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Improvement of Seasonal Efficiency of Residential Heating Systems

Recent studies have shown the seasonal efficiency of oil-fired residential heating systems to be in the range of 55–75 percent compared to steady-state efficiency of 80 percent or greater. A digital model using measured input data was used to compute the on period flue losses, the off period flue losses and the losses due to combustion induced building infiltration. Excess capacity in existing systems and new construction is shown to increase fuel consumption but decrease electric power consumption in oil-fired warm air furnaces. Excessive furnace capacity is generally undesirable, but if the cost of electric power is high compared with the cost of fuel oil, there is an optimum amount of excess capacity. The use of outdoor air for combustion and draft control (sealed combustion system) is shown to be beneficial sometimes, but may incur a cost penalty under other conditions.

Introduction

Improving the efficiency of fuel utilization is a major objective today. The rapidly rising fuel bills for residential heating have spawned numerous ideas for making residential central heating systems more efficient. However, some of these system modifications have unrecognized limitations to their effectiveness.

The steady-state efficiency of properly operating gas furnaces is about 80 percent. Well-adjusted oil burners can, in theory, achieve slightly higher steady-state efficiency because of the lower hydrogen/carbon ratio of fuel oil. However, there are additional system losses for a heating system installed in a home that result in substantially lower "seasonal system efficiency." Field measurements [3, 4][1] and calculations made with the model described in the following have shown that the seasonal efficiency for gas- and oil-fired systems is in the range of 50–75 percent. Oil-fired systems using power (gun type) burners tend to have higher seasonal efficiencies than atmospheric gas burners.

Study of the stack losses for a cycling residential heating system has yielded insight into the effectiveness of methods proposed for improving the seasonal system efficiency.

[1] Number in brackets designate References at end of Paper.

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The general features of the stack loss model and the input data are described. The effectiveness of modifications such as reducing burner overcapacity and using outside air for combustion are then examined.

Stack Loss Model

Furnace efficiency is usually measured by comparing the heat output to the fuel consumed. This ignores certain system losses. A better method for finding the system efficiency is to compare the stack losses with the fuel consumed. Bonne, et al. [1, 2] have derived a model for computing the annual stack losses from a cycling combustion system. The model, based on the stack loss method, can be visualized with the aid of Fig. 1. Fuel and combustion air flow into the furnace combustion chamber. Products of combustion carry sensible and latent heat as well as any unburned fuel out the stack. Additional dilution air for draft control is drawn in the draft diverter (or draft hood in a gas furnace). This air normally enters the structure through cracks and openings in addition to the normal flow of infiltration air through the structure due to wind and temperature effects.

The magnitude of the losses fluctuates with the cycling period of the burner, firing rate, and weather conditions. Thus the losses are influenced by things such as the load and excess capacity of the furnace. Bonne's model employs the dynamic heat transfer characteristics of the furnace, the fuel properties, measured combustion conditions, draft control air flow, the infiltration characteristics of the house, and the local weather profile. The model compares the fuel consumed during the on-periods of the burner to the energy lost through the stack. Heat loss due to air flow through the stack during

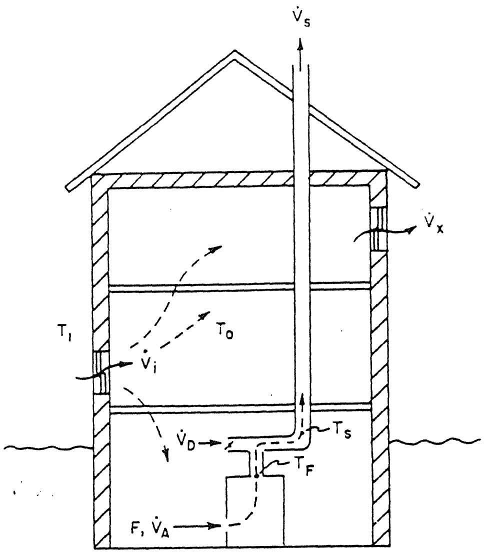


Fig. 1 Schematic of infiltration flow: T = temperature at indicated points; $\dot{V}$ = air flows at indicated points

the burner off-periods is computed also. The weather profile then is used to calculate system loads and weighted average stack losses over an entire heating season. Table 1 defines and categorizes the various losses.

