Advanced Venting Solution to Enable Broad Market Acceptance of High Efficiency Biofuel/Oilheat Appliances

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Executive Summary

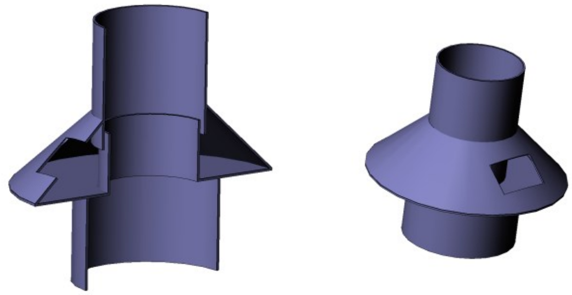
As heating appliances become more efficient, exhaust gas temperature decreases. For very high efficiency, condensing appliances venting using plastic flue pipe is common practice. For appliances which are high efficiency but not condensing current practice is to chimney vent. These near-condensing appliances are significantly lower in cost than condensing appliance, require less maintenance, and have longer life. With chimney venting of these appliances, there is the potential for water condensation in the chimney, leading to water flow into the building and chimney damage. The potential for this is an impediment to market adoption of high efficiency, non-condensing appliances. Often, an expensive chimney liner is required. This project is focused on the use of dilution venting to reduce the flue gas temperature of such appliances, allowing use of low cost plastic vent materials and side wall venting. In addition, the injection of cold air into hot flue gas has the potential to create a negative operating pressure or draft in the heating appliance. Appliances operating under negative pressure have little potential for combustion products to leak out into the living space. Some appliances are designed for operation at a slight positive pressure but these require carful design and maintenance to prevent such leakage. This project, then, had two goals: reduction of flue gas temperature to a level compatible with low cost plastic materials and the creation of negative pressure in the appliance. The approach explored in this work involves a fan which injects outside air into the flue. This approach leads to a fan which is not subject to the corrosive effects of flue gas. The use of outside air prevents increased infiltration heat load in the building.

The work in this program has shown that dilution venting can be implemented with oil firing without creating the potential for water condensation downstream of the mixing device. Under very cold outdoor conditions a low dilution air flow should be used to ensure that such condensation is avoided. To achieve good draft with lower air flow, a higher eductor jet velocity would be needed, which requires a smaller jet open area.

The project scope of work included modeling of the dewpoint of the mixed air and flue gas to identify temperature and mixing ratios where downstream condensation may be possible; simple modeling of several eductor-injector geometries to predict level of draft produced; computational fluid dynamics modeling of selected geometries; and experimental work to evaluate actual performance of specific prototype venters.

Two of the primary approaches (annular injector geometry and central-jet geometry) developed in this work are expected to have potential for field application.

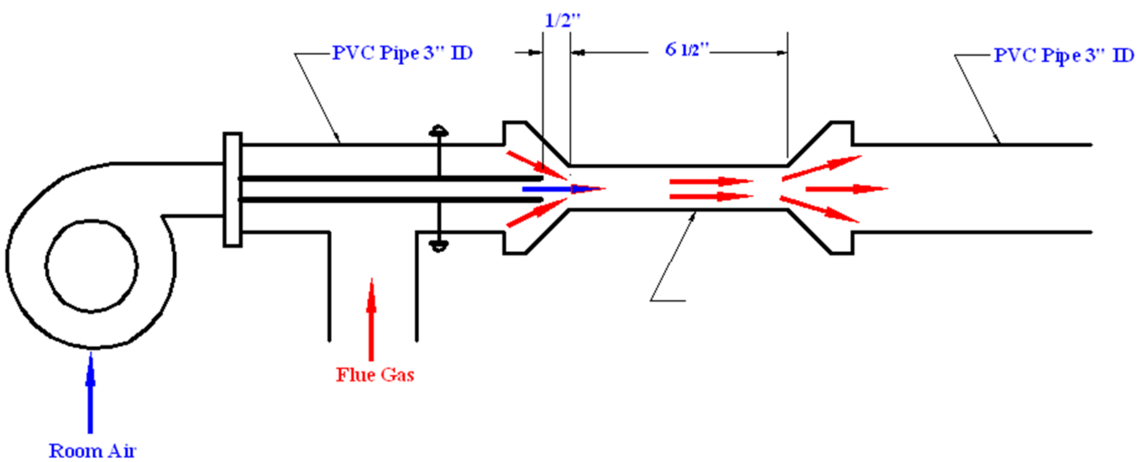
The annular injector geometry, illustrated in Figure ES1, offers the advantage of rapid cooling of the duct wall downstream of the eductor and so a rapid transition to lower cost venting materials. However, this geometry does not perform well under high backpressure conditions associated with long vent lengths. Further, with the annular venter achieving a small jet open area may create challenges achieving dimensional tolerance.



ES Figure 1 Eductor Arrangement with Annular Injector Geometry

Under conditions of relatively low back pressure the annular eductor system was shown to produce good draft levels and achieve mixed gas temperatures below 150 F.

For high back pressure conditions associated with long vent lengths, a central-jet geometry with a relatively small diameter mixing tube was found to produce the draft level needed. There is a concern with this more-restrictive arrangement that startup transient chamber pressure and smoke may be a concern. In work to optimize all parameters it was found this could be mitigated. This is an area which should be considered in general application of this approach. The long vent configuration used here included 30 feet of vent pipe with 5 elbows.



ES Figure 2 Eductor Arrangement with Central Jet Geometry

Introduction

There is significant potential to reduce fuel use by heating equipment through improved thermal efficiency and better matching of equipment sizing and load. The typical residential heating system discards combustion products at temperatures in the 400-600 F range translating to energy loss through the venting system in the 15-20% range. System oversize factors range 1.5 to 5.0 (capacity/peak load) leading to high part-load energy losses. A recent BNL study [1] has shown that the energy savings potential, associated with replacing older hydronic systems with modern systems can reduce fuel use by 20-25% up to 38% in some cases.

High efficiency, condensing boilers, furnaces, and water heaters achieve a much higher level of performance than conventional systems. These systems, however, are significantly higher cost and this limits market penetration. Further, these systems cannot operate in a condensing mode in many common applications (e.g. baseboard hydronic systems) where system temperatures exceed the dewpoint. Where condensing systems can be used, venting of the low temperature combustion products is typically done through low cost, plastic (PVC) pipe.

With more common, non-condensing systems, as the efficiency level is increased and flue gas temperature decreases, there is increasing potential for condensation of water vapor in the chimney. This condensation leads to corrosion and damage of the vent components and the potential for vent system collapse and spillage of combustion products into the home. This is a very significant concern for manufacturers who have little control over the chimney systems into which their appliances will be vented. In the recent national debate over required minimum efficiency levels for boilers and furnaces, issues over chimney condensation potential and the high cost of condensing systems were raised strongly as factors which prevent raising minimum efficiency requirements even to the mid-80% range.

The problems of condensation in ventilation systems at high efficiency levels can be avoided through the use of dilution venting. With this approach, flue gas is diluted with ambient air, reducing its temperature and then the mixture is vented using low cost, condensate-resistant plastic pipe. With the addition of the ambient air, dewpoint is lowered relative to the mix temperature, and potential for downstream condensation is lowered.

This has been demonstrated with gas-fired water heaters where room air is used in some products, while this imposes an additional infiltration energy loss on the building, the annual burner run hours are not very large. At least one gas-fired water heater uses outside air for this dilution.

The application of dilution venting to oil-fired boilers and furnaces, could resolve venting problems that exist even with conventional equipment and promote higher efficiency appliances with lower system cost and safer venting.

The goal of this project is the development of a low cost venting solution which can eliminate impediments to the deployment of high-efficiency, non-condensing boilers and furnaces fired with liquid fuels (heating oil and biofuel/heating oil blends). The problems of condensation in chimney at high efficiency levels can be avoided through the use of dilution venting. With this approach, flue gas is diluted with ambient air, reducing its temperature and then the mixture is vented using low cost, condensate-resistant plastic pipe. With the addition of the ambient air, dewpoint is lowered relative to the mix temperature, and potential for downstream condensation is lowered. With the use of outdoor air for this dilution, an additional infiltration heating load is not imposed on the building.

A second requirement for the dilution venting system developed here is to enable operation of the boiler or furnace under negative pressure. Operating under negative pressure relative to the indoor space ensures that any leakage that occurs will be from the room into the combustion passages, eliminating the potential for reduction in indoor air quality due to flue gas spillage.

This concept of dilution to reduce flue gas temperature and/or create draft has been used in other applications. The A.O. Smith Company uses a concept of this type on gas-fired, moderate efficiency water heaters [2]. In this product both flue gas and dilution air pass through an induced draft fan located at the exhaust port of the water heater.

Some gas-fired water heaters currently on the market use a dilution venting system to specifically allow venting with low cost plastic materials even with lower efficiency appliances. Figure 1 shows one configuration.

