<Original>

A Study on the Thermal and Dynamic Behavior of the Single Combustion Chamber Pulse Burner

Kwan Soo Lee* and Robert J. Schoenhals**

(Received August 29, 1985)

맥동 연소식 열교환기의 열적 및 동적 특성에 관한 해석

이 관 수 · Robert J. Schoenhals

Key Words; Heat Transfer(열전달), Pulsation(맥동), Combustion(연소), Water Heater (급수가열기)

초 록

배기 decoupling chamber를 갖고 있는 맥동 연소 급수가열기에 대한 수학적인 모델링과 관련된 컴퓨터 시뮬레이션을 하였고, 이때 계에 대한 맥동 현상과 관련된 열적 및 동적 특성을 고려하였다.

시뮬레이션결과 시동기간에 계의 각 부분에서 일어나는 열전달, 압력 및 온도를 구하였고 또한 해석의 제한성과 개선의 필요성을 논의하였다.

이 연구에서 얻어진 결과는 도관에서 맥동에 의한 대류열전달을 예측하기 위한 모델을 세우는데 활용될 수 있다.

Nomenclature

A : Cross sectional area (m^2)

C_p : Specific heat at constant pressure (J/kg·K)

C_v : Specific heat at constant volume (J/kg·K)

c : Speed of sound (m/s)

$\overline{gs}$: Total exchange area per unit area

$\overline{GS}$: Total exchange area (m^2)

H : Heat transfer coefficient (W/m^2·K)

h : Enthalpy (J/kg)

ΔH : Heat of combustion (J/kg)

L : Length (m)

m : Mass (kg)

$\dot{m}$: Mass rate (kg/s)

P : Pressure (N/m^2)

P_g : Supply pressure of the fuel (N/m^2)

$\dot{Q}$: Heat loss rate (w)

$\dot{q}$: Heat loss rate per unit area (w/m^2)

r : Air-fuel ratio

S_b : Equivalent flame speed (m/s)

* Member, Assistant Professor, Dept. of Mechanical Engineering, Hanyang University.

** Professor, School of Mechanical Engineering, Purdue University.

T : Temperature (k)
t : Time (s)
U : Velocity (m/s)
u : Internal energy (J/kg)
v : Volume (m^3)
W : The perimeter of the flue tube (m)
x : Axial coordinate in the flue tube (m)
$\bar{x}$: Dimensionless axial coordinate, x/L_f
ε : Emissivity
ρ : Density (kg/m^3)
σ : Stefan-Boltzmann constant, $5.67\times10^{-8} w/m^2\cdot K^4$

Subscript

a : Air
b : Burning
c : Combustion chamber
e : Exhaust chamber
f : Flue tube
g : Gas
gv : Gas flapper valve
l : Exit of the flue tube
o : Entrance of the flue tube
p : Product
r : Reactant
t : Tail pipe
to : Entrance of the tail pipe
w : Wall

1. Introduction

Although combustion-driven oscillation was first reported by Byron Higgins(1) around 1800, it has been well documented only since the early 1900's. A French patent of a pulsation system with flapper valves, which was an attempt to develop an early aircraft engine, was issued to Esnaut-Pelterie in 1906(2). Similar patents were processed in France by Karavodine in 1908 and Marconnet in 1910(3). These patents provided a plausible principle which appeared to have potential for aircraft propulsion and for the generation of the gas pressure for a turbine. However, success was later attained in the propulsion application with "the Schmidt tube" developed by Schmidt in Germany. This device was applied to the engine of the well-known $V-1$ "buzz bomb" in 1942, and represented the first real achievement of a pulse combustion unit in a practical section of technology. After the advent of the $V-1$, the bulk of the work concerning the pulse combustion process was performed in the area of propulsion. With the appearance of the ramjet and the turbojet engines in the aircraft industry, interest in pulse combustion diminished somewhat during 1960's. But, since fossil fuels have become more scarce and expensive, there has been a renewed interest in the pulse combustion as a means of increasing the efficiencies of combustion heating systems for a variety of applications.

In a pulse combustion burner, there is an oscillation of the pressure in the combustion chamber. This allows the device to have several operating and design advantages over conventional burners, including compact design, enhanced heat transfer performance, and automatic rejection of exhaust gases.