The Group I losses are computed from the energy balance equations using measurements of the concentration of combustibles and oxygen (or CO_2) in the flue gas, fuel heating valve, hydrogen/carbon ratio, fuel flow rate, flue temperature and inlet air temperature.

Group II losses result from convective flow through the combustion chamber and draft control device during the burner off period. These are computed from a knowledge of the inlet air temperature, the dynamic flue flow, the draft control air flow, and the flue temperature. A knowledge of the purge air flow rate in oil burners is needed also.

Group III losses account for the effect of the combustion and draft control air on the house infiltration. Group I and II losses are computed on the basis of the flue and room temperatures. However, the air used by the furnace both for combustion and draft control must come ultimately from outside. It is exhausted out the stack at some higher temperature. This contributes to the infiltration loss for the structure. However, under most conditions only a portion of this loss can be charged against the furnace. The reason for this is as follows.

If there were no stack (an electric heated house) there would still be some infiltration of air due to wind and temperature effects. This flow would be balanced by an equal exfiltration air flow. When the stack is added it provides an additional path for exfiltration. This would lower the pressure in the house and increase the infiltration. However, when the pressure is lowered, the normal exfiltration also decreases. Therefore, the stack air flow must come partly from increased infiltration and partly from decreased exfiltration. Under many conditions these effects are about equal; and only 50 percent of the stack air flow can be charged to increased infiltration. Measurements have shown that 70 percent is a good average. This infiltration weighting factor, ϕ, is used to adjust the loss due to heating the stack air flow from outdoor to room temperature.

The Group IV losses and gains account for three miscellaneous items. Gas furnaces generally use a pilot which is left burning during the summer. This loss is not present in oil furnaces.

Distribution ducts or pipes passing through unheated areas such as crawl spaces or attics lose heat. The stack, especially a metal stack, tends to transfer some useable heat to the occupied space. This heat transfer is also included in Group IV.

The input data needed by the model are given in Table 2 [4, 7]. Measurement methods have been described by Janssen, et al. [3].

A gas tracer technique has been used for measuring the various air flow rates and building infiltration rates. Methane has been found to be a convenient trace gas. Building infiltration rates are measured

Table 1 Definition of heating system losses (T_F = flue temperature, T_0 = room temperature, T_1 = outdoor temperature)

Group I Burner on-period flue losses
- L_C — flue combustibles
- L_W — enthalpy of uncondensed water
- L_N — enthalpy of CO_2, H_2O, N_2 between T_F and T_0
- L_E — enthalpy of excess air between T_F and T_0

Group II Burner off-period flue losses
- L_D — enthalpy of draft air between T_F and T_0
- L_P — enthalpy of purge air between T_F and T_0

Group III Furnace contribution to building infiltration losses
- L_A — enthalpy of combustion air between T_0 and T_1
- L_{DA} — enthalpy of draft air between T_0 and T_1
- L_H — enthalpy of draft control air between T_0 and T_1
- L_M — enthalpy of water evaporated to humidify furnace air

Group IV Miscellaneous losses/gains
- L_L — pilot flame energy wasted during the summer
- L_U — energy wasted by the distribution system to unheated space
- G_S — energy gain from stack back to house

with: (*a*) burner on; (*b*) burner off, stack open; (*c*) burner off, stack plugged. The infiltration factor is then computed from equation (1).

$$\phi = \frac{\dot{V}_i' - \dot{V}_i}{\dot{V}_s} \tag{1}$$

$\dot{V}_i'$ = infiltration flow rate, stack open
$\dot{V}_i$ = infiltration flow rate, stack plugged
$\dot{V}_s$ = average stack flow rate, burner on and off