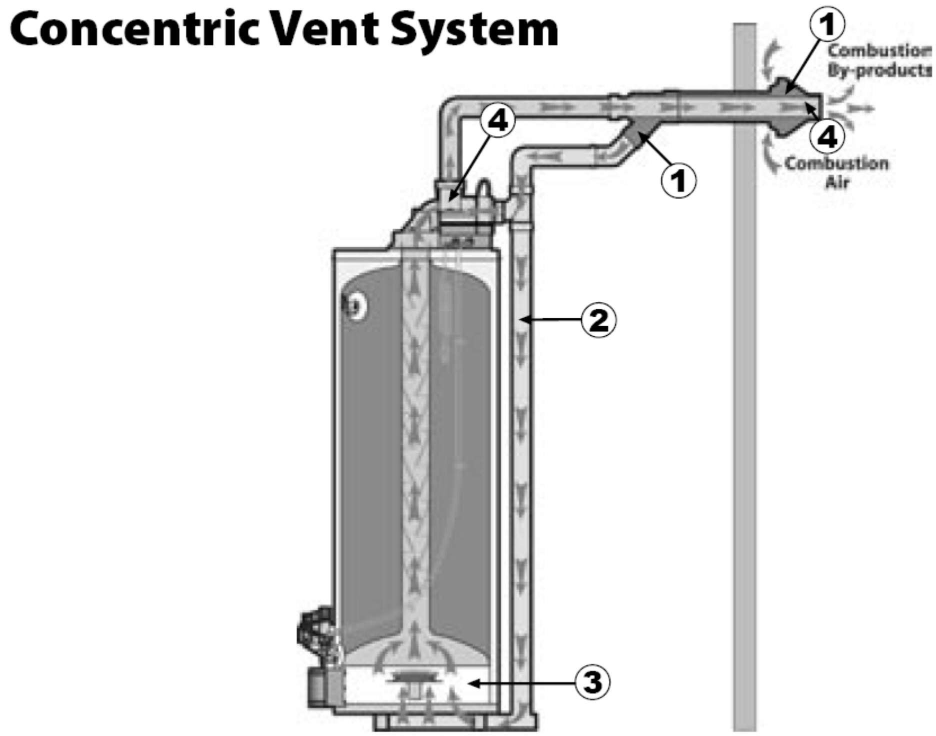


Figure 1 Illustration of the venting arrangement used in a U.S. Craftmaster water heater.

For the appliance shown in Figure 1, outside air comes in through port 1 outside of the home and flows in though a concentric pipe arrangement. This air splits at the appliance – part of it is directed to the burner and part is directed to the exhaust fan intake where it is mixed with flue gas. Relative to the eductor approach this has the negative feature of having flue gas pass through the fan.

One concept which received considerable attention during this project is the use of an eductor to mix clean air from the exhaust of a fan with the flue gas. The eductor would create the draft required. This approach allows the fan to be out of the exhaust gas stream, reducing the potential for corrosion and fouling.

The Quickdraft Company produces an eductor-based venting product for industrial applications including boilers and process applications such as perchloric acid hoods [3]. The products are heavier than seems reasonable for the small boiler application and the tolerances of construction are not very high. Figure 2 provides an illustration. One of these eductors in a small size was obtained and evaluated for use in the development work of this project. In this design, the blower air flow originates in the tee of a straight run, producing a perimeter flow in the straight run.

Figure 2 Illustration of the Quickline draft inducer for industrial applications made by the Quickdraft Company

A draft inducer product is available in Europe which has some relevance to this project. It is intended to mount outdoors, on the top of a chimney. An outdoor fan is used to inject air in an annular eductor arrangement similar to that used in the Quickdraft product discussed above. It can be applied to gas, oil, and wood-fired systems. The "wide-open" central flow passage reportedly does not impose any basic flue gas flow restrictions [4].

Other Applications - The Dyson Company has recently introduced a tabletop fan product to be used for comfort which uses an interesting and related approach. Like the Quickline product is has an annular shaped jet which induces air flow. The product, illustrated in Figure 3, is termed a bladeless fan (the fan with its blades is of course in the base). Other companies are producing related products.

Figure 3 Dyson table top "bladeless" vent fan

Two focus cases have be considered for dilution venting of oil-fired, high efficiency boilers: 1) chimney vent and 2) sidewall vent

1. Chimney Vent

- Gas temperature entering – 300 F
- Gas temperature leaving (to plastic vent) – 160 F
- Firing rate – 0.65 gph
- Excess air – 35%
- Required draft at breech – 0.025"
- Air inlet (for pressure drop calculation) – 20' with 4 elbows, 2" PVC
- Vent connector (from dilution device to wall) – 15', 2 elbows
- Chimney –plastic lined, 30'

2. Sidewall Vent

- Gas temperature entering – 300 F
- Gas temperature leaving (to plastic vent) – 160 F
- Firing rate – 0.65 gph
- Excess air – 35%

- Required draft at breech – 0.025”
- Air inlet (for pressure drop calculation) – 20’ with 4 elbows, 2” pvc
- Vent connector (from dilution device to wall) – 20’, 4 elbows

with a 40 mph direct sidewall wind, maintain -0.015” H_2O at the breech

Two primary dilution vent design approaches have been included in the project. The first is an eductor based approach in which outside air is injected into the flue to create draft and cool the flue gas. The second is a fan which combines flue gas and outside air at it’s inlet. As discussed above the first option offers the potential to have only clean air passing through the fan blades. The work has involved two main parts, design calculations and experimental. Details of the approach and results for both of these parts are presented in the following sections.

Approach to Design Calculations

Temperature and Dewpoint of the Diluted Flue Gas

With either of the two main approaches considered to implement dilution, a key concern is the dewpoint of the gas leaving the mixing device. Too high a dewpoint could lead to condensation in the flue and/or at the vent terminal outside. Condensation outdoors under winter conditions could lead to ice formation and flue blockage. To evaluate the potential for this concern a routine was setup to calculate the mixed temperature and dewpoint of the diluted flue gas as a function of the outside air temperature and relative humidity, flue gas temperature and mole fraction water vapor, and the mass ratio of dilution air to flue gas. Atmospheric pressure is also an input parameter to the calculations.

Antoine’s equation was used to calculate the incoming air water vapor mole fraction from the relative humidity and pressure. This commonly used equation can generally be expressed as:

$$\log_{10} p = A - \frac{B}{C + T}$$

Where:

p = pressure at saturation

T = absolute temperature

A,B,C = constants in Antoine’s equation in appropriate units

The constants used here were adapted for specific temperature ranges from the National Institute of Standards and Technology (NIST) Chemistry WebBook [5]. The correlations of Hyland and Wexler [6] were also used.

It was assumed that the temperature of the air mixing with the flue gas is the same as that of the outdoor air, i.e. there is no tempering of the air by heat transfer from either the room or the flue gas. This is seen as a very conservative assumption as some tempering is likely but depends strongly on the air and flue gas ducting lengths and arrangements.

The mole fraction of water vapor in the flue gas is primarily a function of the fuel composition and the excess air level. Combustion air relative humidity is a secondary factor but this was also considered. Calculations were done for both oil and natural gas although most of the focus has been on oil-fired systems. In these calculations, no condensation of water from the flue gas was assumed. Condensation would occur only if the boiler were a condensing appliance and, if this did occur, it would lower the mix gas dewpoint and lower the potential for condensation to occur in the vent system.

The mixed gas temperature has been calculated from a simple heat balance with an assumption of constant specific heat.

<u>Calculation of an Eductor Dilution Device Pressure Rise</u> (draft created)

The dilution devices under consideration in this project need to both create draft in the boiler and move the flue gas through the downstream exhaust vent duct. These can be combined into a requirement to create a pressure rise across the device. For all of the geometries under consideration in this study, the principle by which these devices create this pressure rise is the same – momentum is transferred from the high velocity driving air flow stream into the low velocity flue gas stream. This momentum transfer ends as a higher pressure in the low velocity combined exhaust stream.

A very simple model of this was developed for use in this study. The simple model combines mass and heat balances with the impulse momentum principle. For this application this principle can be stated as: the sum of the external forces acting on a fluid control volume is equal to the net increase in momentum flux. In this case external forces are taken as pressure. Momentum flux is the product of mass flow and velocity and the net increase in momentum flux is the sum of the mass flow * velocity product over each of the leaving and entering streams.

Figure 4 illustrates a simple eductor vent and in this case the geometry is a central jet of cold air. Hot flue gas leaving the boiler enters the control volume on the left side and this is designated with the subscript 1. The area here is the cross section area of the inlet flue minus the blockage from the eductor air tube. Subscript 2 refers to the eductor flow as it enters the control volume. At this point $P_1 = P_2$. Subscript 3 refers to the mixed flow leaving. This exit area needs to be far enough downstream so that the hot gas and air are well mixed and the velocity profile is nearly flat (i.e. the eductor flow has transferred all of its momentum to the flue gas flow).

For all cases the velocity (V) can be calculated from the mass flow, density, and area and the ideal gas law is used for calculation of density from the pressure and temperature.