In order to design high quality pulse combustion devices, more detailed understanding of pertinent phenomena is required. This can be pursued by combining theoretical studies with meaningful experiments. However, pulse combustion development has been impeded by the considerable complexity of the various physical processes involved. A good summary of previous studies concerning pulse combustion has been provided by Putnam(1), who has suggested three analytical approaches for understanding the complexity of working pulse combustors. Griffiths et al.(4,5,6) have supplied considerable experimental data and also provided design guidelines for Helmholtz-type pulse combustors. Reader(7) has carried out experimental and

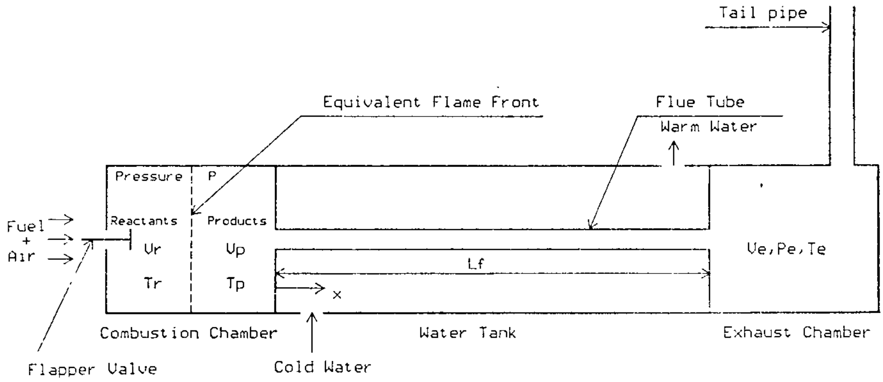


Fig. 1 Single combustion chamber pulse burner for water heating

theoretical investigations of a pulse combustion device, which burned a gaseous methane fuel and operated at a nominal frequency of 40 Hz. A one-dimensional model of a valveless pulse combustor was derived, but no results were presented because the computer program was incomplete. Much of the work in pulse combustion has been limited to analytical research. One of the first exceptions was Ahren et al.[8], who predicted frequency, peak pressure, and firing rate. This model, which was based on the Helmholtz resonator, consisted of a combustor and a tail pipe, but no decoupling chamber. Dhar et al.[9], presented an extension of the model by Ahrens. Lee et al.[10,11] improved the model of Dhar et al. especially in regard to flame front propagation and flapper valve motion. Most of the previous experimental and theoretical research has been focused on characterizing the dynamic behavior in terms of the combustion chamber pressure. However, Huang[12] have presented experimental data associated with the sustained steady state temperature distribution and heat transfer along the flue tube. However, the oscillatory component of temperature was not measurable because the response time (time constant) of the thermocouple probe was excessive for the frequency range involved.

The objective of this research is the understanding of the physical phenomena occurring within a single combustion chamber pulse burner with decoupling chamber. The present work is an extension of that reported in reference[11]. More specifically, the model described in this study predicts thermal behavior, and then couples this information to the prediction of dynamic characteristics.

2. Modeling

Fig. 1 shows a possible arrangement of pulse burner for water heating. It also shows the essential features of four component models developed in this work. These models pertain to the combustion chamber, flue tube, exhaust chamber and tail pipe.

In this pulse combustion burner, the reactant gas mixture is supplied through the inlet flapper valve, and reacts chemically in the combustion chamber. As a result of this combustion, the pressure in the combustion chamber increases, closing the inlet valve and expelling the product gas through the exhaust decoupling chamber. The flue tube is immersed in the water tank. The flue tube and the water tank constitute a parallel-flow exchanger. As the product gas flows down the flue tube, the pressure in the combustion chamber drops and eventually falls below the reactant supply pressure. This causes the inlet valves to reopen and admit fresh reactant gas mixture into the combustion chamber. This new reactant is self-ignited by the residual gas which remains from the previous cycle. This again forces the flapper valve to close, and the entire process is repeated.

2.1 Combustion Chamber

Because of the combustion which occurs in this portion of the system, it is very difficult to predict fluid flow and heat transfer. To obtain a simple model of combustion, it is appropriate to make the following major assumptions:

(1) The gas within combustion chamber consists of two zones, an unburned zone and a burned zone, and the pressures of both zones are equal.

(2) The gases in each zone obey the ideal gas equation of state.

(3) The air-fuel ratio does not vary with time.

(4) The effect of dissociation can be negligible.

