A NOVEL FURNACE DESIGN UTILIZING A LOW TEMPERATURE PLASTIC CONDENSING HEAT EXCHANGER

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SUMMARY

The initial phase of a research and development program for the Consumers' Gas Co. and the Federal Department of Energy, Mines and Resources to design a condensing heat exchanger/gas fired residential air furnace has been completed. Progress to date has resulted in a novel design utilizing a relatively low temperature plastic material for the last stage heat exchanger. To utilize this low temperature plastic, a method of reducing the temperature of the flue gas entering the final heat exchanger was devised using a unique flue gas recirculation process.

Heat transfer calculations and pressure drop prediction methods have indicated that the design is sound and can easily be accomodated in a residential furnace with only moderate increase in cost and space requirements. The existing design is also well suited to incorporation as a retrofit package and this is also being pursued.

Based on the calculated performance, a condensing heat exchanger was sized, fabricated and installed on a conventional 80,000 BTU/hr input gas fired residential furnace. The initial experimental tests have given very encouraging results. Based on a final flue gas exit temperature of 85F with an excess air condition of 25%, these initial tests yielded a furnace efficiency of approximately 97%. Although combustion air preheat has not been employed in these initial tests, this feature is planned as part of the prototype design.

INTRODUCTION

In order to examine a specific furnace design, it is instructive to initially review the various general calculations that quantify the processes taking place on the combustion-flue gas side of a gas fired furnace. Therefore, this paper is divided into two parts. First, a set of generic calculations is presented which can be applied to any gas fired furnace. This set of calculations is a condensed version of material taken from Reference 1. Second, details of a novel, air based, condensing furnace design are presented. This furnace design incorporates a plastic condensing heat exchanger nested in a unique flue gas recirculation loop. The material presented includes both conceptual and quantitative operating and design information.

GENERAL FURNACE CALCULATIONS

The Combustion Process

Throughout these calculations natural gas will be treated as if it consisted of 100% methane, CH_4. A general chemical expression representing the combustion of methane is shown in equation 1.

$$CH_4+2\phi(O_2+3.76N_2) \rightarrow CO_2+2fH_2O_\ell+2(1-f)H_2O_v+7.52\phi N_2+2(\phi-1)O_2 \qquad (1)$$

where: ϕ quantifies the excess air. (% excess air = $100(\phi-1)$)
f is defined as the portion of the water vapour in the products that has condensed. In a conventional furnace, f=0.

Considering the furnace as a simple control volume, as shown in Figure 1, it is apparent that the condensed fraction of water vapour in the flue products, f, can be determined as a function of flue gas exhaust temperature and the amount of excess air supplied. The partial pressure of water vapour in the flue products, P_{H2O}, is shown in equation 2.

$$P_{H2O} = P_{TOT} \cdot \frac{2(1-f)}{1 + 2(1-f) + 7.52\phi + 2(\phi-1)} \qquad P_{TOT}=1 \text{ atm} \qquad (2)$$

For cases when the flue gas temperature is below the dew point, rearranging equation 2 and substituting the saturation pressure of water, P_{SAT}, in place of the partial pressure of water, P_{H2O}, enables one to solve for the fraction of water condensed as shown in equation 3.

$$f = 1 - \{[9.52(\phi-1)]/[2(\frac{P_{TOT}}{P_{SAT}} - 1)]\} \qquad (3)$$

Solving equation 3, the fraction water condensed from the flue gas, f, is shown in Figure 2 as a function of both excess air and flue gas temperature, T_{FG}. P_{SAT} can be found as a function of temperature in standard thermodynamic saturation steam tables. The dew point temperature of the flue gas, T_{DP}, can be found by examining Figure 2 at f=0 or by setting f=0 in equation 2. The result of this substitution is:

$$P_{SAT} = \frac{2}{9.52\phi + 1} \cdot P_{TOT} \qquad (4)$$

Using equation 4, one can use saturation steam table values to obtain the dew point temperature corresponding to P_{SAT}.

Performing an energy balance on the furnace control volume of Figure 1 yields equation 5.

$$\hat{h}_{CH_4} + 2\phi\hat{h}_{O_2} + 7.52\phi\hat{h}_{N_2} = Q + \hat{h}_{CO_2} + 2f\hat{h}_{H_2O_\ell} + 2(1-f)\hat{h}_{H_2O_v} + 7.52\phi\hat{h}_{N_2} + 2(\phi-1)\hat{h}_{O_2} \quad (5)$$

where $\hat{h}_i$ is the enthalpy per mole of the ith component.

In order to make use of equation 5, it was assumed that the methane enters the furnace at 70F (+21.1C) and combustion air is drawn from outdoors at 30F (-1.1C). Under these conditions the amount of thermal energy derived from the cooling of the flue products, per mole of methane consumed, was calculated by solving equation 5. Figure 3 shows the resulting data as a function of the flue gas temperature and excess air in the low temperature range. Equation 5 was solved using tabulated enthalpy data (2). The values of fraction water condensed, f, were calculated as outlined earlier. The efficiency scale shown in Figure 3 is based on equation 6.

$$\eta = (\dot{Q}/\dot{n}_{CH_4})/HHV_{CH_4} \qquad HHV_{CH_4} \cong 383 \times 10^3 \text{ BTU/lbmole} \quad (6)$$

where $\dot{n}_{CH_4}$ is the molal flow rate of methane per unit time and $\dot{Q}$ is the thermal energy release per unit time.

Thus, by measuring the bulk temperature of the flue gas leaving a condensing furnace and knowing how much excess air was used for combustion, it is possible to estimate the thermal efficiency of that furnace using Figure 3.

Design Flow Rates

In order to size the various heat exchangers studied it was decided that the furnace input should be fixed at $\dot{Q}_{in}$ = 80,000 BTU/hr (23.44 kW). In this case, the molal flow rate of methane during burner operation is 0.209 lbmole/hr (0.095 kgmole/hr) and the mass flow rate of methane is 3.35 lbm/hr (1.52 kg/hr).