The mass balance is:

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2$$

For the case where the total area at the inlet is the same as the total area at the outlet, and both = A, the momentum equation can be written as:

$$(P_1 - P_3) * A = \dot{m}_3 * V_3 - \dot{m}_1 * V_1 - \dot{m}_2 * V_2$$

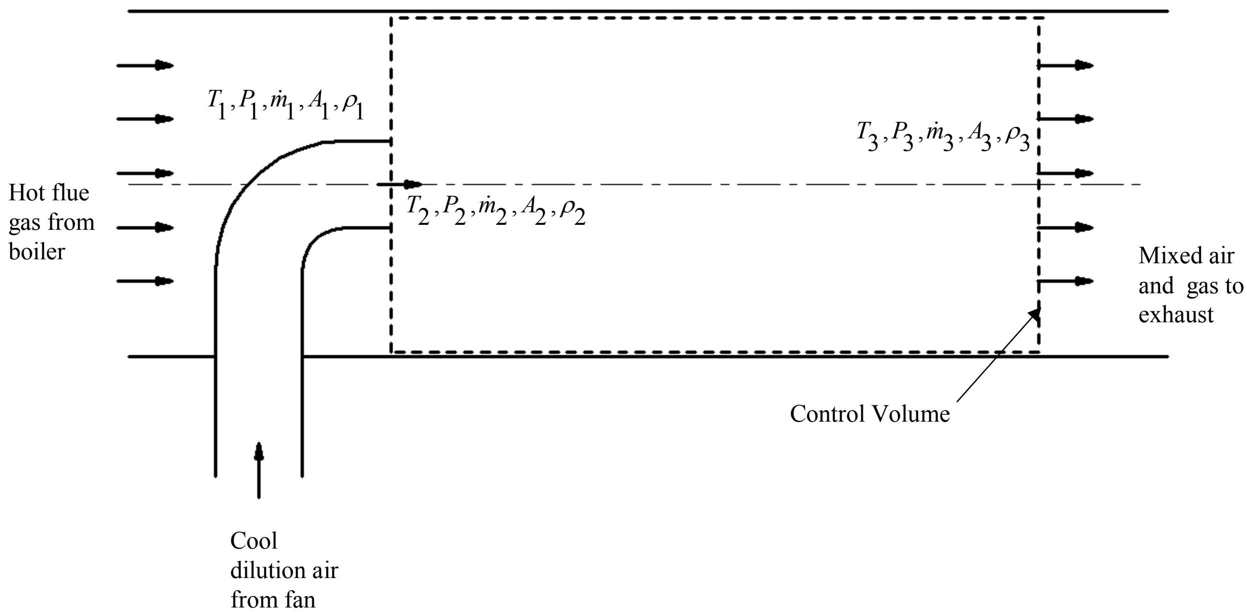


Figure 4 Illustration of simple model to calculate pressure rise across eductor and draft produced.

This can be rearranged to find $P_3 - P_1$, which is the most important output parameter i.e. what draft can be produced by a given geometry and flow set.

The eductor fan pressure (P_{fan}) required to produce a given V_2 can be calculated from the Bernoulli equation:

$$\left(P_{fan} - P_1\right) = \frac{\rho * V_2^2}{2 * g}$$

A heat balance can be used to find the temperature of the mixed gas or the mass flow of dilution air that is needed to produce a target outlet gas temperature. This can be written as:

$$T_1 * \dot{m}_1 * c_{p1} + T_2 * \dot{m}_2 * c_{p2} = T_3 * \dot{m}_3 * C_{p3}$$

where C_p is specific heat.

CFD Studies

Computational Fluid Dynamics (CFD) Studies were done using Ansys/Fluent commercial software. For several eductor geometries a two dimensional, axisymetric solution was done which includes only one half of a cross section. A κ-ε turbulence model was used. The purpose of the CFD studies was to help understand the flow patterns and draft produced as well as the wall temperature profiles with different geometries for evaluation of materials requirements.

Experimental

Two different residential boilers were used for most of the experimental work at BNL. The first is a Thermodynamics LM 75 steel boiler with a horizontal, cylindrical combustion chamber and a horizontal, return-flow, tubular heat exchanger. The second is an Energy Kinetics, high efficiency steel boiler. This has a refractory-lined combustion chamber and a scroll-shaped heat exchanger geometry. The firing rate range explored was 0.55 to 0.85 gallons of oil per hour (gph). This is equivalent to input rates ranging from 75,900 to 117,300 Btu/hr.

For this test, two different venting geometries were explored. The first included a vertical plastic vent pipe after the dilution venter. The second was strictly a horizontal, "sidewall" vent arrangement with 30' of vent pipe and 5 elbows.

A simple illustration of the vertical vent arrangement is shown in Figure 5 and a photo of the vertical vent arrangement outside of the test lab is provided in Figure 6. This setup is at BNL. The air intake piping in this arrangement includes 4 elbows (not shown).

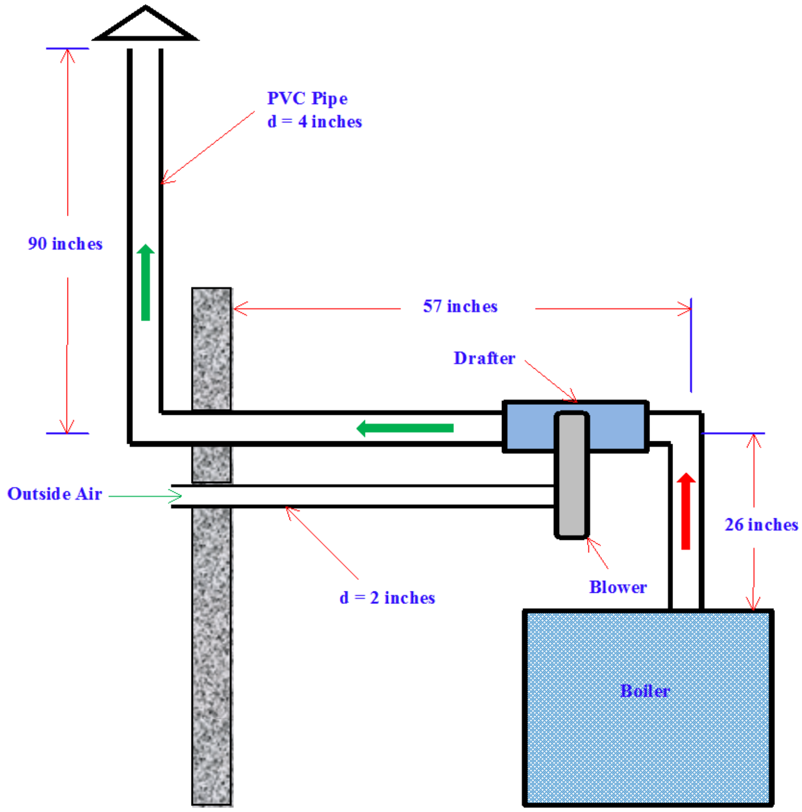


Figure 5 Illustration of the dilution vent test system with a vertical plastic "stack".

Figure 6 Photo of vertical plastic stack

<u>Eductor Geometries</u>

In this project several basic geometries were explored for the eductor section and these are simply illustrated in Figure 7. Arrangement A is a simple central jet. CFD simulations and analysis have shown this to be the most efficient arrangement although it has the following drawbacks: 1) a long mixing length is required and elbows located in the flue within ~ 20 inches of the injection point can reduce the achieved draft 2) the wall temperature downstream of the injection point is still at boiler stack temperature (high) and so the conversion to plastic is delayed until full mixing is achieved and 3) achieving good performance at high back pressure levels requires accurate alignment of the injector tube in the flue duct. Arrangement B offers the advantage of rapid wall cooling and a shorter mixing length. The key disadvantage with arrangement B is associated with high backpressure cases where a small injection area is required. Our earliest tests were done with injector open areas of 2 in^2. As we design for higher backpressure and lower required dilution ratio the required injection area is reduced to 0.3 to 0.5 in^2. We would like to make this geometry from sheet metal, leading to concerns about manufacturing tolerance and uniformity of the small annular air injection gap. A machined option with this geometry would seem to be too expensive and heavy.

Arrangement C in Figure 7 is a compromise between arrangements A and B and provide for a variable injection area with a single injector. It is a simple central jet but with a cone-shaped insert which directs the flow outward towards the wall, speeding mixing. By moving the cone in and out from the end of the injector pipe the open area can be adjusted. The central injector tube and cone have good construction tolerance and finish and so can be used to achieve a uniform air distribution even at small injection areas. This arrangement was built and tested during this period. In general, its performance was fair at best. Further details are provided below.

Arrangement D in Figure 7 is basically a classic venturi ejector approach.

