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MEASUREMENT OF HEATING SYSTEM DYNAMICS FOR COMPUTATION OF SEASONAL EFFICIENCY

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INTRODUCTION

The steady state efficiency of gas fired residential furnaces is usually assumed to be about 75%. Another 5% credit is given for heat recovered from the jacket and flue pipe when the furnace is installed in the heated space. Thus the input and output capacities stated on the furnace name plate normally have a ratio of 0.8. A well adjusted oil burner can achieve equally high efficiency. However, residential heating loads seldom require the furnace to approach steady state conditions. Residential furnaces are normally controlled in an on/off mode. Concern for fuel conservation raised questions about, (1) the applicability of the steady state furnace efficiency to describe the operating efficiency of the whole heating system, (2) the average seasonal system efficiency, (3) identification of the factors which influence seasonal efficiency, and (4) means to improve it.

Bonne[1] has described a computerized combustion model for computing the seasonal stack losses from a cycling residential furnace. The model employs the dynamic heat transfer characteristics of the furnace, the fuel properties, the measured combustion conditions, the infiltration characteristics of the house, and a weather profile. The model computes the fuel consumed during the on-periods of the burner and compares it to the energy lost through the stack. The weather profile is used to calculate furnace loads and the weighted average stack losses over an entire heating season. This paper describes the experimental techniques devised to measure the necessary parameters on installed heating systems. With these data the seasonal performance of a heating system can be computed. The effectiveness of various operating and system modifications can then be predicted. The calculated effects of some modifications are presented in a companion paper.[2]

The parameters to be measured are divided into two groups. The furnace parameters describe the characteristics of the furnace itself. The system parameters refer primarily to the infiltration characteristics of the house.

FURNACE PARAMETERS

Nine of the parameters to be determined are associated with the furnace as shown in Table 1.

We must be able to predict the percent operating time of the furnace for any indoor/outdoor temperature difference. This prediction can be made in two different ways. If we observe the percent on time for the furnace at some particular outdoor temperature then we can compute the balance point temperature (outdoor temperature that would require continuous operation of the furnace) for

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the furnace from Eq. 1.

$$T_B = T_o - \frac{T_o - T_1}{L} \qquad (1)$$

<u>where</u>

T_B = balance point temperature
T_o = indoor temperature
T_1 = outdoor temperature
L = furnace load or % on time

Once the balance point temperature is known Eq. 1 can be used to compute the furnace load at any other indoor/outdoor temperature difference.

A second method, and perhaps a preferred method, for computing the balance point is given by Eq. 2.

$$T_B = 65 - \frac{F \cdot D}{F_s} \cdot \frac{\eta_{ss}}{\bar{\eta}} \qquad (2)$$

<u>where</u>

F = firing rate per day
D = heating degree days per unit time
F_s = actual fuel consumed per unit time
$\bar{\eta}$ = seasonal system efficiency
η_{ss} = steady state system efficiency

In this case we compare the fuel that would be consumed per degree day if the furnace was operating at 100% capacity to that which is actually consumed per degree day over a given period and compute the outdoor temperature that would be necessary to make the furnace operate 100% of the time. The efficiency ratio is of course the unknown we are trying to determine. However, at 100% load this efficiency ratio would have to approach unity. Therefore, if we determine the balance point at several different loads and project it to 100% load we can reach a reasonable estimate of the balance point. If the unit of time for D and F_s is an extended period, e.g. a week or a month, weather and other conditions are averaged to give a more realistic estimate of the balance point.

Typical results calculated from monthly utility bill data (less water heater consumption) are shown in Fig. 1. In this case the maximum indoor-outdoor temperature difference ($65-T_B$) are plotted against furnace load. The scatter at light loads is due to variation in wind conditions and living habits. However, the average of the points is seen to be about 130°F. When it is extrapolated to 100% two asimptotes can be drawn which enclose all of the points. This appears to be a reasonable estimate of the maximum indoor-outdoor temperature difference the heating system can achieve under average weather and living conditions.

