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Computational Fluid Dynamic Analysis of Reduction Gas Emissions Level inTurbine Combustor

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Abstract— **in this paper, (CFD) Computational Fluid Dynamics was used for a detailed burner design by the software Fluent (ANSYS). However, a number of numerical simulations were performed on a generic swirl burner and turbulence flow selection of a non-premixed which using syngas.**

However, non-premixed flames which characterized by relatively high temperatures, high NOx concentrations, and emission indices. The results showed that the presence of methane in syngas increases the peak flame temperature and the thermal NOx, significantly.

Therefore, Investigation showed that effecting of H2, CO, and N2 contents in the fuel mixture level NOx emissions, thus the present compositions for pure methane are respectively influenced on syngas1 by (10% CH4, 45% H2, 45% CO) , syngas2 by (50% CH4, 10% H2, 40%N2) and syngas3 (60% CH4, 20% H2, 20%N2).

Index Terms: **CFD, Non-premixed flame, Swirl flow burner, Combustion Emissions.**

I. INTRODUCTION

Over the last two decades the world has rapidly developed. The population has increased, the economy has grown and the middle class has expanded, all of which contributes to a growing demand for energy. This results in unsustainable climate change. Yet, awareness of our environmental problems creates a demand for sustainable energy sources. New types of fuels are introduced at an increasing rate, creating demand for new technologies[1]. Investments in new technology require a considerable return: cost efficient use of plant assets and fuel resources, legislated reduced emissions (such as NOx, CO, CO2 and others), reduced and simplified maintenance, and modernized operations. Additionally, the new technology should embrace efficiency, sustainability, flexibility and manageability in order to embrace the future[1]

Gas turbine engines derive their power from burning fuel in a combustion chamber and using the fast flowing combustion gases to drive a turbine in much the same way as the high pressure steam drives a steam turbine[2][3].

A swirling flow is defined as a one undergoing simultaneous axial and vortex motions. It results from the application of a spiraling motion, a swirl velocity component (tangential velocity component) being imparted to the flow by the use of swirl vanes, axial-plus tangential entry swirl generators or by direct tangential entry into the chamber [4][5].Swirling flow burners have been essential to both premixed and non-premixed combustion systems because of their significant beneficial influences on flame stability, and combustion intensity, as well as the combustor performance. Until now, gas turbine combustors and industrial systems utilized a high-swirl type of burner in which the swirling motion generated by the injector (or burner) is sufficiently high to produce a fully developed internal recirculation zone at the entrance of the combustor[6].

In many combustion applications, fuel and oxidizer enter separately into the combustion chamber where they mix and burn during continuous mutual diffusion; this can be called non-premixed combustion. Classic examples are combustion in a furnace, diesel engine and some gas turbine applications[7]. The non-premixed combustion reactions occur in the swirl stabilized combustion zone with the reactants being converted into products downstream. They then are diluted by secondary air to reduce the temperature at the exit of the combustor to values that are acceptable for the turbine blade material. In modern stationary gas turbines, liquid fuel is often pre-vaporized and partially premixed before entering the gas turbine combustion chamber. Similar partial premixing occurs with natural gas[8][9]. Models used for partially premixed combustion are more relevant for describing the flame propagation and combustion processes occurring in these engines. Sometimes non-premixed combustion (called diffusive combustion) is used[10].

For conventional non-premixed combustion, the role of the large recirculation zone, also known as the toroidal vortex core, is to promote turbulent mixing of the fuel and air. In non-premixed combustion systems, the

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recirculation zone provides a stable heat source for continuous ignition of the fresh reactants[10][11]

Gas turbine (GT) swirl combustors, such as the general can-type configuration is shown in Figure 1. These combustors rely on the interaction between the turbulent flow field and complex chemical reactions within the primary, intermediate, and dilution zones to generate the required turbine inlet temperature for the given load condition while reducing emissions, pressure loss, and instabilities[12]. The design of the generic swirl burners utilized in this study is intended to replicate this flow through, single can-type combustor operated in a fully non-premixed configuration. This geometry derives its flame stabilization mechanism mostly through the vortex breakdown structures resulting from the tangential velocity imparted on the flow through the swirler in combination with the sudden expansion into the combustor primary zone[13][14].