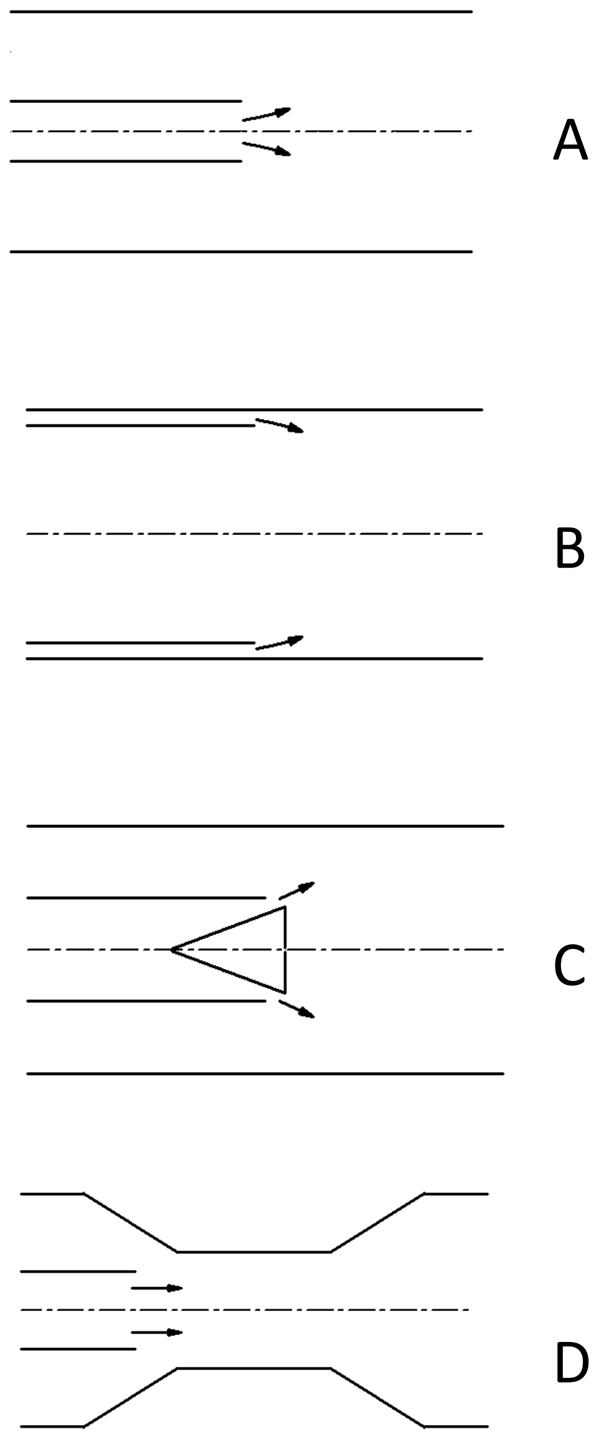


Figure 7 Different educator vent arrangements

One of the geometries evaluated in boiler tests was the annular eductor – arrangement B in Figure 7. Two different units were included. The first was the Quickdraft product illustrated in Figure 2. In this unit the flue diameter at the most narrow part was measured to be 2.625 in. The area of the annular eductor air jet opening was estimated at 1.7 in^2. Because this is a sheet metal design the dimension of the annular air injection opening is not uniform, leading to potential errors in the determination of this open area.

In any eductor design, the area of the air injection opening ("jet area") is a particularly important parameter. A smaller opening will lead to higher air pressure required for a given air flow and the potential for greater draft development. Viewed alternatively, a smaller opening with a given air fan will provide less air flow and a higher mixed gas temperature.

During this project a simpler geometry for the annular eductor approach was designed, built, and tested. The goal of this effort was a lighter eductor section with a direct fan mount. This concept is illustrated in Figure 8, and was produced based on design drawings developed by the project team by Maloya Laser Corp.

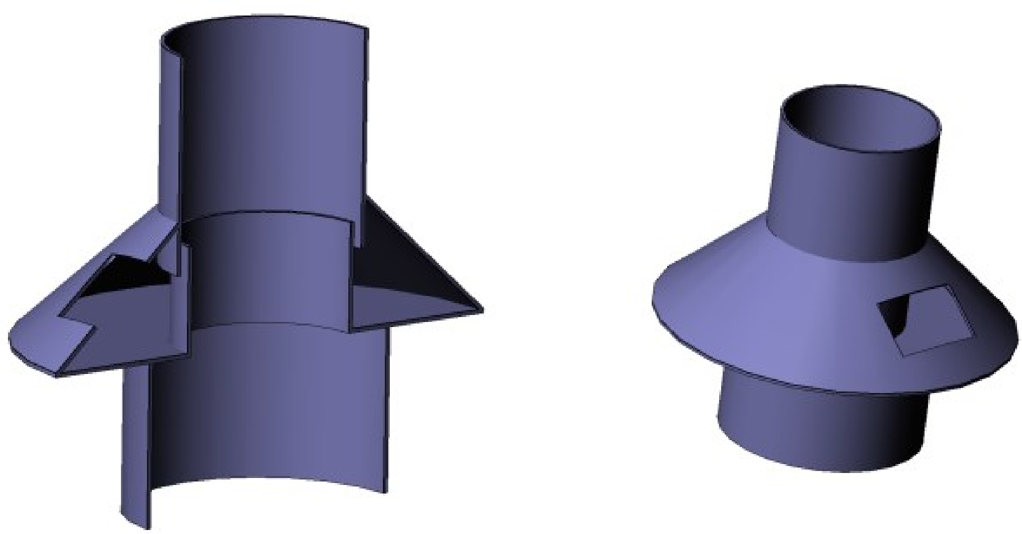


Figure 8 Illustration of alternative design for annular injector

Figure 9 provides a photo of the as-built version of this eductor. The area of the annual space through which the driving air enters is 1.97 in^2. The entrance section has a diameter of 5”. The smallest flue gas flow passage has an inside diameter of 3.4”. The metal used for construction of this eductor has a thickness of 0.070”.

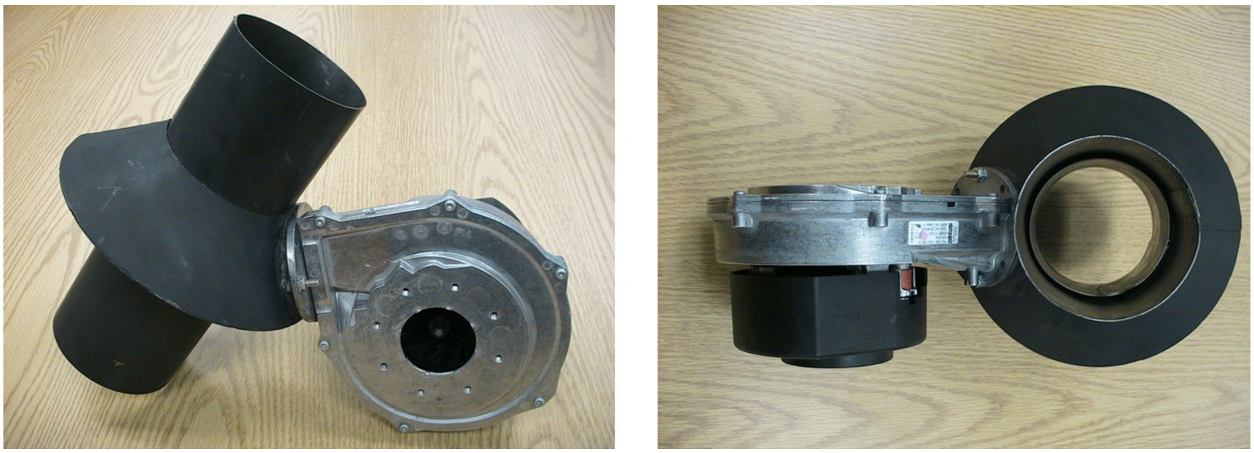


Figure 9 Photo of prototype annular eductor assembly with fan.

Figure 10 provides dimensions of one specific arrangement evaluated which follows the general arrangement “D” in Figure 7. This is a central jet eductor with a converging/diverging mixing section arrangement.

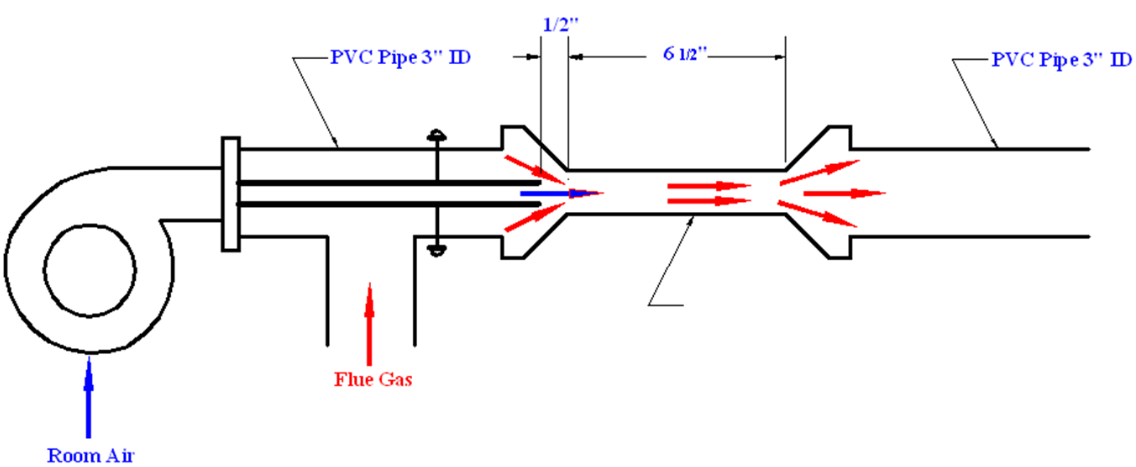


Figure 10 Details of eductor arrangement with central jet

The injection nozzle in this geometry has an inside diameter of ½” providing a jet area of 0.2 in^2. This specific geometry provides a low driving air flow but was used with higher air pressure. An advantage of this geometry is the very simple air injector which can be made at low cost and with good dimensional tolerance.

