

Development of a micro scale pulverised coal test burner

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DECLARATION

I, André Johan Storm, hereby declare that this is my own work and no plagiarism was committed.



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15 January 2016

Date

ABSTRACT

The power demand in South Africa continues to grow which makes it more important to understand different input parameters in power stations. One of these input parameters is the calorific value (CV) of coal. The current type of CV used in power stations is the gross calorific value at constant volume (GCVv). The GCVv is determined with a bomb calorimeter, which does not represent an accurate CV for coal used in a power station, because of the constant volume process of the bomb calorimeter. The constant volume process enables the bomb calorimeter to recover the latent heat of the moisture, which does not happen in a burner of a power station. The net calorific value at constant pressure (NCVp) represents a more accurate CV for coal used in a power station since it does not recover the latent heat of the moisture. To determine the NCVp, a flow calorimeter is required, which is essentially a coal burner that operates at constant pressure. Currently, no small-scale NCVp analysers exist.

The problem is to devise a functioning pulverised coal (PC) micro-burner that discharges into a sized single combustion chamber and complies with the Fossil Fuel Firing Regulations (FFFR) of Eskom. This burner must function as an independent system without the flame support of burners positioned adjacent to or opposite it and without the positive influence of a large common furnace. The burner must be able to sustain stable combustion for such a period of time that all applicable parameters can be measured representatively.

This micro coal burner was designed to operate on different types of coal and to be as small as possible in rating and physical dimensions. Because a minimum of 1.6 g/s PC is fed to this burner, the burner was designed to be 40 kW in thermal rating. To mimic combustion taking place in a full-scale burner on a power station, the micro coal burner is scaled down and operates in a similar manner. A gas burner was designed and manufactured to act as an igniter to the coal and as an experiment to predict the behaviour of the coal. The gas burner will operate in the same manner as the micro coal burner only on an even smaller scale. Different factors affecting combustion were investigated to be implemented on the design of the burner. These factors were swirl, residence time, recirculation zone, fineness and burnout time of PC. The settling of PC inside the transporting pipes was also taken into account.

Firstly, combustion calculations were done to determine the air-to-fuel ratio (A:F) with 20% excess air for minimum, nominal and maximum CV coals to enable proper combustion. Because of the gas igniter, these combustion calculations were also done for liquid petroleum gas (LPG). A safe A:F inside the pulverised fuel/primary air (PF/PA) and primary core air (PCA) tubes was determined for safety purposes to comply with the FFFR.

Taking the burning velocity of 0.4 m/s of coal into account, the dimensions of the tubes were calculated accordingly. For proper drying of coal and better combustion, pre-heating of air was applied for the different airflows in the calculations. The gas burner was 15% in thermal rating compared to the micro coal burner.

At first, it was decided to supply the air to the burner via tangentially placed holes. The gas burner experiment showed how the combustion of gas reacted to these tangentially placed holes. In summary, the tangential placed holes provided sufficient swirl as well as a large recirculation zone. The gas burner achieved complete combustion, however, intense heat was forced back into the tubes by the vortex, which resulted in metal temperature excursions. Computational fluid dynamics (CFD) was used to investigate this problem further.

To prevent this and other problems occurring in the micro coal burner, CFD was used to design swirl generators. These swirl generators have been designed to adequately create swirl, a recirculation zone and ensure safety. The swirl generators were also designed to be dependant upon one another ensuring equal angular and axial velocities upon exiting the swirl generator holes.

After the designing process, the micro coal burner was manufactured and assembled. The burner indicated proper combustion for gas, but not for coal. The gas flame's behaviour and shape correlated well with the CFD done on the design. A portion of the coal burns, but a stable flame cannot be sustained without gas support. This occurs because of improper mixing of the gas flame and coal as well as a small heat zone.

To solve these problems for future modifications, an extension of the PF/PA tube is implemented after the PF/PA swirl generator. This is to contain the PF and to mix it properly with the gas flame. The hot PF will then exit the extended PF/PA tube into an extended secondary air (SA) tube that is required to be attached to the SA swirl generator. The above-mentioned assumptions indicate that the PF will now have adequate air and heat to combust.

The modifications have been implemented on a new design and a CFD analysis was done which correlated well with the assumptions. These modifications will be implemented as the next stage in the quest to develop a micro burner that can ensure sustained combustion for a small-scale NCVp analyser.