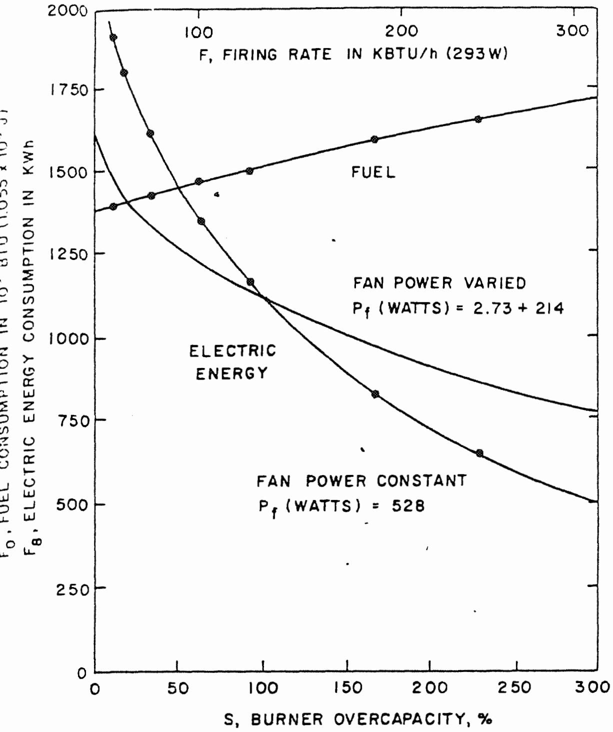
Results

Measurements have been made and analyzed for a number of residential systems [2, 4] which show that losses due to excess combustion air and excess system capacity are the major factors contributing to low seasonal efficiency. In the past operating cost was secondary to comfort and first cost. Since the cost increment of a furnace with some excess capacity was generally small, contractors and heating dealers tended to specify furnaces with sufficient capacity to avoid complaints under the most severe load conditions. These include rapid morning pick-up after night set back, rapid recovery of domestic hot water in boilers with tankless coils and the air flow requirements for air conditioning. Since residential systems are usually sized on the basis of superficial estimates with these factors in mind, generous oversizing has been the rule. A survey of 26 oil fired residential systems in the Boston area showed an average excess capacity of 147 percent [4], i.e., furnace had 2½ times the capacity needed to carry the design load.

Fig. 2 shows the effect of excess capacity on the oil and electric power required for a typical oil fired warm air furnace located in a

Table 2 Input parameters

1. Weather: Temperature versus hours, rh and wind profiles.
2. Building load: Specific heat loss rate (W per °C), indoor temperature and rh, and internal heat sources.
3. Fuel: Heat value, H/C ratio, fuel and electric power costs.
4. Burner: Firing rate, flue oxygen (or CO_2), and combustibles, draft factor, cycling rate, electric power consumption, and gas pilot consumption.
5. Heat exchanger: Cyclic flue temperature.
6. Distribution system: Fan/pump control set points and power consumption.
7. Stack: Oxygen (or CO_2), building heat gain from stack and infiltration factor, ϕ.



ig. 2 **Seasonal energy consumption for an oil-fired warm air furnace. See able 3 for assumed conditions. Constant fan power curve is typical for a 20,000 Btu/hr (35.2 kW) furnace. Variable fanpower curve is typical for ifferent furnaces designed for a given firing rate. Constant draft dilution factor nd maximum flue temperature**

ıorthern climate (Minneapolis). Pertinent operating conditions used n this calculation are given in Table 3. They were chosen on the basis ıf field test data [4].

Fig. 2 shows that the fuel required increases with excess capacity. The electric power required decreases, however. This is due to changes n the fan power demand. As the furnace capacity is increased the ›perating time for the fan decreases. A survey of manufacturers' fan ›ower ratings for a number of different furnaces (both gas and oil) vith firing rates ranging from 50,000 to 210,000 Btu/hr [5] show the ıverage power was given by

$$P_f = 2.37F + 214 \tag{2}$$

P_f = fan power in watts
F = firing rate in kBtu/hr

ndividual furnaces may vary by ±25 percent from this average.