The present study aims to investigate numerically the problem of NOx pollution, carbon monoxide and carbon dioxide using a model swirl burner in an industrial gas turbine utilizing fuel syngas.The importance of this problem is mainly due to its relation to the pollutants produced by gas turbine used widely in generated industrial plants. The swirl burner under investigation is a 100 KW power output, gases fired with methane gas and two syngases.

II. NUMERICAL METHODOLOGY

CFD modeling was used to simulate the isothermal of swirl burner. A 100kW swirl burner constructed from stainless steel was used to examine the flow behave limits at atmospheric conditions (1 bar, 293 K). The nozzle were used with angles 45°, with swirl numbers of 1.05. A single tangential inlet (a) feeds the premixed air to an outer plenum chamber (c) which uniformly distributes the gas to the slot type radial tangential inlets (c). Swirling unburned fuel then passes into the burner body (d), then into the burner exhaust (e) where the gases pass around the flame stabilizing central recirculation zone. The central diffusion fuel injector (b) (which used for fuel during the simulation investigate) Figure1.

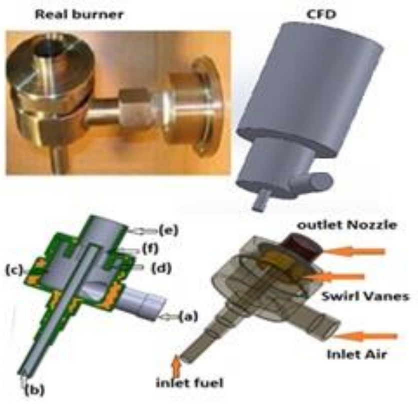

Figure 1. Schematic of the generic burner.

Swirl combustors and burners are usually characterized by their degree of swirl, via a swirl number (S). For this particular paper, the swirl element of S = 1.05 has four tangential inlets which are symmetrically distributed figure 2.

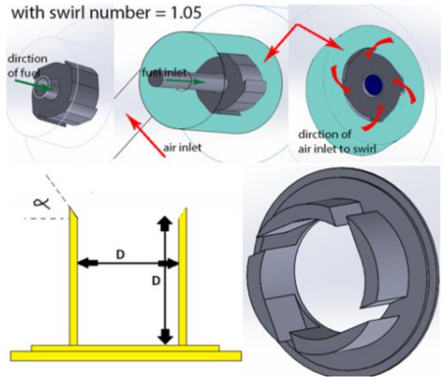

Figure2. Angular nozzle and geometrical swirl respectively.

The swirl burner gives good flame stabilization but produces a CRZ that extends back over the central fuel injector, allowing the flame to propagate into this region. Simulations were conducted using a range of CH4, H2, CO and N2 blends, Table 1.

Table 1. Gas Compositions.

GAS NO	CH4%	H2 %	CO%	N2%
CH4	100	0	0	0
SYNGAS 1	10	45	45	0
SYNGAS 2	50	10	0	40
SYNGAS 3	60	20	0	20

First numerical experiments were run at different equivalence ratios to determine stability trends. Then, numerical modelling were performed using Low Power (27KW), medium Power54KWand high Power81KW, as detailed in table 2

Table2. CFD boundary conditions.

Gas No	P (bar)	T(K)	V air (m/s)	V fuel (m/s)
27KW				
CH4	1	300	31.817	11.440
Syngas1	1	300	14.583	9.101
Syngas2	1	300	37.120	12.511
Syngas3	1	300	31.817	10.562
54 KW				
Ch4	1	300	63.635	22.881
Syngas1	1	300	29.166	18.203
Syngas2	1	300	74.241	25.023
Syngas3	1	300	63.635	21.125
81 KW				
Ch4	1	300	95.453	34.322
Syngas1	1	300	43.749	27.305
Syngas2	1	300	111.362	37.535
Syngas3	1	300	95.453	31.687

III. NUMERICAL TECHNIQUE

Inlet conditions were set at 1 bar and 300 K. Various solvers were investigated with preliminary tests being run with pure methane at flow rates between 0.5 and 0.4g/s. After that the best agreement was observed using the k-ϵ model .To obtain the specific dissipation rate ϵ, and turbulence kinetic energy k, Simulations were performed using all syngases in Table 1 under non-premixed conditions with ANSYS FLUENT 14.5. The pre-processor used to construct the model grid was ANSYS.

