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A Study of the Dynamic Flue-Gas Temperature and Off-Period Mass Flow Rate of a Residential Gas-Fired Furnace

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A STUDY OF THE DYNAMIC FLUE-GAS TEMPERATURE AND OFF-PERIOD MASS FLOW RATE OF A RESIDENTIAL GAS-FIRED FURNACE

Abstract

The flue-gas temperature and mass flow rate through a gas-fired furnace were studied in the laboratory. Temperature profiles were measured under cycling conditions and compared with profiles predicted mathematically using data obtained while the furnace was cooling down from steady-state operation and warming up from equilibrium. The mass flow rates at various flue-gas temperatures were measured using both a vane anemometer and a tracer-gas technique, and these results are compared with the mass flow rate predicted by the theoretical equations. The effect on the off-period flow rate of automatic stack dampers having different sized damper openings was experimentally determined. Theoretical equations are presented for predicting the effectiveness of a stack damper as a function of the ratio of the area of the damper to the area of the stack and a system friction factor.

Key Words: Automatic stack damper; flue-gas temperature profile; gas-fired furnace; off-period mass flow rate; part-load performance; seasonal efficiency.

ACKNOWLEDGMENTS

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NOMENCLATURE

A/F	mass ratio of stoichiometric air to fuel
A_o	orifice area, ft^2
A_S	stack cross-sectional area, ft^2
C_{air}	specific heat of air, Btu/lb-°F
C_o	loss coefficient
$C_{t,OFF}$	correction factor for the effect of cycling during cool-down period
$C_{t,ON}$	corrrection factor for the effect of cycling during heat-up period
D	diameter of the duct, ft
D_F	off-period flue-gas draft factor
D_o	stack damper effectiveness
D_S	off-cycle stack-gas draft factor
f	friction factor
g	gravitational acceleration, ft/sec^2
h	height of the furnace-flue system, ft
H_d	dynamic pressure loss, lbf/ft^2
H_f	friction head loss, lbf/ft^2
HHV	higher heating value, Btu/lb
H_t	total head loss, lbf/ft^2
z	vertical distance, ft
k	system friction factor
$L_{S,OFF}$	off-period sensible heat loss in percent
$\dot{m}_F$	flue-gas mass flow rate, lb/min
$\dot{m}_{S,OFF}$	mass flow rate of the stack gas during off-period, lb/min

$\dot{m}_{S,ON}$	mass flow rate of the stack gas during on-period, lb/min
P	pressure, lbf/ft^2
p'	pressure difference from ambient condition, lbf/ft^2
P_o	ambient pressure, lbf/ft^2
ΔP	pressure drop, lbf/ft^2
Q_F	volumetric flow rate, ft^3/min
Q_{IN}	fuel input rate, Btu/min
$Q_{S,OFF}$	sensible heat loss during the off-period, Btu
Q_T	total heat input, Btu
R_e	Reynolds number
$R_{T,F}$	ratio of actual combustion air to stoichiometric combustion air
t	time, min
t_{OFF}	duration of off-cycle, min
t_{ON}	duration of on-cycle, min
Δt	time interval, min
T_F	flue-gas temperature, °F (°R)
$T_{F,SS}$	flue-gas temperature at steady state, °F (°R)
T_{RA}	room air temperature, °F (°R)
$T_{S,SS}$	steady-state stack gas temperature, °F (°R)
v	average velocity, ft/sec
v_o	velocity at the orifice, ft/sec
θ_F	temperature difference for heat-up period, °F

$\theta_{F,A}$	actual temperature difference for cycling condition during heat-up period, °F
$\theta_{F,P}$	predicted temperature difference for cycling condition during heat-up period, °F
$\theta_{F,O,X}$	predicted temperature difference at the beginning of the heat-up period, °F
$\Delta\theta$	difference in θ, °F
μ_F	dynamic viscosity coefficient of the flue-gas, lb/ft-min
ρ'	density difference from ambient condition, lb/ft^3
ρ_F	flue-gas density, lb/ft^3
ρ_o	ambient density, lb/ft^3
ρ_s	stack-gas density, lb/ft^3
τ_{OFF}	cool-down time constant
τ_{ON}	heat-up time constant
ψ_F	temperature difference for cool-down period, °F
$\psi_{F,A}$	actual temperature difference for cycling condition during cool-down period, °F
$\psi_{F,P}$	predicted temperature difference for cycling condition during cool-down period, °F
$\psi_{F,O,X}$	predicted temperature difference at the beginning of cool-down period, °F
$\Delta\psi$	difference in ψ, °F

A STUDY OF THE DYNAMIC FLUE-GAS TEMPERATURE AND OFF-PERIOD MASS FLOW RATE OF A RESIDENTIAL GAS-FIRED FURNACE

by

Cheol Park, William J. Mulroy, and George E. Kelly

Introduction

In order to conserve fuel in heating, it is necessary for industry to manufacture more efficient furnaces and boilers. Test and calculation procedures must be developed to make this possible through the identification of high-efficiency systems. Test and calculation procedures for gas- and oil-fired furnaces and boilers have been recommended by Kelly et al [1]. The purpose of this study is to provide background information about the equations used by Kelly et al in their calculation procedure.

Flue-gas temperature profiles with respect to time for on-period heat-up and off-period cool-down of the furnace will be described. Detailed derivations of mass flow rate equations will be made. For cyclic patterns of heat-up and cool-down, correction factors $C_{t,ON}$ and $C_{t,OFF}$ will be derived.

These theoretically derived equations will be compared with corresponding experimentally measured values, using data from a gas-fired furnace tested at the National Bureau of Standards (NBS). In addition, experimental set-up and test procedures will be explained and measurements and a theoretical analysis of the stack damper effectiveness will be described.

Experimental Set-up

The up-flow forced warm air gas furnace employed in the experimental study had a 132,000 Btu/hr (38.68 kW) rated bonnet capacity and a 165,000 Btu/hr (48.35 kW) rated input capacity. This furnace had six burners with each burner having a separate heat exchange passage through the furnace. The gas valve switch of the furnace was operated manually instead of by a thermostat for these tests, and the blower was operated by a manual switch with override by the high temperature limit control. A 6-inch (15.24 cm) diameter, 5-foot (1.52 m) high test stack was attached to the furnace throughout the rest of the study, although in a few tests the stack height was increased to 10 feet (3.05 m). The upper portion of the furnace, the warm air supply duct, and the stack were covered by a layer of R7 fiberglass insulation faced with aluminum foil in order to reduce heat loss. The integral draft-diverter relief opening could be open or sealed depending upon the test condition. However, most tests were conducted with the integral draft-diverter relief opening sealed.

Warm air was exhausted into the room, where temperature was maintained at a constant level by a high capacity air-conditioner. Inlet air temperature was kept constant within ± 5°F (± 2.8°C). The overall set-up is shown in Figure 1. Thermocouples, installed at various locations, were used to make temperature measurements. A shielded 24-gage K-type thermocouple was inserted approximately one inch into the outlet of each of the six heat exchange passages. Nine thermocouples, wired in parallel, were installed on a horizontal test plane located one foot from the inlet of the test stack. Hereafter this plane will be referred to as the test plane. Figure 2 shows the locations of thermocouples in the test plane. Electromotive force created by each thermocouple due to temperature deviation from the reference point was channeled into a data acquisition system. There was also a thermocouple near the test plane, connected to a low speed strip-chart recorder, to continuously monitor the variation of temperature during the course of each test.

Volumetric ratio of carbon dioxide concentration was measured by a non-dispersive infrared analyzer [2]. A sample gas was taken through a copper tubing connected to the stack at the test plane. Sampling was also made in the heat exchanger outlets upsteam of the integral draft diverter. Figure 3 shows a schematic diagram of a flow control system for gas concentration measurement. Stack flow rate was determined by using the methane tracer gas method. Methane gas was introduced into each of six heat exchange passages through a supply line incorporating a rotameter and a six-branch feeding manifold. Methane concentration at the top of the furnace stack was measured by an infrared absorption analyzer. During some tests using the methane tracer gas method, electric heaters were installed in place of the gas burners. A vane anemometer was mounted 2 inches (5.1 cm) above the top of the stack for stack flow rate measurement during some cool-down tests. The anemometer had been calibrated under a geometrically similar flow condition. In carrying out this calibration, the air velocity, at the top of a circular duct with the same diameter and height as the test stack, was measured at the test room temperature. A high capacity dry gas meter was installed between the air supply line and this duct to measure the mass flow rate. The anemometer used here was 4 inches (10.2 cm) in diameter with an eight-vane rotor. Its effective cross-sectional area was 9.42 in^2 (60.8 cm^2).

For tests of the effectiveness of automatic stack dampers, manually operated damper plates having various size openings were installed in the stack approximately 18 inches above the inlet of the stack.

Experimental Procedures

Test procedures for steady-state, cool-down and warm-up followed closely the test procedures prepared by Kelly et al [1]. The furnace used natural gas as its fuel for all tests. Room air temperature was

