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Fluid Dynamic Studies on Ejectors for Thermal Applications of Gasifiers

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Abstract

The paper describes the use of biomass gasification technology for meeting the thermal needs in both industrial and the domestic sector. The use of ejectors to draw the hot gas from the reactor by suction, premix it with air and burn it in burners, eliminating the elaborate cooling and cleaning system typically used for the engine applications is described. The gas handling is made effective through control of power and ease of operation. Model analysis and experimental results on the performance of the ejector are also discussed. A typical application for 20 kW (thermal) shows an overall efficiency of about 45 % based on water boiling experiments.

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Introduction

Liquid as well as solid fuels are generally used for thermal applications in ceramic industries, kilns, foundries, etc. In the absence of liquid fuel combustors or non-availability of liquid fuel, generally solid fuels are used for generating heat. Liquid fuels are preferred for their ease in handling and better combustion control. In the field of solid combustion, the commonly adopted technique is grate combustion, either for boiler, kiln or other applications. Controlling the combustion rate and the quality of product gas is not simple and calls for accessories. It is also well known that a gaseous fuel is very convenient in terms of the fuel control and quality of combustion compared to a solid or a liquid system. In this country as well in several others, where gaseous fuel like natural gas is not available in abundance for the industrial sector, liquid fuel like the furnace oil, LDO, etc., and solid fuels like coal, wood, biomass are considered as alternatives. As far as the domestic sector is considered, the existing demand for the liquified petroleum gas is not met with and this situation will continue to be so. As far as the liquid fuel (kerosene) is concerned, it is rationed commodity. Thus, solid fuels are used for large scale cooking especially in canteens, hotels, bakery ovens, etc. Shah[1] summarises two field experiences using the gasifier systems for thermal application. He emphasises on the convenience in handling the gaseous fuel as temperature control could be achieved effectively in an application which required 523 K hot gas. The paper also discusses the economics of operation compared to the conventional methods of using oil burners and LPG. Though technical details of the gasification systems are not sufficiently discussed, figures indicate an elaborate cooling system in one and forced draft without cooling system in another.

In the early development of gasification technology in this country, Government of India encouraged significantly the use in reciprocating engines to save petroleum fuels. Some limited trials are also being made on the usage of gasifiers for thermal applications. Attempts by Sekar et al[2] in trying to do this indicate the problem of the blower getting clogged with tar in a few hours of operation. Attempts to use forced draft to avoid any cooling system calls for a reactor design to hold the gas under pressure without any gas leakage into the ambient. The other issue in such a device would be controlling the air flow rate to achieve the premixedness and allowing it to burn in the burner, as otherwise it would lead to the burning of the gas-air mixture in the pipeline leading to the burner. The current designs for meeting the thermal needs are by generating the combustible gas using the suction of a blower and later leading it to a burner. This calls for an effective cooling and cleaning system as the blower handles the gas to prevent blockage inside the blower and the pipelines. The other basic issue is the handling of premixed gas in the blower and the pipeline leading the burner. The presence of the premixed gas air mixture in the blower calls for a leak proof blower with shaft seals to prevent any gas leakage into the atmosphere (because the gas is at a positive pressure in the blower and any leakage leads to CO poisoning). The downstream piping leading to the burner should also be made leak tight for similar reasons. Based on the above, the following questions arise,

a. Can the gas be used for thermal applications without having to process the gas through cooling and cleaning system ?

b. Can one avoid blower coming in contact with the gas, eliminating the problem of clogging and also avoid premixing inside the blower ?

The first point indicated has a serious impact on the system and site requirement, in case the gas has to be cooled at larger power levels. For most situations, water required for cooling needs careful management.

In the light of the above, the present work is devoted to the description of a technology for gaseous fuel generation and combustion using gasifiers to meet the thermal needs of the industrial and the domestic sector, and accounting for the above issues. Using a simple aerodynamic device, the ejector, the paper addresses the previously mentioned questions in meeting the heat demand. The description to follow includes the analysis of the design of the ejectors for various power levels and an experimental study to demonstrate the feasibility of such a system. Specific study towards cooking applications along with the overall thermal efficiency of the system is discussed.

Ejectors have been generally known in the industrial sector for applications in jet compressors or in thermal plants for steam. In the domestic sector the LPG stoves uses the principle of ejector to mix the fuel with air before the gases are burnt.