Three meshes were examined in order to perform a mesh independency analysis, Table 3 After carrying out the analyses, it was decided to use a low size mesh that consists of 13003 nodes and 52493 elements, and that provided mesh-independent results when compared to the high size mesh. The mesh was designed with a structured grid creating a higher node density in areas where the flow was expected to change considerably, i.e. close to the nozzle, Figure 3.The PRESTO discretization scheme was used for pressure, with the SIMPLE scheme for pressure-velocity coupling using a convergence criterion set at 10^{-4}. Non-slip wall boundary conditions were defined using non adiabatic conditions at 1 bar inlet pressure and inlet temperatures of 300 K for a Steady-state analysis.

TABLE3. Mesh Independency

Mesh density		Swirl number1.05	
A	Low quality (coarse)	Nodes	13003
		Elements	52493
B	Medium quality(medium)	Nodes	15504
		Elements	64575
C	High quality (Fine)	Nodes	28719
		Elements	133699

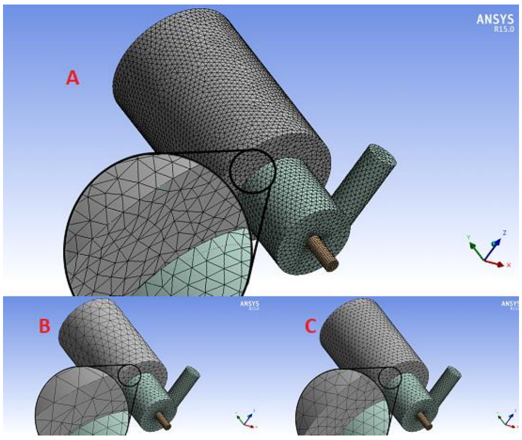

Figure 3. Mesh Distribution in different quality from high to low

IV. BURNER SIMULATION AND RESULTS

The 100 kW swirl burner has been the subject of extensive testing in this simulation study. The effect of changing fuel compositions on the NOx , carbon monoxide , carbon dioxide emissions , and burner stability are investigated - at stable flame region. During the simulations, the main burner geometry and the nozzle are kept constant. The burner is tested with a power output in the range 27 kW up to 81 kW. Swirl burner is optimized using three syngas's as presented in table 1.

In the present simulation, the masses fraction of exhausted gases emitted from the combustion chamber were determined. As well as investigating the mixing of three gases with methane in varying proportions, which purposes to reduce NOx , CO and CO2 as results of CFD simulation.

It is obvious in Figure 4. that the levels of nitrogen oxides rise to the highest level, when the syngas 1 is used at all power outputs equally. Thus, there is a noticeable decrease in nitrogen oxide levels when using methane and the rest of the other gases.

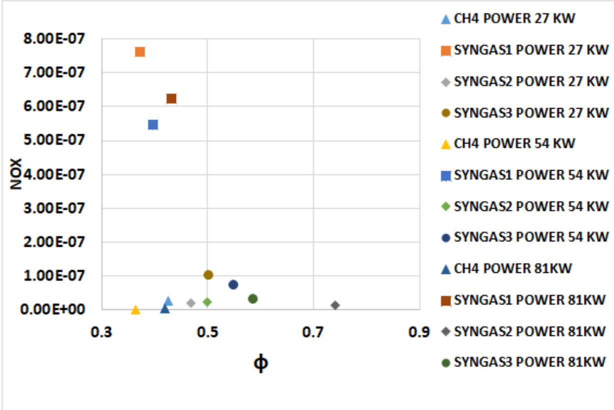

Figure 4.Comparison NOx emissions and equivalent ratio using CH4 , SYNGAS 1 and SYNGAS 2 .

In Figure 5, it appears that when syngas 1 was used by changing the power from 27 to 81 kW, there was a significant increase in the level of carbon monoxide emissions, while there is a noticeable decrease when burning methane gas to the average level at all power outputs, and still a decrease when using the syngas 3. There is a noticeable gradual decrease when the syngas 2 is used and this is due to the high percentage of nitrogen in the gas compositions.