The flow rate of combustion air is proportional to the variable "ϕ". For the case of 35% excess air or ϕ = 1.35, the molal flow rate of combustion air is 2.69 lbmole/hr (1.22 kgmole/hr). The mass flow rate of combustion air under this condition is 77.9 lbm/hr (35.3 kg/hr). Therefore, the total mass flow rate of reactants (or combustion products) is 79.4 lbm/hr (36.0 kg/hr)(@ ϕ=1.35).

Combustion Air Preheat

When considering the use of an additional condensing heat exchanger for the purpose of preheating combustion air (drawn from outdoors), it is of interest to quantify the rate at which the combustion air temperature will rise with respect to the temperature drop of the combustion products. The solution to this problem is not immediately obvious because consideration must be made for the condensation taking place on

the flue gas side. The relative temperature change of the two gas flows, dT_a/dT_{FG}, can be estimated by equating the thermal energy given up by the flue gas (both sensible and latent) to the thermal energy gained by the combustion air.

Assuming that the specific heats of air and flue gas are equal and that the ratio of the molal heat of vapourization of water to the molal specific heat of air is;

$$\hat{h}_{fg}/\hat{C}_{p,a} = 2691F \ (1495C) \tag{7}$$

It can be shown that (1);

$$dT_a/dT_{FG} = (5382 \cdot df/dT_{FG} - 1 - 9.52\phi)/9.52 \tag{8}$$

where df/dT_{FG} is in units of F^{-1}.

Assuming that the temperature change of the flue gas in the combustion preheat section is relatively small, it is reasonable to evaluate df/dT_{FG} at a representative point and to use that value over a range of combustion air temperatures and a small range of flue gas temperatures. Assume flue gas is taken from a condensing furnace at T_{FG} = 100F (37.8C) and combustion has taken place with 35% excess air (ϕ=1.35). Referring to Figure 2, f=0.58 (58% of the water has already been condensed.). The value of df/dT_{FG} is measured graphically from Figure 2, $df/dT_{FG} = 1/74\ F^{-1}$. Using equation 8, $dT_a/dT_{FG} = -4.6$.

Thus, if this flue gas is used to preheat combustion air, the air temperature will rise by approximately 4.6 degrees for each degree that the temperature of the flue gas drops. If combustion air must be heated through a 50F (27.8C) temperature rise, the flue gas temperature will drop from 100F (37.8C) to 89F (31.7C).

Reading Figure 3, it can be seen that the addition of this combustion air preheat would have increased the overall furnace efficiency by roughly one percentage point. Therefore, the savings presented to the furnace user, by the addition of this heat exchanger, will be about 1% of his heating bill.

The use of combustion air preheat also provides a practical benefit for furnace operation. By reducing the flue gas temperature as low as possible before the flue gas leaves the furnace (and removing as much water as possible) the problem of ice formation at or near the outside wall exhaust port is correspondingly reduced.

DESIGN OF THE AIR BASED HIGH EFFICIENCY FURNACE

The Conceptual Design

As part of the initial design consideration, possible methods of cooling the flue gas were considered. It was recognized that the flue gas and air flows must be arranged in an overall counterflow arrangement in order to consistently be able to cool the flue gas below its dew point. This requirement suggested a sectioned heat exchanger or a multiple heat exchanger arrangement. It was also recognized that this concept corresponds well to the shift from radiation to convection

dominated heat transfer and to the shift from non-corrosive to corrosive flue products as the flue gas is progressively cooled. In fact, this situation is well suited to a two heat exchanger system, each being designed for a specific task.

The furnace design presented here is such that the flue gases are first cooled in a conventional furnace heat exchanger (primary heat exchanger) and then are cooled further into the condensing regime in a plastic heat exchanger. Finally, the flue gases are cooled using a combustion air preheat, plastic, condensing, heat exchanger. These plastic heat exchangers are to be constructed of an extruded sheet material called COROPLAST (10% polyethylene, 90% polypropylene). The use of plastic offers a resistance to possible condensate corrosion that few metals can duplicate.

The configuration of the Coroplast extruded sheet is similar to that of corrugated cardboard. Each sheet consists of two thin walls separated by a series of parallel ribs. The end view of this construction is shown in Figure 4 (3). A simple and compact heat exchanger can be constructed from Coroplast by stacking a series of sheets together such that the flow passages of each sheet run perpendicular to the flow passages of the two adjacent sheets. Once connecting ducts are added to four sides of the resulting block, a cross-flow heat exchanger is complete.

Coroplast has a maximum service temperature (at which softening begins) of 250F (121C). The temperature of the flue gas exiting the primary heat exchanger is to be well above this maximum service temperature. Thus, a unique recirculation loop was designed in order to protect the plastic heat exchanger against overheating. The operation of this recirculation loop is illustrated in Figure 5.

Recirculation Flow Calculations

The purpose of the recirculation loop is to mix a sufficient amount of the cool saturated flue gas (which exits the condensing heat exchanger) with the relatively hot flue gas exiting the primary heat exchanger in order that the temperature of the mixed flow entering the condensing heat exchanger, T_5, does not exceed 250F (121C). In order to calculate the required amount of flue gas recirculation, it is useful to use the molal flow rate of CO_2 to quantify the overall flow of constituents at various points in the furnace. The molal flow rate of CO_2 is denoted $\dot{n}_{CO2,i}$ at the ith state point. Performing an energy balance at the mixing junction upstream of the condensing heat exchanger, it can be shown that the ratio of the molal flow of CO_2 in the recirculation flow to the molal flow of CO_2 leaving the primary heat exchanger is given by equation 9. $H(T_i)$ is the enthalpy of the flue gas at the ith state point (per mole CO_2).

$$\frac{\dot{n}_{CO2,R}}{\dot{n}_{CO2,4}} = \frac{H(T_4) - H(T_5)}{H(T_5) - H(T_R)} \tag{9}$$

The ratio of molal flow rates shown in equation 9 can be converted to a similar ratio of mass flow rates using the molecular weights of the two

gas flows and by accounting for the water which is condensed from the combustion gas. This conversion is shown in equation 10.

$$\frac{\dot{m}_R}{\dot{m}_4} = \frac{\dot{n}_{CO2,R}}{\dot{n}_{CO2,4}} \cdot \frac{\hat{M}_R}{\hat{M}_4} \cdot \frac{1 + 9.52\phi - 2f_7}{1 + 9.52\phi} \tag{10}$$

The molecular weight of the flue gas at the ith state point, $\hat{M}_i$, can be calculated (1) as a function of the molecular weights of the various constituents, f_i, and ϕ.

