown DC amplifier, the output of which is controlled through a precision potentiometer operated from outside the test pit at the control console. A further refinement of the power control system lies in a voltage regulation circuit which serves to isolate the power delivery from fluctuations in line voltage.

The traveling heater section is located about midway down the duct (Fig. 7a) and has a maximum traverse distance of about 18 inches in the 9 feet length of the duct. A drive mechanism can continuously vary its axial position. All operations can be performed remotely from outside the test pit using a control console. Under typical operation the heater section automatically reverses traverse direction at the arbitrarily set traverse limits in a continuous fashion. In manual operation, it can be stopped at any position of the traverse region of the tube.

34 Instrumentation

341 Steady flow parameters.

3411. The steady-state pressure in the duct $\bar{P}$ and the pressure in the supply tank P_t are obtained from Bourdon-type Heiss gauges.

3412. The gas stagnation temperature T^0 in the duct is measured by a copper-constantan thermocouple immersed in the flow immediately downstream of the entrance. These data are recorded on speedomax charts.

3413. The mass flow $\dot{m}$ in the duct is determined by means of a sonic orifice located upstream of the plenum chamber. A Bourdon gauge yields the upstream pressure P_u . In addition, the orifice itself serves to insulate the high pressure source from the effects of downstream flow disturbances. Four sonic orifices are available; their diameters are, respectively, $D_u^* = 0.2$; 0.292; 0.746 and 1 inch. The mass flow calculation is classical:

$$\dot{m} = \rho_u^* A_u^* V_u^* = \rho_u^* A_u^* a_u^* = (P_u^*/RT_u^*) A_u^* \sqrt{\gamma RT_u^*} = \gamma P_u^* A_u^* / a_u^*$$

$$a_u^* (\text{ft/sec}) = 49.02 \sqrt{T_u^* (^{\circ}\text{R})} \quad \text{with,}$$

$$T_u^0 / T_u^* = \left(1 + (\gamma - 1) M_u^{*2} / 2 \right) = (\gamma + 1) / 2 \quad \text{for } M_u^* = 1; \text{ hence,}$$

$$a_u^* = 44.7 \sqrt{T_u^0} = 44.7 \sqrt{T^0} . \text{ now}$$

$$P_u^* = (P_u^* / P_u^0) (P_u^0 / P_u) P_u ; \text{ hence}$$

$$\dot{m} = (\gamma / 44.7) (P_u^* / P_u^0) (P_u^0 / P_u) A_u^* (P_u / \sqrt{T^0}) = K P_u / \sqrt{T^0},$$

The coefficient K depends on the orifice which is used.

The ratio $P_u^* / P_u^0 = 0.52828$. Expressing $\dot{m}$ in lb/sec, P_u in psia and T^0 in $^{\circ}\text{R}$, one obtains the following results:

D_u^* (in.)	$A_u / A_u^* = (D_u / D_u^*)^2$	A_u (sq. in.)	P_u / P_u^0	Mass Flow Formula
0.2	39.7	0.0314	0.99980	$\dot{m} = 0.01675 P_u / \sqrt{T^0}$
0.292	18.6	0.067	0.99930	$\dot{m} = 0.0357 P_u / \sqrt{T^0}$
0.746	2.85	0.436	0.97014	$\dot{m} = 0.24 P_u / \sqrt{T^0}$
1.000	1.58	0.787	0.89398	$\dot{m} = 0.47 P_u / \sqrt{T^0}$

3414. The Reynolds number in the duct can easily be deduced from the mass flow:

$$Re = \rho \bar{V} D / \mu = \dot{m} D / A \mu = \dot{m} D / (\pi D^2 / 4) (\mu / \mu_0) \mu_0$$

D being the duct inner diameter, $D=1.5$ inches and μ_0 the viscosity of air at $32^\circ\text{F}=0^\circ\text{C}$, $\mu_0 = 0.350 \times 10^{-6}$ lb. sec/ft². Hence,

$$Re = 0.9025 \times 10^6 \dot{m} / (\mu / \mu_0)$$

$\dot{m}$ being expressed in lb/sec. μ/μ_0 is a function of the gas temperature:

$$\mu / \mu_0 = f(T), \text{ where } T \simeq T^\circ \text{ as } M \ll 1.$$

T (°R)	μ/μ_0
360	0.7742
378	0.8070
396	0.8391
414	0.8706
432	0.9015
450	0.9319
468	0.9617
486	0.9909
504	1.020
522	1.048
540	1.076
558	1.103

3415. The Mach number in the duct can also be deduced from the mass flow:

$$\dot{m} = \rho \bar{V} A = (\bar{P}/RT) Ma A = \gamma \bar{P} A M / a,$$

$$M = \dot{m} a / \gamma \bar{P} A = 0.616 \dot{m} \sqrt{T} / \bar{P}$$

with $\dot{m}$ in lb/sec; T in $^{\circ}R$ and $\bar{P}$ in psia.

In fact, one measures T° and not T ; M can be calculated in the following way:

Since $M \ll 1$, a first approximation is :

$$M_1 = 0.616 \dot{m} \sqrt{T^{\circ}} / \bar{P},$$

and:

$$M = 0.616 \sqrt{T^{\circ} / (1 + M_1^2 / 5)} / \bar{P} \text{ (second approximation)}$$

Note that, as $\dot{m} = K P_u / \sqrt{T^{\circ}}$,

$$M = K' (P_u / \bar{P}) \sqrt{T / T^{\circ}} \simeq K' P_u / \bar{P}.$$

Hence, the Mach number in the duct is proportional to the ratio of what we have called the "upstream pressure" P_u over the "downstream" or duct pressure $\bar{P}$.

The Mach number in the duct is imposed by the "apparent" area of the downstream sonic orifice. This "apparent" area is defined in the following way: when the siren is on, the main flow through the downstream orifice area varies in a periodic fashion, due to the orifices of the siren wheel. Everything happens as if the downstream orifice area was contracting and expanding with time, as shown in sketch g:

Sketch g

The mass flow in the duct corresponds to the average mass flow through the downstream orifice, and hence, to a certain virtual or "apparent" area $\bar{A}_d^*$ of this orifice.

With no intra-orifice gap at all, A_d^* minimum would be equal to zero, because once each period the orifice would be completely closed (Sketch 10).

Sketch 10

Sketch 11

Sketch 12

The effect of the gap is not only to reduce the peak-to-peak amplitude $A_d^* \max - A_d^* \min$, but also to increase $\bar{A}_d^*$ so as to keep $A_d^* \max$ nearly constant ($A_d^* \max$ corresponds to no obstruction at all, and it happens once in a period in all cases, considering no gas inertia). (Sketches 11 and 12)

Hence, considering the duct plus the downstream orifice system, like a diffuser plus a throat, we see that if the orifice is sonic, the average Mach number in the duct M is constant because the area ratio $A/\bar{A}_d^*$ is fixed, A being the duct area.

Actually, the sonic conditions are not obtained exactly at the fixed orifice but rather at a small distance downstream of this orifice, otherwise the perturbations generated by the revolving disc would not propagate upstream i. e., generate waves in the tube. But our approximate treatment only speaks of an "apparent" area, corresponding to a certain flow, and not of the actual area, and hence is still valid.

Most of the present investigations have been carried out with a constant gap of 0.019 inches, which is a reasonable minimum spacing to limit possible contact, and approach the condition for the strongest oscillations.

It should be noted, however, that it has been impossible with the present apparatus to vary continuously the oscillations amplitude in the tube by varying the intra-orifice gap, without changing the Mach number, i. e., one of the parameters we should like to keep constant. This is due to the fact that any change in the gap G , changes $\bar{A}_d^*$ and hence M (Fig. 4).

On the other hand, by varying the gap one could change continuously the Mach number around the values imposed by the choice of the downstream sonic orifice. Unfortunately, however, it would be impossible then to keep the oscillations amplitude constant.

Therefore, the fixed rectangular orifice size is now being altered by movement of one side of the rectangle to maintain a constant Mach number as

the gap is changed. Thus, we may attain the goal of constant M , variable amplitude, or, if so desired, constant amplitude, variable M . (Fig. 3)

The object of the experimentation is to compare the heat transfer with oscillations to the heat transfer without oscillations, under the same steady-state conditions (same M and $\bar{P}$). It is necessary to obtain those conditions in the steady-state case. The position of the rotating disc is adjusted so that the downstream sonic orifice is partially closed. When the "apparent" area is $\bar{A}^*$ (same as in the case of oscillating flow), the same $\bar{P}$ and M are obtained.

342. Heat transfer parameters.

The particular parameter which the wall-to-gas heat transfer unit is designed to investigate is the boundary layer film coefficient, i. e., the heat transfer per unit time, per unit area, per degree of temperature difference between the nichrome ribbon and the bulk of the flowing gas:

$$h = (dQ/dt)/A \Delta t$$

This definition makes clear just which of the measurable parameters must be taken as data:

3421. The quantity dQ/dt may be taken as the power delivered to the nichrome ribbon to maintain it at a sufficient temperature while it transmits energy as heat to the flowing gas (corrected for heat losses by conduction to the heater unit, or by radiation. (See p. 38)

First, the voltage E_h across the ribbon is measured and recorded during each run on a self-balancing potentiometer which is equipped with a zero suppressor, so that the width of the potentiometer chart covers a span of only one volt while there may be as many as eight volts across the ribbon. The ribbon voltage may be measured to within 1% accuracy by this method.

$$E_h (\text{volt}) = \gamma + (E/100)(\delta - \gamma) \quad , \text{ where,}$$

E is the voltage measurement in percent of total scale,

γ is the voltage corresponding to 0% scale, expressed in volts,

δ is the voltage corresponding to 100% scale, also in volts.

In the present experimentation E was of the order of 1.5 volts.

Calibration of the potentiometer is performed before each run and drift has been found to be very slight.

In a similar fashion, the current I_h through the nichrome ribbon is recorded on a self-balancing potentiometer as a voltage drop across a known resistance. Accuracy of the same order as the voltage is possible for the current measurement. This potentiometer is also calibrated from a known source immediately before a test run.

$$I_h (\text{amp.}) = 100(\alpha + (I/100)(\beta - \alpha)) \quad , \text{ where,}$$

I is the current measurement in percent of total scale, and α and β have the same definition as γ and δ . I was about 70 amp during the present experiments.

The product of voltage and current yields the power input to the heater unit:

$$(dQ/dt)_{\text{WATT}} = I_h E_h$$

As the previous investigators (Ref. 5, 7, and 8) have used the English units, we shall also express dQ/dt in English units, i. e:

$$(dQ/dt)(B.T.U./sec.) = 0.948 \times 10^{-3} (dQ/dt)(Watt)$$

Since the voltage is measured from terminals set into the stainless steel strip ends, the power loss in the delivery system is not recorded as a part of the ribbon voltage.

A second method of power measurement is the use of a sensitive wattmeter which measures voltage across the ribbon directly and which measure current through a 40:1 current transformer coil wound on the main power delivery cable. The wattmeter may be read to an accuracy of 5% and is used primarily as a device for holding the power input at a given level during a test run.

Let us point out that, in all the runs we have carried out, E_h and I_h , once set, did not significantly vary during a traverse of the heater section. Thus only one value of the power has been taken for different values of the X position (constant power operation).

3422. The quantity ΔT represents the difference in temperature between the nichrome ribbon and the bulk gas flow:

$$\Delta T = T(\text{ribbon}) - T(\text{gas}) = T_h - T \simeq T_h - T^o$$

The ribbon temperature (usually maintained in the range: $350^{\circ}\text{K} \div 550^{\circ}\text{K}$ i. e. $170^{\circ}\text{F} \div 530^{\circ}\text{F}$) is measured from a calibrated platinum resistance temperature gauge cemented at the back of the heater strip. This has replaced the three spring loaded chromel-alumel thermocouples used in the previous experiments (Ref 5), the life of which was limited to two to ten hours because of the severe conditions of temperature and vibration. The resistance gauge life-time is much longer, but depends on the frequency range of interest; the most severe range, from this point of view is the low frequency range, especially when flow reversal is present and if the duct pressure is large.

3423. The wetted area of the heater strip is:

$$A = \pi R h = \pi \times 0.73 \times 0.5 = 1.15 \text{ sq. in.}$$

Sketch 43

3424. Finally, the heat transfer coefficient is:

$$h = (dQ/dt) / A \Delta T = 0.948 \times 10^{-3} (dQ/dt) / 1.15 \Delta T, \text{ with,}$$

h in $\text{BTU} \times (\text{sec})^{-1} \times (\text{sq. in})^{-1} \times (^{\circ}\text{R})^{-1}$, dQ/dt in Watts and ΔT in $^{\circ}\text{R}$.

3425. The Nusselt number:

$$N_u = hD/k = hD/(k/k_0) k$$

can be deduced from h. D is the duct diameter (D=1.5 inches), k is the thermal conductivity of air, k/k_0 is given as a function of the gas temperature by:

T ($^{\circ}$ R)	k/k_0
360	0.7494
378	0.7846
396	0.8196
414	0.8543
432	0.8885
450	0.9225
468	0.9561
486	0.9894
504	1.022
522	1.055
540	1.087
558	1.119

with $k_0 = 3.23 \times 10^{-7}$ BTU X (sec) $^{-1}$ X (in) $^{-1}$ X ($^{\circ}$ R) $^{-1}$.