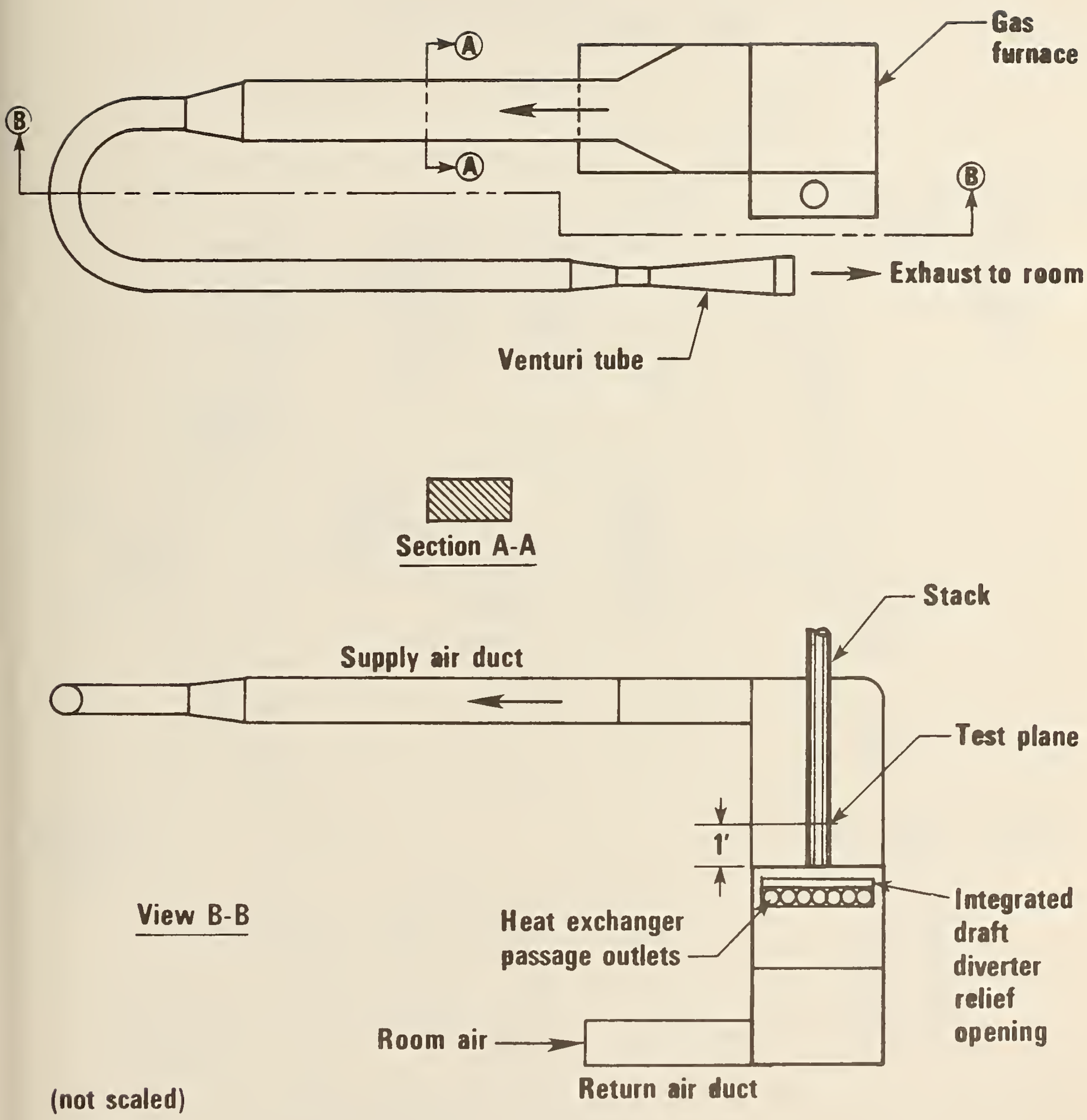


Figure 1. Set-up for a gas furnace testing

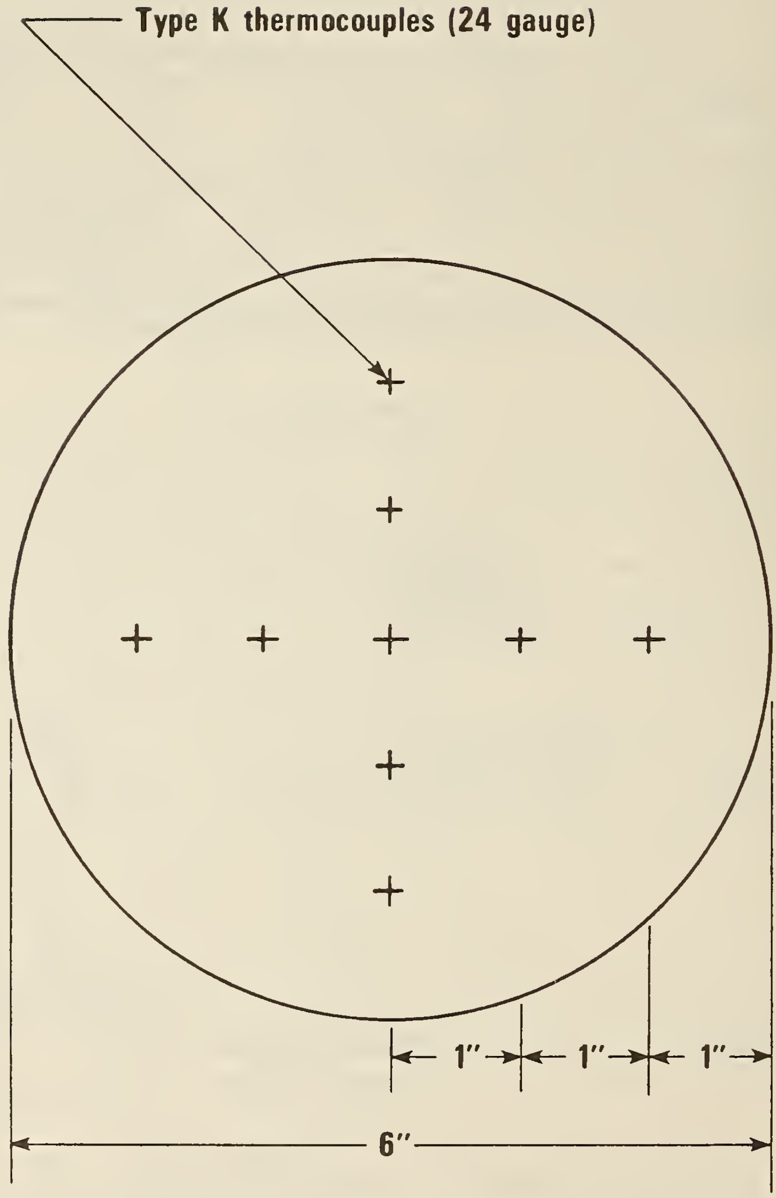


Figure 2. Thermocouples in the test plane

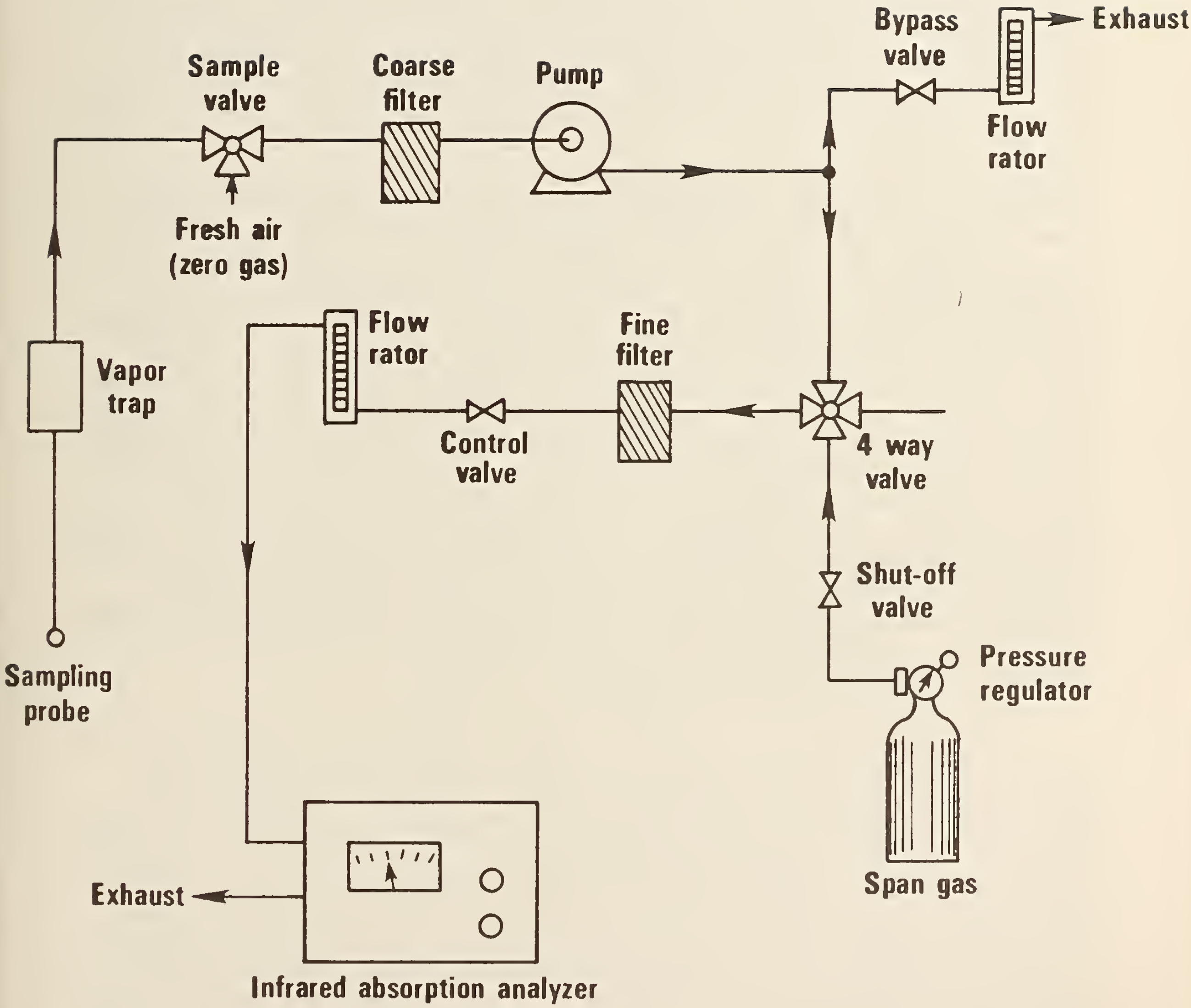


Figure 3. Flow control system for infrared absorption analyzer

maintained at 75°F (24°C) with less than 5°F (2.8°C) deviation. A room exhaust fan was operated to withdraw stack gas from the room. All instruments were warmed up before conducting each test. Flue-gas temperature changes were monitored by a thermocouple located near the test plane, and continuously recorded on a chart recorder during each test.

Steady-state testing was performed by operating the furnace with the burners and the blower turned on until stack temperature variation in three successive readings taken 15 minutes apart was not more than 3°F (1.7°C). The integral draft diverter was open during this test. When steady state had been reached, room temperature and steady-state gas temperature were measured by nine thermocouples located in the stack. The average value of nine thermocouple outputs was taken as stack-gas temperature. A sample of stack gas was pumped out at a rate of 3 ft^3 per hour (85 L/hr) through a sampling copper tube with its opening at the test plane. The sample gas was analyzed by the infrared analyzer to determine the concentration by volume of carbon dioxide present in dry stack gas. After taking the value of carbon dioxide concentration of stack gas, a sample of flue gas was obtained by moving a sampling probe around in the heat exchanger passage outlets into the draft diverter. The average value of carbon dioxide concentration in the flue gas was taken. The furnace was then temporarily turned off to seal the draft-diverter relief opening with a layer of R7 fiberglass insulation backed with aluminum foil. The unit was then turned on again and when the steady-state condition had been re-achieved, the stack outlet was progressively restricted with a piece of metal sheet until the carbon dioxide concentration in the flue gas measured at the test plane agreed within $\pm$ 0.2 percentage points of the previously determined average value measured at the heat exchanger passage outlets. The average temperature then occurring at the measurement plane was assumed to be the representative flue-gas temperature. During steady-state tests, temperature readings from each of the thermocouples in the heat exchanger passages were also recorded.