The energy equations for the two zones are

$$\frac{d}{dt}[m_r u_r] = -P_c\frac{dV_r}{dt} - \Sigma\dot{Q}_r - h_r\dot{m}_b + h_{in}\dot{m}_{in} \tag{1}$$

$$\frac{d}{dt}[m_p u_p] = -P_c\frac{dV_p}{dt} - \Sigma\dot{Q}_p + h_r\dot{m}_b + \frac{\Delta H}{1+r}\dot{m}_b - h_{f,0}\dot{m}_f \tag{2}$$

The equation of state is applied to each zone to obtain

$$P_c V_r = m_r R_r T_r \tag{3}$$

$$P_c V_p = m_p R_p T_p \tag{4}$$

The constraint condition on volume is

$$V_c = V_p + V_r \tag{5}$$

where V_c is the total volume of the chamber. Conservation of mass in each zone is written as

$$\dot{m}_r = \dot{m}_{in} - \dot{m}_b \tag{6}$$

$$\dot{m}_p = \dot{m}_b - \dot{m}_f \tag{7}$$

In order to more readily solve equation (1) through (5), they can be arranged in the following form:

$$\dot{P}_c/P_c = \left[F\frac{\dot{m}_{in}}{m_r} - G\frac{\dot{m}_b}{m_r} - D\frac{R_r}{C_{p,r}}\,\frac{\Sigma\dot{Q}_r}{P_c V_p} - \frac{\Sigma\dot{Q}_p}{P_c V_p} - D\frac{\dot{m}_f}{m_p}\right]/E \tag{8}$$

$$\dot{T}_r/T_r = \left[\frac{\dot{P}_c}{P_c} - \frac{\Sigma\dot{Q}_r}{P_c V_r} - A\frac{\dot{m}_{in}}{m_r}\right]\frac{R_r}{C_{p,r}} \tag{9}$$

$$\frac{\dot{V}_r}{V_r} = \frac{\dot{P}_c}{P_c} + \frac{\dot{m}_r}{m_r} + \frac{\dot{T}_r}{T_r} \tag{10}$$

$$\frac{\dot{T}_p}{T_p} = \left[\frac{\dot{P}_c}{P_c} - \frac{\Sigma\dot{Q}_p}{P_c V_p} + B\frac{\dot{m}_b}{m_p}\right]/D \tag{11}$$

where

$$A = \frac{h_r - h_{in}}{R_r T_r}, \tag{12}$$

$$B = \frac{(h_r - h_p) + \dfrac{\Delta H}{1+r} + (h_p - h_{f,0})\dfrac{\dot{m}_f}{\dot{m}_b}}{R_p T_p}$$

$$D = \frac{C_{v,p}}{R_p} + 1, \quad E = D\left(1 + \frac{V_r}{V_p}\,\frac{C_{v,r}}{C_{p,r}}\right) - 1$$

$$F = D\frac{V_r}{V_p}\left(1 - \mathrm{A}\frac{R_r}{C_{p,r}}\right),$$

$$G = D\left(\frac{V_r}{V_p} - \frac{m_r}{m_p}\right) - B\frac{m_r}{m_p}$$

Studies have shown that in most cases the heat loss in the combustion chamber is primarily due to radiative heat transfer with convective heat transfer providing a smaller contribution. As a first approximation, it is reasonable to assume that the heat loss in the unburned zone is negligible with the heat transfer in the burned zone being due to radiation alone. Kim and Ferguson[13] have presented a simplified method for estimating the radiation heat transfer from a hot inner region of the gas through a transparent outer region to the combustion wall.

The equation used is

$$\dot{Q}=A\sigma(T_g^4-T_w^4)\varepsilon_w^{0.8}f^m(1-e^{-n\tau}) \tag{13}$$

where

$\dot{Q}$: Total heat loss by hot region to walls

A : Total wall area

T_g : Hot gas temperature

ε_w : Wall emissivty

f : Ratio of hot gas volume to chamber volume

τ : Optical depth of the hot gas region

m and n are chosen as follows:

$$m=2/3 \quad \text{and } n=1.0 \text{ if } \tau<1$$
$$m=(2/3)\tau^{-0.33} \quad \text{and } n=0.9 \text{ if } \tau\geqslant 1 \tag{14a}$$

Motivation for the development of this model involved heat transfer in a diesel engine. However, it can be expected to apply equally to gas fired heating. The optical depth is related to the absorption coefficient, α, by

$$\tau=\alpha L_g. \tag{14b}$$

where L_g is the mean beam length. Hottel[14] has presented a method for obtaining the absorption coefficient for a mixture of carbon dioxide and water vapor.

