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NUMERICAL STUDY OF REPLACEMENT OF AN ATMOSPHERIC BURNER WITH A POROUS BURNER IN A GAS PRESSURE REDUCTION STATION

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ABSTRACT

One of the problems of natural gas pressure reduction stations (City Gate Stations; CGS) is blocking the gas passage due to reduction of temperature in the regulator. In order to prevent this problem, gas is heated in a gas heater before pressure dropping. The gas heater includes, gas passage pipes, intermediate fluid and hot fluid passage pipes. Hot fluid is combustion products of an atmospheric burner and has high thermal energy. Since utilizing of porous media technology has many advantages in combustion systems, the idea of applying this type of burners instead of conventional burners has been widely considered. In this study, replacing of an atmospheric burner with a porous burner in the gas heater of a CGS has been studied numerically. First, by modelling the gas heater with an atmospheric burner, the values of produced heat fluxes at the different equivalence ratios have been studied. The results of these simulations determine the condition which leads to maximum efficiency of combustion in the atmospheric burner. The results indicate that maximum thermal efficiency of the atmospheric burner is obtained in 60 percent of the primary air. In the next step, by modelling the gas heater with a porous burner at various sizes and various equivalence ratios, produced heat fluxes were calculated and compared with those of the atmospheric burner. According to the results, although the gas heater of the pressure reduction station has not been designed for a porous burner, but utilizing a porous burner instead of the atmospheric burner, causes the thermal performance of the heater to increase. The increase of thermal performance at lean equivalence ratios is higher.

INTRODUCTION

A heater is used in a pressure reduction station (in City Gate Stations; CGS) in order to raise the gas temperature before pressure reduction in the regulator. The heater is a heat exchanger with three fluids (natural gas, water, gaseous combustion products). Because of largeness of cold natural gas flow rate, dimensions of these heaters are relatively large. Most of the heaters implement natural draft combustion systems. In such systems chimney effect is utilized as driving force for the combustion products. These systems are usually partial premix type which gas is premixed with primary air (50-75% stoichiometric) and finally is burned with sufficient secondary air. The amount of primary air mainly depends on the gas jet velocity and the amount of secondary air depends on chimney effect.

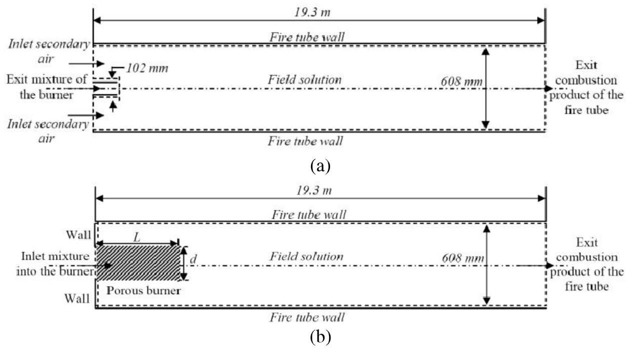
Many experimental studies about combustion in porous media have been performed. Flame stability and Propagation in a sandy medium has bean studied experimentally by Soete [1]. He provided a method for estimating the flame speed and pre-heating effect duo to thermal conductivity of solid mesh. Echigo showed analytically and experimentally that thermal radiation is transferred from the combustion products to the unburned mixture as returning heat in the porous medium [2]. Kawaguchi et al. [3] showed that by increasing the flow velocity from 25 m/s to 10 m/s, the radiation efficiency decreases from 20% to 5%. Mare et al. [4] found that the flammability Limits of Gaseous Mixtures in a porous media dependent more on the porous media geometry than physical properties. Pollutants formation in a household heating system with an atmospheric burner and a porous burner has been compared [5].