Cool-down testing was executed after steady-state testing was completed. As soon as the burner was turned off, temperatures at the test plane and at the heat exchanger were measured in various time intervals. Since the decay rate of the temperature during the early stages of cool-down period was great, the interval in taking data was set to 10 seconds. Near equilibrium, the decay rate was very small, and the data interval was set to 5 minutes. The blower was turned off 3 minutes after the burners shut off, as recommended by the test procedures in [1]. The burners remained off until equilibrium was attained, as indicated by variation in the flue-gas temperature of not more than 3°F (1.7°C) between three successive readings taken 15 minutes apart. It took about 55 minutes in a typical test condition for the unit to reach a state of equilibrium.

After an equilibrium condition was attained, a heat-up test was conducted. The furnace was turned on and flue-gas temperature was measured in a similar way as in cool-down testing. As recommended by the test procedures in [1], the blower was turned on 1.5 minutes after burners had been turned on and temperature measurement was continued until steady state was reached.

Tests were also performed under the condition when the pattern of temperature profile with respect to time is cyclic and the temperature of flue gas reaches neither equilibrium nor steady state. A typical test was made with the duration of the on-cycle chosen as 8 minutes and the duration of off-cycle as 2 minutes. The blower remained on during all of these tests, and temperature data were taken every 10 seconds until three complete cycles of warm-up and cool-down were achieved.

Measurements of the flow rate of stack gas through the insulated circular test stacks as a function of temperature were made by means of a vane anemometer and the tracer-gas method, with the integral draft diverter sealed. Prior to making a test for stack flow measurement, the vane anemometer was calibrated. Compressed air from a supply line at room temperature was passed through a dry gas meter before it was allowed to flow into a geometrically similar calibration duct. The anemometer was placed 2 inches (5.1 cm) above the end of the duct and centered. Air flow velocity was determined by reading the distance indicated on the dial of the vane anemometer for a given interval of time, and flow rate was computed based on air speed and the cross sectional area of the duct. Results were plotted against the calibrated flow rate indicated on the dry gas meter in a given time interval. In order to calculate the mass flow rate through the stack of the furnace, the air speed of stack gas was measured during a cool-down test period, with the vane anemometer installed on the furnace stack in the same manner as was done during its calibration procedures. Flue-gas temperatures both at the test plane and in the heat exchange passage outlets were obtained every 30 seconds from steady state to equilibrium.

In addition to the vane anemometer, the tracer-gas method was also used in flow rate measurement for some tests of air flow through the furnace as a function of the flue-gas temperature. In these tests, the draft diverter was blocked and electric heaters were installed in the furnace in place of the gas burners. The furnace was then operated at different steady-state flue-gas temperatures by varying the voltage to these electric resistance heaters. Methane gas was released into each heat exchanger passage at a predetermined constant flow rate, while a flue-gas sample taken at the top of the flue was continuously monitored on an infrared methane gas analyzer. The flow rate through the furnace was then calculated from methane flow rate and its dilution.

Experiments to determine the effectiveness of automatic stack dampers at blocking flow during the off-period were conducted using manually operated dampers installed in the furnace test stack. The draft diverter was blocked and the furnace was operated at a full-load condition for approximately one hour with its damper open. The furnace was then shut off, the damper closed, and a constant methane flow rate of 5.3 ft^3 per hour (150 L/hr) established. The furnace was allowed to cool naturally and flue-gas temperature and stack methane concentration were measured continuously. The cool-down period usually lasted for one hour. After each test the area of the damper plate was reduced by drilling holes one inch or less in diameter in the damper plate and the steady-state and cool-down tests were repeated. Tests were also made with a damper with a single large hole in it, in order to compare its performance with the damper having a number of small holes. In these comparison tests, a 10-foot insulated test stack was employed instead of the 5-foot stack. Although blocking the draft-diverter relief opening of the furnace results in the stack damper being tested as a flue damper, the effectiveness of the damper was determined at relatively low flue-gas temperature and thus the results should satisfactorily describe the performance of stack dampers.

Temperature Profiles

By examining experimental data, it was possible to develop equations to predict the flue-gas temperature during the heat-up and cool-down periods [1,3]. When the unit is allowed to cool down from steady state to equilibrium, the temperature of flue gas in the furnace decays exponentially. Temperature profile with respect to time may be expressed as:

$$\psi_F(t) = \psi_{F,O,X}\, e^{-\frac{t}{\tau_{OFF}}}, \qquad (1)$$

where $\psi_F(t) \equiv T_F(t) - T_F(\infty)$, $T_F(\infty)$ denotes flue-gas temperature at equilibrium, and $\psi_{F,O,X}$ and τ_{OFF} are constants to be determined from known conditions.

A similar expression can be written for flue-gas temperature during the heat-up period.

$$\theta_F(t) = \theta_{F,O,X}\, e^{-\frac{t}{\tau_{ON}}}, \qquad (2)$$

where $\theta_F(t) \equiv T_{F,SS} - T_F(t)$, $T_{F,SS}$ denotes flue-gas temperature at steady state, and $\theta_{F,O,X}$ and τ_{ON} are constants.

If the flue-gas temperatures at time t_1 and t_2 are experimentally known, the time constant during the heat-up period, τ_{ON} can be obtained as follows:

$$\theta_F(t_1) = T_{F,SS} - T_F(t_1) = \theta_{F,O,X}\, e^{-\frac{t_1}{\tau_{ON}}} \qquad \text{at } t = t_1,$$

and

$$\theta_F(t_2) = T_{F,SS} - T_F(t_2) = \theta_{F,O,X}\, e^{-\frac{t_2}{\tau_{ON}}} \qquad \text{at } t = t_2.$$

Solving both equations simultaneously gives:

$$\tau_{ON} = \frac{t_2 - t_1}{\ell n \left[\dfrac{T_{F,SS} - T_F(t_1)}{T_{F,SS} - T_F(t_2)}\right]} \quad . \tag{3}$$

Similarly if flue-gas temperature during the cool-down period is measured at times t_3 and t_4 after the furnace is turned off, τ_{OFF} can be obtained using

$$\psi_F(t_3) = \psi_{F,O,X}\, e^{-\frac{t_3}{\tau_{OFF}}} \qquad \text{at } t = t_3,$$

and

$$\psi_F(t_4) = \psi_{F,O,X}\, e^{-\frac{t_4}{\tau_{OFF}}} \qquad \text{at } t = t_4.$$

These two equations, when solved simultaneously, yield

$$\tau_{OFF} = \frac{t_4 - t_3}{\ell n \left[\dfrac{T_F(t_3) - T_F(\infty)}{T_F(t_4) - T_F(\infty)}\right]} \quad . \tag{4}$$

The initial constant values, $\theta_{F,0,X}$ and $\psi_{F,0,X}$, may then be defined by using the measured flue-gas temperatures at t_1 and t_3, respectively, to obtain

$$\theta_{F,0,X} \equiv \theta_F(t_1)\, e^{\frac{t_1}{\tau_{ON}}} = [T_{F,SS} - T_F(t_1)]\, e^{\frac{t_1}{\tau_{ON}}} , \tag{5}$$

and

$$\psi_{F,0,X} \equiv \psi_F(t_3)\, e^{\frac{t_3}{\tau_{OFF}}} = [T_F(t_3) - T_F(\infty)]\, e^{\frac{t_3}{\tau_{OFF}}} . \tag{6}$$

Figures 4 and 5 show typical experimental results obtained for the flue-gas temperature as the furnace cooled down from steady-state operation and heated up from equilibrium conditions. Data were taken at the test plane as well as at the heat exchanger passage outlets. The predicted flue-gas temperature cool-down and heat-up profiles are shown by dotted lines. The following values of t_1, t_2, t_3, and t_4, which were recommended by Kelly et al in [1],were used.

t_1 = 0.5 min $\qquad$ t_2 = 2.5 min

t_3 = 1.5 min $\qquad$ t_4 = 9.0 min.

A good agreement between measured values and predicted values is found except near the beginning of the time period.

When the cool-down and heat-up periods are finite, the pattern of the flue-gas temperature changes with respect to time. Due to the finite length of time for on- and off-cycle periods, the initial values of the exponential function, $\theta_{F,0,X}$ and $\psi_{F,0,X}$, become smaller due to the fact that the unit may never heat up to steady state or cool down to equilibrium. It is therefore necessary to introduce correction factors such that for the heat-up period,