Flow through the air and gas valve systems can be treated in the same manner as flow through an orifice, with the valves fully open during only a fraction of the operation cycle. The total inlet mass flow rate (air plus fuel) can be expressed by

$$\dot{m}_{in}=\dot{m}_g+\dot{m}_a=\dot{m}_g(1+r) \tag{15a}$$

with

$$\dot{m}_g=\begin{cases} A_{gv}\sqrt{2\rho_g(P_g-P_c)} & P_g\geqslant P_c \\ 0 & P_g<P_c \end{cases} \tag{15b}$$

The mass burning rate is expressed using the concept proposed by Lee et al.[11]:

$$\dot{m}_b=\rho_r A_c S_b \tag{16}$$

Here, S_b is the equivalent flame speed which is different from the actual flame speed and must be given as input data.

2.2 Flue Tube

The thermal performance of the unit is established by the processes occurring in this portion of the system. To model this section, the following assumptions are made:

(1) The velocity profile is uniform over the flue tube cross sections.

(2) The time variation of the local gas density may be neglected

(3) At any instant, the pressure gradient along the axis of flue tube is constant

The continuity equation is expressed as

$$\dot{m}_f=\rho U_f A_f \tag{17}$$

The momentum equation is

$$-\frac{\partial P}{\partial x}A_f-\tau_0 W=\frac{\partial}{\partial t}(\rho U_f A_f)+\frac{\partial}{\partial x}[(\rho U_f A_f)U_f] \tag{18}$$

The shear stress, τ_0, at flue tube wall can be written as

$$\tau_0=\begin{cases} \left|f\cdot\frac{\rho U_f^2}{8}\right|, & u_f\geqslant 0 \\ -\left|f\cdot\frac{\rho U_f^2}{8}\right|, & u_f<0 \end{cases} \tag{19}$$

If the instantaneous shear stress is considered to be the same as that occurring during steady flow at the same velocity, the friction factor values, f, can be obtained in the usual manner. Equation (18) is now integrated from the entrance of the flue tube to the exit, and is combined with the continuity equation (17) and shear stress equation (19) to obtain

$$L_f\frac{d\dot{m}_f}{dt}=(P_c-P_e)A_f-\frac{|\dot{m}_f|}{8A_f^2}\cdot\dot{m}_f\cdot W\int_0^{L_f}\frac{|f|}{\rho}dx-\frac{\dot{m}_f^2}{A_f}\left[\frac{1}{\rho_L}-\frac{1}{\rho_0}\right] \tag{20}$$

The energy equation is written as

$$\dot{m}_f C_p\frac{\partial T_f}{\partial x}+\rho C_v A_f\frac{\partial T_f}{\partial t}+W\dot{q}_f=0 \tag{21}$$

where $\dot{q}_f$ is the heat loss per unit area in the flue tube due to radiation and convection. Following Hanby[15], the convective heat transfer coefficient for calculating the convection heat transfer is obtained by assuming the quasi-steady-state condition. If the gradient of the axial radiation heat transfer can be considered negligible, the radiation heat transfer between the flue gas and the surface of the flue tube can be expressed by[16]:

$$\dot{q}_f=\overline{gs}\cdot\sigma(T_f^4-T_w^4) \tag{22a}$$

with

$$\overline{gs}=\frac{1}{\frac{1}{\varepsilon_f}+\frac{1}{\varepsilon_w}-1} \tag{22b}$$

The total gas emissivity has been determined from the simplified emissivity equation developed by Hottel[14].

2.3 Exhaust Chamber

The exhaust chamber is treated in the same manner as the combustion chamber. The energy equation, state equation, and conservation of mass equation are written as

$$\frac{d}{dt}[m_e u_e]=\dot{m}_f h_e-\dot{m}_t h_{t0}-\Sigma\dot{Q}_e \tag{23}$$

$$\dot{P}_e/P_e=\frac{\dot{m}_e}{m_e}+\frac{\dot{T}_e}{T_e} \tag{24}$$

$$\dot{m}_e=\dot{m}_f-\dot{m}_t \tag{25}$$

The total heat transfer from the flue gas to the wall of the exhaust chamber is expressed as the sum of the gas radiation and convection.

$$\dot{Q}_e=\overline{GS}\ \sigma(T_e^4-T_w^4)+HA_e(T_e-T_w) \tag{26}$$

The total exchange area $\overline{GS}$ is evaluated as in the case of the flue tube calculation. The convective heat transfer coefficient is obtained from an appropriate correlation.

2.4 Tail Pipe

The continuity, momentum, and energy equation can be written as in the case of the flue tube. For an anechoic pipe, the velocity at the pipe entrance at any instant is given by reference[17]:

$$U_t=\frac{(P_e-P_{atm})}{c\cdot\rho_t} \tag{27}$$

3. Mixture Gas Properties

The thermodynamic and transport properties of gas mixtures required for computation of the behavior of a pulse combustion burner are enthalpy, viscosity, and thermal conductivity.