3426. Hence, with the power input held constant, it is evident that the most sensitive and significant test datum is the ribbon temperature which increases or decreases as the heat transfer falls or rises. In the interpretation of the measurements of this temperature T_h one has to keep in mind the following:

It is impossible to achieve perfect thermal insulation between the heated ribbon, the boron nitride sleeve and the container. Thus, there is a loss of heat to the surroundings not fully accounted for by convective heat transfer. This error may be compensated for by allowing the heater unit to warm up i. e.,

by permitting it to come up to an equilibrium temperature before beginning a test, so that conductive heat transfer from the ribbon to unwanted heat sinks will be a minimum.

Sketch 14

This small conductive heat transfer loss has been compared to the convective heat transfer itself, measuring the powers necessary to bring the strip to the same equilibrium temperature, $T_h \text{ equilibrium}$ (Sketch 14), respectively, without and with flow in the duct. The first measurement was the conductive heat transfer alone; the second measurement, the sum of the conductive and the convective heat transfer. Because of the efforts of previous investigators (Ref 5, 7 and 8) to improve the thermal insulation, this loss has been found to be very small and, as it was planned to draw qualitative rather than quantitative conclusions from the experiments, the correction for this loss did not appear to be imperative. However, the "warm up" method certainly reduces considerably the losses during the run; it even cancels these losses in the case when the axial location of the heater section is fixed (fixed value of $T_h \text{ equilibrium}$) or during a traverse of the heater section in steady-state conditions (because in this case h and, hence, $T_h \text{ equilibrium}$, are nearly constant along x). However, during a traverse in x at finite speed and in oscillating conditions (and this is usually the case), the instantaneous difference $T(\text{strip}) - T(\text{surroundings})$ induces, since $T(\text{strip})$ varies heat transfer by conduction which superimposes itself to the heat transfer by convection. This obviously

introduces an error. But, if the instantaneous $T(\text{strip}) - T(\text{surroundings})$ is kept small, because of the warm-up period, i. e., if one waits until $T(\text{surroundings}) \approx \bar{T}(\text{strip})$, average in time of $T(\text{strip})$ during a traverse, then one can approach the case of perfect insulation.

Even with this method to improve thermal insulation, the heat capacity of the strip itself introduces a thermal lag. The existence of such a lag can be easily shown: To improve the accuracy on the measurement of all the parameters, it has been decided to make the average of the results for two traverses. The heater section travels from $X = \frac{X}{L} = 0\%$ (near the duct end) to $X = 100\%$, and back, from $X=100\%$ to $X=0\%$.

Sketch 15

The two measurements corresponding to the same X position ($X_1=X_2$), but obtained respectively at the first and at the second passage, are in perfect agreement, as far as the oscillating pressure and velocity are concerned. However, the two measurements T_{h1} and T_{h2} of the strip temperature do not agree (the difference $T_{h1} - T_{h2}$ has been exaggerated on sketch). This introduces, when the average is made, a systematic error on T_h . Thus, the effect was to flatten and, even, completely transform the axial T_h variation. In particular, the point E (Sketch 15), respect to which the curve $T_h(X)$ is symmetrical does not coincide with $X=100\%$.

Let us suppose that the system can be described as a linear first order system (with inertia only), characterized by its time constant τ , or its transfer function:

$$1/(1 + \tau p)$$

Let us call the input T_h equilibrium (t), i. e., the temperature that the ribbon should take at time t (when located at position $x=vt$, where v is the traverse speed) if the ribbon had no heat capacity, or if the section was stopped and if the equilibrium temperature was reached. The output is our measurement $T_h(t)$.

Let us represent the function T_h equilibrium (t) with a general sinusoidal wave (Fig. 9 shows that this is not far from reality). Since the maximum and minimum of T_h equilibrium (t) correspond to the minimum and maximum of heat transfer (i. e., to the antinode and the node of oscillating pressure), the period of the sinusoid is:

$$T = \lambda / 2 v$$

where λ is the wavelength corresponding to the frequency of oscillations f , $\lambda/2$ is the distance between a node and an antinode i. e., between two extremes of T_h equilibrium (t), v is the traverse speed,

The angular frequency of the sinusoid is then:

$$\omega = 4\pi v / \lambda \quad (1)$$

The output $T_h(t)$ is also a sinusoid, but its amplitude is reduced by the ratio $k = (T_h)_{\max} / (T_h \text{ equilibrium})_{\max}$ and it presents a phase-lag φ as shown on Sketch 16 and on the classical Nyquist representation (Sketch 17)

Sketch 16

Sketch 17

$$k e^{i\varphi} = 1/(1 + \tau i\omega) = (1 - i\tau\omega)/(1 + \tau^2\omega^2)$$

$$\text{or, } k(\cos\varphi + i\sin\varphi) = 1/(1 + \tau^2\omega^2) - i\tau\omega/(1 + \tau^2\omega^2)$$

$$\text{and hence: } \begin{cases} k = 1/\sqrt{1 + \tau^2\omega^2} \\ \tan\varphi = -\tau\omega \end{cases}$$

Determination of the time constant $\tau = -(\tan\varphi)/\omega$: For a given experiment, ω is known (Equation 1p.41) and one can determine φ by two methods:

A. If point E (Sketch 15) is sufficiently well determined:

$$\varphi = 2\pi(t_E - t_{100\%})/T = 2\pi\theta/T \text{ radians.}$$

B. Another way is to shift the whole $T(t)$ curve toward negative time until the values T_{h1} and T_{h2} agree, i. e., until one obtains:

$$T_{h1}(t - \theta) = T_{h2}(t - \theta)$$

This method is more accurate than method A, but takes longer. It has been performed for one particular sequence (run 5, sequence 1) with $f = 262\text{cps}$ (Fig. 34) θ has been found to be -5.6 sec. and $T = 76$ sec. Hence,

$$\varphi = -2\pi \times 5.6 / 76 = -0.464 \text{ radians.}$$

$$\tan \varphi = -0.5$$

$$\omega = 2\pi / T = 2\pi / 76 \text{ radian} \times (\text{sec})^{-1}$$

$$\tau = -(-0.5) \times 76 / 2\pi \approx 6 \text{ sec.}$$

Elimination of the thermal lag effect:

If one wants to have a reasonable response without any correction, it is obvious that one has to keep φ small i. e., $\tau\omega$ small. As τ is fixed we require a small value of ω , where $\omega = 4\pi\nu/\lambda$. Hence, there is a relation between the driving rate ν of the heater section, and the wavelength of the oscillations in the duct; the smaller the wave length, i. e., the higher the frequency of oscillations, the smaller the required driving rate.

But one can eliminate very easily one part of the heat capacity effect, i. e., the one relative to the phase-lag, using method (A) described previously. Method (B) would prove to be excessively long, although slightly more accurate.

One effect of the heat capacity still remains, it is the amplitude reduction; but, using a driving rate approximate to the wavelength i. e., keeping φ small, we can consider this a second order effect:

$$\text{Since: } |\tan \varphi| = \tau\omega < 1, \tau^2\omega^2 \ll 1,$$

$$\text{and } k = 1 / \sqrt{1 + \tau^2\omega^2} = 1 + (\text{order of } \varphi^2).$$

343. Oscillating parameters

3431. The frequency measurement is determined in terms of rpm of the Varidrive motor. A two pole magnet affixed to the shaft induces a voltage in a coil mounted adjacent to it, as the magnet rotates. The output

of the coil is electronically integrated and recorded as a DC signal on a self-balancing potentiometer. Visual monitoring of the frequency is made possible by a $\%$ rpm meter mounted on the control console. Accurate frequency measurement is made by comparison on the oscilloscope of the pressure signal P'_d (or P'_h) with a signal of known frequency (wave generator and Berkeley counter)

3432. Oscillating pressure:

A Norwood air-cooled straingauge pressure transducer is located just upstream of the sonic orifice at the duct exit. This location corresponds, as we shall see later (p. 57), to a pressure antinode of oscillations, regardless of the frequency ("closed"-end). Hence, the value of $\tilde{P}_d$ yields a measurement of the intensity of oscillations in the duct. Displayed on an oscilloscope screen and measured through a true rms meter, this parameter P'_d is used to adjust the frequency, so as to be at resonant conditions (maximum $\tilde{P}_d$ rms).

A Kistler pressure transducer (sensitivity: $3.52 \times 30 = 105.6$ mV/psi) is located in the heater section itself. The oscillating pressure signal P'_h is sent to a rms meter and a peak-to-peak meter, so that continuous curves of $\tilde{P}_h$ rms and $\tilde{P}_{hPK}$ can be obtained versus duct longitudinal location x , when the heater section makes a traverse (Fig. 13 and 14).

The signal P'_h can also be displayed on an oscilloscope screen and photographed by a polaroid camera. This gives a pictorial record of the wave shapes induced in the duct (Fig. 74).

3433. Oscillating velocity:

The hot-wire anemometer probe is connected externally to a drive mechanism which can continuously vary radial position and indicate the distance of the

wire from the wall or to the center of the duct. In automatic operation the hot-wire anemometer automatically reverses radial traverse direction at the arbitrary set traverse limits and continues traversing until stopped. In manual operation, it can be stopped at any $Y=y/R$ radial location.

The Model HWB2 hot-wire anemometer of the Flow-Corporation was used in this study. It is a multi-purpose precision instrument for use in measuring fluid velocity, including steady velocity, instantaneous velocity (small or large fluctuations) and turbulence intensity. Provisions have been made so that accurate quantitative measurement may be readily obtained; qualitative or semi-quantitative observations of velocity fluctuations are extremely simple to make. This last point was developed in References 7 and 8, hence, it seemed necessary to focus our current study on quantitative measurements which are absolutely necessary if one wants to make a comparison between the results obtained for different sequences.

The simplest hot-wire circuit is shown on Sketch 18:

Sketch 18

Battery B supplies electric current, measured by a meter I, and controlled by rheostat A, to a probe whose working end consists of a thin wire (we have

used a tungsten filament 0.00015 inches in diameter). Because of its small diameter, the wire very rapidly attains an equilibrium state in which its final temperature and resistance depend on the stream velocity. Hence, the resistance of the hot-wire depends on the wire temperature, which is fixed by the amount of stream cooling, so that the hot-wire equilibrium resistance is a measure of the stream velocity. If the hot-wire is, for example, a hundred times smaller than the control resistor A, then the wire current will remain constant, despite changes in the wire resistance, and the voltage E itself becomes a direct measure of stream velocity.

Alternatively, if the heating current is adjusted to produce a chosen fixed resistance of the wire, then, the amount of heating current flowing through the wire will be a measure of stream velocity.

In Model HWB2 anemometer, both techniques are used. The circuit is arranged so that the wire current is independent of hot-wire resistance changes, and depends only on the settings of front panel current controls. For the measurement of steady velocity $\bar{V}_h$, the recommended procedure is to set the current to produce a chosen fixed resistance and use the current as the measure of velocity. For examining instantaneous velocity V_h' in a fluctuating stream, the procedure is to set the constant current to produce some desired time-average wire resistance and use the resistance fluctuations (i. e., voltage fluctuations E; at constant current I) as a measure of fluctuating velocity.

The cooling of a heated filament depends not only on the stream velocity, but also on the state variables, temperature and pressure, which determine the properties of the air stream. Operating the wire, not at a fixed resistance, but at a fixed resistance ratio r , i. e., a fixed ratio of the resistance R of the wire heated, to the resistance R_E of the wire at equilibrium with the stream, eliminates any significant effect of stream temperature on average velocity measurements. The wire heating current then depends only

on the product of pressure and velocity (this holds true for streams where the Mach number is less than 0.3, which is our case).

In the accurate (1% or 2%) measurements of absolute average velocity, an experimental calibration curve is required for each hot-wire filament; but for measurements of moderate accuracy (5%), at low Mach number, a suitable calibration curve can be easily generated from a single measurement of wire current I_0 in still air (temperature T_0), using the formula given in Reference 11 :

$$(2) \quad V = \left\{ \left[\left(\frac{I}{I_0} \right)_r^2 - 1 \right] \frac{T}{T_0} \right\}^2 / CP, \text{ where,}$$

V is the stream average velocity $\bar{V}_h$; I is the hot-wire current, I_0 is the hot wire current at the following conditions: air velocity $V=0$, air temperature $=T_0$; T is the air stream total temperature T^0 ; C is a calibration constant ($C=0.0505$ for our tungsten probe, 0.00015 inches in diameter); r is the resistance ratio; P is the air stream static pressure, in atmospheres:

$$P(\text{atm.}) = \bar{P}(\text{psia}) / 14.7$$

For different values of r , one obtains the family of curves shown on Sketch 19:

Sketch 19

Sketch 20

Measurements of oscillating velocity:

In order to make measurements of the oscillating velocity, we operate at constant current through the wire, hence, we must crossplot the family of calibration curves shown in Sketch 19 to the family shown in Sketch 20.