During the course of this project, a variety of different fans were evaluated for the generation of the driving jet in the eductor. Most of the work, however, focused on variable speed blowers produced by EBM, Inc. and specifically models RG 128 and RG 148. Fan discharge pressures over the range 1.5 to as high as 8 inches of water were explored and fan electrical power typically ranged 80-95 watts.

In tests of different vent arrangements, gas temperatures were typically measured using 1/8”, type K thermocouple probes. Flue pipe surface temperatures were measured using type K, foil, surface mount thermocouple sensors.

For some of the tests studies were done of the combustion chamber pressure transients during startup with different venting arrangements. These measurements were made using a rapid response, capacitive sensor, pressure transducer system – Baratron Model 310 BH-10.

Results

Mixed gas temperature and dewpoint

These parameters were calculated for many different conditions using the approach discussed above. Results are shown in Figures 11 to 15. These figures include the calculated mixed gas temperature, the dewpoint temperature of the mixed gas, as well as Tmix-sat. This last parameter is a measure of how far the mixed gas temperature is above the dewpoint. A larger value for this parameter would provide more of a margin against operation in a condensing mode downstream of the dilution device.

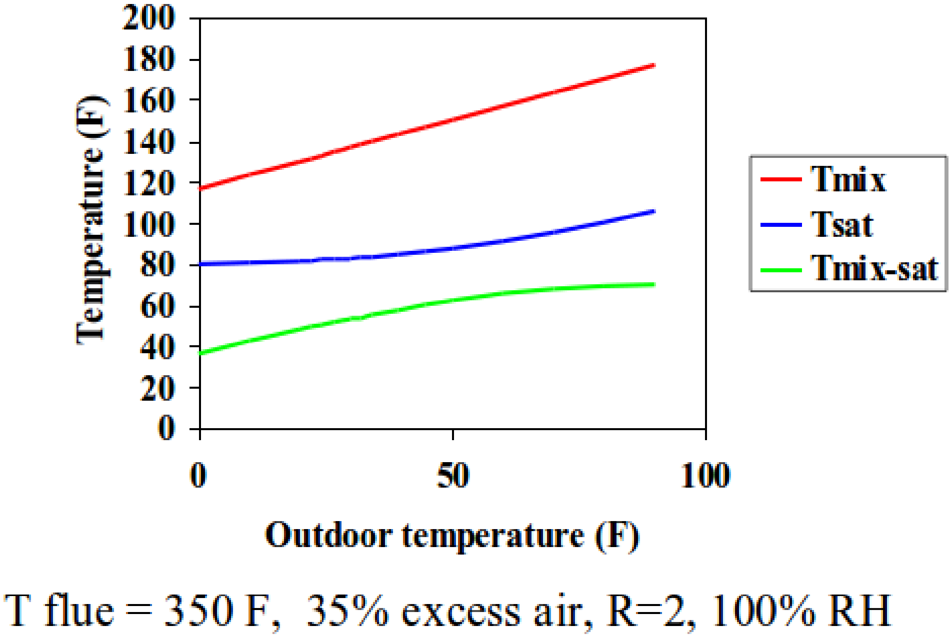


Figure 11 Calculated mixed gas condition as a function of outdoor air temperature. Tflue = 350 F, 35% excess air, dilution/flue gas mass flow ratio = 2, 100% relative humidity.

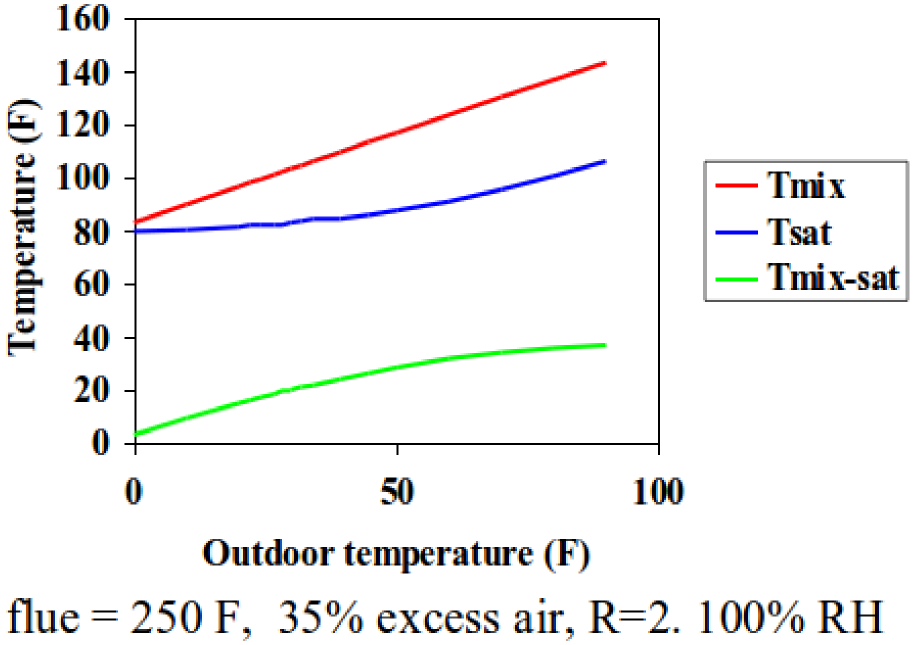


Figure 12 Calculated mixed gas condition as a function of outdoor air temperature. Tflue = 250 F, 35% excess air, dilution/flue gas mass flow ratio = 2, 100% relative humidity.

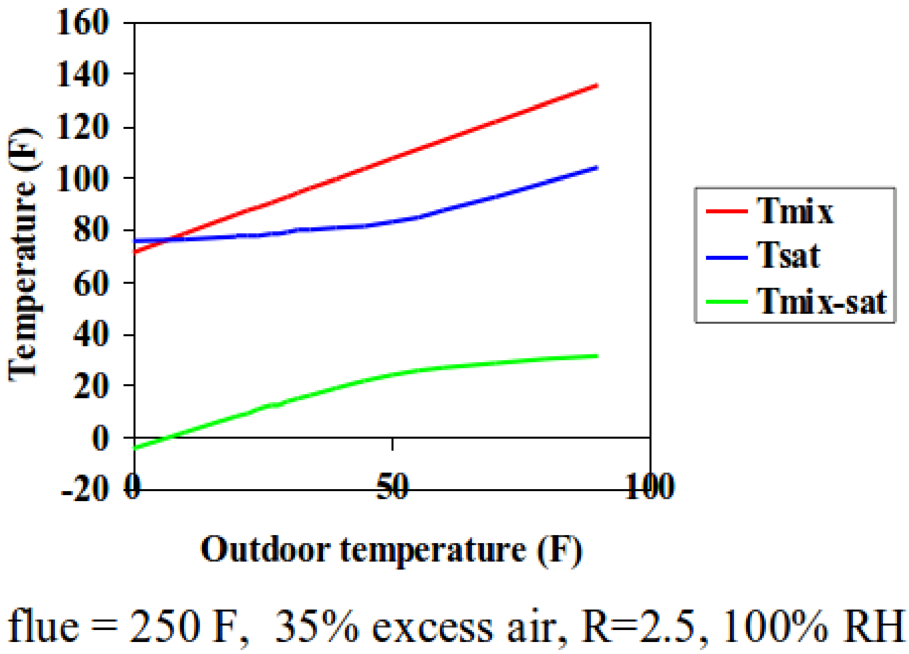


Figure 13 Calculated mixed gas condition as a function of outdoor air temperature. Tflue = 250 F, 35% excess air, dilution/flue gas mass flow ratio = 2.5, 100% relative humidity.

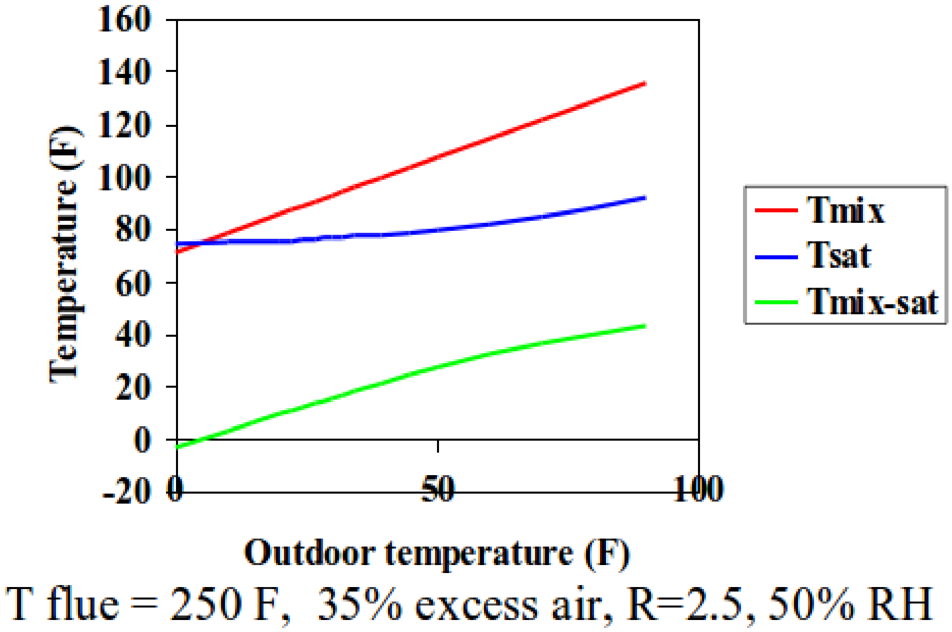


Figure 14 Calculated mixed gas condition as a function of outdoor air temperature. Tflue = 250 F, 35% excess air, dilution/flue gas mass flow ratio = 2.5, 50% relative humidity.