$$\theta_F(t) = C_{t,ON}\theta_{F,0,X}\, e^{-\frac{t}{\tau_{ON}}} , \tag{7}$$

and for the cool-down period,

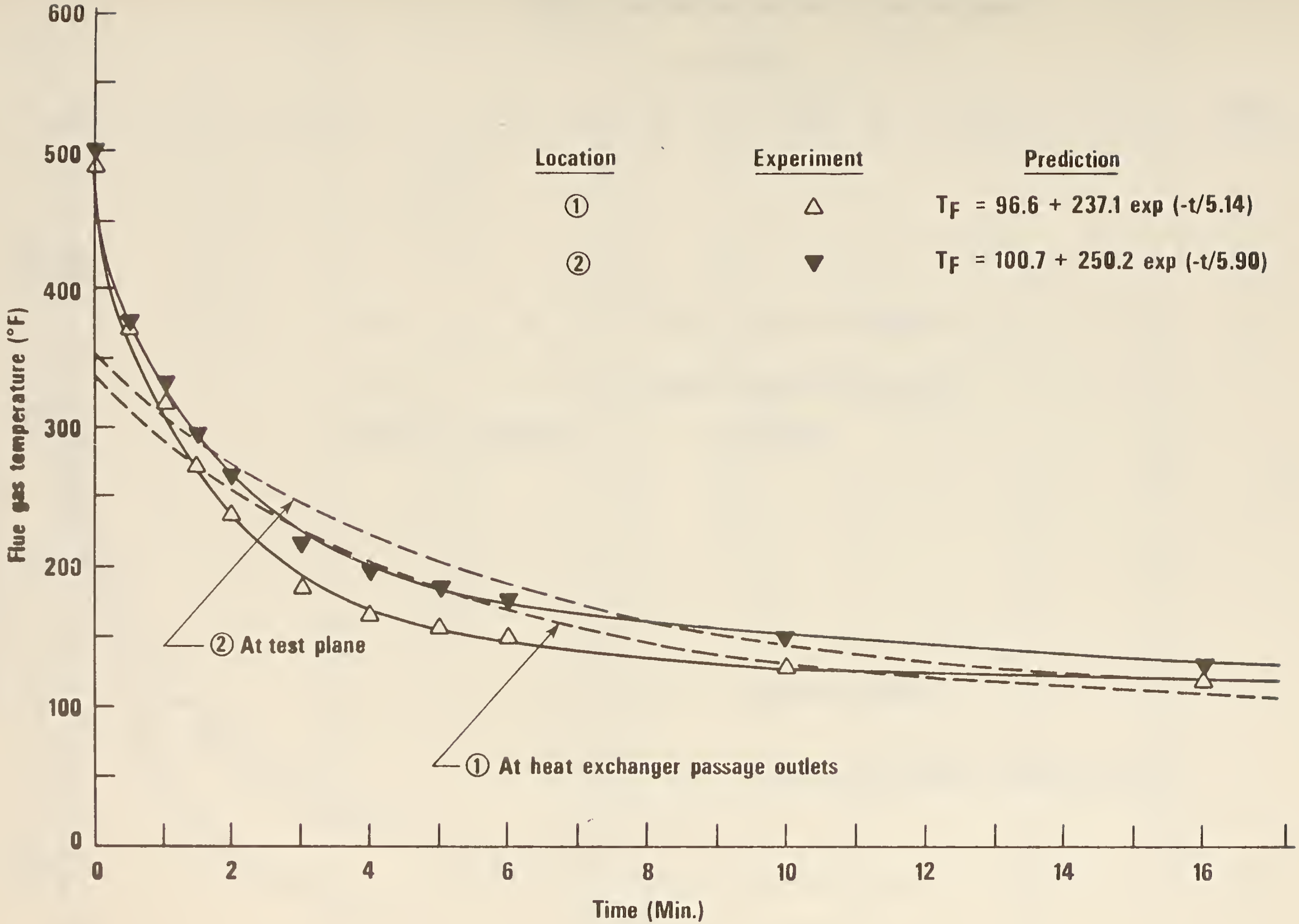


Figure 4. Flue gas temperature at cool-down test

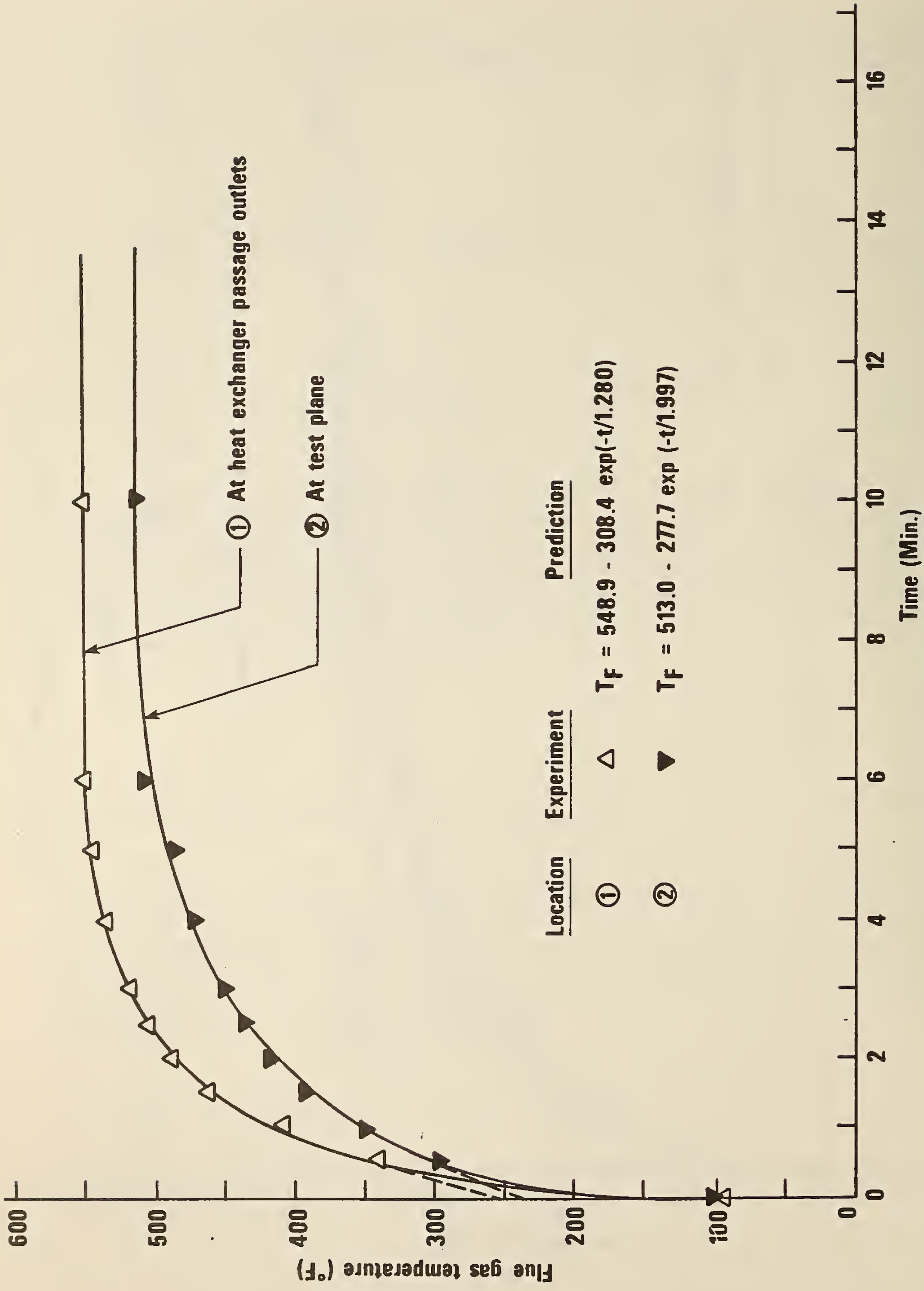


Figure 5. Fluegas temperature at heat-up testing

$$\psi_F(t) = C_{t,OFF}\psi_{F,O,X}\, e^{-\frac{t}{\tau_{OFF}}} . \qquad (8)$$

The quantities $C_{t,ON}$ and $C_{t,OFF}$, which are correction factors for the on- and off-periods respectively, can be determined as follows.

The main assumptions are:

$$\frac{\theta_{F,P}}{\theta_{F,O,X}} = \frac{\theta_{F,A}}{T_{F,SS} - T_F(\infty)}$$ at the start of the heat-up period,

and

$$\frac{\psi_{F,P}}{\psi_{F,O,X}} = \frac{\psi_{F,A}}{T_{F,SS} - T_F(\infty)}$$ at the start of the cool-down period,

where the subscripts P and A refer to the predicted and actual value of $\theta_F(t)$ and $\Psi_F(t)$ under cycling conditions.

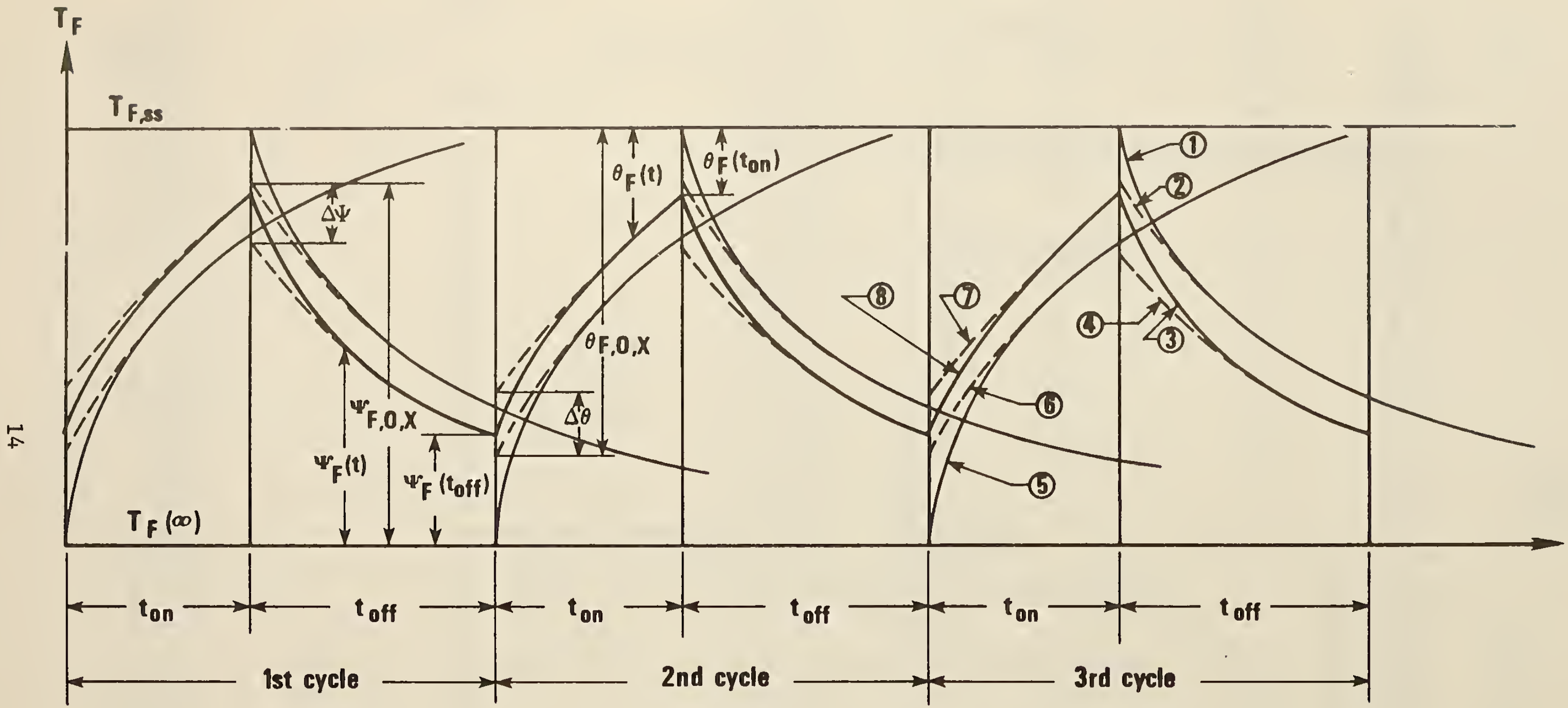
Referring to Figure 6 and defining $\Delta\theta$ to be the difference between the predicted value of θ_F at the beginning of heat-up from cycling condition and the predicted value of θ_F at the beginning of heat-up from equilibrium condition $(\theta_{F,O,X})$, the assumed relation may be expressed as:

$$\frac{\theta_{F,O,X} - \Delta\theta}{\theta_{F,O,X}} = \frac{T_{F,SS} - T_F(\infty) - \psi_F(t_{OFF})}{T_{F,SS} - T_F(\infty)}, \qquad (9)$$

where t_{OFF} is the duration of the off-cycle. The difference $\Delta\theta$ may thus be written as:

$$\Delta\theta = \frac{\psi_F(t_{OFF})\,\theta_{F,O,X}}{T_{F,SS} - T_F(\infty)} . \qquad (10)$$