Choosing design temperatures of T_5=250F (121.1C) and T_7=T_R=96F (35.6C), equation 9 was solved as a function of the exit temperature of the primary heat exchanger, T_4, and the resulting curve is shown in Figure 6. A calculation assuming 35% excess air (ϕ=1.35) and f_7=0.64 (see Figure 2) yields the result, $\hat{M}_R/\hat{M}_4$=1.04. The final term of equation 10 is equal to 0.9. Therefore, by equation 10, the ratio of recirculation gas mass flow to combustion product mass flow at state point 4 can be taken as being about 94% of the analogous mole ratio read from Figure 6. For example, if the flue gas exits the primary heat exchanger at T_4=500F (260C) then Figure 6 shows that (for 35% excess air) for each mole of CO_2 produced in combustion, about 1.9 moles of CO_2 are recirculated. Thus, the mass flow rate of recirculated flue gas is about 1.8 times the mass flow rate of flue gas leaving the primary heat exchanger. The mass flow rate of gas entering the condensing heat exchanger is about 2.8 times the rate at which gases leave the primary heat exchanger.

Sizing the Condensing Plastic Heat Exchanger

In order to size the condensing plastic heat exchanger, the inlet and outlet flue gas design temperatures of this heat exchanger were assumed to be 250F (121.1C) and 100F (37.8C) respectively. Return air enters this heat exchanger at 70F (21.1C). These flue gas design temperatures (T_4=500F (260C)) dictate that 76% of the burner input energy is transferred in the primary heat exchanger and 18% of the input energy is released in the condensing heat exchanger. Therefore, if the total temperature rise of air passing through the furnace is 85F (47C) then the temperature rise of the air passing through the condensing heat exchanger is 16.3F (9.0C).

In order to size the condensing heat exchanger, it was analysed in two sections. First, the section where the flue gas is above the dew point was sized and then the section where condensation takes place was sized. The dew point temperature was estimated by noting that the constituents entering the plastic heat exchanger are:

molal flow at state point 5 (per mole CH_4 combusted) =

$$CO_2 + 2H_2O_v + 7.52\phi N_2 + 2(\phi-1)O_2$$
$$+(\dot{n}_{CO2,R}/\dot{n}_{CO2,4})(CO_2 + 2(1-f_7)H_2O_v + 7.52\phi N_2 + (\phi-1)O_2) \tag{11}$$

Assuming 35% excess air, the dew point temperature of this mixture is about 110F (43C). The design temperatures used to size the dry and wet portions of the condensing heat exchanger are shown in Figure 7.

Since the temperature change of the air passing through the dry portion of the heat exchanger is relatively small, it is valid to calculate heat transfer based on the log mean temperature difference (LMTD) even though the heat exchanger is not arranged in counterflow (LMTD ≅ 89F (50C)). The rate of heat exchange in the dry section of the heat exchanger, $\dot{Q}_d$, can be expressed as:

$$\dot{Q}_d = A_{w,d} U_{w,d} (\mathrm{LMTD}) \tag{12}$$

where $A_{w,d}$ is the wall area of the dry portion of the heat exchanger and $U_{w,d}$ is the heat transfer coefficient from flue gas to air based on $A_{w,d}$. The heat transfer coefficient, $U_{w,d}$, was estimated using equation 13.

$$U_{w,d} = ((h_{FG}(1+\eta_f A_f/A_w))^{-1} + (h_a(1+\eta_f A_f/A_w))^{-1})^{-1} \tag{13}$$

where h_{FG} is the convective film heat transfer coefficient on the flue gas side, h_a is the convective film coefficient on the air side, A_f/A_w is the ratio of fin area to wall area for Coroplast (A_f/A_w = 0.68) and η_f is the fin efficiency of the Coroplast ribs (η_f = 0.24 with thermal conductivity of Coroplast taken as 0.09 BTU/hr ft F (0.15 W/mK)).

It was assumed that the flow on both sides of the heat exchanger is fully developed and laminar. The assumption of laminar flow can easily be shown to be true. The ducts between the Coroplast ribs have a width to height aspect ratio of 1.47. In this case the Nusselt number, Nu, for both flows is equal to 3.4 (4). The convective heat transfer coefficient, h, expressed in terms of the Nusselt number is:

$$h = Nu \cdot k/D \tag{14}$$

where k is the thermal conductivity of the fluid in question and D is the hydraulic diameter of the flow passage (D = 1.41×10^{-4} ft (4.3×10^{-3}m) for Coroplast). Making the approximation that the conductivity of both the flue gas and air is equal to the conductivity of dry air (k = 0.016 BTU/hr ft F (0.028 W/mK)), the convective heat transfer coefficients in the dry section of the heat exchanger are, $h_{FG} = h_a$ = 3.88 BTU/hr ft² F (22 W/m²K), and $U_{w,d}$ = 2.1 BTU/hr ft² F (12.7 W/m²K).

Using the inlet and outlet temperatures of the dry section of the heat exchanger, the rate of heat transfer, $\dot{Q}_d$, was estimated as shown in equation 15.

$$\dot{Q}_d = \dot{m}_5 C_{p,FG} (T_{in} - T_{out}) \tag{15}$$

Knowing that $\dot{m}_5$ is approximately 2.8 times as great as $\dot{m}_4$ and assuming $C_{p,FG}$ to be equal to the the specific heat of air, equation 15 yields, $\dot{Q}_d$ = 7380 BTU/hr (2.16 kW). Substituting this result into equation 12 yields, A_d = 37 ft² (3.4 m²).

Next, the size of the wet or condensing portion of the plastic heat exchanger was estimated. The heat transfer coefficient between condensing flue gas and air can be calculated as in equation 13 after having evaluated an effective wet film coefficient and a wet fin efficiency on the flue gas side of the heat exchanger.