Two observations can be made; (1) even in January the average load on the heating system does not exceed 40%, (2) there is considerable scatter at light loads, but this scatter decreases as the load increases. The normal design temperature difference of 90°F for Minneapolis is shown also. Comparing this to the system results of 130°F gives 45% excess capacity. This is a modest amount of excess capacity compared to many systems.

Most gas meters have a high speed dial calibrated to about 1/10th cubic foot/revolution. Observing the speed of this dial is a good measure of the gas flow rate. However, care must be taken to insure that the water heater and other gas appliances are not operating at the time the measurement is made. Oil flow rate can be measured with the apparatus shown in Fig. 2. Two valves and a calibrated burette allow one to determine the time for a measured amount of oil to be consumed. The heating value of the fuel is best obtained from the utility or oil supplier.

Fig. 3 shows the technique used for measuring the combustion and stack dilution air flows. A methane tracer technique was employed. During the burner off period methane was introduced into the combustion chamber. The concentration of methane was sensed with a non-dispersive infrared analyzer either at the flue outlet or in the stack. The air flow is equal to the measured methane flow divided by the concentration at the detection point, Eq. 3.

$$Q = \left(\frac{Q}{C}\right)_{tracer} \tag{3}$$

The difference in flow at the flue outlet and in the stack is made up by the dilution air entering the draft hood. During the on-period the draft dilution air can be measured by introducing the methane tracer at the draft hood and measuring the concentration in the stack. The air flow through the combustion chamber during the on-period can be computed if the oxygen concentration and unburned hydrocarbon concentration is measured at the flue outlet. Thus combustion air and draft dilution air flow can be measured both during the on-period and off-period of the burner. These are steady state values. The computing model then modifies these steady state flows with the transient temperatures to compute the transient flow.

The transient temperatures of the flue gases must be known in order to compute the energy losses. These temperatures are easily measured with a thermocouple and recorder. Care must be exercised to avoid radiation errors, however. The temperature rise or fall can be described by an exponential function as shown in Eq. 4.

$$(T - T_o) = (T_m - T_o)\ (1 - e^{-t/\tau}) \tag{4}$$

where

T = instantaneous temperature
T_o = initial temperature
T_m = equilibrium temperature
t = time interval
τ = time constant

Eq. 4 can be rearranged to determine the time constant,

$$\tau = \frac{t}{\ell n \left[\dfrac{T_m - T_o}{T_m - T}\right]} \tag{5}$$

If the natural logarithm in Eq. 5 is converted to log base 10,

$$\tau = \frac{.434\ t}{\log_{10} \left[\dfrac{T_m - T_o}{T_m - T}\right]} \tag{6}$$

the slope of the function plotted on semi-log paper will be proportional to the time constant. The data for a typical case are shown in Fig. 4.

Eq. 6 shows that at time t=0 and $T=T_o$ $\log_{10}$ of the temperature parameter would be unity. Thus a single time constant could not describe the data presented in Fig. 4 very well. A weighted, two time constant function given by Eq. 7 can do a much better job.

$$\frac{T_m - T_o}{T_m - T} = \frac{1}{\dfrac{A}{e^{t/\tau_1}} + \dfrac{1-A}{e^{t/\tau_2}}} \tag{7}$$

The very rapid initial temperature rise is associated with ignition and heating of the gases in the combustion chamber. Then there is a more gradual temperature rise associated with the actual warm up of the furnace heat exchanger. These two functions are combined and the curve is seen to fit the data quite well. The weighting factor, A, is readily computed from Eq. 7 if the value for the temperature difference function is determined at some time (about 5 min in this case) after ignition.

SYSTEM PARAMETERS

The building infiltration characteristics and operating temperatures must be defined also. These system parameters are listed in Table 2.

If there were no stack, i.e., an electric-heated house, there would still be some flow of infiltration into the house and an equal flow of exfiltration air out of the house. This would represent a heat loss mechanism. Now if we add a stack to the house and allow exfiltration out of the stack this will reduce the normal exfiltration from the house. However, in order for any exfiltration to exist the pressure in the house must be higher than at some point outside the house and if we allow the stack to take some of the exfiltration then this must tend to reduce the pressure in the house. If the internal pressure is reduced then the infiltration will increase. Thus the effect of stack flow tends to decrease exfiltration and increase infiltration. To make a fair assessment of the importance of stack loss we must determine what the contribution of stack flow to increased infiltration is. This can be done by measuring the infiltration with the burner off and the stack closed, with the burner off and the stack open, and the burner operating.