Also many numerical studies about combustion in porous medium have been done. More of the previous studies in porous medium combustion modelling have been performed using a one dimensional model. But in the recent two decades, by growth in computer modeling, two-dimensional models [6-7] and three-dimensional models [8] were presented. Malico et al. [7] have simulated a porous burner assembled to a heat exchanger, with an axisymmetric two-dimensional model. They compared the temperature profiles for solid and gas and also formation of NO_x and CO with the experimental data. They found that the computed values for CO is less and for NO_x is more than the experimental values. Recently Hayashi et al. [8] studied variations of temperature at the central line of the burner by modelling a two layers porous burner using a three-dimensional model.

No significant study on the heaters of CGS has been done. In a research the study of replacement of conventional water bath gas heater in CGS with linear/electrical heaters has been studied [9]. In this research, with FLUENT software the atmospheric burner in the heater is simulated and the values of produced heat fluxes at different equivalence ratios have been studied numerically. Then with modelling the porous burner in different sizes and equivalence ratios , comparison of produced heat flux is performed with the atmospheric burner.

GEOMETRY OF THE MODEL

The heater has two fire tubes which combustion products pass through them and transfer heat to the water. These fire tubes have the same shape, so one of them is simulated in this study. Since the burner and fire tube is axially symmetric, so a two dimensional axisymmetric model can be used. Figure 1 shows a view of geometry and boundary conditions of the model. In the modelling of the atmospheric burner, the mixture of fuel and primary air inter the solution field through a 102mm diameter tube. The mixture is then mixed with the secondary air which flows around the tube.



Figur1: Schematic of the solution field geometry of the burner and fire tube (a) atmospheric burner (b) porous burner

In the porous burner model, natural gas is fully premixed and inters the porous media combustor. No secondary air inters in this case.

Table 1 shows different dimensions of the porous burner which are studied in this research. In this table, d and L indicate the diameter and length of the porous burner respectively.

Table1: dimensions of the porous burner

L (cm)	d (cm)
50	20
75	30
100	40

according the specification of the burner, in the maximum thermal capacity of the heater, each fire tube must transfer 27600 (W/m^2) heat flux. The input equivalence ratio (ϕ) and fuel-air mixture speed for the porous burner are presented in Table 2. Because the porous burner is a fully premix type, it works only in lean equivalence ration range (ϕ<1). In the atmospheric burner, in addition to the primary fuel-air mixture inlet, secondary air also enters. Therefore in the atmospheric burner, in addition to the primary equivalence ratio in the burner (ϕ_p), there is also a final (primary air +secondary air +fuel) equivalence ratio (ϕ_f).

It is conventional to use 50 to 60 percent of stoichiometry air as primary air in partial premix burners [10]. Therefore, primary equivalence ratio would be between 1.67 to 2.

Table2: input equivalence ratio (ϕ) and fuel-air mixture speed for the porous burner [3]

ϕ	d (cm)	V (m/s)	ϕ	d (cm)	V (m/s)
1	20	16	0.8	40	4.9
	30	7.1	0.7	20	22.2
	40	4		30	9.9
0.9	20	17.6		40	5.5
	30	7.8	0.6	20	25.6
	40	4.4		30	11.4
0.8	20	19.6		40	6.4
	30	8.7			

Different values of primary and final equivalence ratio which have been studied in the atmospheric burner have been shown in Table3. Output mixture speed and secondary air speed are also shown in this table.

Table3: Cases studied in the atmospheric burner modeling

ϕ_p	ϕ_f	Secondary air speed (m/s)	Output mixture speed from the burner(m/s)	ϕ_p	ϕ_f	Secondary air speed (m/s)	Output mixture speed from the (m/s)burner
2	0.7	1.50	35.6	1.8	0.9	0.90	38.9
	0.8	1.21	35.6		1	0.72	38.9
	0.9	0.99	35.6	1.67	0.7	1.34	41.4
	1	0.81	35.6		0.8	1.05	41.4
1.8	0.7	1.41	38.9		0.9	0.83	41.4
	0.8	1.21	38.9		1	0.65	41.4

COMBUSTION MODEL

Various reaction models can be used for Combustion modelling including from a single-step irreversible model to a full detailed chemical kinetics.