For the purpose of estimating an effective wet film heat transfer

coefficient on the flue gas side of the heat exchanger it was assumed that the saturated flue gas would behave in a manner similar to that of saturated air. Calculations were made according to the method presented in reference 5 for the analysis of condensing air cooling coils. In this analysis the effective wet side film coefficient, $h_{o,w}$, is given by:

$$h_{o,w} = ((C_{p,a}/b_w h_{c,o}) + (y_w/k_w))^{-1} \tag{16}$$

where $C_{p,a}$ is the specific heat of moist air, b_w is a coefficient from Figure 12.18 of reference 5 evaluated at the water film temperature, $h_{c,o}$ is the convective film coefficient when no condensation takes place (3.88 BTU/hr ft² F (22 W/m²K)), k_w is the thermal conductivity of water and y_w is the water film thickness.

Considering a mean air temperature of 78F (25.6C) and a mean flue gas temperature of 106F (42C), the b_w coefficient of equation 16 was evaluated assuming a condensate film temperature of 100F (37.8C)(b_w = 1.8 BTU/lbm F). Solving equation 16 using nominal values of $C_{p,a}$, k_w and assuming y_w=4.13x10^{-3} ft (1.26x10^{-3} m)(worst case) yields $h_{o,w}$ = 22 BTU/hr ft² F (125 W/m²K). Calculating a wet side fin efficiency using $h_{o,w}$ instead of the dry film coefficient still yields a fin efficiency of 0.24. Solving for the overall wet heat transfer coefficient, $U_{w,w}$, using an equation similar to equation 13 yields $U_{w,w}$ = 3.83 BTU/hr ft² F (21.7 W/m²K).

In view of this result, it was noted that the controlling thermal resistance in the wet section of the plastic heat exchanger is the air side convective film resistance. Therefore, the wet section of the heat exchanger was sized neglecting the resistance to heat transfer on the flue gas side. Thus, $U_{w,w}$ = 4.5 BTU/hr ft² F (25.5 W/m² K).

An estimate of the mean temperature difference between the flue gas and air sides of the heat exchanger was made as the difference between the mean air temperature of 78F (25.6C) and the mean flue gas temperature of 106F (42.2C). Thus, ΔT_m = 28F (15.6C).

Noting that 18% of the burner input energy is transferred in the plastic heat exchanger and that 9.3% of the burner input is transferred in the dry portion , it follows that 8.7% of the input energy must be exchanged in the wet portion of the plastic heat exchanger $\dot{Q}_w$ = 7020 BTU/hr (2.06 kW)). If the rate of heat transfer in the wet section can be expressed by equation 17;

$$\dot{Q}_w = A_w U_{w,w} (\Delta T_m) \tag{17}$$

then it is possible to solve for the wall area of the wet portion of the heat exchanger, A_w, giving A_w = 56 ft² (5.2 m²). The total heat exchanger wall area required is 93 ft² (8.6 m²). The volume of Coroplast heat exchanger needed to provide 93 ft² of heat exchanger wall area is 1.3 ft³ (0.037 m³).

Sizing the Combustion Air Preheat Plastic Heat Exchanger

The combustion air preheat heat exchanger can be fabricated simply by extending the length (in the direction of flue gas flow) of the plastic heat exchanger examined in the previous section. In addition, duct

connections must be made such that part of the face area seen on the air side of the heat exchanger accepts house return air and the remaining part, nearest the flue gas exhaust end, accepts combustion air.

Earlier, it was shown that flue gas used to preheat combustion air by 50F (27.8C) would be cooled by approximately 11F (6.1C). However, the heat exchanger in question carries about three times the flue gas flow as the one considered earlier. Therefore, the drop in flue gas temperature will only be about 4F (2.2C) and the preheat exchanger flue gas inlet and outlet design temperatures are 100F (37.8C) and 96F (35.6C) respectively. These temperatures are shown in Figures 5 and 7.

Since the flue gas temperature drop is relatively small, this heat exchanger can be sized using a log mean temperature difference (LMTD = 38F (21.1C)). The rate of heat exchange in the preheat exchanger, $\dot{Q}_{ca}$, was calculated using the design air temperatures, giving $\dot{Q}_{ca}$ = 935 BTU/hr (274 W). Using the same flue gas to air heat transfer coefficient as used in the previous wet heat transfer analysis, the required preheat exchanger heat transfer area is 5.5 ft^2 (0.5 m^2).

Including the heat transfer area required for combustion air preheat, the total wall area required in the plastic heat exchanger is 98.5 ft^2 (9.1 m^2). This corresponds to a Coroplast heat xchanger volume of 1.4 ft^3 (0.039 m^3). If the heat exchanger is constructed such that the length of the air flow passages is 3.15 inches (8 cm), then the face area exposed to air flow would be 5 ft^2 (0.49 m^2). If this face of the plastic exchanger is square it would measure 27.5 inches (70 cm) along each side. The actual dimensions that should be used for this heat exchanger depend upon many factors including the size and shape of the furnace in which it is to used and the pressure drop that can be tolerated in each of the gas streams.

Plastic Heat Exchanger Leakage

Inherent in the design of the Coroplast heat exchanger (as in any heat exchanger) is the possibility of leakage between the two gas streams. Although the various layers of Coroplast may be sealed carefully, it is wise to plan a design which allows for the possible occurence of heat exchanger leaks.

To protect against leakage of the flue gas into the house air stream it is proposed that the furnace blowers be placed such that the house air is moved in a forced draft and the flue gas is moved in an induced draft fashion. With this blower configuration, any leak occuring in the plastic heat exchanger will result in leakage of air into the flue gas flow.