The term ϕ is used as a weighting factor in the computational model to introduce this influence of stack flow on increased infiltration. It is defined by Eq. 8:

$$\phi = \frac{Q_i' - Q_i}{Q_s} \tag{8}$$

where

Q_i = infiltration flow rate, stack plugged

Q_i' = infiltration flow rate, stack open

Q_s = average stack flow rate, burner on or off

A methane tracer and an infrared analyzer were found to be quite effective for measuring the infiltration characteristics of the house. The arrangement is shown in Fig. 5. Methane at a controlled flow rate was introduced into the return air of a warm air heating system. The concentration was measured in the plenum. The methane flow was allowed to continue until the house reached a level of about 500-1000 parts per million. After which the methane supply was turned off and the decay in concentration was recorded as a function of time.

The use of methane as a trace gas may at first seem hazardous. However, the concentration was at all times kept 1/10 to 1/100 of the lower flamability limit. The air flow rate in the return duct was estimated. The methane flow rate during the charging period was then controlled at a value of no more than .5% of this value. This was in the range of 10-20 ℓ/m. Since the lower flamability limit of methane in air is about 5.3%, this gave a safety factor of 10. A maximum concentration in the house of 1000 ppm gave a safety factor of 50. Thus the only place the methane could burn was at the outlet of the supply tube.

One must ask, "How accurate is the measurement; is the trace gas uniformly distributed and well mixed?" The use of methane as a trace gas has several advantages. Chemically pure methane is odorless and non-toxic. It is a little lighter than air and thus has no tendency to settle out. It diffuses rapidly in air. It is chemically stable and inactive at room temperature and does not adsorb readily on materials normally present in a room.

The question of how well the tracer is distributed can be answered from observations of the plot in Fig. 6. If the tracer is not well mixed at the start of the decay period, the initial concentration measured will be either higher or lower than the equilibrium concentration. As time progresses mixing will improve. Thus the rate of decay would reflect both the infiltration rate and the mixing rate. This would produce a curve on the semi-log plot of Fig. 6 unless both the mixing rate and infiltration rate have the same time constant. This is highly unlikely. We have found the first 15 or 20 minutes of the decay period may show some mixing effect. Waiting until this period passes yields good measurements.

The concentration of the methane follows an exponential decay. When this concentration is plotted on a semilog plot against time the slope of the line becomes the time constant. The actual infiltration flow rate then is the volume of the house divided by this time constant, Eq. 9.

$$Q_i = \frac{\text{House volume}}{\tau_i} \tag{9}$$

Fig. 6 shows the log of the concentration plotted vs time for the three conditions of interest. Although the flow rate change when the burner is turned on and when the stack is open is not large, it is measurable.

RESULTS

The results of an actual measurement are shown in Table 3. It is seen that when the burner is off about 35% of the stack flow should be charged to increased infiltration. When the burner is on and the stack flow is higher, 52% should be charged to infiltration. The value of ϕ was generally found to be around 60%. Extreme values ranged from 0.25 to 0.95. It is influenced by the relative tightness of the house and wind and temperature effects.

Burner air inlet temperature and outdoor temperature are readily measured by conventional means. These parameters then provide the additional system inputs necessary to make use of the stack loss model.

Table 4 presents the results of measurements on 5 houses located in the Minneapolis area. The data presented in the first 13 lines of the table are the input data for Bonne's[1] stack loss model measured with the techniques described. The resulting balance point (i.e. outdoor temperature for 100% load on the furnace) and seasonal efficiency are presented in the last two lines on the table. An average infiltration factor, ϕ, of 70% was assumed for two systems where infiltration measurements were not made. One of these was a hot water system which does not lend itself to infiltration measurements by the tracer technique.