KEYWORDS

Airflow
Burner
Burner
Calorific value
Coal
Combustion
Mini, micro
Pulverised
Swirl

DEDICATION

This dissertation is dedicated to my loving wife Jana, who supported me throughout the completion of my study.

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NOMENCLATURE

BS	British Standard
CFD	Computational fluid dynamics
CV	Calorific value
DRB	Dual register burner
DTF	Drop tube furnace
FFFR	Fossil fuel firing regulation
GCV _v	Gross calorific value at constant volume
ISO	International organisation for standardization
LEL	Lower explosion limit
LPG	Liquid petroleum gas
NCV _p	Net calorific value at constant pressure
PA	Primary air
PC	Pulverised coal
PCA	Primary core air
PF	Pulverised fuel
PSCTF	Pilot scale combustion test facility
RTD	Research, Testing and Development (Eskom facility in Rosherville).
SA	Secondary air
SCA	Secondary core air
UEL	Upper explosion limit

LIST OF SYMBOLS

A	Area
A:F	Air-to-fuel ratio
C	Carbon
C ₃ H ₈	Chemical formula for propane
C _d	Coefficient of discharge
CO	Carbon monoxide
CO ₂	Carbon dioxide
C _R	Resultant velocity
C _T	Tangential velocity
C _X	Axial velocity
H ₂	Hydrogen
H ₂ O	Water
ID	Inner diameter
IR	Inner radius
kW	Kilo watt
kW _T	Kilo watt thermal
L	Length
LEL	Lower explosion limit
m	Mass
$\dot{m}$	Mass flow
M _i	Inherent moisture
M _S	Surface moisture
M _T	Total moisture
MW	Mega watt
n	Number
N ₂	Nitrogen
NO _x	Nitrous oxide
Ø	Diameter
O ₂	Oxygen
OD	Outer diameter
OR	Outer radius
P	Pressure
q	Specific heat energy
Q	Heat energy
$\dot{Q}$	Heat energy rate
R	Specific gas constant
r	Radius
S	Sulphur
SCA	Secondary core air
SO ₂	Sulphur dioxide
T	Temperature
UEL	Upper explosion limit
V	Volume
$\dot{V}$	Volume flow
η	Efficiency
θ	Swirl angle
μ	Micro
ρ	Density
ϕ	Equivalence ratio
ω	Angular velocity

1. INTRODUCTION

1.1 BACKGROUND

It is a well-known fact that coal is one of the most important sources of energy in the world and will remain one for a significant time in the future (Kurose *et al.*, 2009:144; Zhou & Cen, 2007:718). More specifically, 46% of the world's energy is derived from coal as primary energy resource for electricity generation (IEA, International Energy Agency, Annual report 2013- <http://www.iea.org/topics/coal/>). In South Africa, the figure is even higher, 85-90% (South Africa. Department of Energy, 2010) of the electrical energy generation uses coal as primary energy resource. Therefore, it is very important for plant design, maintenance, the commissioning of a newly built plant, and especially for the operation and optimisation of the process, that coal characterisation is carried out. For plant performance and thermal efficiency, this is vital. To enable this, facilities are necessary for the testing of coal to evaluate its combustion behaviour and to determine its properties. Coal properties vary over a relatively wide range and not all coal is suitable for any steam boiler. These properties are typically coal quality (CV), ash and moisture content, the ratio of volatile matter to fixed carbon (reactivity) and the fineness to which the coal is pulverised, etc.

It is relatively expensive (because of loading and daily energy demand) to perform such tests on full-scale burners at power stations, also owing to their thermal energy consumption. Generally, a large power station consists of six units, where each unit has six mills with four to six burners each. The capacity of these burners usually ranges between 50 and 65 MW each and it can be calculated that the cost to execute a test can range from 1000 – 1500 R/h, on coal cost alone. It is also impossible to test such a burner in isolation. The ignition, combustion, and flow of a single burner are influenced by the flame support of adjacent and opposite burners in the furnace as a whole. The combustion chamber in this case is the large furnace volume; the greater environment thus absorbs the swirling effects of the single burner.

Therefore, smaller scale test burners are required to perform such tests for evaluation that is more representative and lower in cost. The only such pulverised coal (PC) burner in South Africa is a 1 MW pilot scale test burner at Eskom Research Testing and Development (RTD) in Rosherville, Gauteng (Rajoo, 2010:11). This facility is suitable for extensive coal characterisation tests, but at 1 MW thermal rating, it is still relatively large and expensive to operate.