When the fluctuations are very slow, the instantaneous resistance ratio is given by the steady velocity calibration, i. e., as the instantaneous velocity changes, the operating point moves up and down the constant current curve, as seen in Sketch 20. If the amplitude of the fluctuations is small, the calibration wave may be replaced approximately by a straight tangent line. For very rapid fluctuations (above about 500cps), because of the heat capacity of the hot-wire, there is some thermal inertia present. The rate at which the wire resistance changes becomes the true measure of how much the instantaneous velocity differs from the average velocity. For this reason a compensation circuit is included in the amplifier which automatically takes the time derivative of the high frequency components in the wire signal and, thereby, gives a true representation of the high frequency velocity fluctuation.

For the measurement of the large scale velocity fluctuations themselves, the result given in Reference 10, p. 17 has been used:

The "equivalent" change I_r (in current I) at constant resistance ratio $r=R/RE$, corresponding to the voltage deflection e (signal of the hot-wire operated at constant current I), is given by:

$$\Delta I_r = ct. = I \left(\sqrt{1 - (2i/I)(e/e_0)^2} - 1 \right) \quad (3)$$

where e_0 is half the total distance between positive and negative plateaus of the square wave sent through the hot-wire under operating conditions (same r , same I), to eliminate the unknown i. e., the resistance of the wire under those conditions.

i is the square-wave current, given by:

$$i(\text{mA}) = 3.05M / C (1 + Br/400) \quad , \text{ where,}$$

M is the square wave switch position (1 or 4), B the cold balance bridge null dial setting and r the resistance ratio.

Hence, if i and I are known, and e/e_0 is measured from, say, the oscilloscope pattern, the quantity $\Delta I/r = ct$ can be calculated and, through use of the calibration curve, the fluctuation in velocity can be found as shown on Sketch 21:

Sketch 21

$$\bar{V} = \left\{ \left[\left(I/I_0 \right)^2 - 1 \right] T/T_0 \right\}^2 / CP$$

$$\bar{V} + \Delta V = \left\{ \left[\left((I + \Delta I)/I_0 \right)^2 - 1 \right] T/T_0 \right\}^2 / CP$$

But now, one must face the following problem: during the heater section traverse, only the rms and peak-to-peak values of the hot-wire signal e' are recorded. To apply formula (3) one must know the position of the axis t corresponding to $\Delta I = 0$ (Sketch 22).

Sketch 22

With the present apparatus, it is impossible to measure quantities such as e'' and e''' , which may not be equal. Hence, we have made the simplifying assumption that:

$$e'' = e''' = e_{(\text{peak to peak})} / 2 ,$$

and therefore, to obtain the maximum and minimum excursion of the velocity, we have plugged into formula (3) respectively:

$$e = - e_{(\text{PEAK TO PEAK})} / 2$$

, and,

$$e = + e_{(\text{PEAK TO PEAK})} / 2$$

So, we obtain the following results:

$$\Delta I_1 = I \left(\sqrt{1 - (i/I)(e_{PK}/e_o)} - 1 \right) < 0$$

$$\Delta I_2 = I \left(\sqrt{1 + (i/I)(e_{PK}/e_o)} - 1 \right) > 0$$

and, if, $I_1 = I + \Delta I_1 < I$

$I_2 = I + \Delta I_2 > I$, , we obtain,

$$V_1 = \left\{ \left[\left(I_1 / I_0 \right)^2 - 1 \right] T / T_0 \right\}^2 / CP < \bar{V}$$

$$V_2 = \left\{ \left[\left(I_2 / I_0 \right)^2 - 1 \right] T / T_0 \right\}^2 / CP > \bar{V}$$

and finally the peak-to-peak amplitude of velocity fluctuation is:

$$\tilde{V}_{PK} = V_2 - V_1, \text{ as shown on Sketch 23.}$$

Sketch 23

The quantity e_0 appearing in the formulas is shown on sketch 24 for a perfect square wave.

Perfect square wave signal.

Sketch 24

Actual square-wave signal.

Sketch 25

Since the square wave was not perfectly shaped, (Sketch 25) it was felt that e_{opk} as determined by the peak-to-peak meter could be improved upon as a measure of the square wave amplitude. e_{orms} , however, remained well defined for both the cases of the perfect square waves and those possessing sharp spikes because of turbulence as other causes. Therefore, e_o has been taken as:

$$e_o = (e_o \text{ PK})_{ideal} / \sqrt{2} = (e_o \text{ rms})_{ideal} \approx (e_o \text{ rms})_{actual}.$$

In the application of this method of velocity oscillation measurements, one must be aware of three important problems:

A. In the case of large velocity fluctuations, the average velocity indicated by the hot-wire may not coincide with the true time-average of the stream velocity because of the curvature of the calibration curves (Non-linearity). This is shown on Sketch 26 where we have established the correspondence between the hot-wire signal e' , the "equivalent" change in current ΔI and the velocity V :

Sketch 26

In the case of small oscillations, of course, one operates in a linear range, using the tangents of the calibration curves. In that case:

$$(i/I)(e/e_0) \ll 1, \text{ and, from (3),}$$

$$\Delta I|_{r=ct} / I \simeq 1 - (i/I)(e/e_0) - 1 = -\frac{i}{I}(e/e_0)$$

and, from (2),

$$\Delta V/V = 2\Delta[(I/I_0)^2 - 1] / [(I/I_0)^2 - 1] = 2[2(I/I_0)(\Delta I/I_0)] / [(I/I_0)^2 - 1]$$

or, since $(I/I_0)^2 \gg 1$, $\Delta V/V \simeq 4\Delta I/I$.

Hence,

$$\Delta V/\bar{V} \simeq -4(i/I)(e/e_0)$$

This is the simplified formula given in Reference 8 p. 19. In that case $\bar{V} \text{ measured} \simeq \bar{V} \text{ actual}$. But, in our case, we are often dealing with $(i/I)(e/e_0) \simeq 0.3$ or more. Hence, the complete non-linear treatment must be considered and $\bar{V} \text{ measured} \neq \bar{V} \text{ actual}$. This can explain the small discrepancies encountered in the average velocity profiles $\bar{V}(y)$ at different x locations in the duct under oscillating conditions (Fig 56 and 57).

B. Because of the non-linearity, the shape of the oscillating velocity signal, when displayed on the oscilloscope screen, is not the shape of the actual velocity fluctuation. In particular, $\tilde{V}_{rms}$ has been calculated in the following way:

$$\tilde{V}_{rms} = \tilde{V}_{pk} (e_{rms}/e_{pk})$$

The e quantities are recorded on the rms and the peak-to-peak meters). Since:

$$\tilde{V}_{rms} / \tilde{V}_{pk} \neq e_{rms} / e_{pk}$$

the value for $\tilde{V}_{rms}$ thus obtained is only an approximation.

To know how far we are from reality, oscilloscope photographs were taken of the velocity signal $e(t)$:

Sketch 27

$$e(t) = \bar{e} + e'(t) \quad \text{with} \quad \bar{e} = \frac{1}{T} \int_0^T e(t) dt$$

The true velocity oscillation has been determined using the formulas previously presented:

$$\Delta I = I \left(\sqrt{1 - (i/I)(e'(t)/e_0)} - 1 \right)$$

and,

$$V = \bar{V} + V' = \left\{ [(I + \Delta I)/I_0]^2 - 1 \right\} T/T_0 \}^2 / CP$$

Thus, the true peak-to-peak amplitude and rms intensity have been taken as:

$$(\tilde{V}_{pk})_{actual} = V_{max} - V_{min}$$

$$(\tilde{V}_{rms})_{actual} = \left[(1/T) \int_0^T V'^2 dt \right]^{1/2}$$

and have been compared to the values obtained from the coarse method.

Discrepancies up 5% on $\tilde{V}_{pk}$ and 10% on $\tilde{V}_{rms}$ have been encountered.

Hence the measurement of the oscillating velocity can't be done with a high accuracy using the meters alone (unless with electronic devices correcting for the non-linearity). It could be improved if it were directly made on the oscilloscope because such quantities as e'' and e''' (Sketch 22) could be measured. But that would be extremely time consuming. However, the oscilloscope is absolutely needed in case of flow reversal because of the rectifier action of the hot-wire. Both peak-to-peak and rms meters give an incorrect value in that case (p. 75).

C. It must be pointed out that all absolute measurements of oscillating velocity require a correct setting of the resistance ratio, i. e., an adaptation of the hot-wire current I to the average velocity $\bar{V}$ at the point of interest. This setting, which can't be done automatically, must be changed if one changes the location, i. e., if $\bar{V}$ changes. This requirement greatly limits the possibilities of automatic movements of the probe. At a fixed axial location X in the tube, it is impossible to vary automatically and continuously radial position y , while making absolute measurements, because $\bar{V}$ varies with y . However, at a fixed radial location y , it is possible to make a traverse in X , since $\bar{V}$ corresponding to this y varies only slightly with X . In particular, in runs 4, 5, 6, 7 and 9, where the probe was located at $Y=y/R=100\%$, i. e., in the core flow, the resistance ratio setting and the corresponding hot-wire current measurement was required to be done only once for a complete X traverse. In run 10, the current setting has been done for each y location and the traverses in X have been carried out at fixed y . Comparisons between oscillating velocity profiles $\tilde{V}(y)$ obtained at a fixed hot-wire current setting and at correctly adjusted current are presented in Figures 58, 59, 60 and 61.

Conclusions from Figure 60 are that there is practically no difference between the discrete investigation ($Y=0, 5, 10, \dots, 100\%$) and the continuous investigation (probe in-and-out) for a fixed current $I=I(25\%)$. But both methods show a great discrepancy with respect to the accurate method (I adjusted at each Y location). The results agree only for $Y \approx 25\%$ since in the two former methods I was set at that value.

The use of the two former methods can lead to erroneous conclusions: for example, in Figure 60 one could find an "annular effect," i. e., a maximum of $\tilde{V}_{rms}$ near the wall, where there is none (as shown by the latter method).

However, the plots of $\tilde{V}/\bar{V}$ versus y are in good agreement for the three methods (Fig. 61). This may be due to the compensation effect of e and I in equation (3) p. 48, where the ratio e/I appears. Hence, one could think of a simplified method consisting in a continuous investigation with $I=I(25\%)$ and a computation of $\hat{V}$ (actual) by:

$$\tilde{V}(y) \text{ actual} / \bar{V}(y) \text{ actual} = \tilde{V}(y) \text{ measured} / \bar{V}(25\%)$$

or,

$$\tilde{V}(y) \text{ actual} = \tilde{V}(y) \text{ measured} (\bar{V}(y) \text{ actual} / \bar{V}(25\%))$$

where the profile $\tilde{V}(y)$ is known and does not appreciably vary with χ .

4. EXPERIMENTAL PROGRAM

41. Acoustic properties of the flow.

Suppose, first, that we consider the duct acting as an acoustic organ pipe, i. e. that the amplitude oscillations of pressure $\tilde{P}$ and of velocity $\tilde{V}$ are negligible, compared to the steady-state pressure $\bar{P}$ and velocity $\bar{V} \neq 0$ of the flow.

Although there is an orifice at the end of the duct, this end can be considered as a "closed" end ($\tilde{V}=0$) exactly like in the case of the nozzle of a combustion chamber (Sketch 28)

Sketch 28

—— Velocity
- - - - Pressure

In this case, if $\lambda \gg b$, the conditions in plane I are quasi-steady, i. e., the Mach number M is constant. Since $V=aM$, $\tilde{V}/\bar{V}=\tilde{a}/\bar{a}$ in plane I. With

$a \sim T^{1/2} \sim P^{(\gamma-1)/2\gamma}$, $\tilde{a}/\bar{a} = [(\gamma-1)/2\gamma] \tilde{P}/\bar{P}$ and finally:

$$\tilde{V}/\bar{a} = [(\gamma-1)/2\gamma] M \tilde{P}/\bar{P}$$

where $[(\gamma-1)/2\gamma]M$ is small. Hence $\tilde{V}/\bar{a} \approx 0$ in plane I which can be considered as a "closed" end. In the case of our duct end, the condition $\lambda \gg b$ is satisfied (because of the rapid contraction of the stream lines, especially when the duct is driven at low frequencies. (λ))

The upstream extremity of the duct, on the contrary, is an "open" end since $P=\text{constant}=P_+$ (plenum chamber pressure.) Hence, at this end $\tilde{P}=0$.