These data represented furnaces tested in the "as found" condition. In the case of house no. 3, small burner orifices were installed after initial testing. No other changes were made. This reduced the firing rate from 95,000 to 54,500 BTU/hr. The equilibrium flue temperature dropped from 530°F to 340°F, flue oxygen dropped from 8.0% to 6.8% but the draft dilution factor increased from 62% to 100%. The net result was an increase in seasonal efficiency from 57.2% to 71.5%.

The temperature setting for turning off the fan or pump was 90°F in all cases. Experience has shown that this is about the optimum setting. Lower off temperature saves very little heat but does increase electric power consumption. The on temperature of 140-150°F was perhaps a little high. However, the temperature rise in a warm air system is so rapid that there is not much difference between 120°F and 150°F.

While these results are somewhat superficial, they are presented to illustrate the kind of results that can be achieved with Bonne's[1] stack loss model using the measurement methods described to obtain the necessary input parameters.

CONCLUSIONS

We have shown the measurement techniques that can yield the parameters necessary to make use of the stack loss model in computing the seasonal efficiency of existing, installed heating systems. The methods lend themselves to field use and have been used in five different houses. There is a minimum disturbance to the system and the household and a set of measurements can be completed in 4 to 5 hrs. The measurement technique is described as applied to a forced warm-air system. It can be used with other systems if means are provided for good circulation within the structure when making the infiltration measurements.

REFERENCES

1. U. Bonne and A. Johnson; "Thermal Efficiency in Non-Modulating Combustion Systems", Conference on Improving Efficiency in HVAC Equipment and Components in Residential and Small Commercial Buildings, Prudue Univ., October 1974.

2. U. Bonne, J. Janssen, R. Torborg, N. Pearman and A. Johnson; "Digital Simulation of the Seasonal Efficiency of Combustion Heating Systems", Seminar on Efficient Fuel Utilization in Heating Systems, ASHRAE, Boston, June 1975.

Table 1

FURNACE PARAMETERS

1. Furnace load at one outdoor temperature
2. Cycle time at one furnace load
3. Fuel flow and heating value
4. Combustion chamger off-period flow
5. Draft hood on-period flow
6. Flue heating/cooling time constants
7. Circulating fan/pump limit switch settings
8. Flue oxygen
9. Flue combustibles

Table 2

SYSTEM PARAMETERS

1. Building infiltration rate
   a. Burner on
   b. Burner off, stack open
   c. Burner off, stack plugged
2. Burner air inlet temperature
3. Outdoor temperature

Table 3

CONTRIBUTION OF STACK FLOW TO INCREASED INFILTRATION

Stack	Burner	τ_i Min	Q_i Cfm	ϕ
Plugged	Off	145	165	
Open	Off	131	183	0.35
Open	On	121	198	0.52

Table 4. TYPICAL RESULTS

Item	Units	House No. 1	No. 2	No. 3	No. 4	No. 5	No. 3 Capacity Reduced
Measured Input Data:							
Type System		Gas FWA	Gas FWA	Gas FWA	Rot. Oil FHW	Gun Oil FWA	Gas FWA
Firing Rate - Burner	Btu/hr	120,000	94,000	95,000	164,000	100,000	54,500
Pilot	Btu/hr	1,100	1,000	900	0	0	900
Heated Area	Sq ft.	2,100	1,800	1,520	2,800	2,970	1,520
Cycles at 50% Load	Cycles/hr	6	1.6	6	1.25	6	6
Fan or Pump Setting On/Off	°F	150/90	140/90	140/90	150/90	150/90	140/90
Equilib. Flue Temp.*	°F	600	575	530	712	519	340
Min. Flue Temp.*	°F	100	95	90	68	68	90
Flue Oxygen*	%	6.46	6.45	8.0	6.0	8.0	6.8
Flue Combustibles	%	0	0	0	0	0	0
Flue Draft Factor	%	100	100	100	41	37	100
Draft Dilution Factor*	%	120	165	62	125	8	100
Infiltration Factor, ϕ*	%	51	80	70**	70**	80	70**
Outdoor Temp.	°F	71	20	50	18	12	50
Average Wind Speed	Mph	6	5	6	6	6	6
CALCULATED RESULTS:							
Balance Point	°F	-97	-118	-145	-169	-88	-82
Seasonal Efficiency	%	59.6	61.3	57.2	63.5	73.4	71.5