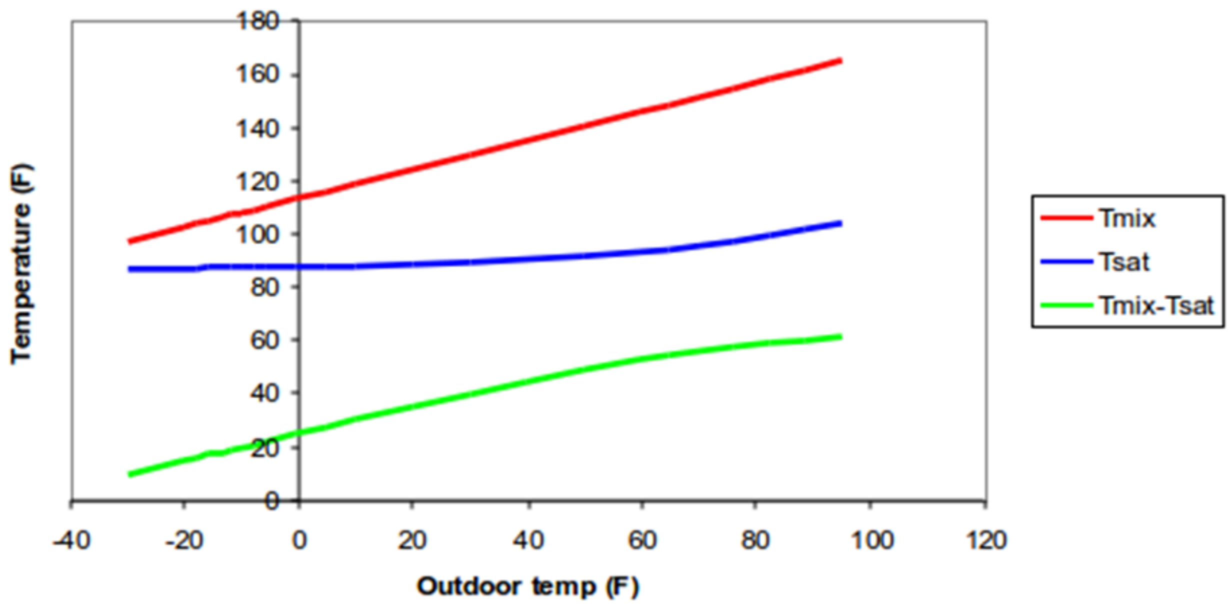


Figure 15 Calculated mixed gas condition as a function of outdoor air temperature. Tflue = 250 F, 40% excess air, dilution/flue gas mass flow ratio = 1.2, 100% relative humidity.

The case illustrated in Figure 11 shows a relatively high flue gas temperature (350°F) but a high dilution ratio. In this case there really is not concern about approaching the dewpoint over the range of outdoor temperature conditions explored. However, this unit firing in the summer time (for domestic hot water) with a very high outdoor temperature (90°F) would have a mix temperature that would be too high for low cost plastic vent materials. Higher temperature plastics or stainless steel would have to be used. This would defeat a significant advantage seen for this venting approach. Generally, this approach does not seem to be as attractive for such higher temperature cases. Of course, higher dilution ratios could be used.

In contrast, Figure 12 shows a case with the same dilution ratio but a 250°F flue gas temperature. Here the mix temperature is lower under the hot summer condition, allowing use of low cost plastic vent materials. However, here at the coldest outdoor temperature explored, near 0°F, the mixed gas temperature approaches the dewpoint. This configuration could be a concern in colder climates. Figure 13 explores this same case but with a higher dilution ratio and here it is shown that the potential for low temperature condensation in the vent is greater at low outdoor temperatures. These results are important in that they lead to the conclusion that a low flue gas entering temperature and a lower dilution air flow could give good performance over a wider range of outdoor temperature conditions.

Figure 14 again explores the condition with a high dilution flow but a much lower outdoor air relative humidity is assumed, 50%. The interesting point here is that the low temperature dewpoint crossover is not very strongly affected by this much lower relative humidity. This reflects that at low outdoor air temperatures the moisture content of the air is very low even at high relative humidity and the mixed gas moisture content, and so dewpoint, is primarily controlled by the flue gas entering water vapor content.

Based on all these results, an attempt was made to find a set of operating parameters that is closer to optimal and which can operate at even lower outdoor temperature conditions. Results of this are shown in Figure 15. Here the flue gas temperature is again assumed to be 250°F and the relative humidity is 100%. The dilution ratio, however, is much lower at 1.2. Here it is shown that the dewpoint can be avoided, even at outdoor temperatures lower than -20°F. At the highest outdoor temperature, 95°F, the mixed gas temperature approaches the upper limit where use of lower cost plastics could be considered.

CFD and Simple Model Analysis

The CFD analyses were completed for all of the geometries under consideration in this project and many detailed results plots were produced including velocities, temperatures, pressures, and turbulence intensities. Figure 16 provides one chart, for example, of the temperature field produced with an annular jet eductor. This serves to illustrate the attribute of this approach in achieving a cold wall temperature very quickly. The wall temperature actually increases further downstream from the jet exit plane as the hot flue gas mixes with the driving air flow.

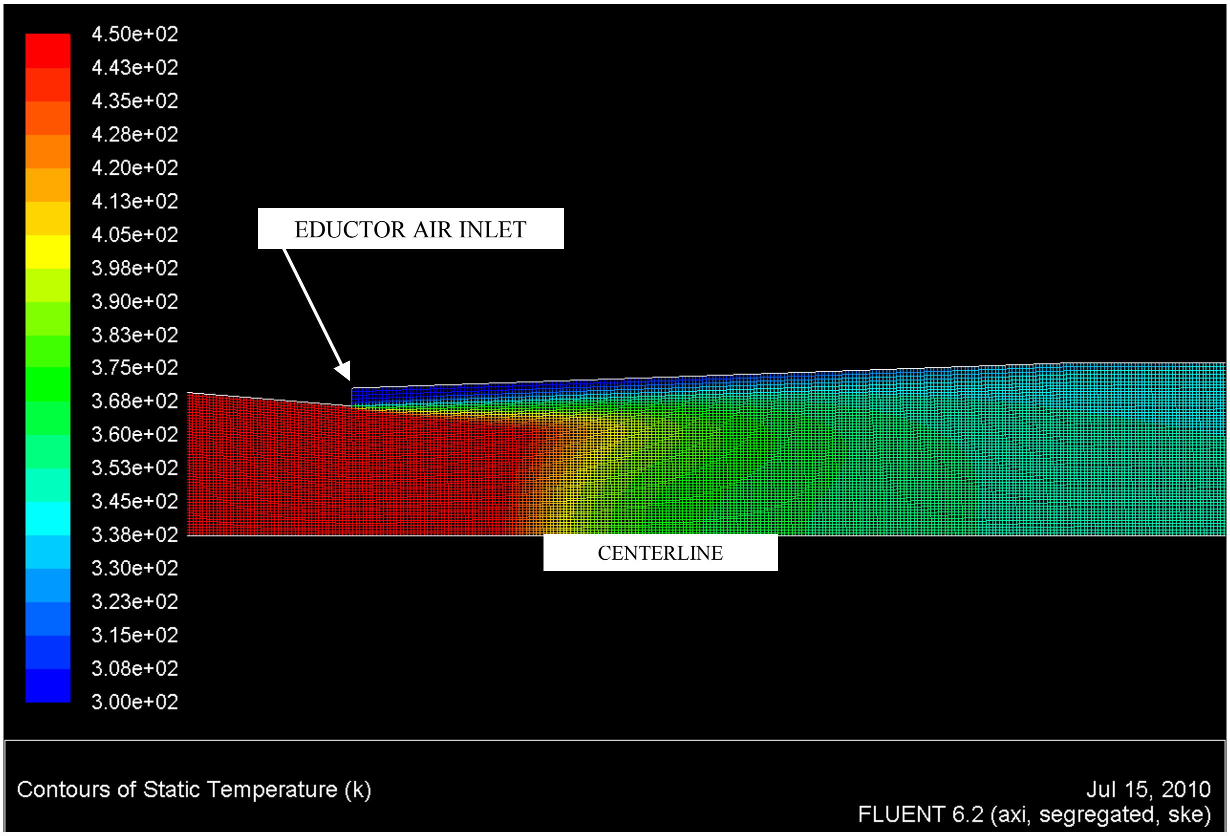


Figure 16 Annular jet eductor CFD simulation, temperature field.