The duct will be driven at resonant conditions when:

$$L = (2k+1) \lambda/4$$

, as shown on Sketch 29

λ is the wavelength = a/f . The resonant frequencies are then given by:

$$f = a/\lambda = (2k+1) a/4L = (2k+1) f_0,$$

f_0 being the fundamental frequency: $f_0 = a/4L$. f_0 depends on the gas temperature T since:

$$a \text{ (ft/s)} = 49.02 \sqrt{T(^{\circ}\text{R})}$$

for air,

and hence the harmonics f_k depend also on T . Let us call f_k the value of the frequency of the $(2k+1)^{\text{th}}$ harmonic at a certain reference temperature of the gas $T=T_{\text{ref}}$ (0°F for example). In order to drive the duct in the same conditions (i. e., for example, to keep the duct in resonant conditions) whatever the value of T , we must have the same wavelength λ , i. e., we must drive the air in the duct at a frequency f such that:

$$\lambda = a_{\text{ref}} / f_{k_{\text{ref}}} = a / f_k.$$

i. e.,

$$\lambda = 49.02 \sqrt{T_{\text{ref}}} / f_{k_{\text{ref}}} = 49.02 \sqrt{T} / f_k,$$

or,

$$f_k = f_{k_{\text{ref}}} \sqrt{(T/T_{\text{ref}})} = f_{k_{\text{ref}}} \sqrt{(T^{\circ}/T_{\text{ref}}^{\circ})} \quad (4)$$

Knowing $f_0 = a_{ref}/4L = 49.02 \sqrt{460}/4 \times 9 = 29.2 \text{ cps}$, and giving to k the integer values: 0, 1, 2, we obtain immediately the resonant frequencies

$f_{k \text{ ref}}$:

k	Order of the harmonic = $H - 2k + 1$	$f_{k \text{ ref}}$ in cps
0	0 (fundamental)	29.2
1	3 (third harmonic)	87.5
2	5	146
3	7	204
4	9	263
---	---	---

Note that in all the plots of the present report, we have used f_{ref} instead of f_{actual} .

For acoustic vibrations (i. e., small oscillations), one obtains in the duct, driven at a resonant frequency, the classical nodes-antinodes pattern; the antinodes of velocity $\tilde{V}_{max}$ corresponding to the nodes of pressure $\tilde{P}=0$, and conversely, $\tilde{V}=0$ corresponding to $\tilde{P}_{max}$.

In our case of large amplitude oscillations the same general result holds (Fig 13 and 23), but because of the non-linearity of the problem, $\tilde{V}$ is no longer equal to zero at the velocity nodes and $\tilde{P}$ is no longer equal to zero at the pressure nodes.

Sketch 30

The curves $\tilde{V}_{pk}(n)$ or $\tilde{P}_{pk}(n)$ are no longer composed of sinusoidal arcs, but have rather the shape indicated on Sketch 30. Any non symmetric wave shape cannot completely cancel at the nodal location. (This non-linear effect seems to give a better explanation of $\tilde{P}_{min} \neq 0$ than the possible presence of higher harmonics which wouldn't show a nodal point at that location).

For all the sequences one can check that the values of $f_{k ref}$, computed from f_k (actual frequency) using equation (4), agree completely with the values we have previously established. This is a proof that the actual length of the duct, $L=9$ ft, can be considered as close to the acoustic length (see, for example, figure 36 where the $\langle \tilde{P}_{rms}/\tilde{P} \rangle$ peaks correspond to correct theoretical values of f_{ref}).

42. "Local" problem and "overall" problem.

The purpose of the experimentation is to study the influence of the steady parameters ($M, \bar{P}$), characterizing a run, and of the oscillating parameters ($\tilde{P}, \tilde{V}, f$, shape of the waves, presence of shocks or of flow reversal), characterizing a sequence, on the heat transfer with and without oscillations. These heat transfer comparisons are made using the ratio Nu/Nu_0 of the Nusselt number in oscillating flow over the Nusselt number in steady flow. It is possible to study these effects from two different points of view:

421. The "local" problem:

The "local" problem corresponds to a fixed axial location in the duct (Sketch 31).

Sketch 31

At that location there exists a certain steady-state ($M, \bar{P}, \bar{V}(y)$) and a certain unsteady state characterized by pressure and velocity oscillation amplitudes ($\tilde{P}, \tilde{V}(y)$), frequency and shape of these oscillations. If the heater strip width W is small, compared to the wavelength λ (Sketch 32), we can reasonable assume that the heat transfer from the wall to the gas, at that location, only depends on the oscillating conditions at that location as if the whole duct was driven with those oscillations (no node - antinode pattern) (Sketch 32).

The "local" problem, hence, is limited to the high λ/W range, i. e., the low frequency range. Note that $W=0.5$ inch and $\lambda/2=1.19$ inches at $f=5285.2$ (upper frequency limit), and hence: $\lambda/2W=1.19/0.5=2.38$. (lower value) But in the range where we have normally operated,

$$263 < f < 1430 \text{ cps, } 24/0.5 > \lambda/2W > 4.41/0.5, \text{ i.e.}$$

$$8.82 < \lambda/2W < 48,$$

and, hence, our approximation is valid.

When increasing the frequency, the heat transfer measured at positions 1 and 2 (Sketch 32) are found to differ. Nonlinear unsteady effects due to the local thickness of the "thermal boundary layer" (if this term can be used here!) would seem to be responsible. At position 1 the boundary layer is developing as the flow moves toward an antinode while thinning takes place at position 2 (moving toward a node). Nonlinear effects thus result in a sort of hysteresis (Sketch 34), i.e., even if the same local oscillating conditions do exist, identical values of the local heat transfer would not be expected. However, if $V=0$ (i.e., no steady-state flow), this hysteresis effect would be eliminated, since it is V which introduces a dissymmetry in the problem.

Sketch 34

For the local problem, we can think of two ways of varying the local oscillating conditions and, consequently, the local heat transfer:

A. Gap variation at fixed axial location x

B. Traverse of the heater section along the duct; from the curves

$Nu/Nu_0 = f(x)$ and $\tilde{v}/\bar{v} = g(x)$, for example, we can obtain the correlations:

$$Nu/Nu_0 = f(\tilde{v}/\bar{v})$$

Solution (A) could not be used with the apparatus as it was designed, because a gap variation altered the Mach number, as well as it varied the amplitude (Fig 4 and p. 33). Hence, it would have been difficult to separate the two effects, unless we could collect enough data so as to cross-plot the curves.

Solution (B) is very attractive, since in a single traverse we can not only have a continuous variation of the wave amplitude, but also we can collect interesting data about the different node-antinode patterns and draw a complete picture of what is taking place in the duct.

However, when doing this we must assume that only the amplitude varies when we go, for example, from a node to an antinode of velocity; neglecting second order effects, such as: variation of the shape of the waves (p.82);

variation of the velocity profiles ($\tilde{V}=\tilde{V}(y)$) along the duct (p. 82) etc....

Hence, for a rapid survey of the first order effects it has been decided to use method (B), without modification of the apparatus. However, on p. 89 we will reconsider method (A).

422. The "overall" problem:

The "overall" problem is another interesting aspect of the oscillating heat transfer which we could call the engineering or applied aspect, i. e., how the overall heat transfer may be increased (or decreased). We shall see that the relative local heat transfer increase $(Nu-Nu_0)/Nu_0$ is maximum at the velocity antinodes of oscillations ($\tilde{V}_{max}$), and minimum at the nodes ($\tilde{V}_{min}$). In any application the interesting quantity is often the average heat transfer increase. For example, in a nuclear reactor, it would be interesting to know how the overall heat transfer from the reactor walls to the propellant can be improved by pulsing the gaseous propellant.

The present apparatus is not exactly appropriate for such a study because, instead of presenting a travelling heater section, the whole wall should be heated up to constant temperature or at a variable temperature ($T(wall)=\text{known function of } X$), as it exists in nuclear reactor design (Sketch 35).

Sketch 35

Sketch 36

Any attempt to treat the "overall" problem as an averaging of "local" problem results, does not provide a proper solution because, as shown in Sketch 35, the heat transfer at B is less than that obtained with our apparatus (Sketch 36) since the flow has been already heated up in the upstream portion of the duct, and the thermal boundary layer has already developed.

However, a correct averaging of the local results provides a reasonable approximation of what one can expect from a more accurate investigation from the overall heat transfer point of view.

By "correct averaging," we mean an averaging of quantities for an distance equal to $k\lambda/2$ and not for the whole traverse span, as shown on Sketch 37. However, when λ becomes small the two averages are quite similar.

Sketch 37

This restriction limits the "overall" problem to the high frequency range, since, to perform our average, we need: $\lambda/2 < l$. Since $l = 18.56$ inches, and, $(\lambda/2)$ inches = 6300 (in $\times \text{sec}^{-1}$) / f (cps), we obtain:

$$f > 6300/18.56 \approx 340 \text{ cps.}$$

However, for $170 < f < 340$ cps, if in the axial survey both a node and an antinodal point are present, the averaging is still possible with:

$$\lambda/4 < l < \lambda/2$$

The conclusion of these considerations is that, to obtain the maximum data both from the "local" and the "overall" points of view, the medium frequency range is the most interesting. It is for this reason that most of the sequences reported here correspond to frequencies in this range ($260 < f < 1400$).

43. Use of the IBM 7090 Computer

In order to keep up with the massive amount of experimental data produced, it appeared to be necessary to handle the detailed calculations on an IBM 7090. This computer can provide not only the figures, but also the plot of the different parameters versus X , and even the correlation curves between these parameters (Fig. 8 to 33).

In the rapid survey approach to define the most interesting regions of variation of the parameters, 21 discrete X positions were picked among the continuum of values stretching over 18 inches. The method for cataloging flow reversal and shock waves was to note the axial locations in the survey when these phenomena were present. The shape of the pressure and velocity waves P_h and V_h was also sketched by hand during the survey. In this way no photograph of the data needed to be taken, and the rapid survey approach was not compromised. However, the principal characteristics of the wave shapes were noted and when unusual importance of these shapes was indicated, further more accurate measurements were made.

44. Dimensionless parameters.

The steady dimensionless parameters are obviously the Mach numbers M and the Reynolds number $Re = \bar{V}D\sqrt{\rho/\mu}$ or $Re = \bar{V}D\rho/\mu$. Since the most interesting results have been obtained for only one value of the Mach number, comparisons of results for different Reynolds numbers are

equivalent to comparisons for different ρ or $\bar{P}$ (duct pressure).

The dimensionless frequency parameter can be taken as $D\sqrt{\omega\rho/2\mu}$, which appears in equations of laminar unsteady viscous flow. Since

$$D\sqrt{\omega\rho/2\mu} = \sqrt{(\nabla D\rho/\mu)(\omega D/\bar{V})} = \sqrt{Re(\omega D/\bar{V})},$$

we can take for the dimensionless frequency parameter the quantity

$\omega D/\bar{V}$ even in turbulent flow ($\sqrt{Re}$ is a characteristic expression

encountered in laminar flow only). At constant Mach number, $\omega D/\bar{V}$

can again be well described by the frequency f or better by the

reference frequency f_{ref} defined on p. 58.

Oscillating parameters have been related to the steady-state values in a ratio form ($\tilde{V}/\bar{V}$, $\tilde{P}/\bar{P}$ etc...). Hence, although in a more elaborate study appropriate dimensionless parameters should be used, results have been presented in terms of the following quantities as characteristic parameters: M , $\bar{P}$, f , $\tilde{V}/\bar{V}$, $\tilde{P}/\bar{P}$ etc...

45. Early experiments with pressure only:

In the early experiments (Run 1, sequences 0 to 66; run 2, sequences 0 to 3; run 3, sequences 0 to 3) corresponding to the conditions: $M=0.07$ (medium orifice), $\bar{P}=54.7$ psia, $\dot{m}=0.294$ lbs/sec, $Re=2.9 \times 10^5$, $200 < f < 3400$ cps, constant intra-orifice gap=0.019 inch, a correct method of oscillating velocity absolute measurement was not yet available. An attempt has been made to correlate the heat transfer increase to the oscillating pressure in the following way:

Saunders (Ref.7,8) has shown that, in a traverse along the duct, the maximum of heat transfer increase corresponded to a node of oscillating pressure and the minimum to an antinode. Hence, plotting Nu/Nu_0 versus $\tilde{P}_{rms}/\bar{P}$, for example, one can expect some decreasing curve, with each point of the curve corresponding to an X value. The correlation

was not very good. A great deal of dispersion was present. However, plotting the curves $Nu/Nu_0 = f(\tilde{P}_{rms}/\bar{P})$ for different resonant (or non resonant) frequencies (87.4 ~~to~~ 1687 cps) as shown in Figure 35, we can draw some interesting conclusions:

When going from a pressure node ($\tilde{P}_{rms}/\bar{P}$) min. to an antinode ($\tilde{P}_{rms}/\bar{P}$) max., the heat transfer decreases monotonously as expected.

The largest local heat transfer increases occur (at the nodes of pressures i. e., antinodes of velocity) for the low resonant frequencies, especially when flow reversal is present.

Both $\tilde{P}_{rms}/\bar{P}$ and Nu/Nu_0 amplitudes of variations (during a traverse in the resonant cases) are reduced as the frequency increases and the curves for the high resonant frequencies (or for the low or high non-resonant frequencies), are located in the left lower region of the plot. These curves indicate small oscillations and small heat transfer increases.

But this can be seen much better when averaging the results in $\bar{x}$. Taking $\langle \tilde{P}_{rms}/\bar{P} \rangle$ as a measurement of the averaged pressure oscillation intensity (this definition is perhaps better than to define such an intensity by $\tilde{P}_{rms}/\bar{P} = (\tilde{P}_{rms}/\bar{P})_{max.}$ because it also takes into account what actually occurs at the nodes of pressure) and taking also $\langle Nu/Nu_0 \rangle$ as the overall heat transfer ratio, we obtain in Figure 36 the variation of these quantities versus frequency.