The effect of various degrees of oversizing on the electric power onsumption of a 125,000 Btu/hr gas-fired warm air furnace is shown n Fig. 2 by the curve with a constant fan power of 528 W. Oil-fired urnaces and furnaces of different size would have curves paralleling he 528-W curve.

Table 3 Operating conditions for an oil-fired, residential, forced warm air system

Weather: Minneapolis 8025° (F) days.
85°F (47°C) design temp difference
Building load: 490 Btu/hr-°F (258 W/°C)
Burner: Power type, 250 W
Draft factor, 0.3 (i.e., ratio of volumetric flue flow during on and off periods of burner cycle)
Excess air, 80.3 percent
Heat exchanger: Steady state flue temperature, 620°F (327°C)
Stack: Draft dilution factor, 0.7 (i.e., ratio of volumetric flow through draft damper to flue flow at steady state)
Building infiltration factor, ϕ = 0.7

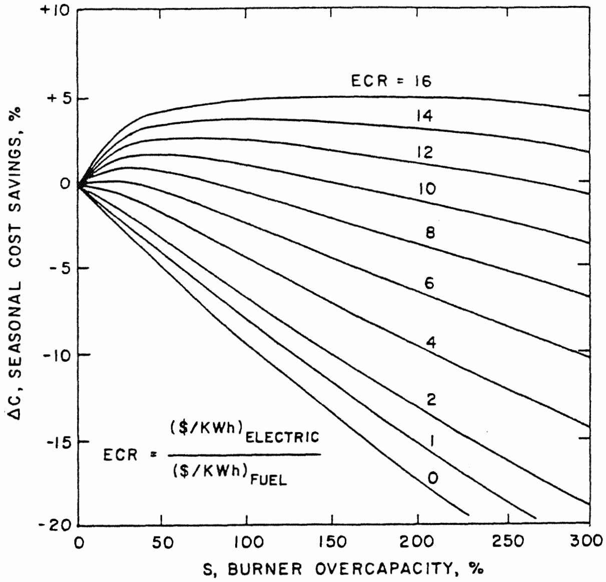


Fig. 3 Effect of sizing on operating cost for an oil-fired forced warm air system—new furnace in a new house (operating conditions same as Fig. 2 and Table 3)

The effect of excess capacity on seasonal operating cost is shown in Fig 3. Figs. 2 and 3 apply to new residential installations where stack, ducts, and furnace are matched so that constant draft dilution and flue temperature can be assumed. If a furnace of some overcapacity is chosen, the effect on operating cost will depend on the local cost of fuel oil and electric power. The parameter ECR (Energy Cost Ratio) is defined as follows:

$$\text{ECR} = \frac{(\$/\text{kWh})\text{ electric}}{(\$/\text{kWh})\text{ fuel}} \tag{3}$$

Electric power is always quoted in cents per kWh, but frequently this is a sliding scale. The cost used is the incremental cost at the maximum use rate (end block rate) since any changes will affect the maximum monthly consumption. However, oil is normally priced in cents per gal. The cost in cents per gal can be converted to \$/kWh by,

$$(\$/\text{kWh})\text{ fuel} = \frac{(\$/\text{gal})(3{,}414\text{ Btu/kWh})}{(140{,}000\text{ Btu/gal})} \tag{4}$$

A higher heating value of 140,000 Btu/gal is assumed here. For Minneapolis where the typical power rate would be \$.03 per kWh and the price of No. 2 oil is about \$.40 per gal,

$$\text{ECR} = \frac{(0.03)(140{,}000)}{(0.40)(3414)} = 3.08 \tag{5}$$

Thus a furnace with 100 percent excess capacity will increase the operating cost about 5 percent in Minneapolis. The line for ECR = 0 reflects the fuel cost only. In Minneapolis any excess capacity will increase both fuel consumption and total operating cost

The price of oil is relatively uniform throughout the United States, but electric power costs vary considerably. ECR varies from a minimum of about 0.95 in Seattle to as high as 9 on the east coast. The average value for the US based on 1976 prices is 4 [6]. Fig. 2 shows that a small amount of excess capacity may be desirable if the energy cost ratio is high. The conclusion is that for new construction it is usually desirable to minimize excess furnace capacity in order to minimize operating cost.