A single-step overall reaction for methane burning for a lean mixture (ϕ<1) is written as:

$$CH_4+2/\phi(O_2+3.76N_2)\rightarrow CO_2+2H_2O+7.52/\phi\ N_2+(2/\phi-2)O_2 \quad (1)$$

This model is easy to use and inexpensive to be applied in a numerical simulation. It is obvious that such simple reaction model can not evaluate pollution emission such as CO, NO etc.

Hsu et al [11] showed that to obtain the accurate profiles of the chemical components, it is necessary to utilize a detailed chemical kinetics. Of course, they expressed that this does not mean that the results of the single-step general reaction is discredited. The single-step general reactions gives acceptable results for lean mixtures 0.7< ϕ<1 [11]. This model gives the burning speed even better than that of some multi-step detailed kinetics model. They expressed that in some cases single-step reaction model is better than a multi-step kinetics model.

Since both the burners operate at lean equivalence ratio, a global kinetics model is used for combustion modelling.

NUMERICAL SIMULATION

A structured mesh is used in the numerical simulation. Some important advantages of a structured mesh is: easy production, fast and easy use of the structured information that causes the calculations speed increases and less memory is required [12].

Various meshes are used for flow modelling in the burner to obtain a suitable simulation. Among these meshes, one mesh is suitable that leads to an acceptable result and the obtained result must be independent of the mesh. This is carried out with refining meshes until the answers almost coincide. Table 4 shows different meshes and their solution time in the atmospheric burner.

It can be seen in table 4 that increasing the number of cells in the solution domain causes solution time to be increase.

Table 4: different meshes used in simulation and required time in the atmospheric burner

Model	Number of point along the fire tube wall	Number of points in the secondary air entry	Number of points in the burner output	Approximate solution time (min)
Model No. 1	1150	5	15	30
Model No. 2	1500	5	25	60
Model No. 3	2000	8	50	210
Model No. 4	2000	15	50	240
Model No. 5	2300	15	75	360

Table 5 shows error percentage of the tube heat flux of the models. This error is obtained by comparison of calculated heat flux with experimental thermal load of the tube.

Table 5: error percent of the meshes

Model	q_m= calculated heat flux in the model (W/m^2)	q_a= experimental heat flux (W/m^2)	Error percent: $\frac{q_a - q_m}{q_a}\times 100$
Model No. 1	22180	27600	19.6
Model No. 2	22700	27600	17.7
Model No. 3	24500	27600	12.2
Model No. 4	27800	27600	0.7

This comparison shows that model No.4 has minimum error relative to others. This may be from suitable locating of the points near the wall, which strongly affects the low of the wall in turbulent modelling.

The same procedure is applied for the porous burner and appropriate mesh is selected. The procedure is not repeated here for briefness.

BOUNDARY CONDITIONS

There are 3 boundary conditions in the problem:

a) Input boundary: this condition is applied at the flow inlet. In the atmospheric burner there is two input boundary; the partially premix condition and the entered secondary air. The flow rate and mass fractions of methane and air are given. For the porous burner, geometry and condition of the inlet flow is given in table 2.

b) Wall: constant wall temperature at outside of the fire tube is used as a thermal wall boundary condition. This arises from this fact that the outside of the fire tube is in water, and temperature of the bath water is nearly constant.

c) Output boundary: at the exit boundary pressure is set to atmosphere pressure (1atm).

RESULTS

In this section, the performance of each burner is studied individually and optimum point of operation for each of them is obtained. Absorbed heat flux by the fire tube is the base parameter for comparison. A comparison is then made between the atmospheric burner and the porous burner in the heater. Heat flux of the burner is calculated in various cases for both burners. Table 8 shows the obtained results. It can be seen that in all cases the porous burner provides heat fluxes greater than that of the atmospheric burner. This difference in equivalence ratios Less than 0.8 is more evident.