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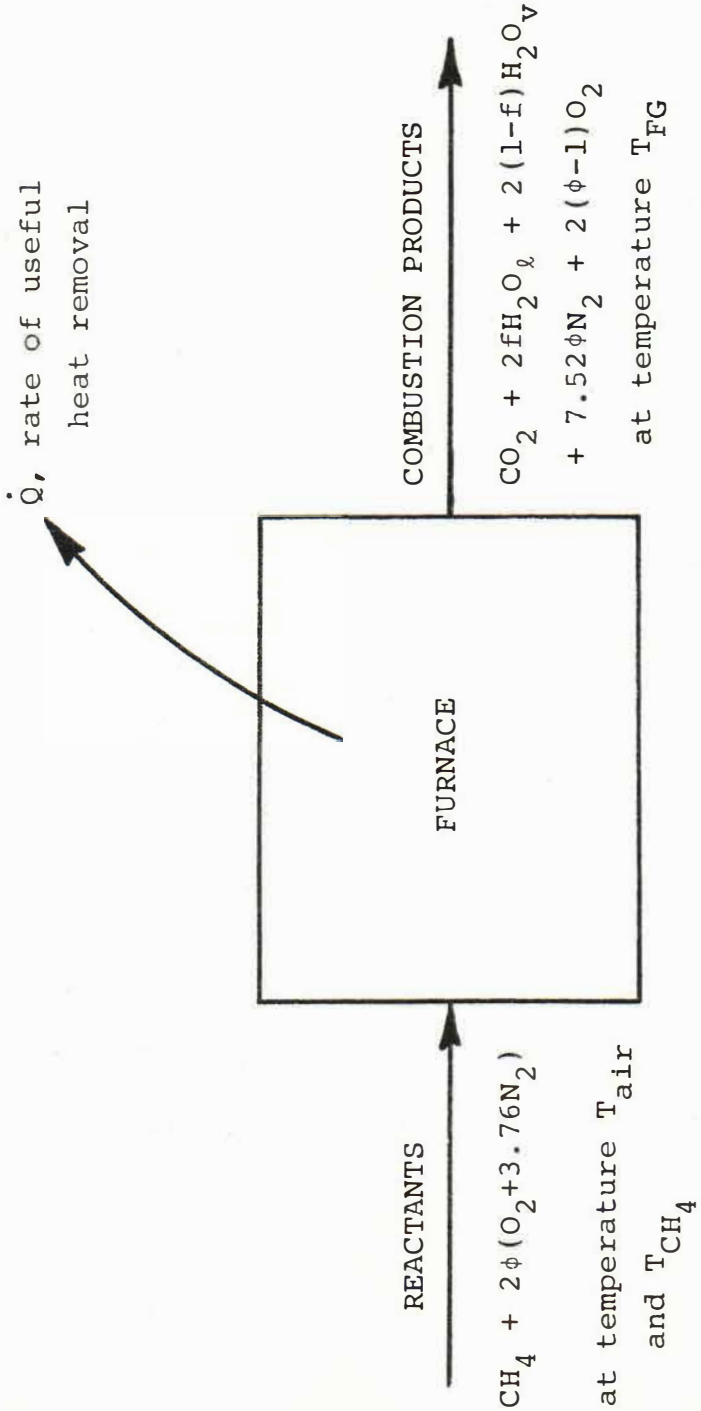