* Measured at, or corrected to outdoor temperature

**Assumed

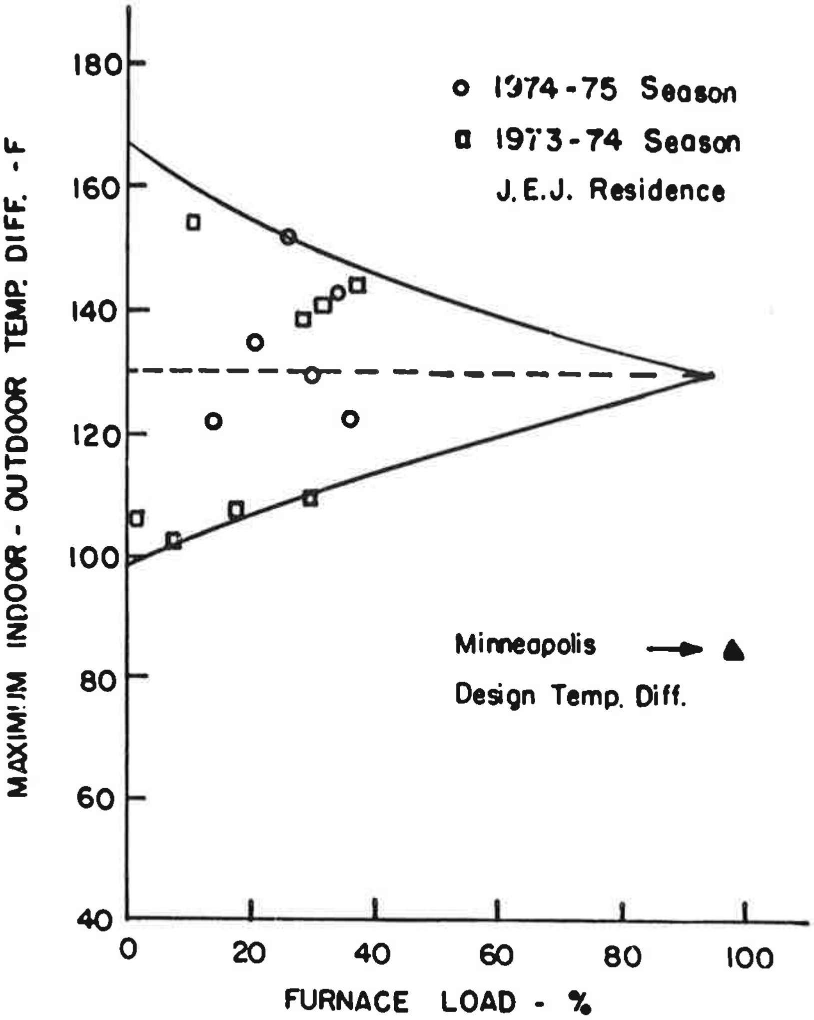

Fig. 1 Estimate of furnace capacity