For the calculation of these properties, the fuel is assumed to be methane and the effects of dissociation are assumed to be negligible when the combustion occurs. The chemical reaction for the combustion of 1 mole of methane can be written as

$$CH_4+2(1+X)(O_2+3.76N_2)\rightarrow CO_2+2H_2O+2XO_2+7.52(1+X)N_2$$

where X is excess air. The relationship between excess air X and air-fuel ratio r is

$$r=\frac{2(1+X)(1+3.76)(28.97)}{16}$$

The enthalpy of mixture gas is determined from

$$h_{\mathrm{mix}}=\sum_{i=1}^{n} y_i h_i \tag{28}$$

where y_i is mole fraction of each component and h_i is molar enthalpy of each component. The reference temperature used for those calculations is 298K, the enthalpy of each component being set equal to zero at this temperature.

The viscosity of a gas mixture composed of n components can be calculated from Herning and Zipperer's formula[18]:

$$\mu_{\mathrm{mix}}=\sum_{i=1}^{n}(y_i\mu_i M_i^{1/2})/\sum_{i=1}^{n}(y_i M_i^{1/2}) \tag{29}$$

where μ_i is the viscosity of constituent i and M_i is the molecular weight of constituent i.

The thermal conductivity of a gas mixture composed of n components may be estimated with a relationship given by Burgoyne and Weinberg[19]:

$$k_{\mathrm{mix}}=0.5\left[\sum_{i=1}^{n} y_i k_i+\left[\sum_{i=1}^{n}(y_i/k_i)\right]^{-1}\right] \qquad (30)$$

where k_i is the thermal conductivity of constituent i.

The viscosity and thermal conductivity of each component are required for the calculations indicated by equations(29) and(30). Unfortunately, the absence of reliable experimental data for transport properties(viscosity and thermal conductivity) becomes a serious limitation as temperature increases. Therefore, to avoid being restricted by an upper limit of temperature, the viscosity and thermal conductivity can be estimated using expressions derived from the rigorous kinetic theory of gases. Then, for viscosity[18].

$$\mu_i=26.693\times10^{-6}\frac{(M_iT)^{1/2}}{\sigma^2\Omega^{(2,2)}} \qquad (31)$$

and for thermal conductivity[20].

$$k_i=0.08321\frac{(T/M_i)^{1/2}}{\sigma^2\Omega^{(2,2)}}\left[1+\frac{0.352}{1+\alpha}\left(\frac{C_{\mathrm{int}}}{R}\right)-\left(1.030-\frac{0.544}{1+\alpha}\right)^2\frac{C_r}{R}\ \frac{1}{Z_r}\right] \qquad (32)$$

Here, σ is the low velocity collision diameter[21], Ω is the collision integral[22], α is resonant correction. C_r and C_{int} are, respectively, due to the rotational modes and all the internal modes associated with the molar heat capacity. Z_r is the collision number for rotational relaxation.

4. Computer Simulation

Four ordinary differential equations were used to describe conditions in the combustion chamber, one ordinary differential equation and one partial differential equation were used for the flue tube, and two ordinary differential equations were used for the exhaust chamber. These equations constitute a coupled set which must be solved simultaneously for each time step. It is required that thermodynamic and transport properties, which vary with temperature, be calculated at each step. The nonlinear and coupled nature of these equations make it unlikely that a closed form analytical solution can be obtained. Therefore, a numerical method consisting of a sixth order Runge-Kutta method was used to solve the ordinary differential equations, and a finite difference method was used for the partial differential equation.

The computer simulation used the system parameters associated with the apparatus employed in prior experimental work at Purdue University[2,12,23], with the exception of part of the tail pipe. In the actual experimental apparatus, the exhaust gas leaving the exhaust chamber was expelled through a muffler installed to reduce the noise before entering the tail pipe. Numerical values for the parameters used are given in the Appendix. For the initial condition, it was assumed that the combustion chamber was fully charged with a uniform mixture of reactant gas. It was also assumed that the inlet flapper valve was initially closed. Ignition was imposed at the center of the combustion chamber.

Eight equally spaced nodes were located in the flue tube. Several different time increments were used, with the very small time increments leading to sizable round-off errors and somewhat larger time increments causing numerical instabilities. The most consistent and stable results, probably incorporating a small amount of round-off error, were obtained with a time increment of 0.0001 sec. All of the results presented in the following section were obtained with this time increment. It was found that time increments differing from this by some nominal factor(for example, 0.00005 seconds) yielded results that were almost identical to those obtained with 0. 0001 seconds.