Modifications to existing furnaces present a somewhat different problem. In this case furnace and fan size are fixed. We assume the electric power consumption rate will be unaffected by decreasing the firing rate. The firing rate for oil burners can be decreased, within limits, by using a nozzle with a smaller orifice. Fig. 4 shows the savings

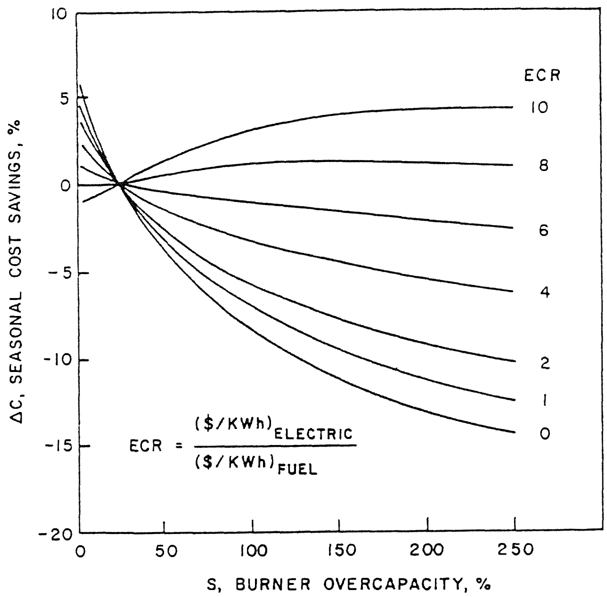


Fig. 4 Effect of sizing on operating cost for an oil-fired forced warm air system—existing furnace. Constant fan power. Draft dilution factor and maximum flue temperature varied with firing rate and overcapacity

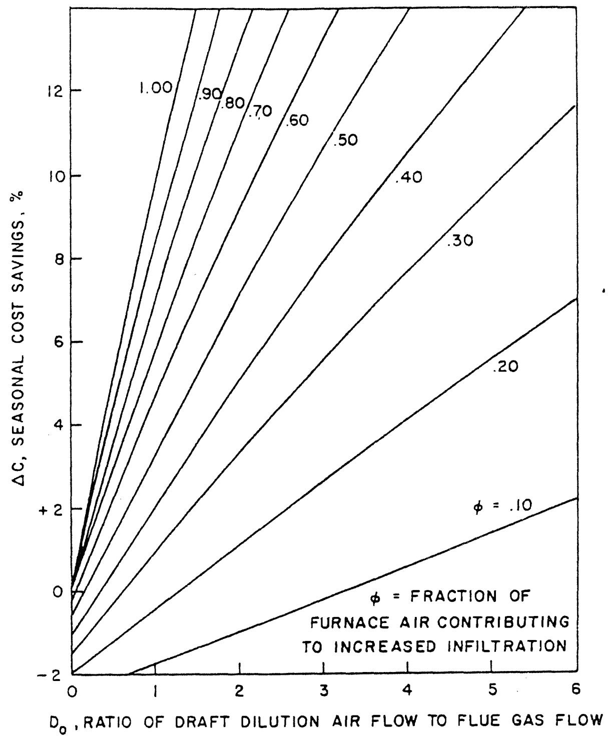


Fig. 5 Effect of sealed combustion chamber on operating cost—no heat exchange between flue gas and combustion air—load conditions as given in Table 2; energy cost ratio (ECR) = 3.28; overcapacity, 175 percent

that can be realized by decreasing the firing rate for a typical oil fired warm air furnace. The reference point is 25 percent oversize. The Boston study [4] revealed an average excess capacity of 147 percent for the 26 burners surveyed. Five systems were found to be 200–300 percent oversize. The amount of excess capacity was estimated from the annual fuel consumption, heating degree days for the area and firing rate [4]. As before, the ECR = 0 curve gives the fuel savings. Reducing the firing rate from 225 percent excess capacity to 25 percent excess capacity would reduce the oil consumption 14 percent per year if excess combustion air remained constant. Operating cost would drop only about 8 percent if ECR = 3 due to the increased fan operation with lower firing rate.