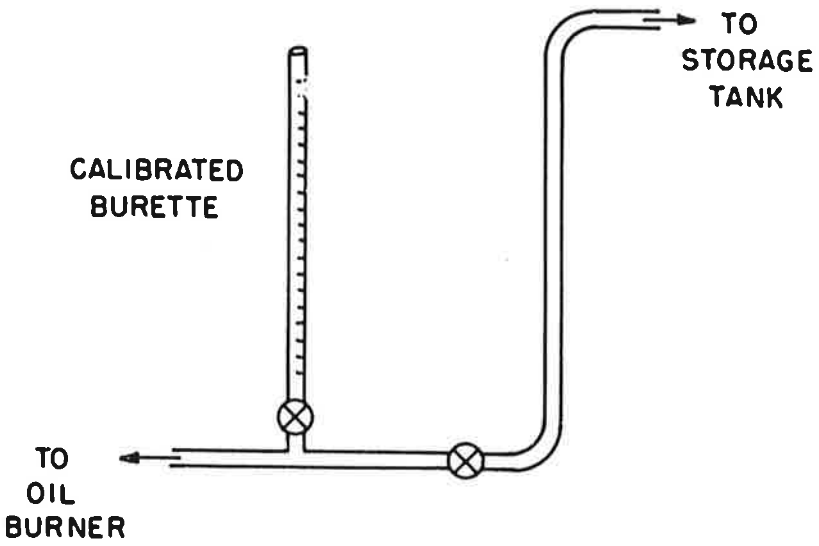

Fig. 2 Fuel flow measurement

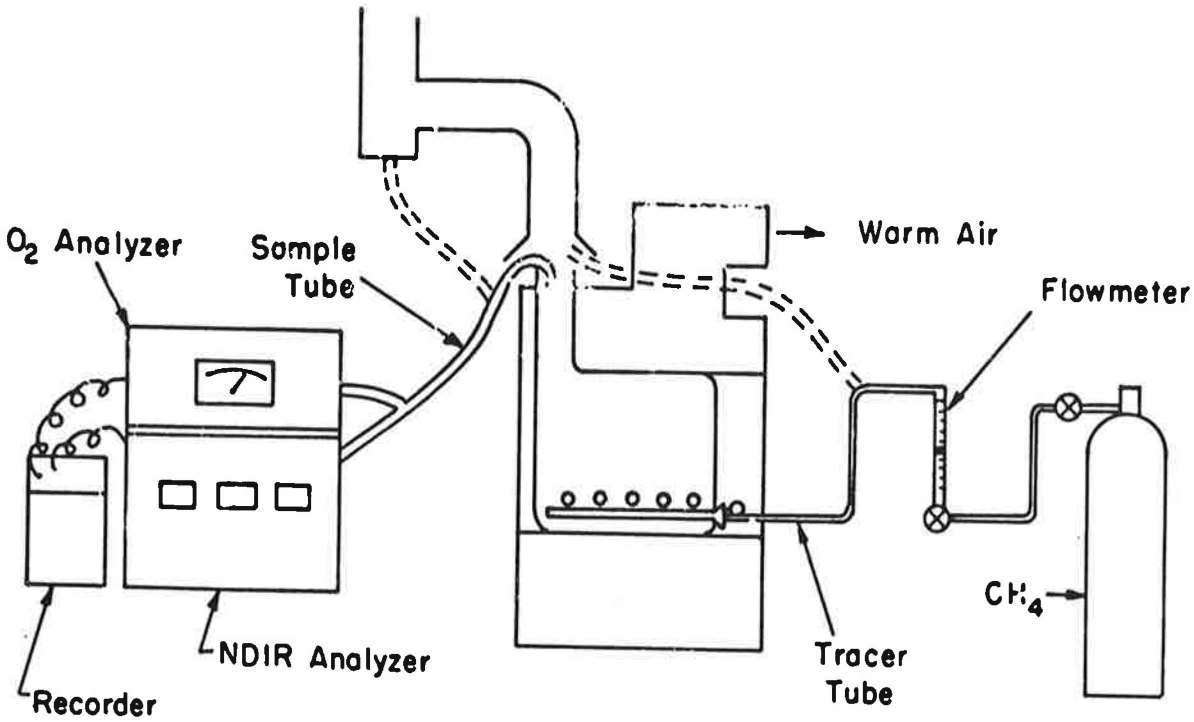