5. Results and Discussion

The results of computer simulations for a single chamber pulse burner with a decoupling chamber terminating into an anechoic pipe are discussed.

Fig. 2 illustrates pressure vs time for the combustion chamber and exhaust chamber. The phase angle between the two is essentially 180°. The amplitude of the pressure oscillation in the exhaust chamber is small in comparison with that in the combustion chamber. Thus, the exhaust chamber attenuates the upstream pressure pulsations as far as the remainder of the exhaust system (tailpipe) is concerned. These results are in reasonably good agreement with prior measurements of steay state frequency,

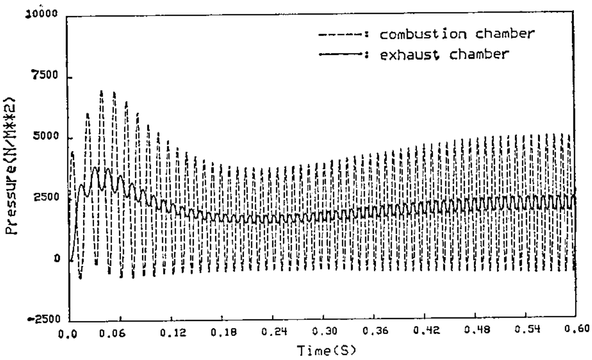


Fig. 2 Pressure in the combustion chamber and exhaust chamber

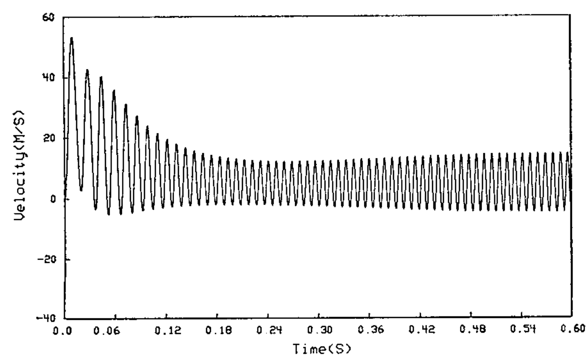
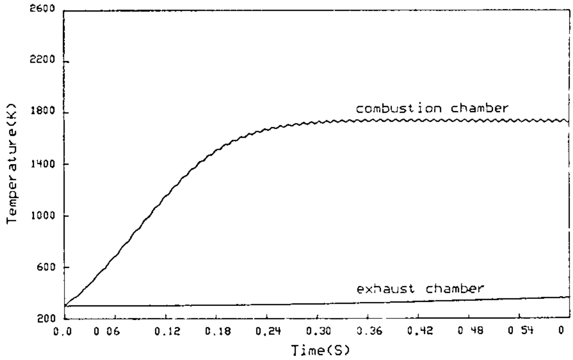


Fig. 3 Velocity in the entrance of the flue tube



Fif. 4 Temperature in the combustion chamber and exhaust chamber

amplitudes, and phase. However, the mean pressure values are somewhat higher than those observed experimentally. For example, the simulation yields a final mean pressure in the combustion chamber of approximately 2150N/m^2(see Fig. 2) in comparison with an experimentally observed steady state value of 1300 N/m^2. It is speculated that the omission of the muffler in the simulation may be the major cause of this discrepancy. The other deviations were found to be somewhat smaller. For example, the simulation yields a steady state frequency of 109 Hz, whereas the corresponding experimentally observed frequency was approximately 90 Hz—a deviation of 21%.

The gas velocity in the entrance of the flue tube is shown in Fig. 3. It was found that, for the larger time values shown, this velocity lags the combustion chamber pressure by approximately 90°.

Temperatures in the combustion chamber and exhaust chamber are illustrated in Fig. 4. The combustion chamber temperatures reached a sustained, nearly constant value after 25~30 cycles of the pressure oscillation. The very small temperature oscillation was found to possess the same frequency and phase as the pressure. The eventual mean temperature is