The Analysis

Figure 1 shows a schematic of a typical ejector. The ejector consists of a nozzle through which air is pumped, a constant area mixing section and a diffuser. The mixture passes through a constant area section for mixing, later through the diffuser. The high velocity primary air through the nozzle entrains the fluid surrounding it, drawing the gas from the plenum around the nozzle and creating vacuum in that zone. The primary air and the entrained low velocity fluid mix in a constant area duct. The mixed fluid is passed through a diffuser, wherein the static pressure is increased to the exit conditions. The continuity and momentum equations at cross sections 1–3 are as follows,

$$\rho_{a1} U_{a1} A_{a1} = \dot{m}_a \tag{1}$$
$$\rho_{g1} U_{g1} A_{g1} = \dot{m}_g \tag{2}$$
$$\rho_2 A_2 U_2 = (\dot{m}_a + \dot{m}_g) \tag{3}$$
$$P_{tg} - \frac{1}{2}\rho_{g1} U_{g1}^2 = P_{ta} - \frac{1}{2}\rho_{a1} U_{a1}^2 = P_1 \tag{4}$$

where $\dot{m}_a$ and $\dot{m}_g$ are the air and gas flow rates respectively and P_a and P_{tg} are the total pressure of air and gas respectively, with the equation of state $P = \rho RT$.

Neglecting the contribution of kinetic energy of the two streams to the energy content of the stream, (low Mach number approximation)

$$\dot{m}_a C_{p_a} T_a + \dot{m}_g C_{p_g} T_g = (\dot{m}_a + \dot{m}_g)\bar{C}_p T_2 \tag{5}$$

where T_a and T_g are the temperatures of air and gas respectively. T_2 is the temperature at section 2 and C_{p_a} and C_{p_g}, the constant pressure specific heats of air and gas. The momentum flow at sections 1 and 2 can be written as,

$$M_1 = \frac{\dot{m}_a^2}{\rho_{a1} A_{a1}} + \frac{\dot{m}_g^2}{\rho_{g1} A_{g1}} + A_2 P_1 \tag{6}$$
$$M_2 = \frac{(\dot{m}_a + \dot{m}_g)^2}{\rho_2 A_2} + A_2 P_2 \tag{7}$$

In writing equations (5) and (6) the velocity profiles in the section have been assumed to be flat. For an ideal ejector with complete mixing, $M_1 = M_2$. However, because of incomplete mixing and friction, M_2 would be less than M_1. Hence it is written as

$$M_1 = M_2 + M_F \tag{8}$$

where M_F is the momentum loss due to friction and incomplete mixing and is written as $M_F = C_{\mathrm{mom}} \frac{(\dot{m}_a + \dot{m}_g)^2}{\rho_2 A_2}$, where C_{mom} is the momentum loss coefficient. Chitillapilly et al[3] experimentally obtained C_{mom} for a ramjet secondary combustion chamber mixing section which in many ways is similar to the ejector mixing section. The value of C_{mom} obtained by them was in the range of 0.5–1.5 for primary to secondary pressure ratio in the range of 4–6. The pressure ratio being close to unity in the present case, we expect the value of C_{mom} to be smaller. The value of C_{mom} has been obtained by the analysis of the experimental data (see later) is in the range of 0.1–0.40 for small ejectors considered here. For the diffuser between the section 2 and 3

$$P_2 + \frac{1}{2}\rho_2 U_2^2 = P_3 + \frac{1}{2}\rho_3 U_3^2 \tag{9}$$

$$T_3 = T_2 \tag{10}$$

If the exit of the ejector is open to atmosphere, $P_3 = P_a$. Solving the above set of equations, $\dot{m}_a$ and $\dot{m}_g$ can be obtained as functions of P_{ta} and P_{tg} for fixed ejector dimensions. In equations (1)–(4) the values of P_1, $\dot{m}_a$, $\dot{m}_g$ and the velocities are the unknowns with the geometric dimensions at different sections (A_{a1}, A_{g1}, A_2 and A_3 are the area of cross section for the air flow, gas flow, the constant area mixing zone, and the diffuser exit area), the total pressure the blower (P_{ta}) can deliver are known. With an initial guess of $\dot{m}_a$, P_1 can be evaluated from equations (1) and (4) and from equations (2) and (4), $\dot{m}_g$ can be calculated. Then from equations (6),(7) and (8), the momentum and pressure at section 2 are evaluated. Using these values at section 2 and equation (9) along with equation (3), the momentum and pressure at section 3 can be evaluated. If $P_3 = P_a$, then the guessed value of $\dot{m}_a$ is correct, otherwise it is calculated using corrected value of $\dot{m}_a$. The corrections have been made using Regula-falsi method.