There are, however, limitations to firing rate reduction. Oil burners are designed to achieve a certain degree of turbulence in a given combustion chamber with a design firing rate. Substantial reduction in firing rate can reduce turbulent mixing and require more excess air for clean combustion. The Boston study showed that the excess air had to be increased 36.9 percent on the average for a 27.1 percent decrease in firing rate. The losses due to increased excess air offset the advantages of capacity reduction. The average fuel consumption decreased about 3.1 percent. However, in a few cases, an actual increase in fuel consumption was observed when the firing rate was reduced due to the increase in excess air needed to prevent smoke. Thus the results of a firing rate reduction should be carefully measured. There probably is an optimum amount of reduction for any given set of circumstances. Fig. 4 gives an estimate from which one can start. Firing rate reduction may also require a relative increase in the draft control air. This will be discussed in a subsequent paper [8].

Another modification frequently proposed is to draw the combustion air from outdoors. The theory is advanced that this will avoid withdrawing warm air from the heated space for combustion. A sealed combustion system is one in which outdoor air is used for both combustion and draft control. In this case the effective infiltration factor, $\phi = 1.00$. The combustion and draft control air are heated from outdoor temperature to stack temperature. All of this loss must be charged against the furnace. The loss can be minimized if provision is made for heat exchange between the stack gases and the combustion air. Figs. 5 and 6 show the savings to be realized if a sealed combustion system is substituted for the normal system. Fig. 5 is the case with no heat exchange between stack gas and inlet air. In Fig. 6 it is assumed the outdoor air is heated to an average room temperature by the stack gases before entering the combustion chamber. The curves are plotted against draft dilution factor, D_0, for various values of the infiltration factor, ϕ. The draft dilution factor is defined for steady-state conditions:

$$D_0 = \frac{\dot{V}_{D0}}{\dot{V}_{fg}} \tag{6}$$

where

$\dot{V}_{D0}$ = volume flow of air through draft diverter
$\dot{V}_{fg}$ = volume flow of flue gas from combustion chamber

The fuel and power costs assumed were No. 2 oil = 0.0085 \$/kWh (0.35 \$/gal) electric power = 0.028 \$/kWh so that the ECR = 3.3.

The draft dilution factor for an oil (power burner with a draft diverter) burner is generally low. The Boston oil burner study revealed an average of 0.7 with a range from 0 to 7. Fig. 5 shows that a sealed combustion system may introduce a cost penalty.[2] However, if heat exchange is provided, Fig. 6 shows a substantial savings under all conditions.[2] Thus a sealed combustion system should include provision for heat exchange between the stack gas and combustion air.

Conclusions

The seasonal system efficiency of residential heating systems is substantially lower than the steady-state furnace efficiency. This is due to the effect of off-period drafts and of excess combustion air and draft control air on the infiltration loss for the structure. Excess furnace capacity amplified these losses. Excess capacity should be minimized in new installations unless the ratio of electric power cost to fuel cost is high (ECR >6). In that case an optimum, nonzero amount of excess capacity exists.

The operating cost of existing oversized furnaces usually can be reduced by decreasing the firing rate. This benefit may be offset by

[2] These calculations did not include the effect of off period cooling of the furnace below room temperature by the outdoor air.