Similarly, defining $\Delta\psi$ as the difference between the predicted value of ψ_F at the beginning of cool-down from cycling condition and the predicted value of ψ_F at the beginnning of cool-down from steady-state condition $(\psi_{F,O,X})$ results in



Cool-down period

① Actual from steady state, monotonic
② Predicted from steady state, monotonic
③ Actual cyclic
④ Predicted cyclic

Heat-up period

⑤ Actual from cold state, monotonic
⑥ Predicted from cold state, monotonic
⑦ Actual cyclic
⑧ Predicted cyclic

Figure 6. A relation between monotonic and cyclic patterns

$$\Delta\theta = \frac{\theta_F(t_{ON})\ \psi_{F,O,X}}{T_{F,SS} - T_F(\infty)} , \tag{11}$$

where t_{ON} is the duration of the on-cycle. A sketch of the heat-up and cool-down curves from steady state and equilibrium conditions as well as under cycling condition is shown in Figure 6.

Boundary conditions for the equations predicting the heat-up and cool-down flue-gas temperature profiles under cyclic condition can be imposed such that the predicted temperature at the end of each on- and off-period approximately coincides with the actual values. The imposed condition causes a discontinuity in the temperature at the beginning of each on- and off-period which violates the physical intuition, but gives results which are in good agreement with measured data, except near the beginning of each period. These boundary conditions are:

$$\theta_F(t_{ON}) = [\theta_{F,O,X} - \Delta\theta]\ e^{-\frac{t_{ON}}{\tau_{ON}}} , \tag{12}$$

and

$$\psi_F(t_{OFF}) = [\psi_{F,O,X} - \Delta\psi]\ e^{-\frac{t_{OFF}}{\tau_{OFF}}} . \tag{13}$$

Substitution of Eqs. (10) and (11) into Eqs. (12) and (13), respectively, yields:

$$\theta_F(t_{ON}) + \frac{\theta_{F,O,X}}{T_{F,SS} - T_F(\infty)}\ e^{-\frac{t_{ON}}{\tau_{ON}}}\ \psi_F(t_{OFF}) = \theta_{F,O,X}\ e^{-\frac{t_{ON}}{\tau_{ON}}} , \tag{14}$$

and

$$\frac{\psi_{F,O,X}}{T_{F,SS} - T_F(\infty)}\ e^{-\frac{t_{OFF}}{\tau_{OFF}}}\ \theta_F(t_{ON}) + \psi_F(t_{OFF}) = \psi_{F,O,X}\ e^{-\frac{t_{OFF}}{\tau_{OFF}}} . \tag{15}$$

In order to find the unknown variables $\theta_F(t_{ON})$ and $\psi_F(t_{OFF})$, the system of equations consisting of Eqs. (14) and (15) must be solved simultaneously. This results in the following equations, as introduced by Chi and Kelly [3]:

$$\theta_F(t) = \theta_{F,O}\, e^{-\frac{t}{\tau_{ON}}}, \tag{16}$$

and

$$\psi_F(t) = \psi_{F,O}\, e^{-\frac{t}{\tau_{OFF}}}, \tag{17}$$

where $\theta_{F,O} = C_{t,ON}\, \theta_{F,O,X}$, (18a)

and $\psi_{F,O} = C_{t,OFF}\, \psi_{F,O,X}$. (18b)

The correction factors are expressed as

$$C_{t,ON} = \frac{1 - \dfrac{\psi_{F,O,X}\, e^{-\frac{t_{OFF}}{\tau_{OFF}}}}{T_{F,SS} - T_F(\infty)}}{1 - \dfrac{\theta_{F,O,X}\psi_{F,O,X}}{[T_{F,SS} - T_F(\infty)]^2}\, e^{-\left(\frac{t_{ON}}{\tau_{ON}} + \frac{t_{OFF}}{\tau_{OFF}}\right)}}, \tag{19a}$$

and

$$C_{t,OFF} = \frac{1 - \dfrac{\theta_{F,O,X}\, e^{-\frac{t_{ON}}{\tau_{ON}}}}{T_{F,SS} - T_F(\infty)}}{1 - \dfrac{\theta_{F,O,X}\, \psi_{F,O,X}}{[T_{F,SS} - T_F(\infty)]^2}\, e^{-\left(\frac{t_{ON}}{\tau_{ON}} + \frac{t_{OFF}}{\tau_{OFF}}\right)}}. \tag{19b}$$

Figure 7 shows a set of curves obtained by plotting the experimental flue-gas temperature profile measured at the test plane previously described and the predicted flue-gas temperature profile given by

$$T_F(t) = T_{F,SS} - C_{t,ON}\theta_{F,O,X}\, e^{-\frac{t}{\tau_{ON}}} \qquad \text{for on-cycle,}$$

and

$$T_F(t) = T_F(\infty) + C_{t,OFF}\, \theta_{F,O,X}\, e^{-\frac{t}{\tau_{OFF}}} \qquad \text{for off-cycle,}$$

with the values of $C_{t,ON}$ and $C_{t,OFF}$ obtained from equations (19a) and (19b), respectively.

A curve of the average flue-gas temperature measured at the six heat exchanger passage outlets is shown in Figure 8, along with the predicted flue-gas temperature profile. The results presented in both figures show that the predicted flue-gas temperature profiles at both locations closely approximate the measured heat-up and cool-down profile although there exist discontinuities at the beginning of the heat-up and cool-down periods.*

Mass Flow Rate

The mass flow rate through a furnace or a boiler during the off-period can be obtained from a consideration of hydrostatic pressure difference between flue gas and ambient air and the equations describing turbulent flow through a circular pipe having a finite length.

The differential equation governing the pressure inside the furnace may be expressed as

$$\frac{dp'(z)}{dz} + \rho'(z)g = 0, \tag{20}$$

where z is the vertical distance inside the furnace above the point where combustion takes place,

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* It would be possible to construct a set of equations having no discontinuities, but this would require a considerably more complex mathematical treatment.