These set of equations can also be used to design an ejector. That is, the ejector dimensions can be obtained for required gas and air flow rates. These equations, however, give only the diameters of various sections. Other dimensions of the ejector such as the length of the mixing zone, the divergence angle of the diffuser and the spacing between the air nozzle and the inlet to the mixing need to be optimised using experiments. In this regard, a fair amount of experimental work has been carried out by Francis et al[4] and Watanabe[5], who have given approximate guidelines for selecting these parameters.

The Experiments

Experiments have been conducted on the ejector fixing on to a 3.7 kW wood gasifier. *Figure 2* shows the experimental set up of the 3.7 kW system. It consists of a ceramic shell reactor, the ejector, burners and a blower. Two venturimeters one each on the gas and air line were used to measure the gas and the air flow rate. The areas of the air nozzle can be varied by axial movement of the pintle (identified as V_a). A provision is made at the gasifier outlet to control the amount of gas flow rate by operating a butterfly valve V_g. Complete control of gas flow and mixture ratio are possible with these two valves. The hot gases thus ejected are mixed with air in the mixing zone and the premixed fuel-air mixture is led into the combustion device. By controlling the amount of air by varying the nozzle area and the gas valve V_g, the right mixture ratio to attain the peak temperature at any desired gas flow rate can be achieved. Thus power and mixture ratio can be independently controlled.

Cold flow runs were carried out to determine the performance of the ejector. In evaluating the performance of the ejector, the following measurements were made: primary air flow rate through the nozzle, secondary air flow rate, the ejected fluid flow rate (hot gas in the gasifier run) and static pressure at various locations as shown in *Figure 1*. The delivery of the blower (0.74 kW blower, 4200 Pa), was connected to the primary air inlet. Measurements were conducted at various openings of the primary air valve (V_a) and gas valve (V_g).

Gasifier runs were carried out by fixing the ejector at the gasifier outlet as shown in *Figure 2*. The system consists of a ceramic reactor with allumino silicate insulation 75 mm thick, ejector, blower and a burner. The gas is taken out at the bottom below the grate as shown. Similar measurements as in the cold flow along with gas outlet temperature were made. The premixed gas-air mixture was led into two burners of design like a domestic stove. This was done to determine the overall efficiency of the system by performing water boiling tests. Apart from the primary air control, the two burners have individual butterfly valves to control the flow rates independently. The gas path leading to the burners was insulated to prevent heat loss. In designing the burners sufficient care is taken to prevent flame flash back. The gas velocity in the pipeline near the burner zone is maintained at about 2–2.5 m/s which is above the flame speed of the gas-air mixture ($\sim$ 0.5–0.75 m/s) at temperature of about 500 K and stoichiometric conditions. The burner consists of steel tubes of 6 mm diameter distributed over the entire cross section of 100 mm diameter pipe. The diameter of the steel tubes is chosen so as to avoid flame flash back. The ignition temperature for gas air mixture is around 800 K, and hence there is a little chance of pre-ignition of gas-air mixture in the plenum between the burner and the ejector.

For starting the experiment, procedure similar to that indicated in Dasappa et al[6] is adopted. After switching the blower on, the gasifier is lighted and the primary flow is adjusted to obtain the combustible mixture at the burner. Steady flame can be obtained generally in less than 15 minutes. This initial startup period to get a steady flame is high compared with that indicated in Dasappa et al[6] and is partly because of more CO can be expected to be generated from the gasifier because it is largely running on the charcoal. As the air requirement for the combustion of CO is less than that for H_2, one tends to have lean mixture during this period resulting in a weak flame. Another reason is the poor gas quality resulting from the low gasifier temperature in the initial period. The temperature build up time of the ceramic shell gasifier is larger (typically, an hour) compared to that of the metal version reported earlier (typically about 15 minutes) because of the large thermal mass of the ceramic version. These factors may not be apparent in the swirl burner recommended for use in earlier work because of its high flame stability. A burning wick placed on the burner ensures that any combustible products generated is consumed during this period. By tuning the air and gas flow, the stoichiometric mixture ratio is achieved. Visual inspection of the flame at the burner and a thermocouple located inside the burners assist in indicating the performance.