At least for the low frequencies, typical resonant and non-resonant conditions are presented. $\langle \tilde{P}_{rms}/\bar{P} \rangle$ shows well defined peaks for resonant conditions up to $f \approx 1000$ cps. Averaged reduced pressure oscillation intensities up to $\langle \tilde{P}_{rms}/\bar{P} \rangle = 0.05$ have been achieved.

For higher frequencies, there is not a great difference between the resonant and the non-resonant cases. The conclusion is that there is not a high proportion of standing waves at these frequencies, but rather a condition exists where travelling waves are predominant.

This becomes quite obvious when plotting the pressure standing wave ratio, $(\tilde{P}_{rms})_{max.} / (\tilde{P}_{rms})_{min.}$ versus frequency. This ratio goes to 1 as f goes to infinity. An interesting result is obtained for low frequencies: the pressure standing wave ratio is greater for non-resonant conditions than for resonant conditions, although the intensity of oscillation is smaller (the same result will be found in runs 4, 5 and 6, p. 71).

Considering now the overall heat transfer ratio, $\langle Nu/Nu_0 \rangle$, we see that it is a maximum for resonant conditions and frequencies close to $f \approx 650$ cps, and goes to 1 as f goes to infinity. Hence, the high frequency oscillations do not appear to be interesting from the standpoint of increasing heat transfer. This does not mean that the high frequency oscillations are not effective. The ratio $\langle Nu/Nu_0 \rangle$ decreases because, for a given intra-orifice gap, the ratio $\langle \tilde{P}_{rms}/\bar{P} \rangle$ decreases as f goes to infinity. But, if we characterize the "effectiveness" of a given oscillation by the ratio $\langle Nu/Nu_0 \rangle / \langle \tilde{P}_{rms}/\bar{P} \rangle$, forgetting about the increase itself, we obtain the results shown in Figure 37. The resonant frequencies are less effective than the non-resonant frequencies; at low frequencies, both converge toward an intermediate value at the high frequencies

$$\langle Nu / Nu_0 \rangle / \langle \tilde{P}_{rms} / \bar{P} \rangle \approx 60 \text{ for } f = \infty.$$

46. Pressure waves and velocity waves in the duct.

As soon as a correct method was found to measure the oscillating velocity, a rapid survey was decided, in the medium frequency range (i. e., $300 < f < 1500$ cps).

Seven frequencies, i. e., seven sequences, were selected for for each run, of which two were non-resonant. The steady-state conditions for the runs were:

run 4 ($M=0.033$, $\bar{P}=54.7$ psia); run 5 ($M=0.034$, $\bar{P}=39.7$ psia);
run 6 ($M=0.030$, $\bar{P}=74.4$ psia); run 7 ($M=0.102$, $\bar{P}=39.7$ psia);
run 9 ($M=0.101$, $\bar{P}=54.7$ psia; four sequences only are available for this run).

Prior to the study of the heat transfer one must determine the characteristics of the oscillating flow, i. e., node-antinode patterns of pressure and velocity oscillations along the duct and their correlation. All this must be accomplished using certain steady-state conditions (M , $\bar{P}$) and oscillating conditions (frequency, siren gap which governs the amplitude of oscillations).

461. Node-antinode patterns.

Results are given, for each sequence, as a plot of $\tilde{P}_{rms}/\bar{P}$, $\tilde{P}_{pk}/\bar{P}$, $\tilde{V}_{rms}/\bar{V}$, $\tilde{V}_{pk}/\bar{V}$, versus the axial location $X=x/l\%$ along the duct, l being the total traverse length. The oscillating velocity is taken in the core flow, i. e., at $Y=y/R=100\%$. Some general results can be given:

The node-antinode pattern is similar to the pattern existing in the case of oscillations taking place in an acoustic organ-pipe, although by comparison, these oscillations are large. However at nodal points, the intensity of oscillations is not zero (Sketch 30 p. 60), and the shape of the curves $\tilde{P}/\bar{P}$ or $\tilde{V}/\bar{V}=f(x)$ is far from being regular, i. e., sinusoidal, as in sound theory.

Defining a velocity standing-wave ratio as $(\tilde{V}_{rms})_{max.}/(\tilde{V}_{rms})_{min.}$ (in the same way as we have defined the pressure standing-wave ratio

on p.63), Figures 48, 49 and 50, show that, for a given run and a given intra-orifice gap (i. e., a given excitation), both standing wave ratios decrease with frequency and go to 1 as f goes to infinity. This occurs for resonant or non-resonant conditions (compare with the results of p.69 and Figure 36). Hence, as the frequency increases, there is a greater and greater proportion of travelling waves instead of standing-waves in the duct.

Although the intensity of oscillations is smaller in non-resonant conditions, the standing-wave ratio for pressure is greater in the same frequency range. For example, in run 1 (Fig.36), we see a maximum of pressure standing wave ratio equal to 4 at a non-resonant frequency around $f=470$ cps; but for the resonant conditions on both sides, the pressure standing-wave ratios are only equal to 3.15 and 2.9. In run 6 (Fig. 50), where $f=420$ cps, pressure standing wave ratio in non-resonant conditions is equal to 3.9 instead of 2.5 and 3.3 in resonant case.

The velocity standing-wave ratio, however, is smaller in non-resonant conditions than in resonant conditions (Fig. 50). For the lowest frequency reached so far, a velocity standing-wave ratio of 11 has been encountered (Fig.49).

462. "Local" problem: correlation between oscillating pressure and oscillating velocity (Non-linear velocity-pressure correlation).

Plotting $\tilde{V}_{rms}/\bar{V}$ versus $\tilde{P}_{rms}/\bar{P}$ and $\tilde{V}_{pk}/\bar{V}$ versus $\tilde{P}_{pk}/\bar{P}$ (Fig.38 and 39) we can draw the following conclusions:

The reduced amplitude (or intensity) of pressure oscillations is always much smaller than the reduced amplitude (or intensity) of velocity oscillations.

$\tilde{P}_{rms}/\bar{P} \leq 0.02$, while $\tilde{V}_{rms}/\bar{V}$, although limited by the hot-wire rectifier action to measured values of < 1 (see p. 75), can actually reach even higher values.

$\tilde{P}_{pk}/\bar{P} \leq 0.07$, but, $\tilde{V}_{pk}/\bar{V}$ reaches 2.5 (again limited by the instrumentation). When one takes into account the reversed flow, this last value can actually reach 5 and more.

With the exception of the low frequencies, $\tilde{V}/\bar{V}$ is monotonic in $\tilde{P}/\bar{P}$, the concavity facing the $\tilde{V}/\bar{V} < 0$ direction (Fig. 38 and 39).

Sometimes, some "hysteresis" is present, i. e., there are in fact two curves, depending on whether we go from a node to an antinode or conversely.

The two former results are similar to those predicted by the sound theory. In that case, the maximum pressure and velocity variations are related by:

$$(\delta P)/P = \pm \gamma (\delta V)/a = \pm \gamma (\delta V)/V (V/a) = \pm \gamma M (\delta V)/V$$

which, with our notations, gives:

$$(\tilde{P}_{pk}/\bar{P})_{max} = \pm \gamma M (\tilde{V}_{pk}/\bar{V})_{max}.$$

With $(\tilde{P}_{pk}/\bar{P})_{max} = 0.07$, $\gamma = 1.4$, $M = 0.03$, we obtain:

$$(\tilde{V}_{pk}/\bar{V})_{max} \simeq 1.7,$$

and, hence, we are not far from flow reversal. The γM being small for the low Mach numbers we are dealing with, large velocity oscillations can be driven with relatively low pressure oscillations.

Furthermore, in a resonant case of the acoustic theory, the pressure and velocity amplitude envelopes C_1 and C_2 are sinusoidal (Sketch 37)

Sketch 37

Hence, if we plot $\tilde{V}/\bar{V}$ versus $\tilde{P}/\bar{P}$ we obtain a Lissajous pattern which is an arc $\widehat{AB}$ of an ellipse ($\lambda/4$ phase-lag). (Sketch 38)

Sketch 38

We find again that the concavity of $\widehat{AB}$ faces $\tilde{V}/\bar{V} < 0$. For large oscillations, however, we have seen that, because of the non-linearity, $(\tilde{V})_{\min}$ is $\neq 0$ and $(\tilde{P})_{\min}$ is $\neq 0$ (p. 60), and hence we do not have an arc $\widehat{AB}$ but, rather an arc $\widehat{A'B'}$.

47. Correlation between local heat transfer increase and local amplitude of velocity oscillations.

On page 67 we have seen that the maximum increases of local heat transfer occur at the nodes of oscillating pressure, which correspond to the antinodes of oscillating velocity (p. 70). Hence, the driving parameter appears to be not the amplitude or intensity of pressure oscillation, for which no satisfactory correlation was found (p. 68), but rather the amplitude or intensity of velocity oscillations.

4FI. Heat transfer increase - oscillating velocity correlation:

From the data, let us take a typical example: run 4, sequences 1 to 7 (Fig. 40). Using the IBM 7090 Computer technique described on page 66, we directly obtain the curves $(Nu - Nu_0)/Nu_0$ and $\tilde{V}_{pk}/\bar{V}$ versus the axial location X along the duct (Fig. 12 and 24), and also the correlation curve, Nu/Nu_0 versus $\tilde{V}_{pk}/\bar{V}$ (Fig. 28).

As a first approximation, one can say that the relationship between Nu/Nu_0 and $\tilde{V}_{pk}/\bar{V}$, when moving between the node and antinode positions, is linear. The dispersion is probably due to the fact that the reduced amplitude $\tilde{V}_{pk}/\bar{V}$ is not the only parameter acting on the heat transfer increase (see p. 81).

However, let us consider this first order correlation in more detail:

If we plot the segment NA (Sketch 39) for different sequences of a same run (Fig. 40), we see that there is definitely an effect of the frequency and also of the conditions under which the duct is driven (resonant or non-resonant). However, the slope $d(Nu/Nu_0)/d(\tilde{V}_{pk}/\bar{V})$ is always positive which means that the local heat transfer increases with the local amplitude of velocity oscillations. This is not true for V_{rms} , Fig. 44.

A particularly interesting case is given by run 4, sequence 1 or 3. Flow reversal occurs when $\tilde{V}_{pk} \approx 2\bar{V}$ (Sketch 40).

Sketch 40

Sketch 41

The fact that the exact relation $(\tilde{v}_{pk})_{\text{limit}} = 2\bar{v}$ does not hold is because velocity oscillations are not of symmetrical shape with respect to $\bar{v}$.

Since the hot-wire anemometer acts like a rectifier (the hot-wire cannot tell velocity direction), the amplitude $\tilde{v}_{pk}$ of the response that one measures electronically remains nearly constant, i. e., on the plot N_u/N_{u0} versus $\tilde{v}_{pk}/\bar{v}$, we stay on a vertical line $\tilde{v}_{pk}/\bar{v} \approx \text{constant} \approx 2$ (Sketch 41 and 42).

Sketch 42

There is an important increase of Nu/Nu_0 from that limiting point. If we assume the same linear relationship between Nu/Nu_0 and $\tilde{V}_{pk}/\bar{V}$ applies, then we cannot completely account for this increase; this is true even when we use oscilloscope photographs (Fig. 47) to establish the true peak-to-peak velocity that exists in the core flow at the velocity antinode location (). Using the linear relationship and the true velocity value $\tilde{V}_{pk}$, predicts point K on Sketch 42 instead of P.

Hence, when flow reversal occurs, we are obliged to assume another law relating heat transfer increase and amplitude of oscillating velocity in the core flow, which grows much faster than the linear law previously found (curve QA instead of QJ). Runs 5 and 6 (same Mach number $M=0.034$, but different duct pressure $P=39.7$ psia and $P=74.7$ psia respectively instead of $P=54.7$ psia in run 4) show the same general results (Fig 46 and 47). Before concluding that the flow reversal is responsible for the highest local heat transfer increases, reaching values of 200% as in run 6, sequence 1 (Fig. 47), one must check that the results which have been obtained do not depend on the fact that the oscillating velocity has been measured in the core flow. On page 85 we shall see that the degree of flow reversal changes for various radial locations in the duct, as well as for different axial locations. These factors must be taken into account in determining the contribution of flow reversal to the heat transfer increase. Further experiments are necessary to better establish this particular point, and the gap-variation method (p. 89) appears to be more appropriate for this study. Flow reversal effects would tend to agree with the results of Ref. 12.

472. Influence of frequency on local heat transfer increase associated with the local amplitude of velocity oscillations:

Let us take the same examples as above (runs 4, 5 and 6, sequences 1 to 7). We have plotted, versus frequency, in Figures 45, 46 and 47 :

A. Two parameters characterizing the position of the line NA (Sketch 39) :

— The ordinate $(Nu/Nu_0)_0$ of NA intersect with $\tilde{V}_{pk}/\bar{V}=0$ (this parameter has no physical meaning -- it is just a way of defining the position of line NA).

— The slope $d(Nu/Nu_0)/d(\tilde{V}/\bar{V})$.

B. Two parameters characterizing the range of heat transfer increase variation:

(Nu/Nu_0) min. (ordinate of N)

(Nu/Nu_0) max. (ordinate of A)

Sketch 43

Let us consider first the resonant conditions:

In the three runs of interest, $(Nu/Nu_0)_0$ reaches a maximum at a frequency around 900 to 1000 cps and the slope continuously decreases to zero.