Figure 1: The Furnace Analysed as a Control Volume

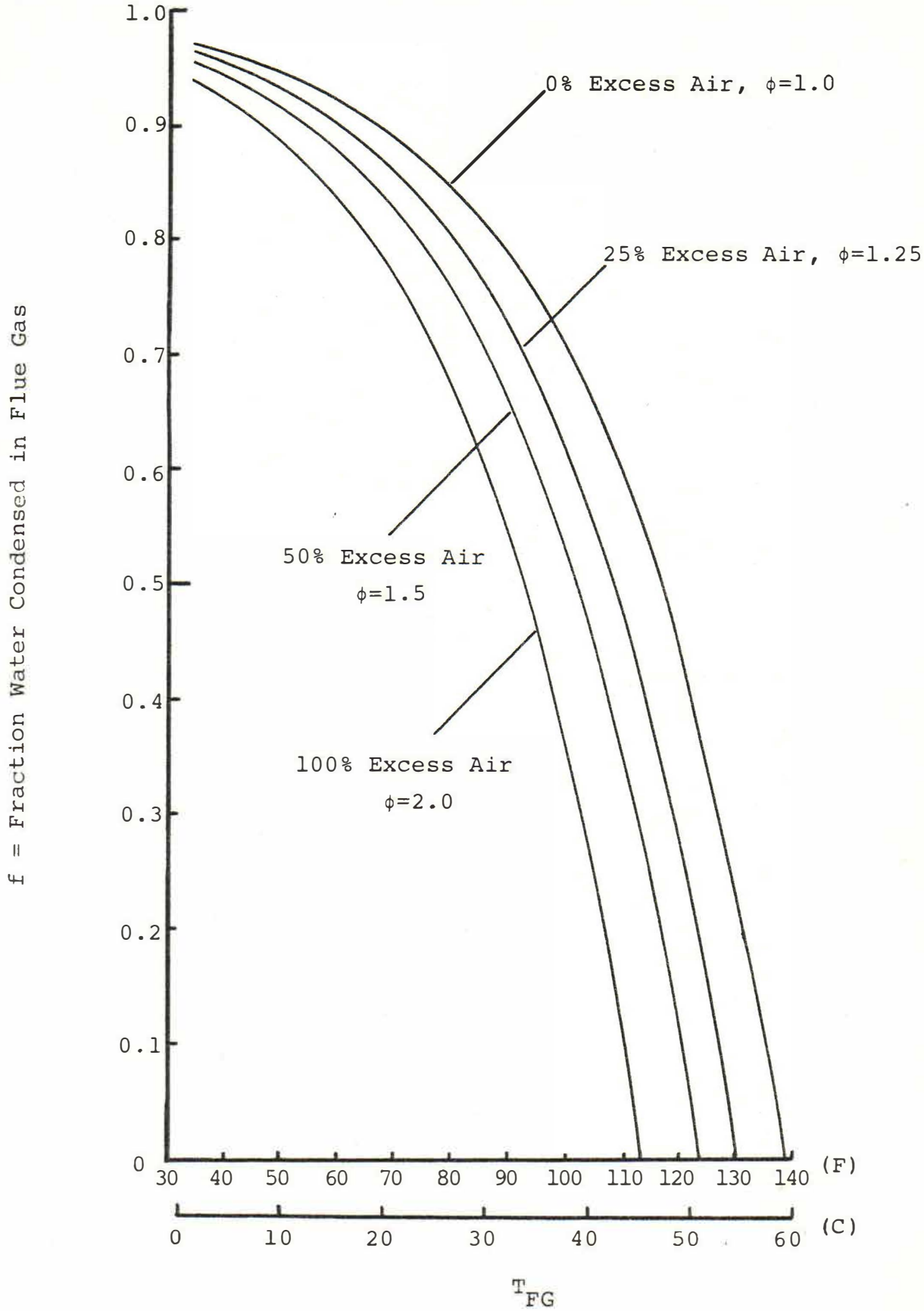


Figure 2: Fraction Water Condensed vs. Flue Gas Temperature

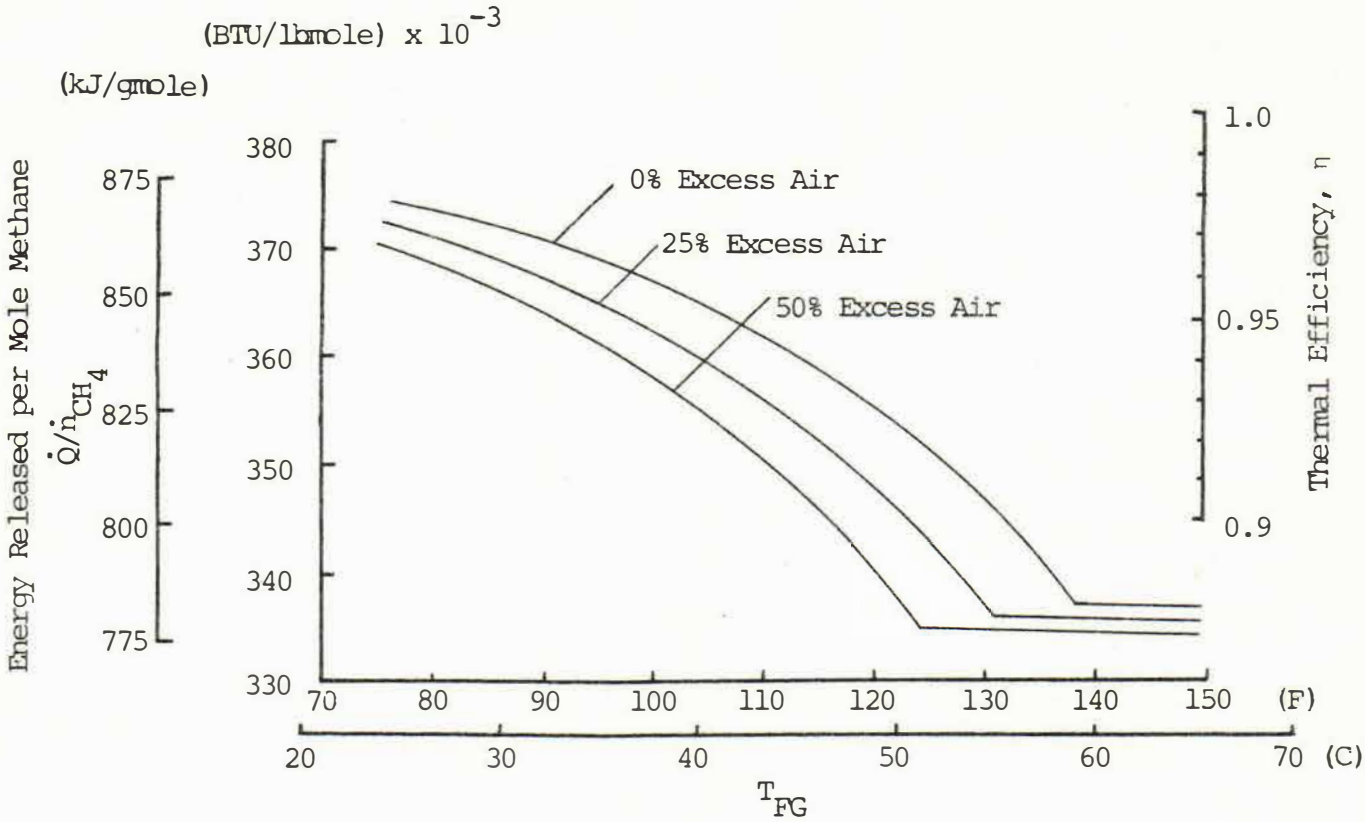
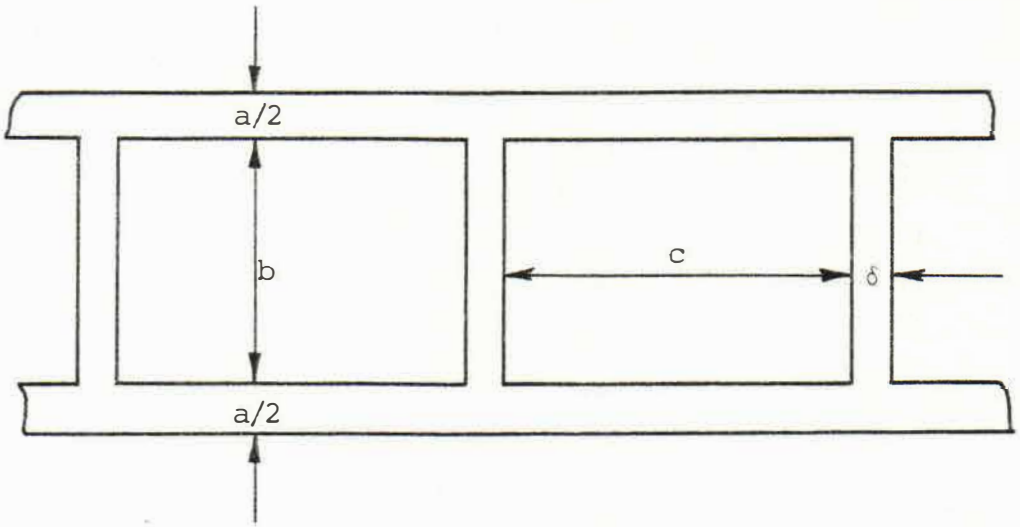