Results and discussion

A typical plot of the wall static pressure at the wall in the mixing portion of the ejector for a given primary air flow rates and different gas flow rate is shown in the *Figure 3.* It is indicative of the performance of the ejector, in particular, in the mixing portion. The maximum suction created is about 2500 Pa at zero flow of the ejected fluid. The suction pressure created at different gas flow rates is also shown in the *Figure 3.* It is expected that the static pressure at p_5, beyond the mixing zone i.e., in the diffuser section should increase to the exit conditions as indicated by Watanabe[5]. But in the present study, this behaviour is not seen. This could be due to the fact that the joint between the end of the mixing zone and the entry into the diffuser, creates frictional loss. It is clear that the profile nearly flattens at station 4, indicating that the mixing is complete. This means that mixing is complete in about 7 diameters of the constant area section.

Figure 4 shows the ejector efficiency, and the experimental data on the suction pressure for the secondary air flow and the secondary air flow rates. From the analysis and the experimental data on ejector by Watanabe, it appears that at higher primary air pressures the efficiency drops. The ejector efficiencies are computed similar to Watanabe[5] as

$$\eta = \frac{C_{p_g}\dot{m}_g T_g[(\frac{P_2}{P_1})^{\frac{\gamma-1}{\gamma}} - 1]}{C_{p_a}\dot{m}_a T_a[(\frac{P_{1a}}{P_a})^{\frac{\gamma-1}{\gamma}} - 1]} \tag{11}$$

In the above equation the denominator is the isentropic work that can be done by expanding air from P_{ta} to P_a and the numerator is the actual isentropic work done on the gas by compressing it from P_{tg} to P_a. γ is the ratio of specific heats. *Figure 4* shows the plot of efficiency at various suction pressure for a given air flow rate (7.1 g/s). Efficiency peaks to about 16 % at high suction pressures, for the area ratio of 2.7, defined as the ratio of areas of the constant area mixing zone to that of the air nozzle exit area. Similar efficiency results are also obtained by Watanabe for the area ratio in the range of 2.14–2.93. The efficiency of an ejector is low because of the irreversible mixing between the two streams. If one computes the efficiency of an ideal mixer without considering any losses due to friction and incomplete mixing, the peak efficiency is 60 % at 1500 Pa suction pressure, and gradually drops to about 5 % at 100 Pa. The lower bound in efficiency is also evident from the experimental result.

Based on these measurements, if one calculates the coefficient of momentum loss due to friction and mixing C_{mom}, as indicated by equations (6), (7) and (8) earlier, one obtains a value of 0.1–0.40. Variation of C_{mom} with respect to suction pressure is shown *Figure 4*. It can be seen that the value of C_{mom} is nearly constant (= 0.1) until the suction pressure of 1000 Pa, beyond which the value increases drastically. The value of C_{mom} at low gas flow rates is high (0.40) compared with high gas flow rate (0.1). This could be due to the separation which could be occurring in the mixing zone at low gas flow rates. The value of C_{mom} calculated above is due to just the loss in the mixing section. Significant loss can also occur in the diffuser. However, for ease of calculation, all the losses have been accounted for in one section (mixing) in the model. Since the diffuser section is much larger in terms of surface area compared to the mixing section, one can expect the total C_{mom} to be much larger. The actual value was varied to determine if a single value would produce good predictions of mass flow rate of gas against gas total pressure. It turned out that $C_{\mathrm{mom}} = 0.25$ upto 1000 Pa and 0.35 beyond would produce good correlations. The sudden jump is, as already argued, due to the separation in the mixing section itself (due to large adverse pressure gradient). It may be noted that the values of C_{mom} are consistent with expectation noted earlier. Hence in numerical computations, momentum loss coefficient has been assumed to be two different constants as 0.25 and 0.35 depending on the suction pressure being less than or more than 1000 Pa. The model predictions on the gas flow rates for different suction pressure are in agreement with the experimental results in the entire range of the flow rates thus validating the model.