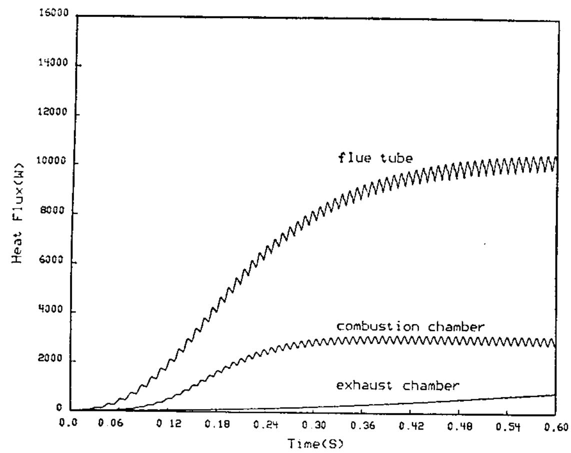


Fig. 5 Heat transfer in the combustion chamber, flue tube, and exhaust chamber

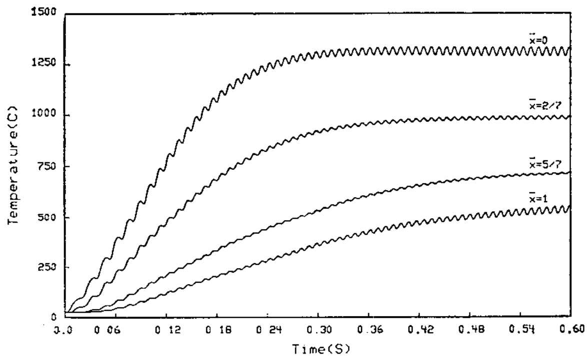


Fig. 6 Gas temperature at various positions along the flue tube

1731K, and the amplitude is only 9°C. The predicted exhaust chamber temperature can be seen to take a longer time to reach a sustained value.

The model developed in this work predicts a longer time to reach steady state operation than the model used by Lee et al.[11]. The latter model yielded steady-state after only 6～8 cycles. The longer times obtained with the present model are more in line with past experimental experience. The model used by Lee et al. does not account for the thermal inertia associated with the heat transfer, and therefore predicts a much shorter time to reach steady state. Fig. 5 illustrates the predicted heat loss in the combustion chamber, flue tube, and exhaust chamber. It can be seen that the flue tube is the most significant element as far as heat loss from the combustion chamber is concerned as it is seen to be only about one-fourth that of the flue tube. Gas temperatures at various positions along the flue tube are given in Fig. 6. Since heat transfer occurs from the hot gas to the water as it flows through the tube, its temperatrue decreases from inlet to outlet. It was found that the frequency of the small temperature oscillation is the same as that of the pressure oscillations discussed previously.

6. Concluding Remarks

The results presented here indicate that the simulation is successful in providing reasonable predictions of a number of experimentally observed features. Consequently, it is anticipated that this model will aid in establishing improved future predictions of convective heat transfer in the presence of pulsations in the flue tube. However, a number of limitations of the present model should be mentioned in order to identify items where future efforts are definitely needed. First, it was found that the predicted results are very sensitive to modest changes in flame speed. This suggests that an improved burning rate model should be developed. Second, it can be seen from some of the plots presented here that true steady state conditions were not really achieved over the observed time period of 0.6 seconds. Thus, to achieve a true steady state, considerably longer computer times would be required with the present simulation program. Since substantial computer time was required to simulate 0.6 seconds of system response time in this work, it is evident that the recurring problem of balancing stability, round-off error, and computer time is a major issue which is deserving of future effort. The third item involves the heat transfer modeling. Future efforts should include an accounting for the thermal inertia of the various containing components (combustion chamber housing, etc.). This, in turn, will surely lengthen the predicted time to reach steady state and will aggravate the problem discussed above. Another aspect of the heat transfer was mentioned briefly, namely, the development of a more sophisticated and realistic model of the flue tube heat transfer in the presence of oscillating pressure, velocity, and temperature.

Acknowledgment

This study was supported by Korea Science and Engineering Foundation, and the School of Mechanical Engineering at Purdue University. Appreciation is expressed to Professor Soedel for his comments and suggestions. Special thanks are expressed to the staff of the Ray W. Herrick Laboratories for their helpful assistance. Gratitude is also expressed to Professor Cohen, Director and to Sandy Stephens for her aid.

References

(1) Putnam, A.M. "General Survey of Pulse Combustion", Proceedings of the First International Symposium on Pulsating Combustion, Sheffield, England, 1971

(2) Vogt, S.T. "Performance Characteristics of a Pulse Combustion Water Heater", MSME Thesis, Purdue University, December, 1978

(3) Blomquist, C.A., "Experimental Gas-Fired Pulse Combustion Studies", Argonne National Laboratory Report No: ANL/EESIM-214, September, 1982

(4) Griffiths, J.C., "Some New or Unusual Methods for Heating Water with Gas", American Gas Association Research Bulletin 97, September, 1963

(5) Griffiths, J.C., and Niedzwicke, R.W., "An Evaluation of Some New or Different Ways to Heat the Home", American Gas Association Research Bulletin 101, May, 1965