(Nu/Nu_0) max. is always maximum at low frequencies probably because of the flow reversal. If we plot not only (Nu/Nu_0) max. corresponding to the ordinate of point A (Sketch 42), but $(Nu/Nu_0)_J$ or K corresponding to the increase which should take place assuming the same law of variation of Nu/Nu_0 versus $\tilde{V}_{pk}/\bar{V}$ as when no flow reversal occurs (i.e., separating the

effects of flow reversal from the effects of amplitude increase), we see that $(Nu/Nu_0)_j$ does increase little at low frequencies and sometimes even decreases. Finally (Nu/Nu_0) goes to 1 at high frequencies.

$(Nu/Nu_0)_{min.}$ reaches a maximum at about $f \approx 900-1000$ cps and also goes to 1 as f increases.

Hence, the range of variation of Nu/Nu_0 along the duct (shown on Figures 45, 46 and 47) narrows as the frequency increases, i.e., what we have called the heat transfer standing wave ratio $(Nu)_{max.}/(Nu)_{min.}$ goes to 1 as f goes to infinity. Thus, we obtain more and more travelling waves instead of standing waves.

Non-resonant conditions reduce the values of all the parameters we have considered with respect to their value under resonant conditions in the same frequency zone (Figures 45, 46 and 47).

Conclusion: If we are looking for a high local wall to gas heat transfer, no matter what the duct pressure $\bar{P}$, we must drive the duct in a low frequency range at resonant conditions, where we can expect flow reversal. Further experiments must be done to survey the frequency range from 0 to 250 cps which seems of high interest.

The high frequency range ($f > 1500$ cps) with the same siren excitation, appears uninteresting from this point of view, although higher amplitudes in this frequency range may prove promising since the efficiency is quite high. We arrive at the same conclusions when considering the overall problem (p. 80).

473. Influence of duct pressure $\bar{P}$:

As we can see, going from run 5 to run 4 and run 6, i.e., increasing $\bar{P}$ from 39.7 to 54.7 and 74.7 psia (Figures 41, 40 and 42), all curves are shifted towards $Nu/Nu_0 > 0$, i.e., an increase of steady duct pressure not only increases the steady heat transfer Nu_0 but also increases the "increase"

of heat transfer due to oscillations. The maximum local heat transfer increase obtained up to now is $(Nu/Nu_0)_{\max.} = 3$ (200% increase) for $P = 74.7$ psia.

Hence, further experiments must be done increasing the duct pressure over 80 psia since high $\bar{P}$ seems very attractive from the standpoint of increasing the local heat transfer contribution due to oscillations.

474. Effect of the Mach number:

The early experiments (run 1, 2, and 3) were conducted with measurements of heat transfer and pressure only, with the medium downstream sonic orifice ($M = 0.07$) and a duct pressure of $\bar{P} = 54.7$ psia. Local heat transfer ratios up to $Nu/Nu_0 = 2.9$ were encountered (Figure 35). Comparing this with the corresponding result of run 4 ($\bar{P} = 54.7$ psia, $M = 0.033$ corresponding to the small orifice), where the maximum heat transfer ratio was $Nu/Nu_0 = 2.1$, one could have concluded that an increase of $(Nu/Nu_0)_{\max.}$ with M at constant $\bar{P}$ was present. However, since in the two cases, the frequency was not the same, and since in run 4, the low frequency range had not been investigated, such a conclusion was premature. To clarify this point, it was decided to go to the higher Mach number case ($M = 0.1$) using the large sonic orifice.

At this higher Mach number, the surprising results was that the local heat transfer increases were very low ($Nu/Nu_0 = 1.2$) and that there was sometimes even a decrease of heat transfer with respect to the steady-state (Figure 43).

The difficulty of obtaining high heat transfer increases may be due to the impossibility of reaching high $\tilde{V}_p/\bar{V}$ since, for the same duct excitation, $\bar{V}$ increases with the Mach number.

The maximum $\tilde{V}_{pk}/\bar{V}$ obtained was 1.1, much less than in the case of lower Mach number, at the same frequency.

However, the decrease of heat transfer with oscillations is more

surprising. Can that fact be attributed to the presence of shock-type waves which were encountered for the first time? Further experiments must be undertaken to define more closely this Mach number effect which is in agreement with Croke's experiments (Ref. 5).

48. The overall problem

The same study as for runs 1, 2 and 3 (Fig. 36) has been undertaken for runs 4, 5 and 6, adding this time the results obtained for the oscillating velocity, but considering only seven frequencies in the medium frequency range $300 < f < 1500$ cps. Figures 48, 49 and 50 show how the different "average" parameters vary versus frequency in resonant conditions. Two non-resonant frequencies have also been considered in Figure 50.

The overall heat transfer ratio $< Nu/Nu_0 >$ seems to be maximum for some medium frequency, around 600 to 900 cps. However, before concluding that this is the best frequency range of operation if one wants to increase the average heat transfer, further experiments must be undertaken in the low frequency range ($f < 300$ cps) where the averaging is quite impossible with the present apparatus (see p. 65). This result is in good agreement with that of runs 1, 2 and 3 (Fig. 36) obtained at a higher Mach number (0.07 instead of 0.03). When the duct pressure $\bar{P}$ increases, the maximum overall heat transfer ratio increases also. For $\bar{P} = 74.7$ psia, an average heat transfer ratio up to $< Nu/Nu_0 >_{\max.} = 1.9$ has been obtained, i.e., 90% average increase. At constant excitation (constant gap) both average pressure and velocity oscillations indicate a maximum in the same range (600 cps to 900 cps). Past this maximum, the overall heat transfer ratio Nu/Nu_0 decreases as the frequency increases, since, for the same siren excitation (constant gap), $< \tilde{V}_{pk}/\bar{V} >$ itself decreases. However, the overall effectiveness $< Nu/Nu_0 > / < \tilde{V}_{pk}/\bar{V} >$ of the velocity oscillations increases as it

can be seen on Figure 48, comparing the slopes of the curves $\langle \text{Nu}/\text{Nu}_0 \rangle$ and $\langle \tilde{V}_{pk}/\bar{V} \rangle$ versus frequency.

The three standing wave ratios based on pressure, velocity and heat transfer go to 1 as f goes to infinity, showing that as the frequency increases, more and more travelling waves are present.

In non-resonant conditions (Fig. 50) both overage pressure and velocity amplitudes of oscillations have smaller values compared to the resonant case, and so does the heat transfer. As previously established the velocity standing wave ratio decreases at low frequencies. Although the overall heat transfer decreases (because of the decrease of the average intensity of oscillations) the oscillations are more effective, i.e., the ratio

$\langle \text{Nu}/\text{Nu}_0 \rangle / \langle \tilde{V}_p/\bar{V} \rangle$ is greater.

Hence, in both cases of non resonant conditions (compared to resonant conditions) and of resonant conditions at high frequency (compared to resonant conditions at low frequency) there is an increase of the overall effectiveness of the oscillations. A possible explanation of this increase will be given on page 87.

49. Second order effects

Because of the dispersion sometimes encountered when trying to correlate Nu/Nu_0 and $\tilde{V}_{pk}/\bar{V}$, one might think that the amplitude of velocity oscillations is not the only parameter to act on the heat transfer increase. In our method of obtaining the variations of $\tilde{V}/\bar{V}$ by a traverse in x , we are forced to introduce other factors than pure oscillations amplitude, such as: shape of the waves, velocity profiles (i.e., variation of velocity parameters with radial location). All these factors depend on the relative position of the heater section with respect to the nodes and

antinodes. Prior to studying the possible influence of these factors on heat transfer, let us define them somewhat more accurately.

491. Shape of the waves as a function of the x location.

4911. Shape of pressure oscillations.

Characterizing the shape of the wave by the ratio $\tilde{P}_{pk}/\tilde{P}_{rms}$, let us consider the plot of this quantity versus $\tilde{P}_{rms}/\bar{P}$ which is a measurement of the position of the heater section with respect to the nodes and antinodes. This has been done in run 4 and run 6 (seven frequencies for each) (Fig. 51 and 52). Although the correlation is not always as good as in sequence 10 for example, (Fig. 32), we can draw the following general conclusions:

$\tilde{P}_{pk}/\tilde{P}_{rms}$ decreases with $\tilde{P}_{rms}/\bar{P}$ (if flow reversal is not present).

This indicates that the pressure waves are steeper at the pressure nodes. We would expect this to occur since fundamental sinusoidal components of the pressure waves are cancelled at the node location while the steep portions of the wave (or higher harmonics) fail to cancel at that point.

4912. Shape of velocity oscillations in the core flow

(Fig. 53):

Defining again the shape by the ratio $\tilde{V}_{pk}/\tilde{V}_{rms}$ and plotting this quantity versus $\tilde{V}_{rms}/\bar{V}$, we obtain, with sometimes much more dispersion than in run 4, sequence 10 (Fig. 33), the qualitative result: velocity waves are usually steeper at the velocity nodes. This is due to the same explanation as offered in the pressure wave shape case.

494. Velocity profiles at different axial locations:

4921. Average velocity profiles $\bar{V}(y)$

Average velocity profiles in oscillating flow at different axial locations are given in Figure 56, and 57. The differences which exist when X varies in the oscillating case are usually greater than in the steady-state case (Fig. 55). This can be due to the non-linear response of the hot-wire

anemometer as analysed (p. 52). In Figure 54, classical turbulent and laminar profiles have been shown indicating that both in steady-state and in oscillating flow the profiles are closer to the turbulent profile than to the laminar one.

4922. Oscillating velocity profiles $\tilde{V}_{pk}(y)$ and $\tilde{V}_{rms}(y)$:

Figures 58, 59 and 60 show the importance of using a correct method of measurement of the oscillating velocity (constant resistance ratio measurement; see p. 55). Figures 68 - 70 show the velocity profile at different axial locations, obtained by cross-plotting the oscillating velocity measurements at different radial positions during an axial traverse (Fig. 62 and 63.).

Both $\tilde{V}_{pk}(y)$ and $\tilde{V}_{rms}(y)$ profile are not universal, i.e., their shapes vary with the axial position. In other words, for example, the curve $\tilde{V}_{pk}(y)/\tilde{V}_{pk}(y = 100\%) = \varphi(y)$ depends on the x position as shown on Sketch 44:

Sketch 44

$\varphi(y)$ functions show maximums occurring between the wall and the core flow. These maximums are greater and occur nearer the wall at the velocity nodes. This phenomenon must be compared with the "Annular Effect" (Ref. 4), although that effect takes place in viscous, laminar incompressible flow, for small oscillations and influences the rms value of the oscillating

velocity as well as its amplitude.

Figure 70 cross plotted from Figure 63 show the intensity profiles $\tilde{V}_{rms}(y)$. These profiles also vary with the axial location (Sketch 45)

Sketch 45

The "Annular Effect" seems to be absent, although near the velocity antinode $\psi(y)$ shows light maximums: one near the wall and the other in the core flow.

4923. Profile of reduced oscillating velocity $\tilde{V}_{pk}(y)/\bar{V}(y)$ and $\tilde{V}_{rms}(y)/\bar{V}(y)$:

On Figure 68 the average velocity profile $\bar{V}(y)$ has been shown. One can notice that $\bar{V}(y)$ decreases much more rapidly than $\tilde{V}_{pk}(y)$ when approaching the wall. Taking the ratio of the two quantities, we obtain at different axial locations, the reduced amplitude profiles $\tilde{V}_{pk}(y)/\bar{V}(y)$ (Fig. 71).

The curves show a marked increase when approaching the wall. However, a universal profiles does not exist, i.e., the profile $\eta(y) = \tilde{V}_{pk}(y)/\bar{V}(y) / (\tilde{V}_{pk}/\bar{V})_{y=100\%}$ depends on the axial location. Since the curves on Figure 71 are almost parallel, $\eta(y)$ increases more rapidly at the velocity node when $y \rightarrow 0$.

$\eta(Y)$

Sketch 46

Now it becomes obvious, since the approximate condition for flow reversal is $\tilde{V}_{pk} \approx 2\bar{V}$ or $\tilde{V}_{pk}/\bar{V} = 2$, that the flow reversal occurs more easily near the wall than in the core flow. For example Figure 71 shows that at the velocity antinode ($X=30\%$) we are very near the flow reversal at the wall since $\tilde{V}_{pk}/\bar{V}(Y=5\%; X=30\%) = 2$, but very far from the flow reversal in the core flow since $\tilde{V}_{pk}/\bar{V}(Y=100\%; X=30\%) = 1.4$. At the velocity node ($X=100\%$) we are nearer the flow reversal condition near the wall $\tilde{V}_{pk}/\bar{V}(Y=5\%; X=100\%) = 1.45$ than in the core flow, $\tilde{V}_{pk}/\bar{V}(Y=100\%; X=100\%) = 0.64$. This is well illustrated on Figure 74 where flow reversal occurs both in the core flow and near the wall but is more important near the wall.