Prediction of gas flow rate at different air flow rates for a given resistance across the gasifier system is shown in *Figure 5*. For a typical composition of the gas generated in the reactor[6] the air to fuel ratio is in the range of 1.2–1.3 for peak flame temperature. This mixture ratio could be obtained for a given gas and air flow by varying the valves V_a and V_g. In an actual gasifier operation the resistance across the gasifier could vary between 100–800 Pa depending upon the gas flow and the periodicity of cleaning of the grate. Under these conditions, the necessity of obtaining the right mixture ratio is important. *Figure 5* shows the experimental results and the model predictions for gas flow rate at different flow rates of air at suction pressures of 100 and 800 Pa. The model results seem to be in good agreement with the experimental results within the accuracy of measurements. In an experiment, for a given air flow rate, varying the gas valve V_g the right mixture ratio could be achieved to attain peak flame temperature.

Measurement of the temperature of the hot gas at the exit of the reactor indicated an average of about 550–600 K, the temperature of the gas at the burner being in the range of 1500–1550 K. This peak temperature was found for an A/F of 1.2–1.3. *Table 1* indicates the gas compositions and the temperatures. The gas composition shown in rows 1 and 2 of *Table 1* indicate the normal range of variation in composition during the gasifier operation. These compositions are on dry basis obtained after cooling the gas to ambient temperature so that all the H_2O vapour would condense and the gas would contain moisture corresponding to saturation at the ambient temperature. In this case the percentage of moisture is negligible. The composition in the third row corresponds to that in second row when H_2O is uncondensed (*i. e.*, the gas is not cooled). For a typical gas compositions (dry basis) mentioned above, the theoretical adiabatic temperature is in the range of 1935–1941 K. *Figure 6* shows a plot of adiabatic flame temperature vs the air to fuel ratio for different compositions and initial temperature. There is only small difference in the adiabatic flame temperature for the compositions in rows 1 and 2.

One would normally argue that if one draws hot gas from the reactor without cooling one would get higher enthalpies in the combustor/burner because extra energy from the sensible heat of the gas is available. However, it is necessary to recognise that uncondensed moisture will be a part of the ejected gas at high temperature (600 K).

In the present case as the gas is ejected at high temperature (600 K), moisture from the gasifier would also be present upto about 10 %[7]. With the composition as indicated in row 3 of the table, the adiabatic flame temperature is 1982 K which is only 41 K more than the dry gas at ambient temperature, even though the initial temperature of the gas is higher by 300 K. Thus by drawing the hot gas from the reactor, the adiabatic flame temperature is not significantly enhanced. Measurement of flame temperature in this work which has used combustible gas directly and in Shrikant et al[8] (to be presented in this meeting, 1993) where gas-air mixture at ambient condition was burnt in premixed mode for stochiometric mixture showed peak temperature of 1540 ± 25 K, indicating that the drawing the hot gas does not make any difference to the thermal performance. This could be generally true even in case of a updraft or a forced draft systems.

Measurement of overall efficiency of the system was carried out using procedure similar to the one used earlier[9], with vessel diameters of 300 mm. *Figure* 7 shows the view of the gasifier with the ejector along with the burner. Wood consumption rate, gas flow rate, gas temperature, temperature rise of water with time along with the static pressure measurements were noted on a four hour experiments. The wood consumption and the gas flow rate are indicative of the power level of the gasifier system. The turn down ratio for the ejector on the gas and air flow rate at stochiometric mixture is about 3, *i. e.*, between 5 and 16 kW range. Typical water boiling efficiency recorded on the system at various power levels, is in the range of 44 ± 3 %. The efficiency seems lower because of the diameter of the vessel being small. Tests on the emissions from the stove were carried out to determine the CO levels. It was found that both CO level less than 6 ppm which were much lower compared to the case of direct combustion of solid biomass, which is in the range of 20-90 ppm. Periodic cleaning of the ejector and the burner has revealed the presence of only small amount of dust in the pipeline and no tar which could clog the burner.