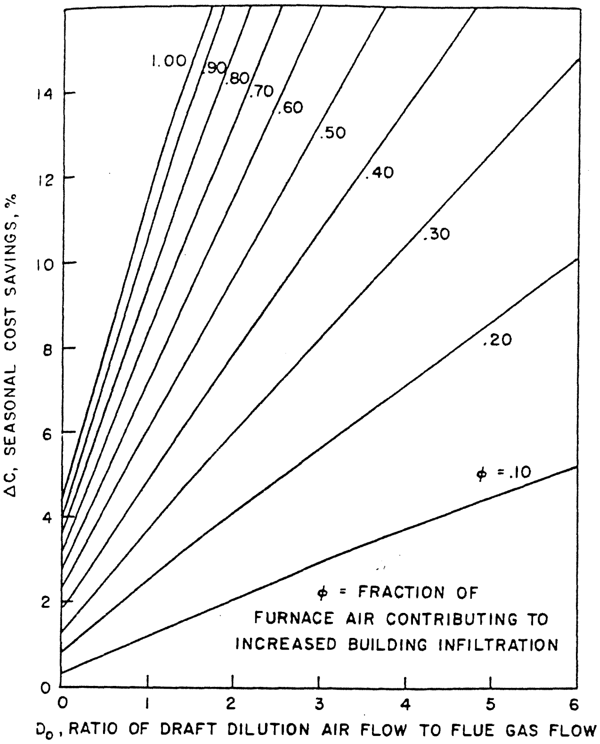


Fig. 6 Effect of sealed combustion chamber on operating cost—with heat exchange (furnace and other operating conditions same as Fig. 5)

the need to increase excess combustion air and it generally increases relative draft diverter flow.

Sealed combustion systems offer operating cost savings if provision is made for heating the outdoor air used for combustion, with the stack gases. If heat exchange is not provided, a cost penalty may exist.

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DISCUSSION

R. E. Barrett[3] and D. W. Locklin[3]

The authors have made a definite contribution to the quantitative understanding of the complex situation of overall efficiency in intermittently operated residential oil burners.

Concerning Figs. 5 and 6 and the conclusion that "a sealed combustion system should include provision for heat exchange between the stack gas and combustion air," there is a tradeoff between the increased savings associated with the heat exchanger and the potential problems it offers. Such a heat-exchanger would be exposed to very low temperature outdoor air on one side and moist flue gas on the other. Thus, there is a high potential for condensation of water (and dilute sulfuric acid) on the heat exchanger and corrosion is likely to result. Also, the physical arrangement of a heat exchanger between the furnace outlet and the stack may be awkward, especially in retrofit installations.

Offsetting these problems in the cost savings such a heat exchanger offers. Comparing the curves in Fig. 5 (sealed combustion system, no heat exchanger) and Fig. 6 (sealed combustion system with heat exchanger) shows that the cost savings associated with the heat exchanger is 2–4 percent for $\phi = 0.5$–0.9. If D_0 is greater than 1.0, the cost savings is 2–3 percent.

The suggestion that the use of a sealed combustion system with a heat exchanger may produce a cost penalty seems questionable as the negative cost savings values do not appear likely for practical structures. The only curves in Fig. 5 that go significantly below the $\Delta C = 0$ line are for $\phi < 0.5$. Earlier the authors suggested $\phi = 0.7$ as a practical value and we doubt that ϕ can ever be less than 0.5 in a real structure. For $\phi = 0.5$ and 0.6, the curves go below the $\Delta C = 0$ line if $D_0 < 0.2$; again, this is unlikely for many real structures. Thus, we believe the authors should reconsider the inference that a heat exchanger is required to make a sealed combustion system beneficial. Such an inference might lead some persons to not consider using sealed combustion units unless a heat exchanger can be included. In reality, it appears that, for many conditions, substantial savings can be realized with a sealed combustion system without a heat exchanger, and that a sealed combustion system never creates a penalty in a practical structure, with or without a heat exchanger.

A further point worthy of consideration and comment by the authors is the possible effects of the low-temperature ambient environment that the oil burner can experience when supplied with unheated outside-air. The firing rates of nozzles used in gun-type oil burners often are sensitive to the oil temperature, with higher mass delivery at colder oil temperature. Where the combustion air adjustments are set to be close to stoichiometric during equilibrium firing, an upward shift in firing rate could increase smoke and pollutant emissions. There is the further question of ice crystals in the oil line or pump with normal amounts of moisture in the fuel system. These could result in filter clogging.