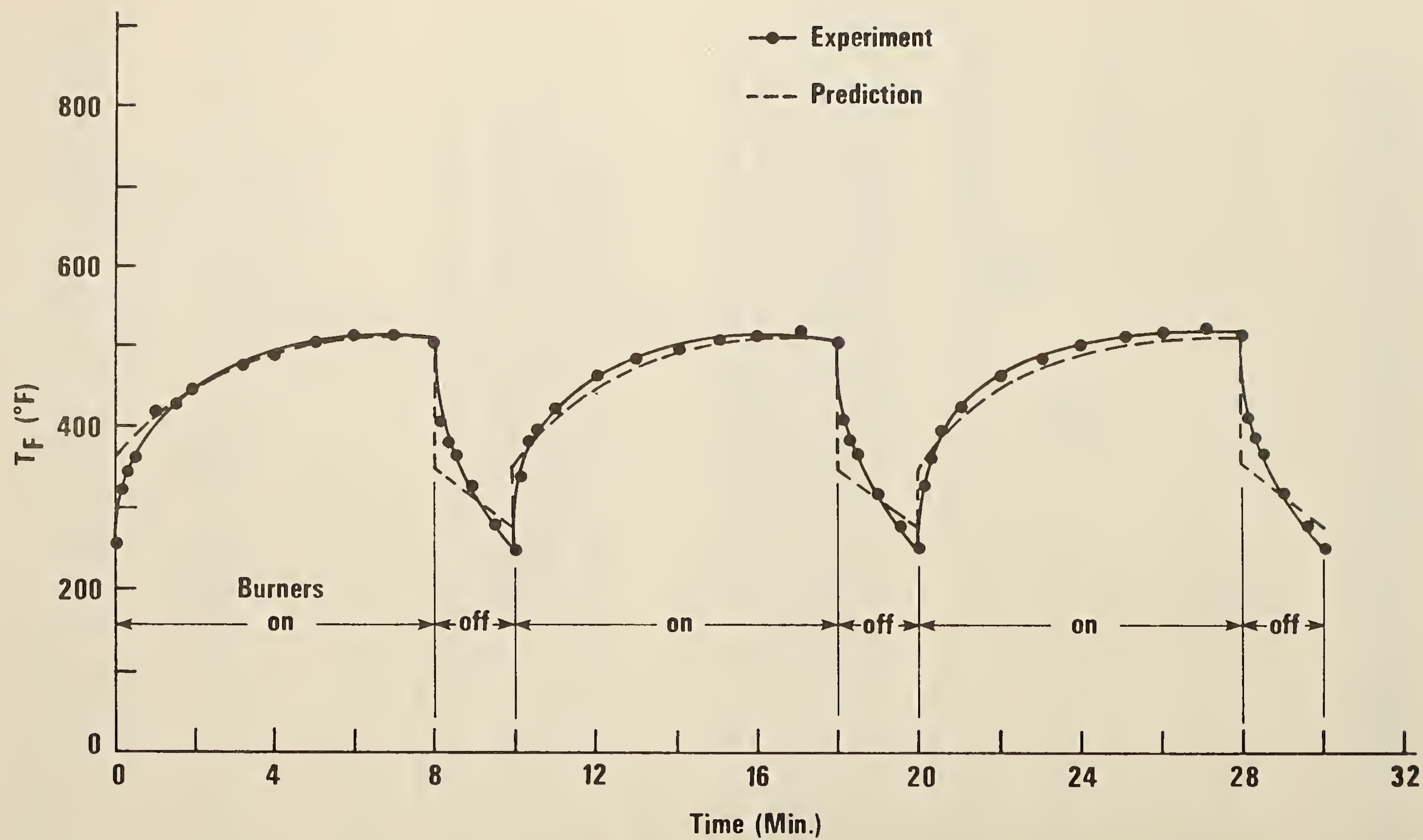


Figure 7. Cyclic pattern of flue gas temperature at the test plane

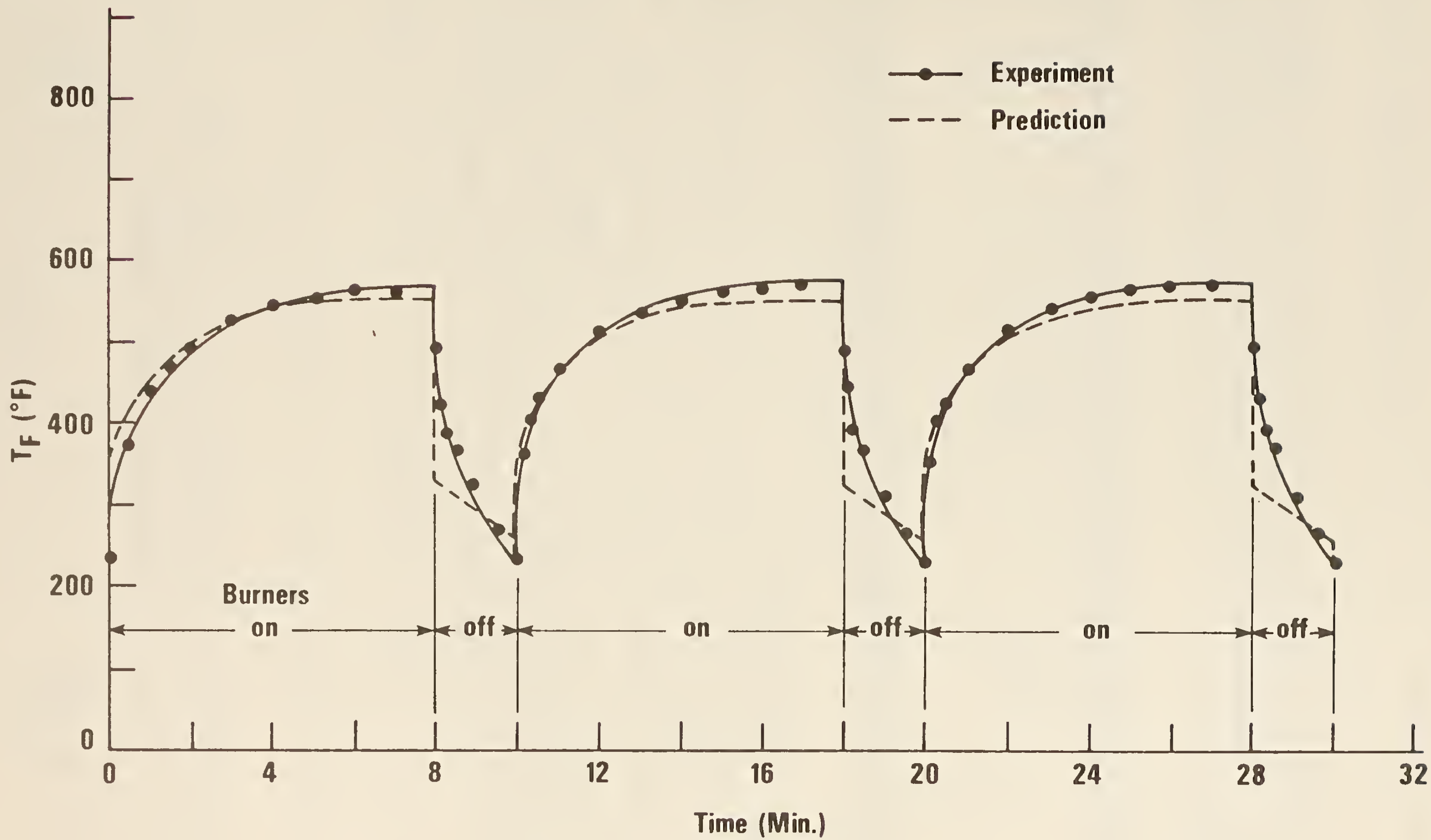


Figure 8. Cyclic pattern of flue gas temperature at the heat exchanger passage outlets

$$p'(z) = p(z) - p_o,$$

and $\rho'(z) = \rho(z) - \rho_o$.

The quantities $p(z)$ and $\rho(z)$ are the pressure and density, respectively, inside the stack at height z; p_o and ρ_o are the pressure and density, respectively, outside the stack and may be assumed independent of z for the purpose of this calculation; and g is the gravitational acceleration.

Making use of the ideal gas law results in the following relationship between ρ' and the flue-gas temperature, $T_F(z)$ in absolute units:

$$\rho'(z) = \rho_o\left(\frac{\rho(z)}{\rho_o} - 1\right) = \rho_o\left(\frac{T_{RA}}{T_F(z)} - 1\right) , \qquad (21)$$

where T_{RA} is the room air temperature.

Inserting this expressing into Eq. (20) and assuming that the variation of the flue-gas temperature with z can be ignored yields:

$$\Delta p = \rho_o g h\left(\frac{T_F - T_{RA}}{T_F}\right) , \qquad (22)$$

where h is height of the furnace-flue system, and Δp is the pressure drop between the inlet and outlet of the furnace-flue system.

The pressure drop between the inlet and outlet of the furnace-flue system should be balanced by the friction forces created by the turbulent flow on the surface of the circular duct. If the inner diameter of the duct is D and the average velocity of steady flow is v, the force balance equation is given by Bird et al [4]:

$$(\pi D h)\left(\frac{1}{2}\rho_F \cdot \frac{v^2}{g}\right) f = \Delta p\left(\frac{\pi D^2}{4}\right) , \qquad (23)$$

where f is a friction factor.

Using the mass flow rate expression: $\dot{m}_F = \rho_F v\left(\frac{\pi D^2}{4}\right)$ and solving for the pressure drop, Δp, Eq. (23) becomes

$$\Delta p = \frac{32\ h f \dot{m}_F^2}{\pi^2 D^5 \rho_F g} . \qquad (24)$$

The empirical equation for the friction factor in turbulent pipe flow is given by a simple expression [5]:

$$f = \frac{0.046}{R_e^{0.2}} \tag{25}$$

in the range of $30,000 < R_e < 10^6$.*

The Reynolds number, R_e, is based upon flue-gas density, average gas velocity, diameter of the duct and the dynamic viscosity coefficient of the flue gas, μ_F, and is given by

$$R_e = \frac{\rho_F v D}{\mu_F} \quad . \tag{26}$$

The dynamic viscosity of air varies with temperature and is determined from the table provided by Ekert and Drake [6] as being approximately proportional to the 0.75th power of T_F.

Setting Eq. (22) equal to Eq. (24) and substituting the above relationship for f and μ_F given by Eqs. (25) and (26), a final expression for the mass flow rate of flue gas through the furnace is obtained.

$$\dot{m}_F = K \cdot \frac{(T_F - T_{RA})^{0.56}}{T_F^{1.19}} \, , \tag{27}$$

where K is a constant.

Under steady-state condition this becomes:

$$\dot{m}_{F,SS} = K \cdot \frac{(T_{F,SS} - T_{RA})^{0.56}}{T_{F,SS}^{1.19}} \quad . \tag{28}$$

---

* Note that the well-known Blasius formula is

$$f = \frac{0.0791}{R_e^{0.25}}$$

in the range of $2.1 \times 10^3 < R_e < 10^5$.

Combining Eqs. (27) and (28) results in the following equation:

$$\frac{\dot{m}_F}{\dot{m}_{F,SS}} = \left(\frac{T_F - T_{RA}}{T_{F,SS} - T_{RA}}\right)^{0.56} \left(\frac{T_{F,SS}}{T_F}\right)^{1.19} . \tag{29}$$

Using Eq. (29), the ratio of mass flow rate during off-cycle to the mass flow rate during on-cycle is easily obtained.

$$\frac{\dot{m}_{F,OFF}}{\dot{m}_{F,ON}} = D_F \left(\frac{T_{F,OFF} - T_{RA}}{T_{F,SS} - T_{RA}}\right)^{0.56} \left(\frac{T_{F,SS}}{T_{F,OFF}}\right)^{1.19} , \tag{30}$$

where D_F is an off-period flue-gas draft factor, which has been introduced to account for the effect of power burners or flue dampers. It should be noted that temperature has absolute units of °R (°K). This expression was introduced by Chi and Kelly [3] with a brief explanation.

Under full-load condition, $\dot{m}_{F,ON}$ is the sum of the fuel flow rate and the combustion air flow rate:

$$\dot{m}_{F,ON} = \frac{Q_{IN}}{HHV} [1 + R_{T,F}(A/F)] , \tag{31}$$

where Q_{IN} is fuel input rate, HHV the higher heating value of the fuel, $R_{T,F}$ the ratio of actual combustion air to stoichiometric combustion air, and A/F the stoichiometric air-to-fuel ratio. The quantity $R_{T,F}$ is determined from the carbon dioxide concentration under steady-state operation with a particular fuel. Eq. (31) is also assumed by Chi and Kelly [3] to adequately describe the flue-gas mass flow rate during the heat-up period, since for most furnaces and boilers T_F rises quite rapidly and variation in $\dot{m}_F$ can usually be ignored.

It is also possible to derive another form of Eq. (29) for the mass flow rate using an assumption that the flue-gas density is the same as that of air at the same temperature. The following formula for calculating the flow rate induced in the stack is provided in the ASHRAE Equipment Handbook [7]:

$$Q_F = K_1 D^2 \sqrt{\frac{T_F \Delta p}{k\ p_o}} \quad , \tag{32}$$

where Q_F is the volumetric flow rate, k is a system friction factor, and K_1 is a constant. Using the quantity K with a subscript to indicate constants having different values in the equations presented below, and employing the fact that the theoretical draft is due to pressure differences Δp gives [7]:

$$\Delta p = K_2\ p_o\ h \left(\frac{1}{T_{RA}} - \frac{1}{T_F} \right), \quad \text{and} \tag{33}$$

$$Q_F = K_3 D^2 \sqrt{\frac{h(T_F - T_{RA})}{k\ T_{RA}}} \quad . \tag{34}$$

From Eq. (21) and Eq. (34), the mass flow rate expression may be written as

$$\dot{m}_F = \rho_F Q_F = \left(\rho_o \frac{T_{RA}}{T_F} \right) K_3 D^2 \sqrt{\frac{h(T_F - T_{RA})}{k\ T_{RA}}} \quad .$$

The mass flow rate at a flue-gas temperature T_F divided by the mass flow rate at the steady-state temperature, $T_{F,SS}$, is then given by

$$\frac{\dot{m}_F}{\dot{m}_{F,SS}} = \left(\frac{T_F - T_{RA}}{T_{F,SS} - T_{RA}} \right)^{0.5} \left(\frac{T_{F,SS}}{T_F} \right) \quad . \tag{35}$$

It is interesting to see that Eq. (29) and Eq. (35) are almost identical in form except for different values of power. However, the difference has a negligible effect on the ratio $\dot{m}_F/\dot{m}_{F,SS}$, as shown in Figure 9.