Similar principles are used in the design of the ejector for a thermal input of 300 kW using the 100 kW gasifier system developed for engine application. Preliminary tests indicate similar performance compared to the 20 kW version, in terms of combustion temperature and quality.

Conclusions

The paper addresses the use of gasification technology for thermal application, in a unique way, by using ejectors. The simplicity in the design and the ease of operation with the gaseous fuel has been discussed, with respect to control on the power level and quality of combustion. The advantage of drawing hot gases from the gasifier to the burner eliminating the elaborate cooling and cleaning system is highlighted. Based on the analysis, a procedure for designing ejector for different power levels as been evolved.

Nomenclature

U	Velocity (m/s)
A	Area of cross section (m^2)
ρ	Density of gas (kg/m^3)
$\dot{m}$	Mass flow rate (kg/s)
P	Pressure (Pa)
M	Momentum ($kg\ m/s^2$)
C_p	Specific heat (kJ/kg/K)

Subscripts

1,2,3	Different section on the ejector as shown in *Figure 1.*
a	Air
g	gas

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Table 1: Measured gas composition and theoretical adiabatic temperature

Sl.No	Exit gas Temp. K	CO_2 %	H_2 %	CO %	CH_4 %	N_2 %	H_2O %	Adiabatic Temp. K
1	300	10	20	20	2	48	0	1935
2	300	11	26	17	1.6	45.4	0	1941
3	600	10	23.6	15.5	1.45	42.2	9.95	1982

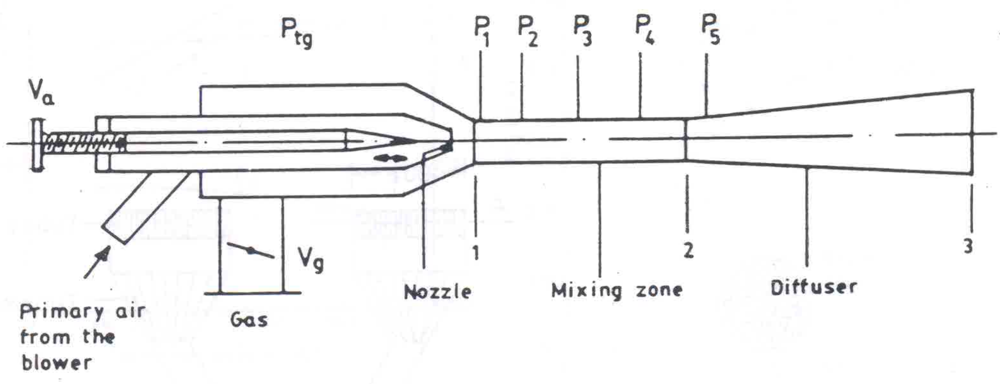

Figure 1: Schematic of a ejector indicating different sections and the static pressure points P_1 - P_5.