Since coal is the primary energy fuel resource used in power stations (van der Merwe, 2014:1), it is very important to know the effect of the burners, mills, and combustion for design specification and optimisation thereafter. The quality or rank of coal is characterised by its volatile matter and carbon (C) content, and the calorific value (CV) (Kitto & Stultz, 2005:9.5). The elemental or ultimate analysis determines the nitrogen (N_2), oxygen (O_2), total C, ash, sulphur (S), hydrogen (H_2), and moisture content by using an ultimate analyser (Kitto & Stultz, 2005:9.8). The proximate analysis is determined manually according to British Standard (BS ISO 17246:2010), American standard of testing and materials (ASME) procedure D3172 or a thermo-gravimetric analyser. These proximate analyses provide the values of volatile matter, fixed C, moisture, and ash content (Kitto & Stultz, 2005:9.5). The CV can be determined by a calculation approximation, by using the Dulong equation, or determined physically by using a bomb calorimeter (Kitto & Stultz, 2005:9.8; Kitto & Stultz, 2005:10.10; Obert, 1973:91).

Of these properties, the most important parameter required is the correct CV, dictated by the applicable process. The CV of a substance specifies the amount of energy available when it combusts completely (Kitto & Stultz, 2005:9.8). However, there are two methods of measuring CV: gross calorific value at constant volume (GCV_v) and net calorific value at constant pressure (NCV_p) (Kitto & Stultz, 2005:9.8; Obert, 1973:91). Both the NCV_p and GCV_v can be calculated theoretically by means of the Dulong method (Kitto & Stultz, 2005:10.10; Storm, 1998:A-56). Calculation by means of the Dulong method utilises the specific CV of each pure combustible element (C, S, H) in the coal, which was laboratory-determined, for constant volume and constant pressure processes. The coal CV (consist of these elements) is calculated on a weighted average of these element CVs with the ultimate analysis percentages of these elements (Storm, 1998:F1).

The GCVv is experimentally determined with a bomb calorimeter (Obert, 1973:91), since the latent energy of the moisture is recovered over time (Kitto & Stultz, 2005:10.10). The bomb calorimeter is an artificially ignited, high pressure, closed system process (constant volume) where coal is ignited electrically in a pure O₂ atmosphere and the energy is measured in MJ/kg (Kitto & Stultz, 2005:10.7; Obert, 1973:91). This is the method used at mines and power station laboratories, although it is not representative of the actual combustion of coal in a burner. A flow calorimeter is needed to determine the NCVp experimentally, since it functions in a constant pressure process mode and does not recover any latent energy of the moisture vapour (Kitto & Stultz, 2005:10.10; Obert, 1973:92). Currently there are no flow calorimeters available for solid fuels in practice, only for liquid and gaseous fuels (Obert, 1973:91).

Kitto and Stultz (2005:10.10) state: Although the GCVv can be accurately determined, it is difficult to establish the NCVp since there is no international standard for calculating the NCVp from the measured GCVv. The following comments can motivate this statement:

Firstly, a bomb calorimeter does not even determine the GCVv accurately enough for thermal efficiency calculations of such a plant. The accuracy is normally within ± 0.25 MJ/kg. The number of significant figures renders the efficiency answer accuracy inadequate for performance evaluation and optimisation purposes (Storm, 1998:A56-58). The following are some of the arguments that can demonstrate the statement:

- A 5 MPa pressure pure O₂ atmosphere is used for combustion.
- The PC is artificially ignited (electrically) in ambient temperature as point of departure; no preheating of air and coal takes place not allowing natural ignition and sequential combustion with time.
- The test sample is not representative of the actual process. It consists of ± 5 grams of PC < 75 micron μm . In practice, the PC fineness ranges from 300 - < 75 micron.
- The bomb calorimeter is calibrated with a benzoic acid tablet, which has a CV of 29 MJ/kg, which is higher than the range of most coal CV's (14 – 26 MJ/kg for RSA coal).
- Etc.

(Storm, 1998:A56-58)

Secondly, international standards for converting GCVv to NCVp do exist (BS and ASME), but the results differ to an unacceptable degree in respect of certain reference temperatures, etc. However, the GCVv determination is not accurate enough for the pulverised fuel (PF) burner process, since among others (Storm, 1998:A56-58):

- The bomb calorimeter process takes place in a closed system and thus constant volume.
- The burner in a power station represents a constant pressure process.
- The latent energy of moisture vaporisation is not recovered (the flue gas exits the furnace at > 120 °C).