The oscillating velocity reduced intensity profiles, $\tilde{V}_{rms}(y)/\bar{V}(y)$ present the same characteristics, although now the profiles are very close to a universal profile: variations of $\zeta(Y) = (\tilde{V}_{rms}/\bar{V})Y / (\tilde{V}_{rms}/\bar{V})Y=100\%$ are shown on Sketch 47:

$\zeta(Y)$

Sketch 47

4924. Profiles of the shape parameter $\tilde{V}_{pk}(y)/\tilde{V}_{rms}(y)$

Figure 73 shows these profiles at different axial locations. They have been obtained by cross plotting Figure 66 data. The shape parameter usually increases when approaching the wall, i.e., the velocity oscillations are generally steeper near the wall. The profiles are not universal and both in the core flow and near the wall the steepest oscillations are obtained at the velocity nodes. (Sketch 48)

493. Effect of the shape and the radial profiles of the velocity oscillations on the heat transfer. (Sketch 49)

Considering again the plots of the heat transfer ratio Nu/Nu_0 versus the reduced amplitude of velocity oscillations $\tilde{V}_{pk}/\bar{V}$, we see that although the heat transfer increase is less important at the velocity node (because of the low amplitude of oscillations), the effectiveness $(Nu/Nu_0)/\tilde{V}_{pk}/\bar{V}$ of nodal oscillations is greater (i.e., $\alpha > \beta$). This may be due to a condition where at the velocity nodes the oscillations "penetrate" nearer the wall than at the antinodes (Sketches 44 and 46). In particular $\tilde{V}_{pk}(y)$

Sketch 49

shows a marked maximum near the wall (Sketch 44).

But another factor seems to have also a great importance: the velocity oscillations are steeper both in the core flow and (mainly) near the wall, at the velocity nodes. That the steepest oscillations are the most efficient oscillations can be seen not only in this local problem but also in the overall problem. We have noticed that, under resonant conditions, the overall effectiveness of the velocity oscillations increases with the frequency, and that, at the same time, the velocity waves become steeper (i.e., $\langle \tilde{V}_{pk}/\bar{V}_{rms} \rangle$ increases) (p. 80).

We have also seen previously that, under non-resonant conditions, the oscillations were in average more effective, but also steeper than in the corresponding resonant cases (p. 81).

Hence, the shape of the waves seems to play an important role in the heat transfer increase but in the present apparatus its effect superimposes on the main effect of the amplitude of oscillations. The two parameters are very difficult to separate when using an axial survey to vary the amplitude of oscillations.

For that reason, it was decided to obtain the variation of amplitude at a fixed axial location, by intra-orifice gap variation (p. 25).

Prior to that, however, the experimental apparatus had to be transformed to prevent the variation of the Mach number during a gap variation (p. 33).

4.1. Gap variation method.

This method has been described on p. 33. Preliminary tests have shown that the Mach number could be maintained constant through the use of a movable "curtain" which partially closes the downstream sonic orifice of the duct (p. 34 and Figure 3). For example, in run 13 (Fig. 75), the Mach number has been kept constant and equal to 0.038 for four different values of the gap. For each of these gap values, a survey of both heat transfer and oscillating velocity in the core flow has been made along the duct so that at a fixed axial location four values of the oscillating velocity amplitude and of the corresponding heat transfer are available. This is not sufficient to draw a conclusion concerning the correlation between these two parameters; however, it is interesting to notice that, at least at the velocity antinodes, the shape parameter $\tilde{V}_{pk}/\tilde{V}_{rms}$ does not appreciably vary when the gap is altered, i. e., when the local amplitude of oscillations varies. This may allow the separation of the two effects, amplitude of oscillations and wave shape, which were previously linked, in the traverse method (p. 87).

5. Conclusion

The local heat transfer increases produced by oscillations imposed on a steady flow through a duct have been found to be maximum at the velocity antinodes and minimum at the velocity nodes. A good correlation has even been found between the heat transfer increase and the amplitude $\tilde{V}_{pk}$ (rather than the intensity $\tilde{V}_{rms}$) of velocity oscillations for surveys between velocity nodes and antinodes. This correlation is affected by the frequency of oscillations (which determines, for this duct, whether standing or travelling waves exist), the duct steady-state pressure (which has been shown to increase the heat transfer increase), and the duct Mach number (the effect of which has to be more closely analysed).

The flow reversal occurring in the case of very large velocity oscillations appears to be responsible for the highest heat transfer increases (up to 200%). Since flow reversal is more easily attained at resonant low frequencies (longer periods and higher amplitudes) and low Mach number flow (where the oscillating velocity can more easily overcome the steady-state velocity), the highest heat transfer increases have been observed under these conditions.

The shape of the wave also seems to be important, the steepest waves being the most effective for increasing the heat transfer. These steeper waves are favored by higher frequencies, non-resonant conditions and proximity of a velocity node.

Finally, the overall heat transfer increase in the duct, obtained by the axial averaging of the local data, appears to be maximum for medium frequencies around 1000 cps when the siren excitation is kept constant as the frequency varies. Overall heat transfer increases up to 90% have been obtained.

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Figure 1

*The revolving orifices
(Low, medium and high frequency)*

Intra-orifice gap drive and exit orifice area adjustment.

Variation of the Mach number M in the duct
as the gap G is altered

Figure 4

Figure 5

Traversing heater section internal detail

a - Traversing heater section & related auxiliary equipment

b - Close up of rotating disc & damping system

c - Close up of traversing heater section

DATA REDUCTION FOR JP-17 HEAT TRANSFER AND PRESSURE, BY L. HOFFMAN, 3/15/63.
 RUN NUMBER= 4. SEQUENCE NUMBER= 1.
 DOWNSTREAM PRESSURE= 54.7 PSI. MACH NUMBER= 0.034 FLOW RATE= 0.1375E-00
 REYNOLDS NUMBER= 0.1265E 06 ACTUAL FREQUENCY= 270.0 CPS. FREQUENCY AT REF. TEMP.= 264.5 CPS.
 AVERAGE HEAT TRANSFER IN STEADY STATE= 0.4574E-04

AVERAGE NUSSELT NUMBER IN STEADY STATE= 0.2136E 03
 AVERAGE $(H-H(SS))/H(SS)$ = 0.2014E-00
 AVERAGE NUSSELT NO. / STEADY STATE NUSSELT NO.= 0.1223E 01
 AVERAGE R.M.S. PRESSURE = 5.531 PSI. AVERAGE P-P PRESSURE = 20.673 PSI.
 AVER. R.M.S./DOWNSTREAM PRESS. = 0.1011 AVER. P-P/DOWNSTREAM PRESS. = 0.3779
 AVERAGE P-P /R.M.S. PRESSURES = 3.8352

	X	HT. TRANS.	NUSS. NO.	(H-HO)/HO	N/NO	10X P(RMS)	10X P(P-P)	10X P(RMS)/PBARD	10X P(P-P)/PBARD	P(P-P)/P(RMS)
B	0.	0.7409E-04	0.3520E 03	0.6240E 00	0.1652E 01	0.4710E 01	0.2122E 02	0.8610E-01	0.3879E-00	0.4505E 01
C	5.	0.7426E-04	0.3527E 03	0.6430E 00	0.1671E 01	0.3000E 01	0.1490E 02	0.5484E-01	0.2724E-00	0.4967E 01
D	10.	0.7514E-04	0.3569E 03	0.6602E 00	0.1689E 01	0.2600E 01	0.1521E 02	0.4753E-01	0.2780E-00	0.5849E 01
E	15.	0.7415E-04	0.3522E 03	0.6371E 00	0.1665E 01	0.3732E 01	0.1499E 02	0.6823E-01	0.2740E-00	0.4016E 01
F	20.	0.7068E-04	0.3358E 03	0.5495E 00	0.1576E 01	0.4746E 01	0.1630E 02	0.8676E-01	0.2980E-00	0.3434E 01
G	25.	0.6586E-04	0.3129E 03	0.4408E-00	0.1466E 01	0.5626E 01	0.1933E 02	0.1028E-00	0.3534E-00	0.3437E 01
H	30.	0.6086E-04	0.2691E 03	0.3323E-00	0.1355E 01	0.5965E 01	0.2177E 02	0.1091E-00	0.3980E-00	0.3650E 01
I	35.	0.5624E-04	0.2672E 03	0.2286E-00	0.1250E 01	0.6186E 01	0.2258E 02	0.1131E-00	0.4128E-00	0.3650E 01
J	40.	0.5275E-04	0.2506E 03	0.1549E-00	0.1175E 01	0.6207E 01	0.2272E 02	0.1135E-00	0.4154E-00	0.3661E 01
K	45.	0.5013E-04	0.2381E 03	0.9812E-01	0.1117E 01	0.6155E 01	0.2260E 02	0.1125E-00	0.4132E-00	0.3672E 01
L	50.	0.4798E-04	0.2279E 03	0.4817E-01	0.1066E 01	0.5098E 01	0.2248E 02	0.1115E-00	0.4111E-00	0.3688E 01
M	55.	0.4653E-04	0.2210E 03	0.2006E-01	0.1038E 01	0.5975E 01	0.2214E 02	0.1092E-00	0.4047E-00	0.3705E 01
N	60.	0.4521E-04	0.2148E 03	-0.3854E-02	0.1008E 01	0.5881E 01	0.2170E 02	0.1075E-00	0.3968E-00	0.3691E 01
O	65.	0.4625E-04	0.2197E 03	0.3982E-02	0.1021E 01	0.5899E 01	0.2136E 02	0.1078E-00	0.3904E-00	0.3620E 01
P	70.	0.4461E-04	0.2129E 03	-0.2260E-01	0.9943E 00	0.5831E 01	0.2120E 02	0.1075E-00	0.3875E-00	0.3604E 01
Q	75.	0.4425E-04	0.2102E 03	-0.3895E-01	0.9776E 00	0.5874E 01	0.2120E 02	0.1074E-00	0.3875E-00	0.3608E 01
R	80.	0.4420E-04	0.2100E 03	-0.3856E-01	0.9780E 00	0.5959E 01	0.2147E 02	0.1089E-00	0.3925E-00	0.3603E 01
S	85.	0.4322E-04	0.2053E 03	-0.6184E-01	0.9544E 00	0.6165E 01	0.2207E 02	0.1127E-00	0.4035E-00	0.3580E 01
T	90.	0.4401E-04	0.2091E 03	-0.4457E-01	0.9719E 00	0.6360E 01	0.2255E 02	0.1163E-00	0.4123E-00	0.3546E 01
U	95.	0.4549E-04	0.2161E 03	-0.9890E-02	0.1007E 01	0.6524E 01	0.2306E 02	0.1193E-00	0.4216E-00	0.3535E 01
V	100.	0.4780E-04	0.2270E 03	0.3973E-01	0.1058E 01	0.6616E 01	0.2328E 02	0.1209E-00	0.4256E-00	0.3519E 01

Y CO-ORDINATE IS HEAT TRANSFER WITH EACH DIVISION = 1.0000E-05 BTU/SEC/SQ.IN.

Figure 9

DATA REDUCTION FOR JP-17 HEAT TRANSFER AND PRESSURE, 3/15/63

RUN NUMBER 4. SEQUENCE NUMBER 1.

Y CO-ORDINATE IS THE NUSSLET NUMBER WITH EACH DIVISION = 0.5000E 02

Figure 10

DATA REDUCTION FOR JP-17 HEAT TRANSFER AND PRESSURE, 3/15/63

RUN NUMBER 4. SEQUENCE NUMBER 1.

Y CO-ORDINATE IS (H-PC)/H₀ WITH EACH DIVISION = 1.0000E-01

Figure 11

DATA REDUCTION FOR JP-17 HEAT TRANSFER AND PRESSURE, 3/15/63

RUN NUMBER 4. SEQUENCE NUMBER 1.

Y CO-ORDINATE IS (N-NO)/NO WITH EACH DIVISION = 0.2000E-00

1	1	1	1	1	1	1	1	1	1	1	1	0.200E 01
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	0.180E 01
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	0.160E 01
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	0.140E 01
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	0.120E 01
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	0.100E 00
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	0.800E 00
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	0.600E 00
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	0.400E-00
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	0.200E-00
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	0.269E-06

0. 10. 20. 30. 40. 50. 60. 70. 80. 90. 100.
PERCENTAGE DISTANCE. (X%)

Figure 12

DATA REDUCTION FOR JP-17 HEAT TRANSFER AND PRESSURE, 3/15/63

RUN NUMBER 4. SEQUENCE NUMBER 1.

Y CO-ORDINATE IS R.M.S. PRESSURE WITH EACH DIVISION = 1.0000E-01 P.S.I.

Figure 13

DATA REDUCTION FOR JP-17 HEAT TRANSFER AND PRESSURE, 3/15/63

RUN NUMBER 4. SEQUENCE NUMBER 1.

Y CO-ORDINATE IS PEAK TO PEAK PRESSURE WITH EACH DIVISION = 0.5000E 00 P.S.I.

Figure 14

DATA REDUCTION FOR JP-17 HEAT TRANSFER AND PRESSURE, 3/15/63

RUN NUMBER 4. SEQUENCE NUMBER 1.

Y CO-ORDINATE IS R.A.S. PRESS. / DOWNST. PRESS. WITH EACH DIVISION = 0.2000E-02

Figure 15

DATA REDUCTION FOR JP-17 HEAT TRANSFER AND PRESSURE, 3/15/63

RUN NUMBER 4. SEQUENCE NUMBER 1.