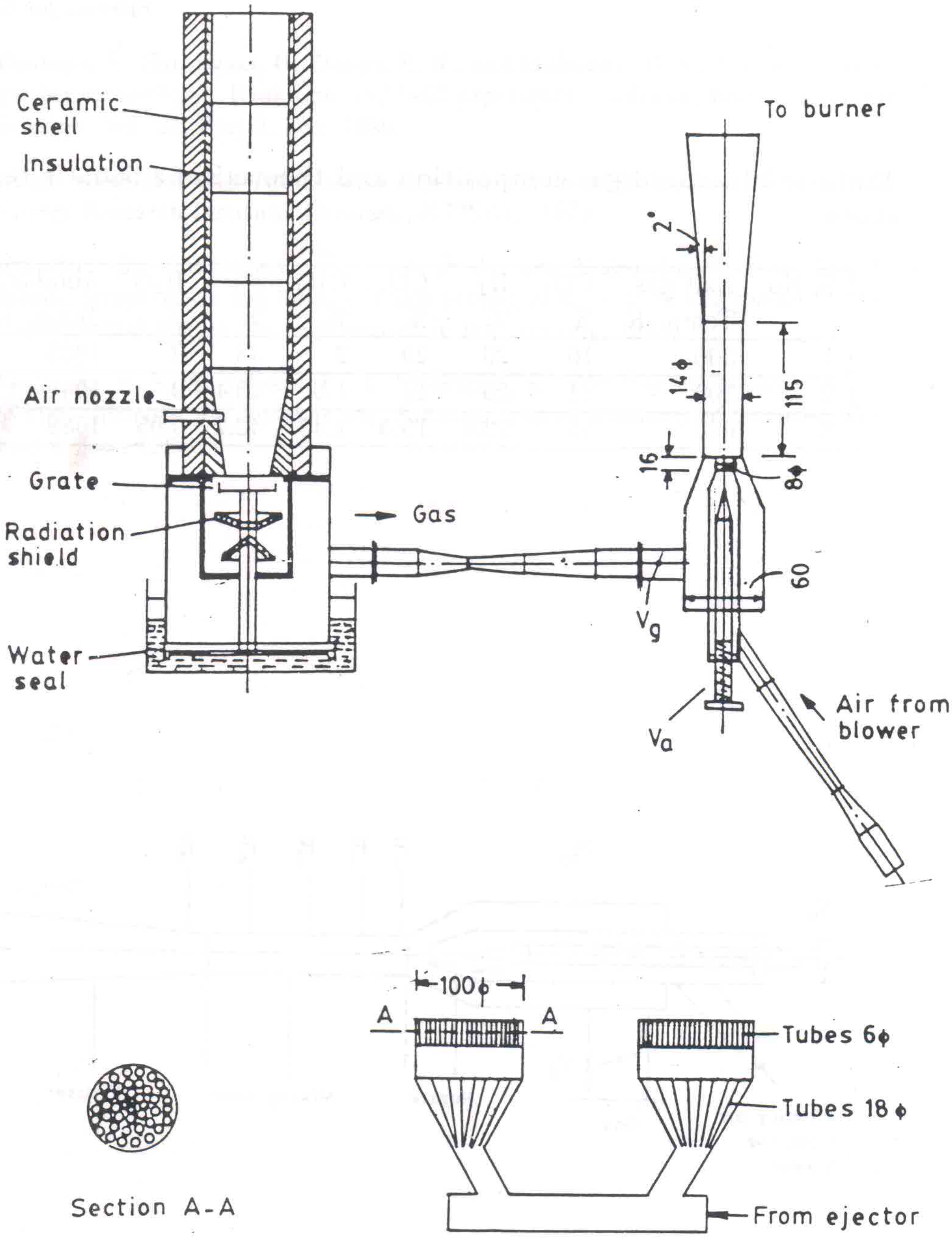

Figure 2: Experimental set up of the gasifier system with the ejector

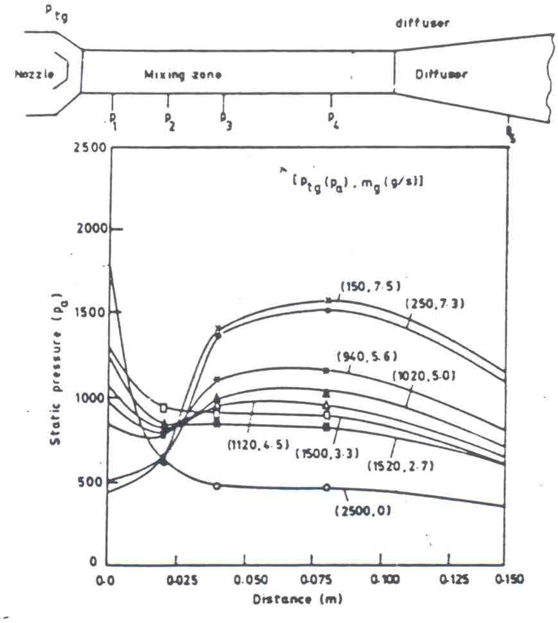

Figure 3: Experimental data on the performance of the mixing portion of the ejector

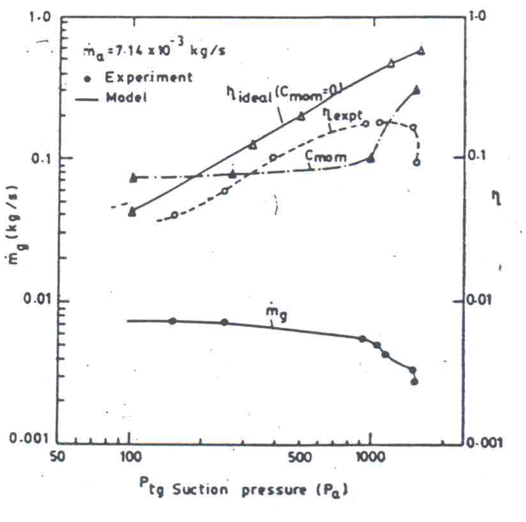

Figure 4: Model prediction and experimental results on the gas and air flow rates along with the ejector efficiency and C_{mom}