Table 6: Comparison of heat flux of the atmospheric and porous burner

Atmospheric burner					
ϕ_p	ϕ_f	Q_{total} (W/m^2)	ϕ_p	ϕ_f	Q_{total} (W/m^2)
1.67	0.7	24100	1.8	0.9	26390
1.67	0.8	25600	1.8	1	27200
1.67	0.9	27100	2	0.7	22580
1.67	1	27890	2	0.8	24100
1.8	0.7	23400	2	0.9	25840
1.8	0.8	24700	2	1	26790

Porous burner							
ϕ_p	L (cm)	d (cm)	Q_{total}(W/m^2)	ϕ_p	L (cm)	d (cm)	Q_{total}(W/m^2)
0.7	50	20	25300	0.9	75	30	26300
0.8	50	20	26630	1	75	30	26990
0.9	50	20	27300	0.7	100	40	24460
1	50	20	27850	0.8	100	40	25660
0.7	75	30	24250	0.9	100	40	26450
0.8	75	30	25500	1	100	40	27220

Figure 3 shows the heat flux as a function of the final equivalence ratio in the atmospheric burner. This Figure demonstrates that increasing ϕ_f to 1 causes the heat flux to increase. Hence, with a stoichiometric fuel air mixture maximum heat flux obtains. On the other hand, with increasing ϕ_p to 2, the absorbed heat flux decreases. This shows that enrichment the primary mixture with fuel decreases heat flux. It should be noted that heat transfer to fire tube in the atmospheric burner is mainly convection type. Emissivity coefficient of gases is small, so radiation heat transfer is negligible in comparison with convective mode.

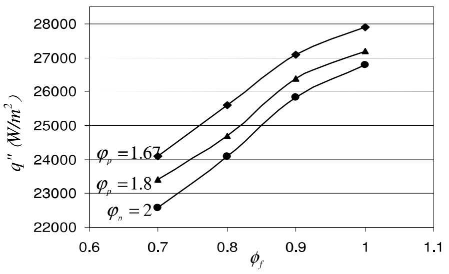


Figure 3: heat flux of the fire tube versus primary and final equivalence ratio in the atmospheric burner

When a porous burner is used, because of high radiative characteristic of it, radiation heat transfer from the burner becomes important. Radiation heat transfer of the burner depends on surface temperature, emissivity factor of the porous media and size of the burner.

Table 10 shows radiation heat transfer (Q_{rad}), convective heat transfer (Q_{con}) and total heat transfer from the porous burner in different equivalent ratios of the inlet mixture and different sizes of the burner. Both Q_{rad} and Q_{con} increase when the inlet mixture becomes fuel rich (i.e. ϕ increases to 1). In this table, L and D are length and diameter of the burner respectively.

Table 7: radiative, convective and total heat transfer in the porous burner

L (cm)	D (cm)	ϕ	Q_{rad} (W/m^2)	Q_{con} (W/m^2)	Q_{total} (W/m^2)
50	20	1	680	26770	27850
		0.9	620	26500	27300
		0.8	550	26080	26630
		0.7	466	24840	25300
		0.6	370	23450	23820
75	30	1	1300	25690	26990
		0.9	1120	25180	26300
		0.8	1010	24490	25500
		0.7	820	23430	24250
		0.6	660	22640	23300
100	40	1	2540	24680	27220
		0.9	2200	24250	26450
		0.8	1860	23800	25660
		0.7	1450	23010	24460
		0.6	1160	22190	23350

It can be seen in Table 10 that maximum heat flux is in equivalence ratio 1 and in L = 0.5m, d = 0.2m, which is the minimum size of the burner. Variation of radiative heat flux with respect to equivalence ratio in different sizes of the burner is shown in Figure 4. As expected, radiation heat flux increases with increasing burner sizes. It should be noted that in the simulations, maximum temperature of porous surface is almost 1300 K.

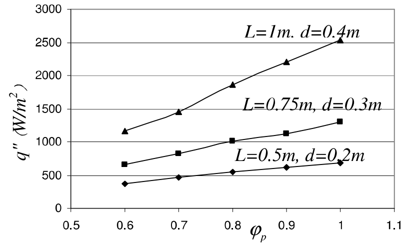


Figure 4: radiative heat flux versus equivalence ratio in different sizes of the porous burner

Variation of convective heat flux with respect to equivalence ratio in different sizes of the burner is shown in Figure 5. Convective heat flux increases with decreasing burner dimensions.