Y CO-ORDINATE IS P-P PRESS. / DOWNST. PRESS. WITH EACH DIVISION = 0.5000E-02

Figure 16

Y CO-ORDINATE IS P-P / R.M.S. PRESS. WITH EACH DIVISION = 1.0000E 00

Y CO-ORDINATE IS P-P / R.M.S. PRESS. WITH EACH DIVISION = 1.0000E 00

Figure 17

DATA REDUCTION FOR JP-17 HEAT TRANSFER AND PRESSURE, 3/15/63

RUN NUMBER 4. SEQUENCE NUMBER 1.

(N/NO) VS. (R.K.S. PRESS. / DOWNST. PRESS.), X DIV.= 0.20E-02 Y DIV.= 0.20E-00

1	1	1	1	1	1	1	1	1	1	1	1	0.200E 01
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	0.180E 01
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	0.160E 01
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	0.140E 01
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	0.120E 01
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	1.000E 00
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	0.800E 00
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	0.600E 00
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	0.400E-00
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	0.200E-00
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	
1	1	1	1	1	1	1	1	1	1	1	1	0.269E-06

LETTERS REFER TO THE X POSITION (SEE FIG. 8); N/N₀ IS Nu/Nu₀.

RUN NUMBER 4. SEQUENCE NUMBER 1.
(N/N0) VS. (P-P PRESS. / DOWNST. PRESS.), X DIV.= 0.50E-01 Y DIV.= 0.20F-00

DATA DESCRIPTION FOR (1) HEAT TRANSFER AND PRESSURE, 0715783
 RUN NUMBER 4. SEQUENCE NUMBER 1.
 (N/N0) VS. (P-P PRESS. / DOWNST. PRESS.), X DIV.= 0.50E-01 Y DIV.= 0.20F-00

[illegible]

Figure 19

DATA REDUCTION FOR JP-17 HEAT TRANSFER AND PRESSURE, 3/15/63

RUN NUMBER 4. SEQUENCE NUMBER 1.

P-P PRESS. / R.M.S. PRESS. VS. R.M.S. PRESS. / PBAR, X MAX. = 0.20E-01 X MIN. = 0.20E-02

Figure 20

JP-17 DATA REDUCTION FOR VELOCITY MEASUREMENTS

RUN NUMBER 4. SEQUENCE NUMBER 1.

PEAK-PEAK VEL. AVERAGED IN X = 0.4901E 02 R.M.S. AVERAGED IN X = 0.1704E 02
 PEAK-PEAK / MEAN VEL. AVERAGED IN X = 0.1354E 01 R.M.S. / MEAN VEL. AVERAGED IN X = 0.4708E-00
 RATIO OF PEAK-PEAK VEL. TO R.M.S. VEL. AVERAGED IN X = 0.2863E 01
 R = 0.1400E 01 S = 0. MEAN VELOCITY = 0.3620E 02

X	V(P-P)	V(R.M.S.)	VP/VBAR	VRMS/VBAR	VP/VRMS
0.	0.84529E 02	0.27128E 02	0.23352E 01	0.74943E 00	0.31160E 01
5.	0.85515E 02	0.26368E 02	0.23624E 01	0.72845E 00	0.32431E 01
10.	0.84529E 02	0.25903E 02	0.23352E 01	0.71558E 00	0.32633E 01
15.	0.84775E 02	0.28705E 02	0.23420E 01	0.79301E 00	0.29533E 01
20.	0.84529E 02	0.30022E 02	0.23352E 01	0.82939E 00	0.28155E 01
25.	0.83543E 02	0.34664E 02	0.23079E 01	0.95762E 00	0.24101E 01
30.	0.88966E 02	0.36104E 02	0.24578E 01	0.99740E 00	0.24642E 01
35.	0.83739E 02	0.31149E 02	0.23148E 01	0.86053E 00	0.26899E 01
40.	0.63820E 02	0.21306E 02	0.17631E 01	0.58860E 00	0.29954E 01
45.	0.51552E 02	0.15999E 02	0.14242E 01	0.44198E-00	0.32223E 01
50.	0.37292E 02	0.12532E 02	0.10302E 01	0.34622E-00	0.29756E 01
55.	0.28812E 02	0.94214E 01	0.79596E 00	0.26028E-00	0.30581E 01
60.	0.22715E 02	0.74279E 01	0.62919E 00	0.20520E-00	0.30662E 01
65.	0.17013E 02	0.59328E 01	0.47000E-00	0.16390E-00	0.28676E 01
70.	0.12074E 02	0.47552E 01	0.33355E-00	0.13137E-00	0.25391E 01
75.	0.87808E 01	0.37330E 01	0.24258E-00	0.10313E-00	0.23522E 01
80.	0.87808E 01	0.39997E 01	0.24258E-00	0.11049E-00	0.21954E 01
85.	0.14269E 02	0.47996E 01	0.39419E-00	0.13259E-00	0.29729E 01
90.	0.18385E 02	0.53328E 01	0.50790E 00	0.17495E-00	0.29031E 01
95.	0.25794E 02	0.88585E 01	0.71258E 00	0.24472E-00	0.29117E 01
100.	0.39625E 02	0.12746E 02	0.10947E 01	0.35211E-00	0.31089E 01

FLOW REVERSAL - THESE VALUES CAN'T BE TRUSTED
 (INSTRUMENTAL LIMITATION)

Figure 21

DATA REDUCTION FOR VELOCITY MEASUREMENTS OF JP-17, 3/15/63
 RUN NUMBER 4. SEQUENCE NUMBER 1.
 Y CO-ORDINATE IS PEAK TO PEAK VELOCITY.

Figure 22

DATA REDUCTION FOR VELOCITY MEASUREMENTS OF JP-17, 3/15/63
 RUN NUMBER 4. SEQUENCE NUMBER 1.
 Y CO-ORDINATE IS RMS VELOCITY.

THESE VALUES CAN'T BE TRUSTED

Figure 23

DATA REDUCTION FOR VELOCITY MEASUREMENTS OF JP-17, 3/15/63

RUN NUMBER 4. SEQUENCE NUMBER 1.

Y CO-ORDINATE IS PEAK TO PEAK VELOCITY DIVIDED BY THE MEAN VELOCITY.

Figure 24

DATA REDUCTION FOR VELOCITY MEASUREMENTS OF JP-17, 3/15/63
 RUN NUMBER 4. SEQUENCE NUMBER 1.
 Y CO-ORDINATE IS R.M.S. VELOCITY DIVIDED BY THE MEAN VELOCITY.

Figure 25

THESE VALUES CAN'T BE TRUSTED

DATA REDUCTION FOR VELOCITY MEASUREMENTS OF JP-17, 3/15/63
 RUN NUMBER 4. SEQUENCE NUMBER 1.
 Y CO-ORDINATE IS PEAK TO PEAK VEL. DIVIDED BY R.M.S. VELOCITY.

Figure 26

(N/NO) VS. R.M.S. VEL. / VBAR., X MAX. = 1.00E 00 X MIN. = 1.00E-01

Figure 27

DATA REDUCTION FOR VELOCITY MEASUREMENTS OF JP-17, 3/15/63

RUN NUMBER 4.

SEQUENCE NUMBER 1.

(N/NO) VS. PEAK TO PEAK VEL. / VBAR., X MAX. = 0.50E 01 X MIN. = 0.20E-00

Figure 28

DATA REDUCTION FOR VELOCITY MEASUREMENTS OF JP-17, 3/15/63

RUN NUMBER 4.

SEQUENCE NUMBER 1.

R.M.S. VEL. / VBAR VS. R.M.S. PRESS. / PBAR, X MAX. = 0.20E-01 X MIN. = 0.20E-02

1	1	1	1	1	H1	1	1	1	1	1	1.000E 00
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	G	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	FLOW REVERSAL	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	0.910E 00
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	F	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	0.820E 00
1	1	1	1	1	E	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	B	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	D	1	1	1	1	1	0.730E 00
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	0.640E 00
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	0.550E 00
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	0.460E-00
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	0.370E-00
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	0.280E-00
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	0.190E-00
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1.000E-01

Figure 29

DATA REDUCTION FOR VELOCITY MEASUREMENTS OF JP-17, 3/15/63

RUN NUMBER 4.

SEQUENCE NUMBER 1.

P-P VEL. / VBAR VS. P-P PRESS. / PSAR, X MAX. = 0.50E-01 X MIN. = 0.20E-01

1	1	1	1	1	1	1	1	1	1	1	1	0.500E 01
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	0.452E 01
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	0.404E 01
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	0.356E 01
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	0.308E 01
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	0.260E 01
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	1
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1	1	1	1	1	1	1	1	1	1	1	1	1
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1	1	1	1	1	1	1	1	1	1	1	1	1
1	1	1	1	1	1	1	1	1	1	1	1	0.200E-00

ED

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FLOW REVERSAL

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Figure 30

DATA REDUCTION FOR VELOCITY MEASUREMENTS OF JP-17, 3/15/63

RUN NUMBER 4.

SEQUENCE NUMBER 1.

P-P VEL. / R.M.S. VEL. VS. R.M.S. VEL. / VBAR, X MAX. = 1.00E 00 X MIN. = 1.00E-01

Figure 31

DATA REDUCTION FOR JP-17 HEAT TRANSFER AND PRESSURE, 3/15/63

RUN NUMBER 4. SEQUENCE NUMBER 10.

P-P PRESS. / R.M.S. PRESS. VS. R.M.S. PRESS. / PBAR , X MAX. = 1.00E-02 X MIN. = 0.20E-02

Figure 32

DATA REDUCTION FOR VELOCITY MEASUREMENTS OF JP-17, 3/15/63

RUN NUMBER 4.

SEQUENCE NUMBER 10.

P-P VEL. / R.M.S. VEL. VS. R.M.S. VEL. / VBAR, X MAX. = 0.20E-00 X MIN. = 1.00E-01

Figure 33

Graphical determination of the heater strip time lag

Figure 34

Local heat transfer increase vs local pressure oscillation relative intensity

Figure 35

Average heat transfer increase, mean oscillating pressure reduced intensity and standing wave ratios, vs. frequency.

Figure 36

Overall "effectiveness" of pressure oscillations vs frequency

Figure 37

Non-linear pressure-velocity experimental correlation

Figure 38

Non linear pressure-velocity experimental correlation

Figure 39

Heat transfer increase versus amplitude of velocity oscillations

Figure 40

Heat transfer increase versus amplitude of velocity oscillations

Heat transfer increase versus amplitude of velocity oscillations

Local heat transfer increase versus local amplitude of velocity oscillations

Figure 43

Heat transfer increase versus intensity of velocity oscillations

Figure 44

Frequency effect on $Nu/Nu_o = f(\tilde{V}pK/\bar{V})$ relation

Figure 45

Frequency effect on $Nu/Nu_o = f(\tilde{V}pK/\bar{V})$ relation

Figure 46

Frequency effect on $Nu/Nu_o = f(\tilde{V}pK/\bar{V})$ relation

Figure 48

Figure 49

Figure 50

Shape of pressure oscillations

Figure 51

Shape of pressure oscillations

Figure 52

Shape of velocity oscillations

Figure 53

Comparison between oscillating flow and steady-state average velocity profiles at one x location

Figure 54

Steady-state average velocity profiles $\bar{V} = \bar{V}(y)$
at three axial locations

Figure 55

Comparison between oscillating flow "average velocity profiles"
at 3 different x locations
 $f = 760 \text{ cps}$

Differences in the average velocity profiles $\bar{V}(y)$
at three axial locations in the duct.

Figure 57

Profiles of oscillating velocity peak to peak amplitude at three axial locations

Figure 58

Profiles of oscillating velocity intensity at three axial locations

Figure 59

Comparison between three methods of oscillating velocity measurement

Figure 60

Comparison between three methods of oscillating velocity measurement

Figure 61

Oscillating velocity amplitude at different radial locations for an axial survey

Figure 62

Oscillating velocity intensity at different radial locations
for an axial survey

Figure 63

Oscillating velocity reduced amplitude at different radial locations
for an axial survey

Figure 64

Oscillating velocity reduced intensity at different radial locations
for an axial survey

Figure 65

Shape of the velocity waves at different radial locations
for an axial survey

Figure 66

Oscillating velocity amplitude profiles at different axial locations

Figure 67

Figure 68

Oscillating velocity intensity profiles at different axial locations

Figure 69

Oscillating velocity intensity profiles at different axial locations

Figure 70

Oscillating velocity reduced amplitude profile at
different axial locations

Figure 71

Oscillating velocity reduced intensity profiles at
different axial locations

Figure 72

Velocity waves - shape profiles at different axial locations

Figure 73

Effects of reversed flow along a radius at axial point
 $X/l = 60\%$, frequency 322 cps, $\bar{P} = 46.7$ psia,
 $\dot{m} = 0.238$ lbs/sec, $M = 0.068$, $Re = 2.22 \times 10^5$

V_H' at $Y/R = 20\%$

P_H' at wall (typical)
small shock present

V_H' at $Y/R = 60\%$

V_H' at $Y/R = 100\%$
center of duct

Gap variation method

Figure 75

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