Figure 17 shows for comparison the temperature field produced with the central eductor jet geometry. Here the wall temperatures just after the jet exit plane are at the flue gas temperature and only cool as the driving air mixes into the flue gas flow stream.

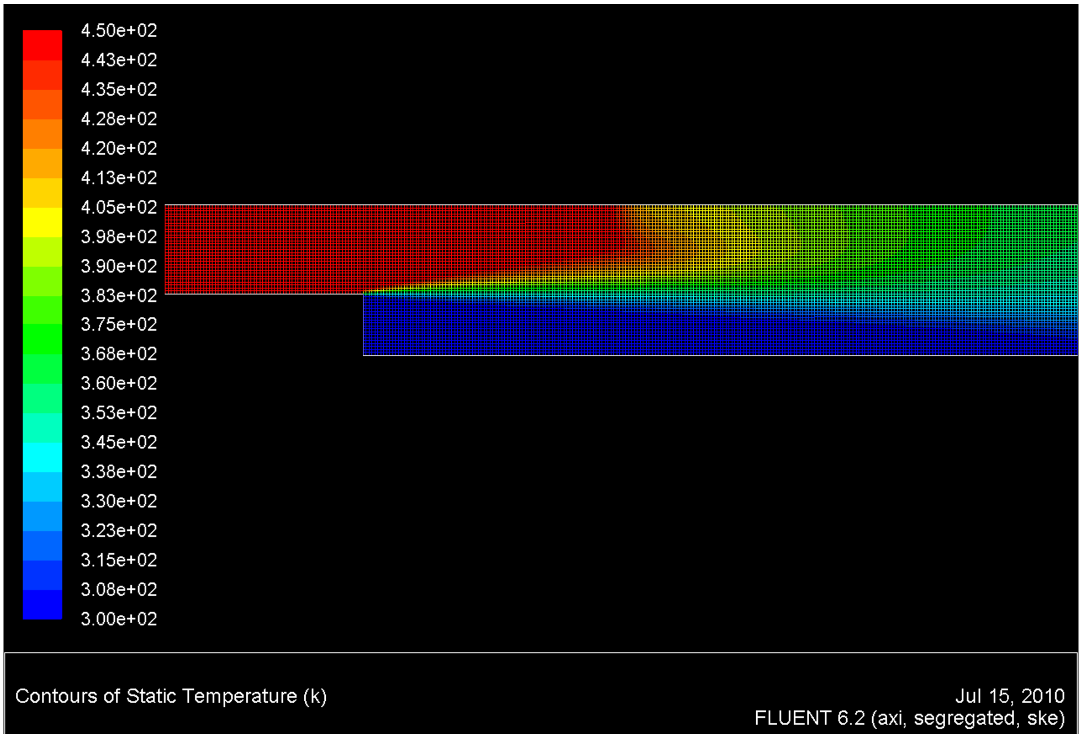


Figure 17. Central jet eductor - CFD simulation, temperature field.

In Table 1 is presented a comparison of the ideal model and the CFD simulation results for three different geometries.

Table 1 Comparison of Three CFD Case Runs

	Draft Produced – in H_2O		Ratio – draft /draft ideal
Case	Simple Model (ideal)	CFD results	
Central Jet	0.256	0.252	0.984
Annular Jet	0.249	0.214	0.860
Central Jet with Cone	0.256	0.185	0.722

The central jet case is geometry A in Figure 7. The annular jet is geometry B and the central jet with cone is geometry C. These results show that with the central jet geometry the simple model comes close to predicting the draft produced and so this has the least friction losses. The annular jet is also fairly close but losses due to wall friction lead to less draft production. The results in Table 1 show that central jet with the cone geometry is predicted to have the lowest level of performance by this metric and this was confirmed in lab testing. The goal of this geometry was to achieve a unit which could be manufactured at low cost and high tolerance while also achieving the rapid wall cooling of the annular jet. These goals were achieved but at a cost of low performance. Further development effort on the central jet with cone approach was not pursued in this project.

Boiler Test Results

Table 2 shows the results of tests done with the commercial annular eductor illustrated in Figure 2. These were done with a firing rate of 0.6 gph, with a conventional burner firing in the Thermodynamics LM75 boiler. The post-eductor venting system was very simple in these tests with direct sidewall exhaust. The impact of backpressure (duct pressure downstream of the eductor) was simulated by simply partial blockage of the exhaust. The results in this table show that this eductor geometry is very effective at producing draft and reducing the gas temperature to well below the level safe for low cost plastic venting, the arrangement has limited capacity for overcoming backpressure. This is indicated by the strong reduction in draft with increasing backpressure.

Table 2 Results of Tests with Annular Eductor (Quickdraft), RG 148 Fan, Variable Backpressure

backpressure	draft	P_{fan}	Flue Gas Temp.	Mixed Gas Temp.	O_2 boiler exit	O_2 after eductor	Fan power
inches water	inches water	inches water	F	F	%	%	watts
0.087	-0.38	1.22	344	145	11.2	18.5	97
0.23	-0.25	1.34	360	143	10.2	18.2	96
0.52	0.0	1.50	410	134	2.9	18.2	96

One configuration using this annular eductor was operated over a winter period periodically at BNL to evaluate performance under different outdoor conditions. The EBM RG 128 fan was used in this configuration. The vent was configured with a 7’ vertical 4” PVC outdoor “chimney”. The eductor air fan was fed with outdoor air using 2” PVC pipe. One of the concerns was confirmation that the mixed gas would not approach a saturated condition. Figure 18 shows the results of measurements over a temperature range from 8 to 60 F (outdoor temperature). On days in which this was tested, the boiler system was operated for 6 hours with the burner operating continuously.

During all of these tests no exhaust vapor plume was observed. What was observed, however, was condensate “drippage” from the body of the eductor. Internal surface temperature measurement indicated that the cold eductor air was reducing the eductor metal temperature below the dewpoint before the two streams mixed leading to the condensation. To eliminate this situation, a thin (1/8”) layer of moldable sheet refractory insulation was added. Figure 19 illustrates the placement of this thin insulation layer. This solution was found to be fully effective in eliminating internal eductor condensation.

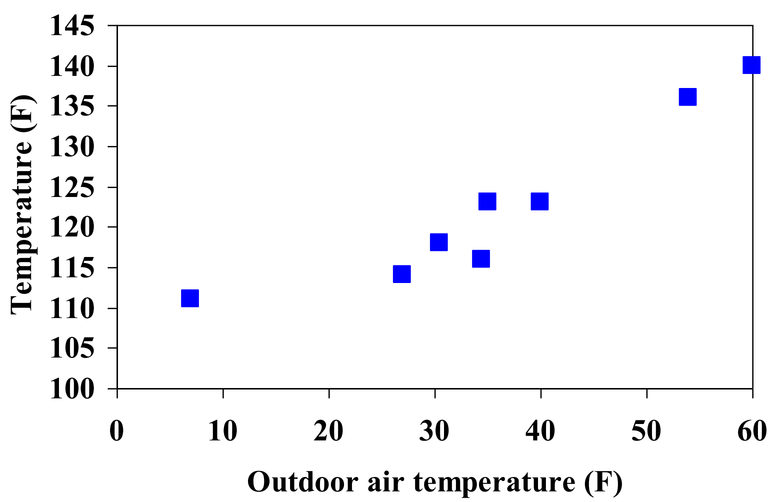


Figure 18 Results of periodic tests over a winter period at BNL. Mixed gas temperature as a function of outdoor air temperature.

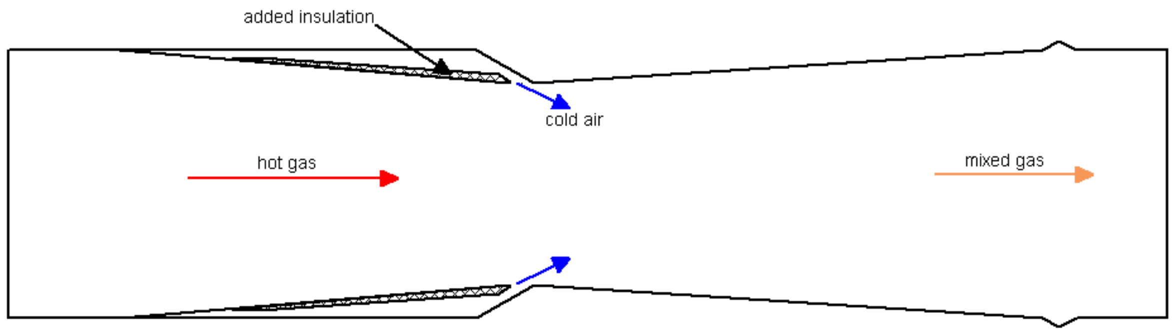


Figure 19 Added insulation on inside surface of annular eductor to eliminate internal condensation

The new configuration of an annular eductor, illustrated in Figures 8 and 9, was tested on the Energy Kinetics boiler. The installation arrangement is shown in Figure 20. These tests were conducted with a short (~ 10'), horizontal vent arrangement. The EBM RG128 blower was used and basic test results are presented in Table 3. As with the Quickdraft annular eductor a thin internal layer of insulation was added to prevent condensation on the eductor surfaces and this was found to be fully effective.