Both the low-temperature effects are minimized if there is some tempering of the outside air delivered to the burner. A heat exchanger between the flue gas and combustion air accomplishes some tempering of outside air, but not during the critical burner start-up portion of the cycle. For this purpose, the indoor-air offers a more nearly constant nozzle environment throughout the cycle. It also minimizes the flue condensation problem mentioned earlier.

While sealed combustion systems have some attractive characteristics, the problem of unequal wind-pressure effects and air intake and exhaust can upset the combustion air/exhaust flow system.

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Special care appears to be needed as to the design and location of intake and exhaust openings.

The authors comments on these questions of burner temperatures and wind effects would be useful.

M. Cooperrider[4]

My disagreement with the ASME paper relates only to the third paragraph in the left-hand column on page 4.

In this paragraph a sealed combustion system is described as one in which outdoor air is used for both combustion and draft control. "In this case the effective infiltration factor is $\phi = 1.00$. The combustion and draft control air are heated to stack temperature. All of this loss must be charged against the furnace." Quote from the paper.

It is with these last two sentences of the phrase that I disagree.

First a residential heating system must be defined. In the case of this paper it is the heating appliance and the building that is being heated.

Second a clear definition must be made as to what is considered a stack temperature. The way the first sentence is phrased the stack temperature means the temperature of the gases in the flue pipe after draft control (dilutant) air has been introduced.

It is the generally accepted practice of the heating industry to define stack temperature as the temperature of the combustion gases as they are discharged from the heating appliance and before any draft control (dilutant) air is introduced into the flue pipe.

From the standpoint of appliance efficiency only, it does not make any difference if the CO_2 and stack gas temperature are taken upstream or downstream of the draft control. In the case where the readings are taken downstream, an accurate method of determining the volume of draft control (dilutant) air must be used to determine the appliance efficiency. In either instance the draft control air has no effect on the appliance efficiency.

In the case of a heating system (Heating Appliance and Building) where both the combustion and draft control air are taken from within the building and both are heated from outdoor to room temperature then these losses must be charged against the system.

When a sealed combustion heating appliance is used in a heating system (Heating Appliance and Building) it is only the combustion air that can be charged against the system. The draft control air is not being heated by the system. It is only being heated by the flue gas loss of the appliance and cannot be charged against the system.

In either of the foregoing two examples the appliance efficiency would be the same. The system efficiency would be lower in the case where the combustion and draft control air are taken within the building. This is due to the tremendous volume of draft control air that escapes through the chimney during the heating season.

I have prepared a series of calculations to support my position, which can be obtained on request from the discusser.

[4] Chief Engineer, R. W. Beckett Corp., Abbe Branch, Elyria, Ohio.

Authors' Closure

Thank you for pointing out two areas that can stand some clarification; I don't believe that we disagree on any basic technical aspects. Let me explain:

1 The main emphasis of the paper is in considering the thermal efficiency of a complete heating system such that the involved seasonal efficiency and total operating cost can be compared to the cost of operating, e.g., a solar or an electric system (without a stack). The appliance or furnace efficiency is "only" a contributing part to the system efficiency.

2 We use the terms flue and stack to designate vent sections upstream and downstream of the draft control relief air opening (draft hood or diverter).

3 Under sealed combustion conditions, heating flue gases from outdoor to flue temperature requires the same energy (and therefore represents the same loss) as heating stack gases (i.e., flue gases + draft control air) from outdoor to stack temperature, since the draft control air only dilutes the flue gases, but does not remove any energy from the furnace. (It probably reduces the stack gain G_S however.)

4 If confronted with an option to convert from indoor to outdoor air for combustion and draft control, we agree with you that taking draft control air from outside generally results in increased system efficiency and fuel savings. Taking combustion air from outside, however, only results in savings if it can be preheated with stack gases, but in increased losses if preheating is not possible or $\phi < 1$ initially.