Since the difference between the GCVv and NCVp values for a grade of coal is normally 0.5 – 2.5 MJ/kg, the accuracy is inadequate for efficiency calculations and optimisation of the process (Storm, 1998:A56-58). In addition, since the Dulong method is theoretical, some losses are not accounted for. Combustion takes place at constant pressure and therefore the NCVp is the correct one (Kitto & Stultz, 2005:10.9). Thus, the CV applicable to a burner on a power station is the NCVp where it is determined by a flow calorimeter, which is essentially a mini PC burner. Research indicates that the smallest PC burners that were developed are in the range of 100 kW, 500 kW, and 1 MW in thermal rating (Orfanoudakis *et al.*, 2005:2; Vega *et al.*, 2013:1).

Thus, with an NCVp analyser in mind, the first challenge is to devise a micro PF burner, as small as possible, that can function independently of the flame support of adjacent burners and the hot environment of a large furnace. It has to have its own small single combustion chamber and sustain combustion, eventually without the flame support of an ignition medium such as fuel oil or gas. Evidence exists of various PF burners with a magnitude of 100 – 500 kW thermal, used for the drying of tobacco, etc., but they operate in an open environment, without a closed combustion chamber where the necessary parameters can be controlled.

Research indicate that no evidence of a micro coal burner, which is required as described above, that can be successfully constructed, without the flame support of another more volatile fuel (Orfanoudakis *et al.*, 2005:3). There are various obstacles in building such a micro coal burner. This is because dimensional analysis does not allow most parameters and dimensions to be scaled down linearly from the full-scale burners. This includes many of the flow, chemical and physical processes concerned (Orfanoudakis *et al.*, 2005:3). The most significant obstacle, however, remains maintaining a stable flame without gaseous fuel support. Some characteristics of combustion that need to be noticed include the correct amount of air needed for complete combustion, velocity for flame propagation, fluid dynamics for better mixing of air and PF (turbulence and vortices), metal temperature excursions, the feeding of coal into the burner, and burner dimensions.

1.2 PROBLEM STATEMENT

Thus, in summary, emanating from the above background the problem is:

- To devise a functioning PC micro-burner that discharges into a sized single combustion chamber and complies with the Fossil Fuel Firing Regulations (FFFR) (Eskom, 2012:35).
- This burner must function as an independent system without the flame support of burners positioned adjacent to or opposite it and without the positive influence of a large common furnace.
- The burner must be able to sustain stable combustion for such a period of time that all applicable parameters can be measured representatively.

1.3 OBJECTIVES

This study will endeavour to achieve the following:

- This PC burner should be as small as possible in physical size and thermal rating kW_T .
- The combustion process of this burner should function in a constant pressure process mode.
- It is endeavoured to ignite the PC with the gas (in such a mode that will make it possible to enable sustained combustion without gas support for future development).
- Measurement of all applicable parameters must be possible to enable mass-energy balance calculations.
- The design and functioning of this micro-burner must be such that it would enable future testing of a set range of coal qualities.

1.4 ANTICIPATED RESEARCH METHODOLOGY

- Perform a literature survey to uncover any existing technology on this application.
- From the minimum PF flow as independent variable, determine the sequential airflows as dependent variables for a required combustion criterion.
- These airflows (core air, primary air and secondary air) and the resultant combustion also have to comply with safety regulations, namely the FFFR (Eskom, 2012:35) with an appropriate A:F.
- Based on these airflows, the burner tubes and combustion chamber dimensions are to be designed to accommodate fuel flame velocity and accompanying swirl angles for adequate vortex generation.
- Simultaneously, this scale model should operate on a similar philosophy to mimic combustion on a full-scale power station furnace.
- The size of the burner's thermal rating should not be influenced by varying coal quality.
- The burner design must be such as to accommodate a range of coal qualities (used in South Africa).
- The resulting different A:F's should be evaluated and accommodated in the design.
- The design should incorporate a suitable ignition fuel (gas/liquid).
- The design should take into account the available manufacturing facilities, materials and cost.
- Testing should be carried out to evaluate the degree of the achieved flame sustainability.
- Future recommendations will be made for modifications and the required improvement of this experimental model to a possible prototype.