All of the test results with the annular eductors showed these to be effective for short vent lengths but not capable of providing negative combustion chamber pressures with the backpressure associated with long vent lengths. As discussed in the previous section, a model vent configuration of 30' of 3" PVC pipe with 5 elbows was set as the target, high-backpressure configuration.

Figure 20 Annular eductor mounted on prototype, high efficiency Energy Kinetics boiler

Table 3 Results of Tests Done with New Prototype Annular Eductor

Parameter	UNITS	
Flue gas O2	%	5.7
Flue gas temperature	F	235
Flue draft	in water	0.10
Chamber draft	in water	0.06
O2 after dilution	%	17.5
Temperature after dilution	F	114
Boiler combustion efficiency	%	90.0

For the high back pressure applications, the eductor vent configuration illustrated in Figure 10 was developed. This includes a central jet eductor and a relatively long, 2" diameter mixing section. This smaller diameter mixing section prevents recirculating flows under high back pressure conditions which can occur with the more open, annular eductor configurations.

Tests with this arrangement were conducted mounted on the Energy Kinetics boiler at two different firing rates. The EBM RG 148 blower was used and results are summarized in Tables 4 and 5. Because of the small area of the eductor jet the eductor fan discharge pressure was high but the use of the high efficiency blower led to an eductor power consumption under 90 watts. Overall this configuration was found to provide very strong draft in steady state even with the high back pressure condition.

Table 4 Central Jet Eductor - Test Results at 0.62 gph Firing Rate

Parameter	Units	Value
Firing rate	gph	0.62
Length of Mixing Section	"	5
Vent Length	feet	30
Vent Elbows	No	5
Vent Termination	-	Large Nozzle
Stack O_2	% dry	7.3
Stack Temp	F	226
Vent O_2 (after dilution)	% dry	14.7
Vent Temp	F	132
Stack Draft	in H_2O	0.185
Chamber Draft	in H_2O	0.17
Fan Inlet Pressure	in H_2O	-0.165
Vent Pressure (after dilution)	in H_2O	0.56
Fan Discharge Pressure	in H_2O	8.3
Fan Power	watts	82

Table 5 Central Jet Eductor - Test Results at 0.79 gph Firing Rate

Parameter	Units	Value
Firing rate	gph	0.79
Length of Mixing Section	"	7
Vent Length	feet	30
Vent Elbows	No	5
Vent Termination	-	Large Nozzle
Stack O_2	% dry	2.6
Stack Temp	F	249
Vent O_2 (after dilution)	% dry	12.3
Vent Temp	F	143
Stack Draft	in H_2O	0.1
Chamber Draft	in H_2O	0.05
Fan Inlet Pressure	in H_2O	-0.23
Vent Pressure (after dilution)	in H_2O	0.63
Fan Discharge Pressure	in H_2O	8.4
Fan Power	watts	89

While this geometry is very attractive, based on the steady state performance reported in Tables 4 and 5 above, a concern was raised during testing. During startup there is always a transient draft loss or pressure transient in the combustion chamber. If a burner is poorly setup, this transient will be greater as the time between the start of fuel injection and ignition is delayed and more fuel is in the chamber prior to ignition. This is termed a "hard start" and is an unacceptable condition. However, even with a properly setup and operating burner there is a small pressure transient creating a momentary fuel-rich condition and smoke, typically lasting only about 1-2 seconds. This transient leads to reduced air flow at startup and elevated smoke number. It can contribute to overall heat exchanger sooting and should be minimized. Restrictions in the flue pipe/vent system impede the relief of this startup pressure transient and can increase startup smoke numbers. With the relatively restrictive geometry of this eductor there can be a high startup pressure transient and resulting high startup smoke number. Analyses of this general phenomenon have been presented in earlier reports [7] [8].

Startup smoke numbers in the range of 2-3 are considered acceptable. With this eductor arrangement startup smoke numbers in the range of 4-5 were consistently observed. This led to an exploration of firing rate, vent length, fuel pressure, excess air, burner nozzle spray angle, eductor geometry, and other basic parameters on the smoke transients. In combination with this very fast time resolution measurements were made of the pressure transients in the combustion chamber that drive the high startup smoke number. For these measurements we use a very fast time response pressure transducer (discussed above) and a digital oscilloscope linked to a computer. In our initial exploration of the 2" diameter mixing section we stayed with a 7" length. The pressure transient measurements showed that a peak combustion chamber pressure of 1.2 inches of water was achieved at startup and this pressure pulse duration was very close to one to two seconds. The time resolution of these measurements was 4 milliseconds yielding 250 measurements per second.

Another area that was explored is vertical venting. For this a PVC chimney was arranged with a 15' vertical rise. For this setup the exhaust from the boiler travels horizontally about 6' to exit the building, passes a single elbow and then rises the 15' In our setup of this vertical vent we explored different options for the eductor/mixing section geometry. Through this process we found that a longer mixing section with the 2" diameter improves performance and settled on a 12" length as optimal.

Table 6 below provides one set of measurement data with this vertical vent arrangement. Because there is less back pressure with this arrangement the startup smoke numbers are quite low.

With this arrangement the inducer works very well although the chamber draft might be considered too high. This made it difficult to reduce the O_2 although it was reduced to ~ 6% for longer term cycling tests.

The vertical vent studies summarized above were part of the effort to optimize the performance with the long horizontal vent. Based on all of the experience gained an optimized set of conditions for the long horizontal vent were developed. This included somewhat higher fuel pressure and a wider nozzle spray angle. Results of these tests with the long horizontal vent are presented in Table 7, below

It should be noted that "cold" in this case represents following an 80 minute idle period and warm represents following a 10 minute idle period. Combustion chamber pressure transient measurements were made during many startups during these tests. The peak pressure is in the range of 0.87 to 1.0 inches of water and the total duration of the pressure transient is 2 sec.

Table 6 Results of Vertical Vent Test with Central Jet Eductor

Parameter	Units	Measurement
Firing rate	gal/hr.	0.77
Inducer fan	-	EBM RG 148
Chamber steady state draft	Inches of water	0.51
Fan power	W	91
Startup smoke from cold	-	3
Startup smoke from warm	-	2
Stack O_2	% dry	9.2
Vent O_2	% dry	14.9
Stack Temp	F	235
Vent Temp	F	134

Table 7 Test Results after Optimization of Burner, 30' Vent

Parameter	Units	Measurement
Firing rate	gal/hr	0.77
Nozzle.	-	Hago .65 80 B
Inducer fan	-	EBM RG 148
Chamber steady state draft	Inches of water	0.28
Fan power	W	91
Startup smoke from cold	-	3
Startup smoke from warm	-	2
Stack O_2	% dry	6.3
Vent O_2	% dry	14.4
Stack Temp	F	253
Vent Temp	F	131

Discussion

The work in this project focused on the development of an eductor-type venting system which will keep a boiler or furnace under negative pressure during operation but allow for low cost venting materials based on the blending of outside air with the flue gas. A key advantage of the eductor vent system is the ability to keep the fan blades outside of the flue gas, enabling use of lower cost materials for the fan construction also.

The alternative design which was in development by Energy Kinetics prior to this project was the use of an induced draft fan downstream of the mixing section which would draw in both flue gas and the outside air. Some type of flow control geometry is required to ensure the correct relative flow of the two streams. Development work on this geometry continued at Energy Kinetics during this project and the key advantage is seen as very robust performance under very long vent length conditions. Further, this geometry allows for startup pressure relief through the dilution air connection, reducing the transient draft loss to the extent where it compares very favorably with chimney start up characteristics.

It is the opinion of the authors that both arrangements have the potential to serve different market opportunities with the common goal of eliminating chimneys as a constraint to increasing efficiency.

Conclusions

The work in this program has shown that dilution venting can be implemented with oil firing without creating the potential for water condensation downstream of the mixing device. Under very cold outdoor conditions a low dilution air flow should be used to ensure that such condensation is avoided. To achieve good draft with lower air flow, a higher eductor jet velocity would be needed, which requires a smaller jet open area.

The annular injector geometry offers the advantage of rapid cooling of the duct wall downstream of the eductor and so a rapid transition to lower cost venting materials. However, this geometry does not perform well under high backpressure conditions associated with long vent lengths. Further, with the annular venter achieving a small jet open area may create challenges achieving dimensional tolerance.

Under conditions of relatively low back pressure the annular eductor system was shown to produce good draft levels and achieve mixed gas temperatures below 150 F.

For high back pressure conditions associated with long vent lengths, a central-jet geometry with a relatively small diameter mixing tube was found to produce the draft level needed. There is a concern with this more-restrictive arrangement that startup transient chamber pressure and smoke may be a concern. In work to optimize all parameters it was found this could be mitigated. This is an area which should be considered in general application of this approach. The long vent configuration used here included 30 feet of vent pipe with 5 elbows.

Both of the primary approached developed in this work are expected to have potential for field application.

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