Fig. 3 Flow measurement with tracer

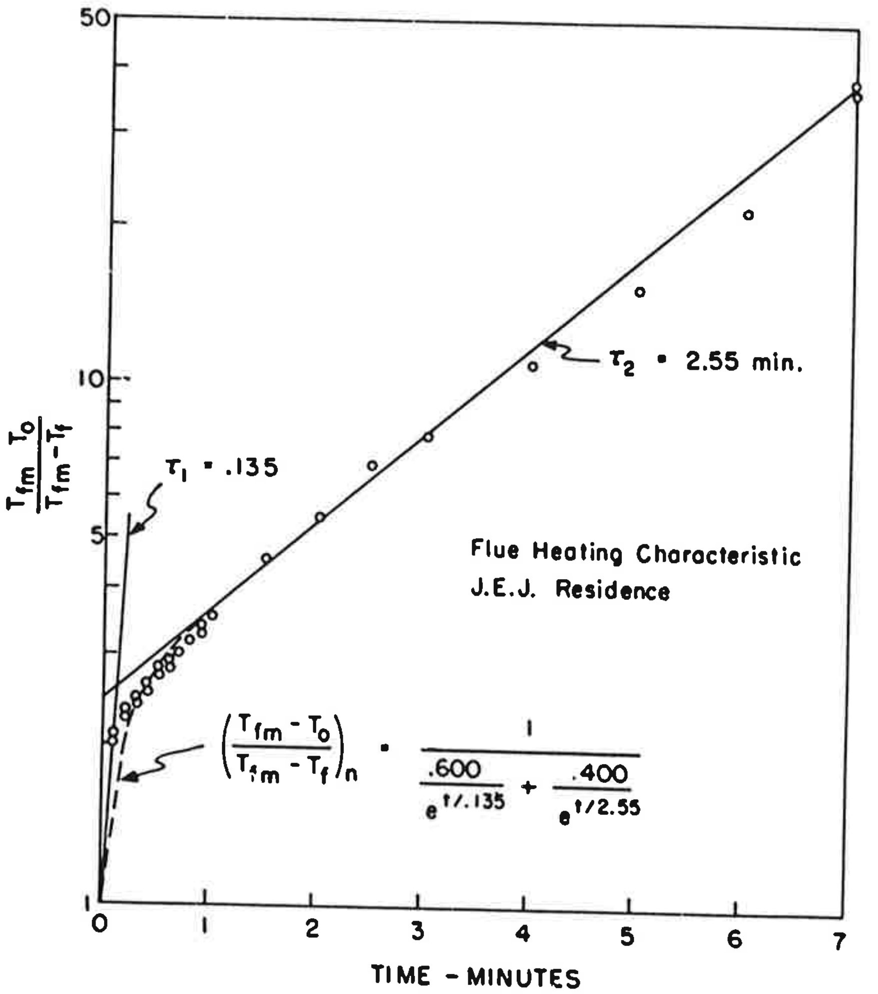

Fig. 4 Flue heating characteristic