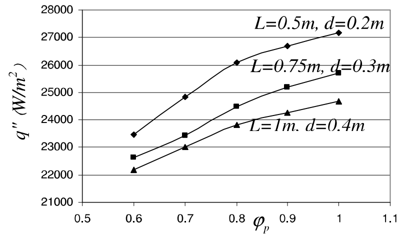


Figure 5: convective heat flux versus equivalence ratio in different sizes of the porous burner

Reduction of the convective heat flux due to increasing the burner size may result from two factors. In a same equivalence ratio, at a larger porous burner, the speed of hot products at the exit of the burner is smaller. This causes the heat transfer coefficient to reduce. Another factor is the larger contribution of radiative heat transfer in the larger porous burners. In this case hot products emit some of their energy by radiation, so the temperature of products reduced more.

Figure 6 demonstrates total heat flux of the porous burner versus equivalence ratio for different burner sizes. An interesting behaviour of total heat flux with respect to dimension of the burner can be observed. It can be seen that in spite of convective heat flux, the amount of total heat flux with dimensions L = 1m, d = 0.4m in comparison with dimensions L = 0.75m, d = 0.3m is larger. This arises from increasing radiative heat transfer in larger dimension.

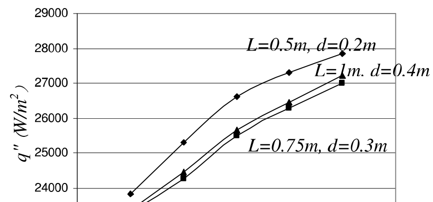


Figure 6: total heat flux from the porous burner versus ϕ in different sizes of the burner

Operation of the porous burner and the atmospheric burner is compared in Figure 7. Total heat flux is sketched versus final equivalence ratio for the both burners. This figure shows that the performance of the porous burner is more than the atmospheric one, specially in lean inlet mixture (smaller equivalence ratio).

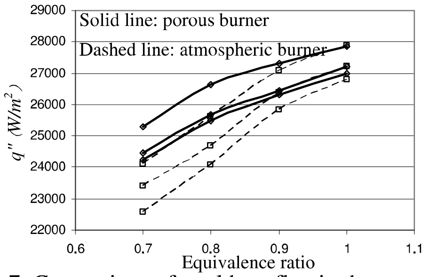


Figure 7: Comparison of total heat flux in the porous burner and the atmospheric versus ϕ

It is important to note that the fire tube is very long (≈20m) relative to the burner length (max. ≈1m). Hence the effect of the burner is limited to a small portion of entire the fire tube. In spite of this fact, the effect of burner replacement is evident in conventional lean mixtures.

CONCLUSION

In this study, replacing the atmospheric burner in the heater of a CGS by a porous burner has been studied with numerical simulation. The performance of each burner is studied individually and optimum point of operation for each of them is obtained. Absorbed heat flux by the fire tube is the base parameter for comparison. The results of this study are as follow:

1- Maximum heat flux in the atmospheric burner is obtained in 60% stoichiometric primary air (ϕ_p=1.67) and stoichiometric final air in the mixture (ϕ_f=1).

2-Reduction of ϕ_f in the atmospheric burner causes large reduction in the absorbed heat flux. Hence making the mixture to be lean for completeness of combustion must be at least.

3- Maximum heat flux in the porous burner is also obtained in a stoichiometric fuel air mixture and leaning the mixture causes heat flux to decrease.

4- In the porous burner, radiative heat flux increases and convective heat flux decreases when size of the porous burner increases. Maximum total heat flux is obtained at the minimum size.

5- Comparison between two burners shows that absorbed heat flux by fire tube in the porous burner case is greater than that of the atmospheric burner. The difference becomes larger in smaller ϕ (leaner fuel mixtures)

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