(6) Griffiths, J.C., and Weber, E.J., "The Design of Pulse Combustion Burners", American Gas Association Research Bulletin 107, June, 1969

(7) Reader, G.T., "Aspects of Pulsating Combustion", *SAE*, No. 789251, pp. 548~557, 1978

(8) Ahren, F.W., Kim, C., and Tam, S.W., "An Analysis of Pulse Combustion Burner", Symposium Paper, Vol. 84, Part 1, At-78-2, No. 4, pp. 488~507, 1978

(9) Dhar, B., Huang, H.C.G., Lee, J.H., Soedel, W., and Schoenhals, R.J., "Dynamic and Thermal Characteristics of a Pulse-Combustion Gas-Fired Water Heater", Proceedings of Symposium of Pulse-Combustion Applications, Paper 4-1 to 4-28, March, 1982

(10) Lee, J.H., "Computer Simulation of Pulsation in a Gas Fired Pulse Combustion Device and Predictions of their Exhaust Noise for Single and Dual Combustion Chamber Designs", Ph. D. Thesis, Purdue University, December, 1983

(11) Lee, J.H., Dahr, B., and Soedel, W., "A Mathematical Model of Low Amplitude Pulse Combustion Systems Using a Helmholtz Resonator-Type Approach", Journal of Sound and Vibration, Vol. 98, No. 3, pp. 379~401, 1985

(12) Huang, H. S. G., "Heat Transfer in a Pulse Combustion Water Heater", Ph. D. Thesis, Purdue University, May, 1984

(13) Kim, D.M., and Ferguson, C.R., "A Simple Yet Sound Model of Radiation Heat Transfer in Diesel Engines", Presented to SAE, 1985

(14) Hottel, H.C., "Radiation from Carbon Dioxide, Water Vapor and Soot", presented to American Flame, April, 1985

(15) Hanby, V.I., "Convective Heat Transfer in a Gas-Fired Pulsating Combustor", ASME Journal of Engineering for Power, Vol. 91, pp. 48~52, 1969

(16) Hottel, H.C., and Sarofirm, A.F., *Radiative*

Transfer, McGraw-Hill Book Company, New York, 1967.

(17) Soedel, W., "On the Simulation of Anechoic Pipes in Helmholtz Resonator Models of Compressor Discharge Systems", Proceedings of the 1974, Purdue Compressor Technology Conference, pp. 136~140, 1974

(18) Touloukian, Y.S., Saxena, S.C., and Hestermans, P., "Thermophysical Properties of Matter", TPRC Data Series, Viscosity, Vol. 11, 1973

(19) Touloukian, Y.S., Liley, P.E., and Saxena, S. C., "Thermophysical Properties of Matter", TPRC Data Series, Thermal Conductivity Nonmetallic Liquids and Gases, Vol. 3, 1970

(20) Brokaw, R.S., "Predicting Transport Properties of Dilute Gases", I & EC Process Design and Development, Vol. 8, No. 2, pp. 240~253, 1969

(21) Svehla, R.A., "Estimated Viscosities and Thermal Conductivities of Gases at High Temperature", NASA Technical Report 132, Lewis Research Center, Cleveland, Ohio, 1962

(22) Neufeld, P.D., Janzen, A.R., and Aziz, R. A., "Empirical Equations to Calculate 16 of the Transport Collision Integral for the Lennard-Jones(12-6), Potential", J. Chemical Physics, Vol. 57, No. 3, pp. 1100~1101, 1972

(23) Vogt, S.T., Yen, M.S., Schoenhals, R.J., and Soedel, W., "Performance of a Pulse Combustion Gas-Fired Water Heater", Paper, No. 2563, ASHRAE Transactions, Vol. 86, Part 1, pp. 126~141, 1980

Appeedix

Values Used in Computer Simulation

$A_c=7.306\times10^{-2}\text{m}^2$

$A_f=7.918\times10^{-3}\text{m}^2$

$A_{gv}=8.871\times10^{-5}\text{m}^2$

$A_t=3.959\times10^{-3}\text{m}^2$

$\Delta H=5.0\times10^{7}\text{J/kg}$

$L_f=1.829\text{m}$

$r=21.819$

$$S_b=\begin{cases}0.1234\text{m/s}\,(P_g\leqslant P_c)\\ 0.0617\text{m/s}\,(P_g>P_c)\end{cases}$$

$T_w=26°\text{C}$

$V_c=6.649\times10^{-3}\text{m}^3$

$V_e=2.867\times10^{-2}\text{m}^3$

$\rho_g=0.6785\text{kg/m}^3$

$\varepsilon_w=0.8$