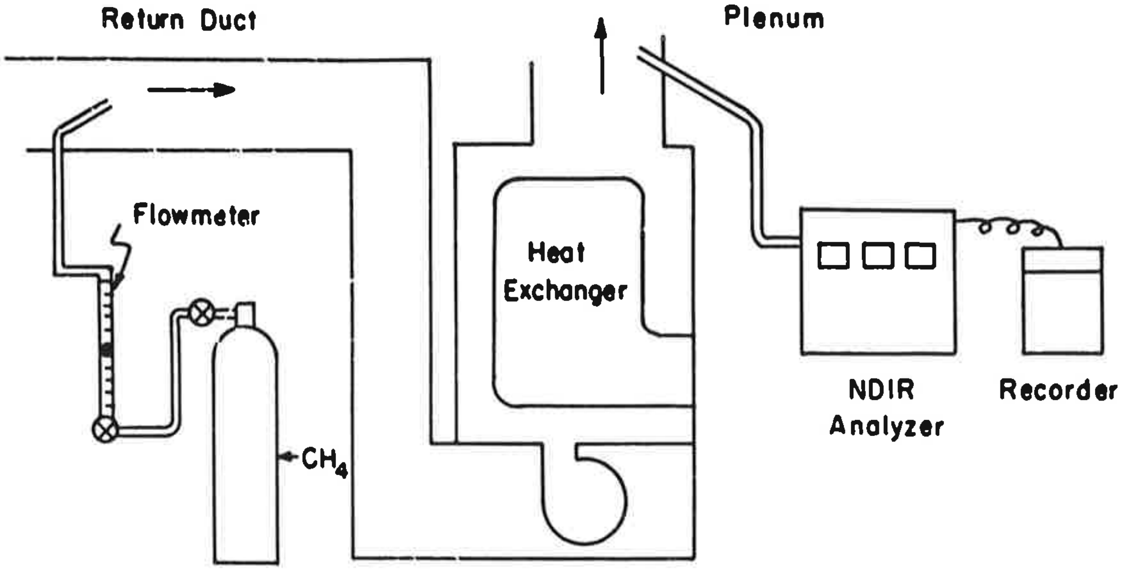

Fig. 5 Infiltration measurement with tracer

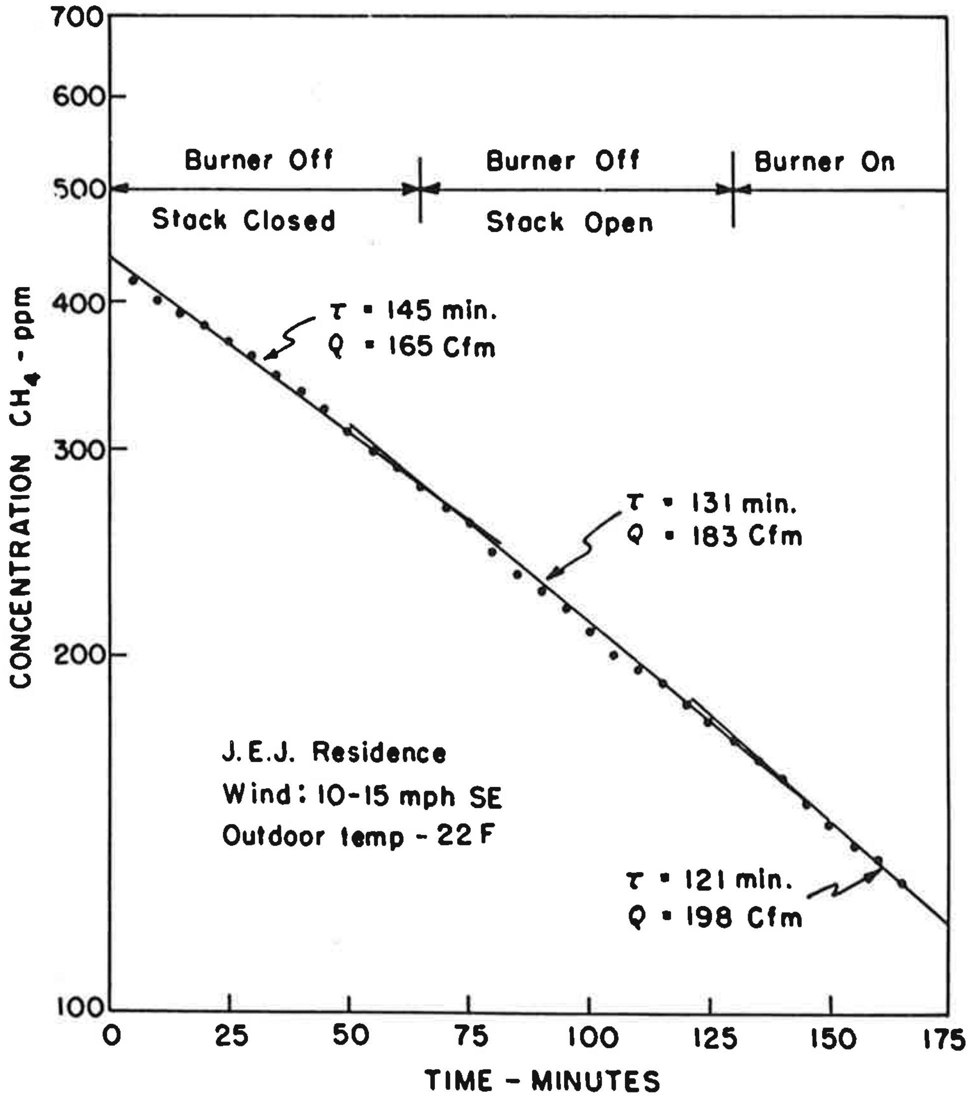

Fig. 6