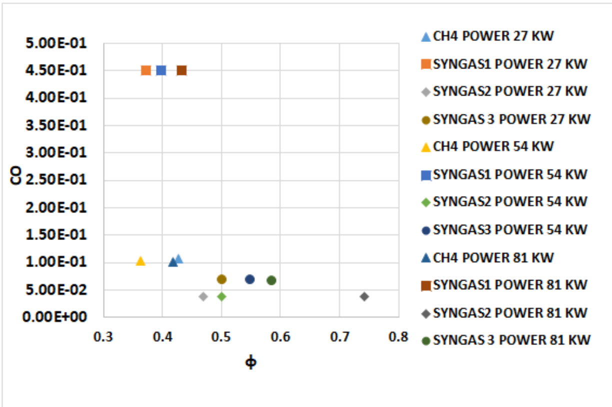

Figure 5 .Comparison CO emissions and equivalent ratio using CH4 and SYNGAS 2.

It is manifest in figure 6 that the emission rate of carbon dioxide, reaches the highest level when using pure methane gas at the all power output .However, the emission rates of carbon dioxide reach their lowest levels when using syngas 1 with all power output levels. Despite the retention of the second and third syngas at the average level of the total emissions.

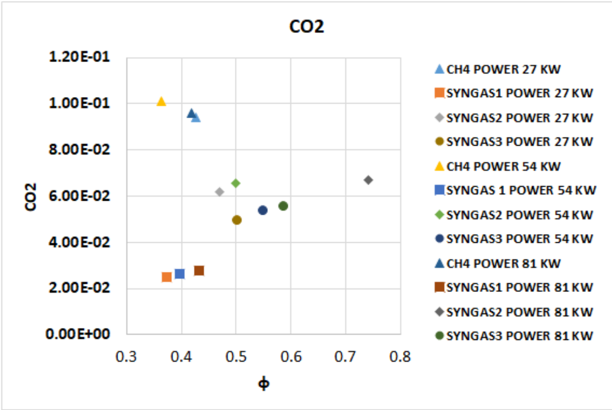

Figure 6. Comparison CO2 emissions and equivalent ratio using CH4 and SYNGAS 2

The effect of changing the amount of nitrogen gas from 20% to 40% in fuel mixtures on NOx emissions is significantly evident in Figure 7. For the case using 20% nitrogen gas there is a significant drop at the power of 27 kW. and reached the lowest level when producing the highest energy at 81 kW. On the contrary, when using 40% nitrogen gas higher levels of nitrogen oxide pollutants are emitted despite the noticeable decrease at 54 kW, and remained constant at 81kw.

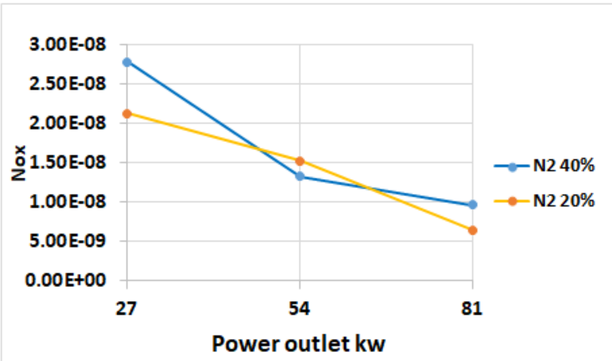

Figure 7 . Effects of N2 percentage on NOx emissions level using different power output at SYNGAS 2.

In the figure 8 the comparison of O_2 percentage consumed using different gases CH_4 and syngas 2 at all power output was presented, which shows that the syngas 2 consumed more oxygen compared to methane gas. The relationship between the amount of work generated and the stoichiometric equivalence ratio of the gas mixture, was observed reavling that higher power output requires less amount of oxygen for complete combustion.The difference between power output 27 and 54 becomes manifested with a slight rising in O_2 when the power output increased further.

Figure 9 shows the range of the comparison of temperatures using syngas 2 at various energy output 27, 54 and 81 KW. The observation is that the temperature is directly proportional to the energy extracted i.e. the more the energy produced the more the temperature as shown by the results respectively (27- 54- 81 kW produces 1800 K, 1910 K, 1950 K).

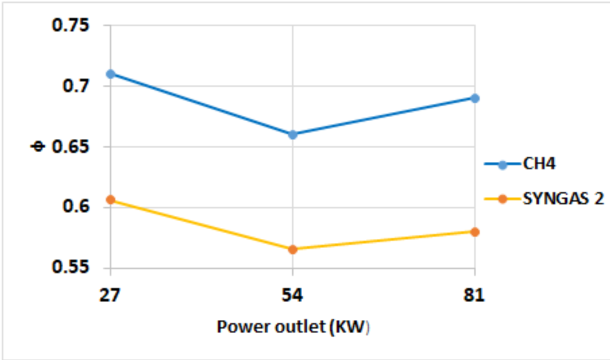

Figure 8Comparison of O2 percentage using CH4 and syngas 2 at all power output.