Figure 9 shows actual data points measured by a vane anemometer and by the tracer-gas method for mass flow rate through a furnace. It also shows two theoretical curves given by Eq. (29) and Eq. (35). In calculating the flow rate using the vane anemometer measurements, the air speed of the flue gas at flue temperature T_F was corrected for the fact that the vane anemometer was calibrated at temperature T_o by means of the following formula [8]:

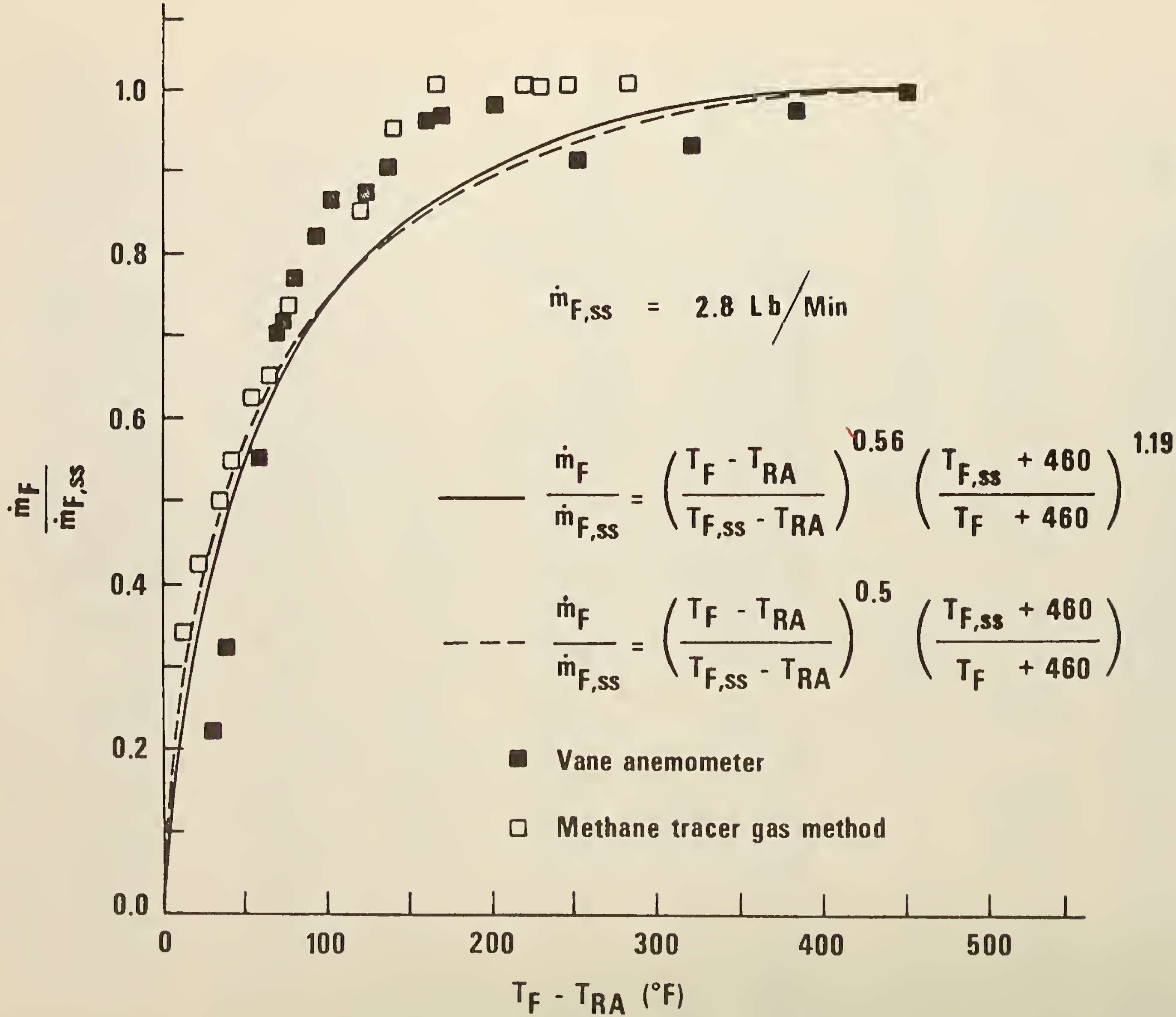


Figure 9. Mass flow rate of flue gas through the stack of the furnace

$$\frac{v_1}{v_o} = \sqrt{\frac{\rho_o}{\rho_1}} = \sqrt{\frac{T_1}{T_o}} \quad . \tag{36}$$

Figure 9 indicates that both Eq. (29) and Eq. (35) adequately describe the mass flow rate through the particular furnace tested. The slight deviation of the experimental data from the theoretical curves for $T_F - T_{RA}$ in the mid-range temperature could reflect the fact that it is not rigorously correct to make the assumption that T_F is independent of z.

Damper Effectiveness

During the cool-down period, heat loss from a furnace or boiler occurs mainly due to the flow of air through the furnace draft control device and up the stack. For units which use indoor air for combustion, the off-period sensible heat loss is given, in percent, by the following relation:

$$L_{S,OFF} = \frac{Q_{S,OFF}}{Q_T} \times 100 ,$$

where $Q_{S,OFF}$ is the sensible heat loss during the off-period and Q_T is the total heat input equal to the product of the fuel input rate Q_{IN} and the time during which the burners are turned on, t_{ON}. The sensible heat loss during the off-period in a certain time interval, Δt, may be expressed as

$$\Delta Q_{S,OFF} = C_{air} \, \dot{m}_{S,OFF} \Delta t \quad \Delta T \quad , \tag{37}$$

where C_{air} is the specific heat of air, $\dot{m}_{S,OFF}$ the mass flow rate of the stack gas, and $\Delta T \equiv T_{S,OFF} - T_{RA}$ is the temperature difference between the stack gas and room air. The mass flow rate through the stack can be shown to be given by the equation:

$$\dot{m}_{S,OFF} = D_S \dot{m}_{S,ON} \left(\frac{T_{S,OFF} - T_{RA}}{T_{S,SS} - T_{RA}} \right)^{0.56} \left(\frac{T_{S,SS}}{T_{S,OFF}} \right)^{1.19} , \tag{38}$$

by carrying out an analysis similar to the one used to derive Eq. (30).

In the above equation, D_S is the off-cycle stack gas draft factor which accounts for the effect of a stack damper, $T_{S,SS}$ the steady-state stack-gas temperature, and $\dot{m}_{S,ON}$ the mass flow rate through the stack during the on-period. $\dot{m}_{S,ON}$ is the product of $\dot{m}_{F,ON}$ and the ratio of the stack-gas mass flow rate to the flue-gas mass flow rate, S/F.

During the cool-down period the total sensible heat lost up the stack (ignoring the infiltration loss) may be obtained by integrating Eq. (37) with respect to time to obtain

$$Q_{S,OFF} \cong C_{air} D_S \dot{m}_{F,ON} (S/F) \int_0^{t_{OFF}} \left(\frac{T_{S,OFF} - T_{RA}}{T_{S,SS} - T_{RA}} \right)^{0.56} \left(\frac{T_{S,SS}}{T_{S,OFF}} \right)^{1.19} (T_{S,OFF} - T_{RA}) dt, \tag{39}$$

where t_{OFF} is the length of the off-period. The factor D_S depends upon the type of system under test. For furnaces and boilers with stack dampers, and draft diverters on draft heads, D_S is the same as the stack damper effectiveness, D_O. (Refer to Table 1 in [1].)

In order to determine the value of stack damper effectiveness factor, D_o, an equation is first theoretically derived, then compared with experimental data.

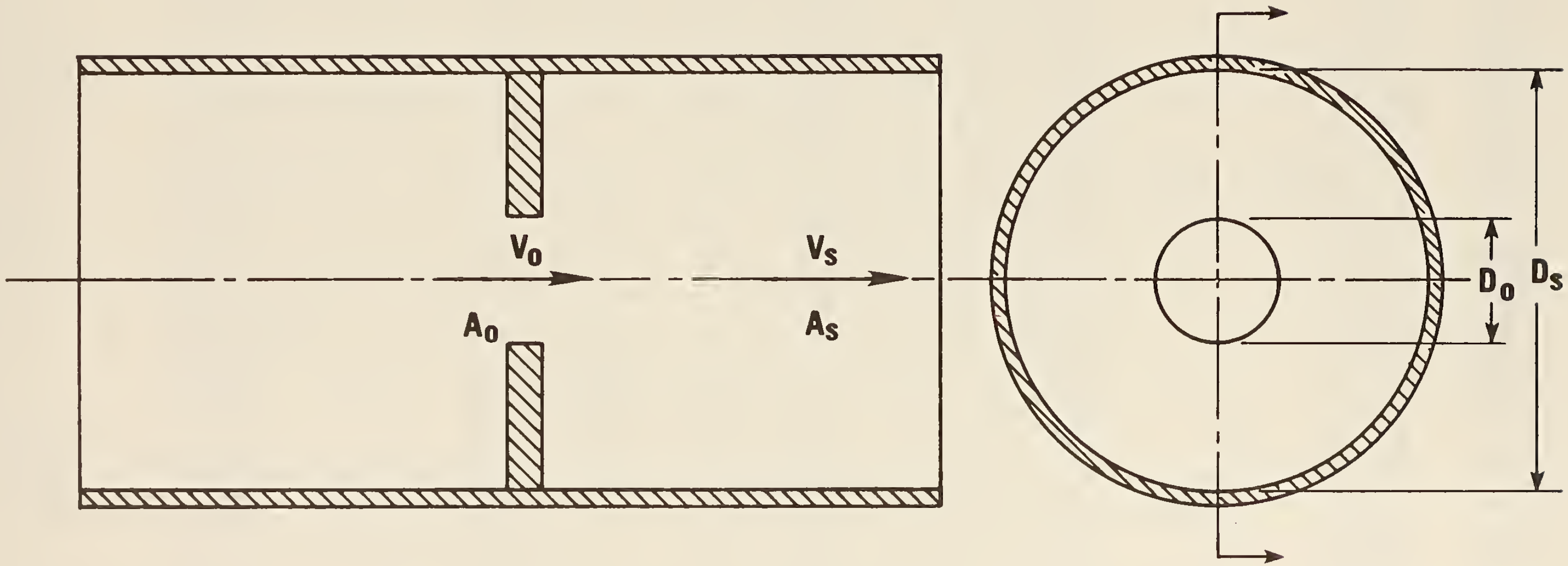
For a system employing a stack damper, the gas flow through the damper may be considered similar to the flow through a square-edged orifice in a duct (Figure 10). Due to contraction of area of the duct in this type of flow, there exists considerable dynamic pressure loss. This dynamic pressure loss, H_d, can be expressed in terms of the stack-gas density, ρ_s, and the gas velocity at the orifice, v_o, using

$$H_d = C_o \frac{1}{2} \rho_s v_o^2 , \tag{40}$$

where C_o is a loss coefficient.