Figure 3: Thermal Energy Release vs. Flue Gas Temperature



$a/2 = 0.014$ inches $= 3.5 \times 10^{-4}$ m

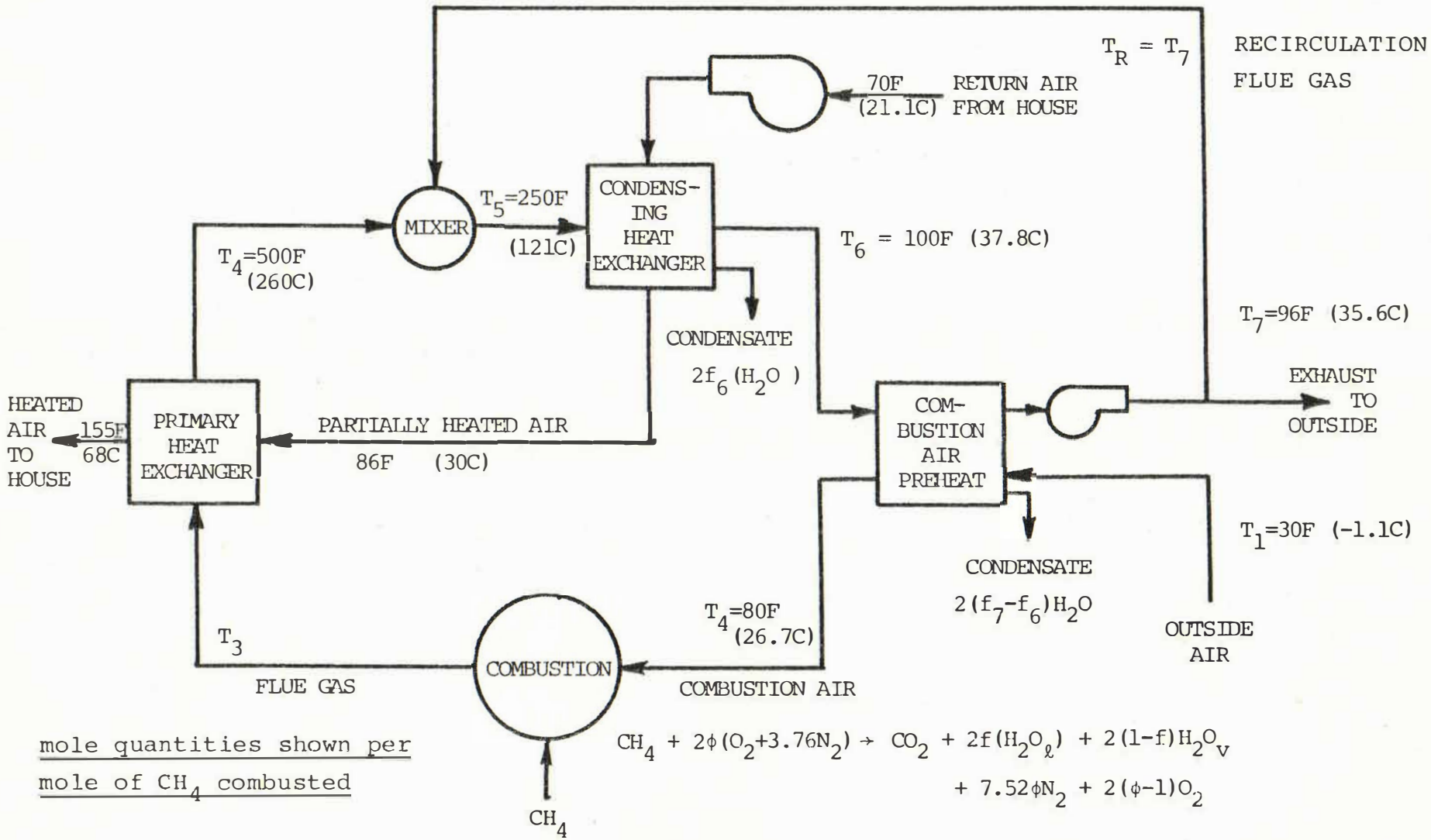
$b = 0.143$ inches $= 3.63 \times 10^{-3}$ m

$c = 0.210$ inches $= 5.34 \times 10^{-3}$ m

$\delta = 0.012$ inches $= 3.0 \times 10^{-4}$ m

Figure 4: End View of Coroplast Extruded Sheet

STATE 5: $CO_2+H_2O_v+7.52\phi N_2+2(\phi-1)O_2+\frac{\dot{n}_{CO2,R}}{\dot{n}_{CO2,4}}(CO_2+2(1-f_7)H_2O_v+7.52\phi N_2+2(\phi-1)O_2)$



mole quantities shown per mole of CH_4 combusted

$$CH_4 + 2\phi(O_2+3.76N_2) \rightarrow CO_2 + 2f(H_2O_\ell) + 2(1-f)H_2O_v + 7.52\phi N_2 + 2(\phi-1)O_2$$