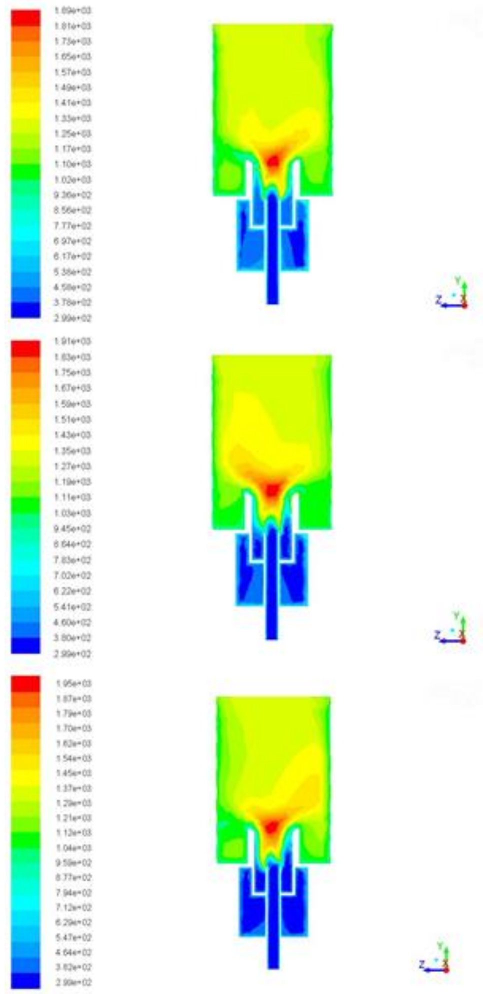

Figure 9 Contours of the temperature using syngas 2 at 27, 54 and 81 KW.

V. CONCLUSION

The combustion emissions are an item of interest during an investigation of the complex flows within a combustor. Emission of pollutants caused by combustion includes various volatile organic compounds harmful chemical compounds such as unburned hydrocarbons (UHC) and carbon monoxide (CO), and oxides of nitrogen. These components have been found to be environmentally detrimental in numerous ways. Carbon monoxide is a poisonous gas that can be harmful in very low quantities. The simulation provided insight on the correlation between maximum burner temperatures, burner average temperatures and average exhaust gas temperatures on NOx concentrations.

The results have shown that the increase in excess air at a given air mass flow rate results in a reduced exit temperature and reduced NOx concentrations. The dilution with nitrogen of the fuel mixture shows considerable effects on reduction of levels of nitrogen oxides. Therefore, the results of these simulations could help in improving the predictive emission monitoring techniques, more effective instrumentation and monitoring of the combustion conditions and better control of the swirl burner. Although considered more of a by-product of combustion than a pollutant, carbon dioxide has been classed and monitored from the graphes. It is noticeable from the amount of carbon dioxide produced, as opposed to the amount of carbon monoxide produced at the same mixture for the all power output. NOx formation during the combustion process occurs mainly through the oxidation of nitrogen in the combustion air (thermal NOx) and through oxidation of nitrogen with the fuel (prompt NOx). The simulation study provided the NO distribution in the combustion chamber and in the exhaust gas at various operating conditions of fuel to air ratio with varying either the fuel or air mass flow rate.In particular, the simulation provided more insight on the correlation between the maximum power output 81 kw and medium power output 54 kw, and lowest power output and the thermal NOx concentration.

An evaluation of turbulence models and combustion models suitable for studying the Swirl Burner by computational fluid dynamics has been carried out. For this evaluation, a 3D computational model of the 100-kW burner has been used. For closure of the Reynolds Averaged Naiver-Stokes equations for turbulent flow, two models have been evaluated. These are the standard k-ε model, the kε- SST model. While for modelling of combustion, one model has been evaluated, namely the non-premixed model. For studying the swirl burner, a combination of the k-ε model, and the k-ε SST model were found to be most suitable for modelling of turbulence and combustion respectively. Computational results with the 100-kW burner indicate that fuel gases are recirculated into a central toroidal recirculation zone downstream of the burner exit. The higher the production capacity, the more the central recycling area distances. This makes the fuel and air mixture more homogeneous.

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