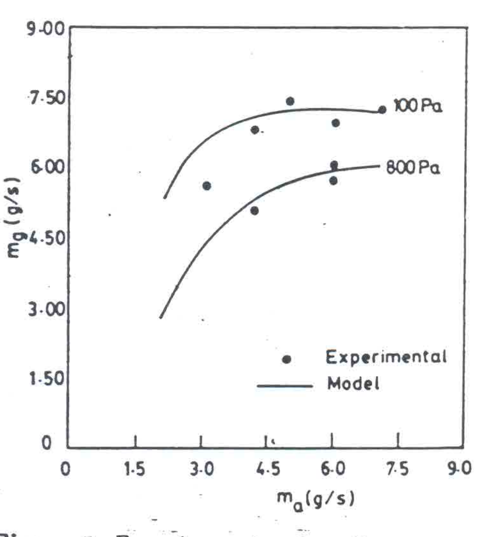

Figure 5: Experimental and prediction of gas flows rates

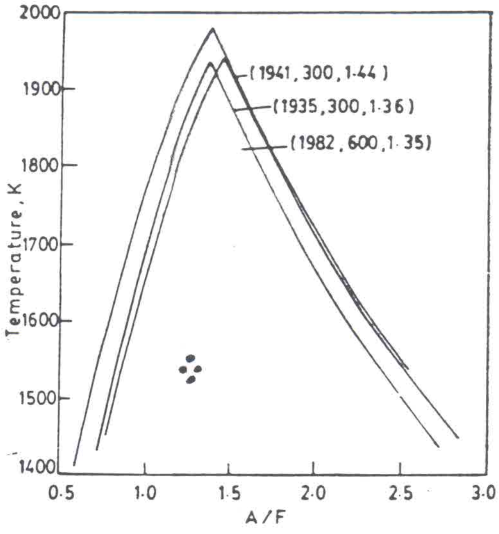

Figure 6: Adiabatic flame temperature for different gas compositions and temperature along with experimental results.(Figures in the bracket indicate Adiabatic flame temperature, K, Gas temperature, K, and A/F)

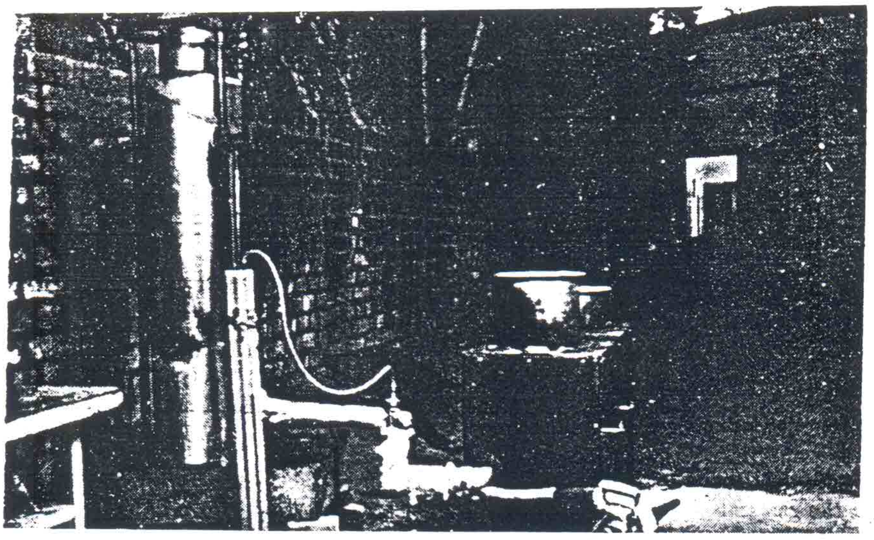

Figure 7: A view of the ejector based gasifier system for thermal application.