Figure 5: High Efficiency Furnace with Flue Gas Recirculation

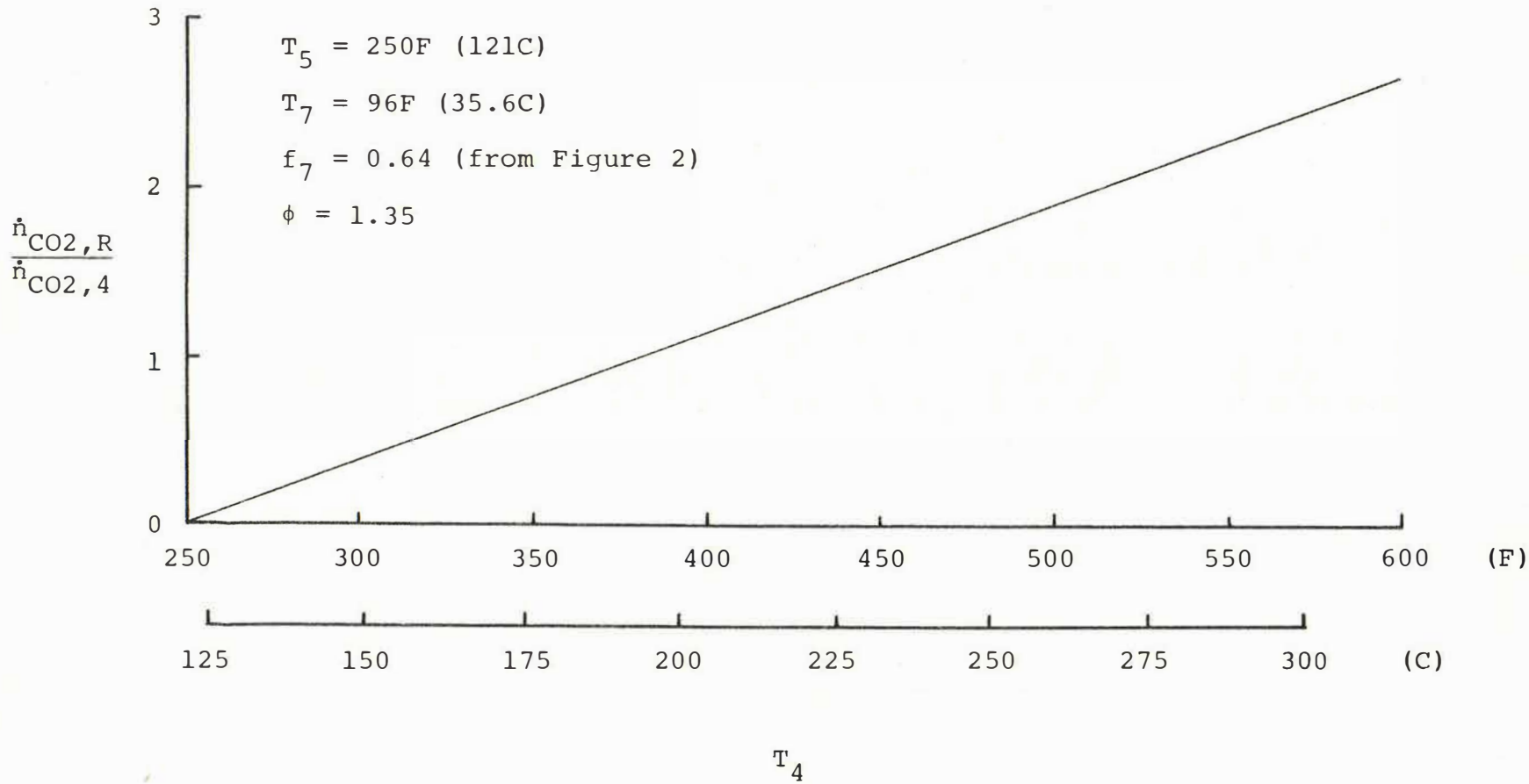


Figure 6: Flue Gas Recirculation as a Function of Primary Heat Exchanger Exit Temperature

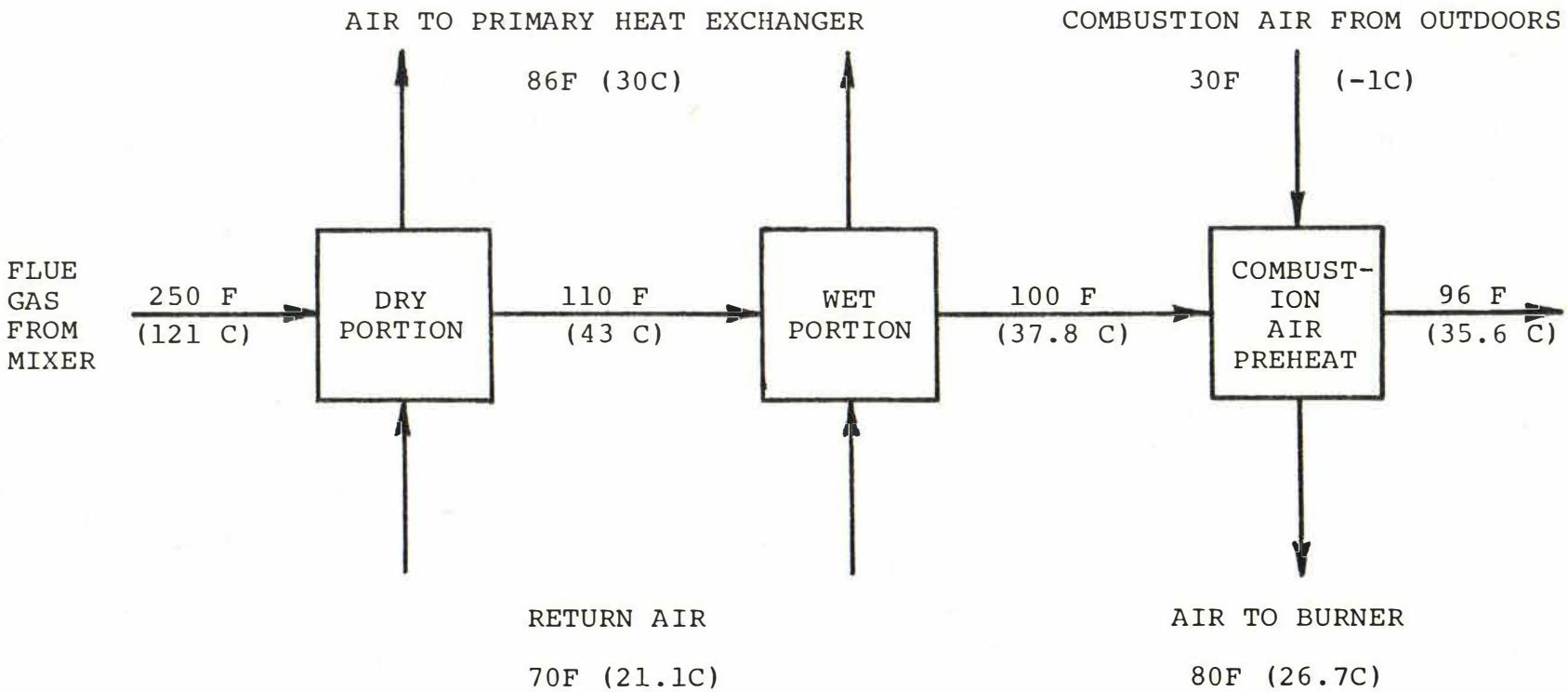


Figure 7: The Dry and Wet Portions of the Condensing Heat Exchanger