By means of the equation for mass continuity it is possible to rewrite this equation as

$$H_d = C_o \left(\frac{A_s}{A_o} \right)^2 \frac{1}{2} \rho_s v_s^2 , \tag{41}$$



$$A_s = \frac{\pi D_s^2}{4}, \quad A_0 = \frac{\pi D_0^2}{4}$$

Figure 10. A square edged orifice in a circular duct

where v_s is the velocity in the large cross-sectional area of the stack, A_s is the stack cross-sectional area, and A_o the orifice area.

For the flow through a square-edged orifice, the coefficient C_o can be obtained from the values tabulated in the ASHRAE Handbook of Fundamentals [9] and expressed approximately (within an error of ± 5%) by

$$C_o = 2.6 \left(1 - \frac{A_o}{A_s}\right)^{1.58} . \tag{42}$$

On the other hand, the friction loss through the stack may be written by

$$H_f = k \frac{1}{2} \rho_s v_s^2 , \tag{43}$$

where the friction factor, k, is a dimensionless constant containing a scale factor for the size of the stack. The total loss of pressure head, H_t, through a stack containing a square-edged orifice is then the sum of the dynamic pressure loss and friction loss and is given by

$$H_t = H_d + H_f = C_o \left(\frac{A_s}{A_o}\right)^2 + k \frac{1}{2} \rho_s v_s^2 . \tag{44}$$

For a stack without the orifice, total loss is just equal of the friction loss and may be written as

$$H_t = H_f = k \frac{1}{2} \rho_s v_s^2 . \tag{45}$$

Defining the damper effectiveness, D_o, as the ratio of the flow rate with the damper, to that of a flow rate without the damper at the same pressure differential, D_o is obtained by dividing the square root of Eq. (45) by the square root of Eq. (44) as

$$D_o = \sqrt{\frac{k}{k + C_o \left(\frac{A_s}{A_o}\right)^2}} . \tag{46}$$

After substituting Eq. (42) into Eq. (46) and expressing the result in terms of the damper area, A_D, the equation for the effectiveness of the stack damper at off-period stack flow becomes

$$D_o = \sqrt{\frac{k}{k + \dfrac{2.6\left(\dfrac{A_D}{A_S}\right)^{1.58}}{1 - \left(\dfrac{A_D}{A_S}\right)^2}}} \quad . \tag{47}$$

Figure 11 contains experimental results obtained using the tracer-gas method of flow measurement during cool-down from steady state to equilibrium condition. Data were obtained for 5-foot-high and 10-foot-high test stacks as described in the section entitled Experimental Procedures. The values of D_o as given in Eq. (47) are also plotted against A_D/A_S for k = 5 and k = 10. A reasonable agreement is achieved between experimental and theoretical results for K=5. D_o = 0 corresponds to a completely effective damper, while D_o = 1.0 corresponds to a damper that does not restrict the flow at all during the off-period.

Effects of Temperature Probe Locations on Calculated Furnace Efficiencies

As mentioned earlier, measurements of temperature were performed at two different locations, the heat exchanger passage outlets and the test plane. The average temperature measured at heat exchanger passage outlets had a higher value during heat-up periods and a lower value during cool-down periods. After conducting a series of tests and computing furnace annual efficiencies, using the procedures described by Kelly et al [1], it was found that there was less than 1% difference on the average between the annual efficiency calculated from data obtained at the heat exchanger outlets and those obtained at the test plane for the particular furnace tested. The same result was also found for the seasonal efficiencies calculated using data from the two different temperature measurement locations. Thus, as long as one's interest is in annual or seasonal efficiency computation, the use of temperature measurements at the test plane described in the section in Experimental Set-up appears to give satisfactory results.

Conclusions

There exists, in general, good agreement between experimental and predicted results for flue-gas temperature profiles and mass flow rates under cycling conditions for a typical gas-fired furnace. Two forms of the equation for predicting the flue-gas mass flow rate during the

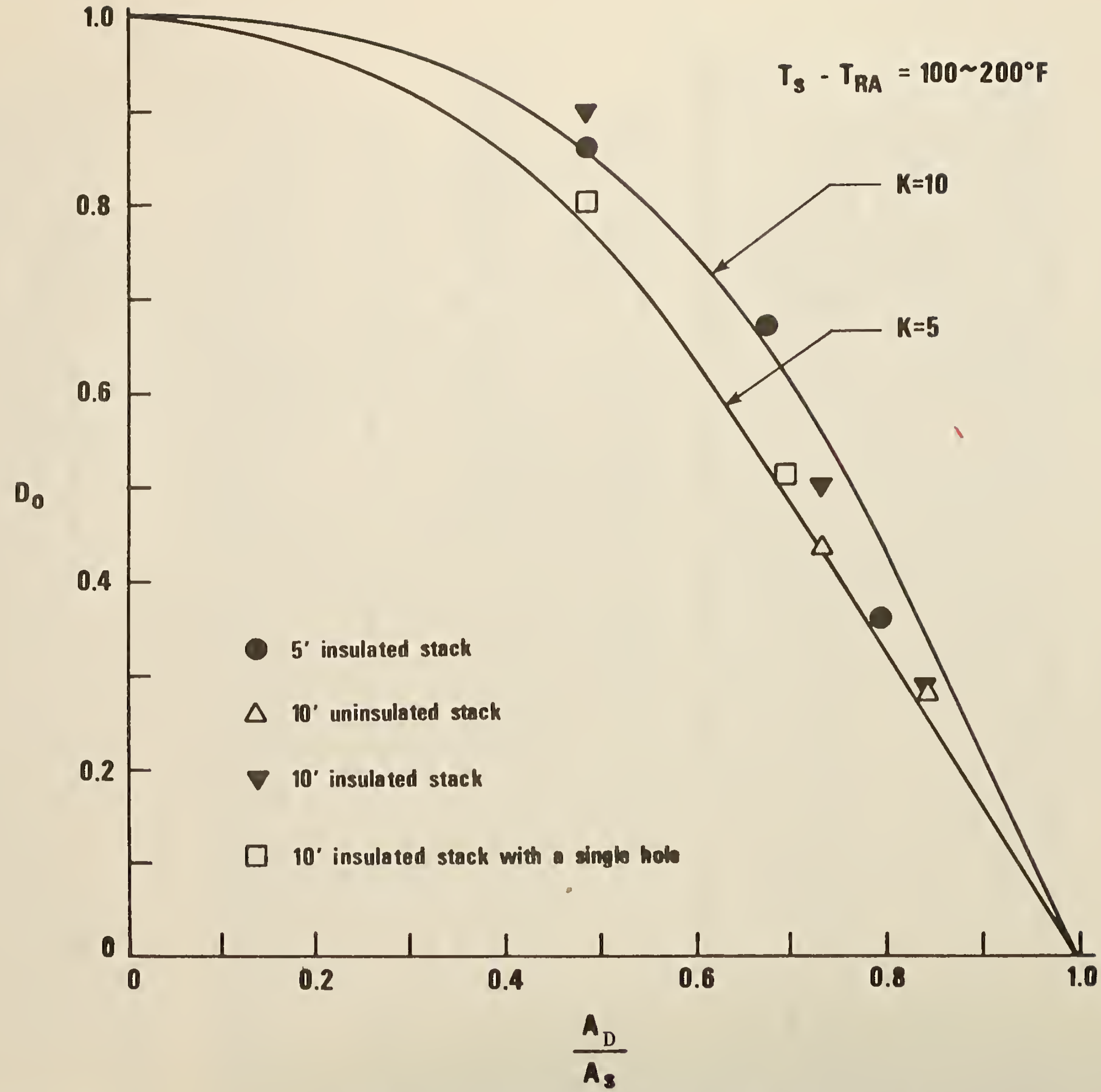


Figure 11. Variation of stack flow with damper area

off-period are presented which give almost identical results. It is shown that the effectiveness of a stack damper is dependent upon the ratio of the damper area to the stack area and upon a system friction factor, k. For a furnace or boiler installation with an unknown system friction factor, it is suggested that a value of k = 5 be used. Seasonal efficiencies calculated using the flue-gas temperature measured in an insulated test stack, one foot from the furnace outlet, were found to agree within 1% with values calculated using temperature data obtained at the heat exchanger outlets.

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15. SUPPLEMENTARY NOTES

☐ Document describes a computer program; SF-185, FIPS Software Summary, is attached.

16. ABSTRACT (A 200-word or less factual summary of most significant information. If document includes a significant bibliography or literature survey, mention it here.)

The flue-gas temperature and mass flow rate through a gas-fired furnace were studied in the laboratory. Temperature profiles were measured under cycling conditions and compared with profiles predicted mathematically using data obtained while the furnace was cooling down from steady-state operation and warming up from equilibrium. The mass flow rates at various flue-gas temperatures were measured using both a vane anemometer and a tracer-gas technique, and these results are compared with the mass flow rate predicted by the theorectical equations. The effect on the off-period flow rate of automatic stack dampers having different sized damper openings was experimentally determined. Theorectical equations are presented for predicting the effectiveness of a stack damper as a function of the ratio of the area of the damper to the area of the stack and a system friction factor.

17. KEY WORDS (six to twelve entries; alphabetical order; capitalize only the first letter of the first key word unless a proper name; separated by semicolons) Automatic stack damper; flue-gas temperature profile; gas-fired furnace; off-period mass flow rate; part-load performance; seasonal efficiency.

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