1.5 LIMITS AND SCOPE

This will be a joint project where multiple studies will be conducted in parallel for completion and functioning of an eventual flow calorimeter. Therefore, certain aspects will not be included in this study, since they have already been completed or are being performed in another study. These are aspects such as:

- Design and manufacturing of the PC feeder.
- Manufacturing of the combustion chamber.
- Cooling of the combustion chamber.
- Cooling of flue gas.
- Regenerative air pre-heaters.
- Gas analysis.
- Extraction fan, bag filters, and ash hopper.

This study will be limited to the following:

- Initially, the design will be intended to accommodate South African coal qualities. South Africa uses coal that is of the lowest grade internationally, as well as medium to reasonably high-grade coal. It is only in certain areas of Europe and the United Kingdom where higher-grade coal is used.
- This implies coal qualities with a range of 14-26 MJ/kg heating value.
- Only PF coal will be tested, no raw coal.
- Computational fluid dynamics (CFD) will be used in a superficial manner as a guideline to eliminate unnecessary manufacturing for trials. Because of its magnitude and intensity, an in-depth study will be avoided.

1.6 DISSERTATION STRUCTURE

This dissertation consists of seven chapters which is outlined below:

1.6.1 CHAPTER 2: LITERATURE SURVEY AND EXISTING TECHNOLOGY

This chapter is a literature survey on the concepts used to develop the micro coal burner and research previously done on this topic.

1.6.2 CHAPTER 3: CONCEPT DESIGN OF THE RESEARCH FACILITY

The decisions made and concepts used to design the micro coal burner is discussed. It also includes combustion and burner dimension calculations.

1.6.3 CHAPTER 4: GAS BURNER EXPERIMENT

The initial design of both the gas and coal burners are discussed. It shows the results and explains how the gas combusts within the gas burner. Positives and negatives created by the tangential placed holes are addressed by CFD and solutions are provided.

1.6.4 CHAPTER 5: FINAL DESIGN

This chapter shows the process taken in designing the micro coal burner by using CFD, combustion and burner dimension calculations.

1.6.5 CHAPTER 6: RESULTS

The manufacturing process, combustion results and explanation of occurring problems of the micro coal burner are shown and addressed.

1.6.6 CHAPTER 7: CONCLUSION AND RECOMMENDATIONS FOR THE FUTURE

Discussion and possible solutions of problems occurring in the micro coal burner is explained in this chapter. It also concludes the dissertation and suggests future work to be done on this topic.

1.6.7 REFFERENCES AND APENDICES

After chapter 7 the references and appendices follow.

2. LITERATURE SURVEY AND EXISTING TECHNOLOGY

2.1 BACKGROUND OF COAL

The need for energy, electricity and fossil fuels continues to grow worldwide because of their use for power generation. In 2000 the countries exhausting most energy were: the United States of America (USA), China, and the former Soviet Union. These countries used 41% and delivered 38% of the total global energy. The energy consumption in developing states is projected to double from 1999 to 2020 (Kitto & Stultz, 2005:9.1)

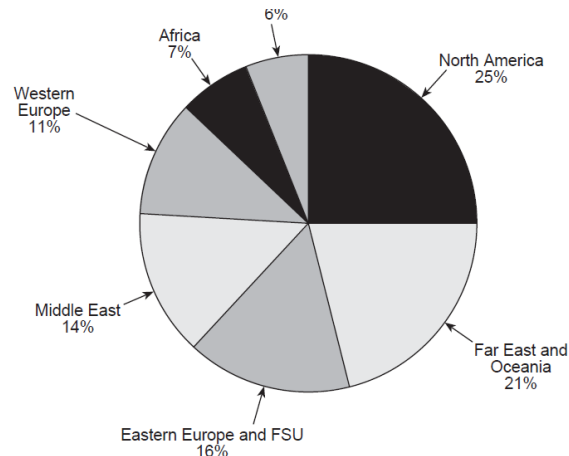


FIGURE 2.1: WORLD PRIMARY ENERGY PRODUCTION BY REGION
(Kitto & Stultz, 2005:9.1)

Coal is one of the most important sources of energy (Kurose *et al.*, 2009:144; Zhou & Cen, 2007:718). Considering only the USA, in 1976 coal production was 26%, oil 29%, and gas 32%, for use in energy generation. The production of coal has since increased to 32%, whereas those of oil and gas have decreased to 17% and 28% respectively (Kitto & Stultz, 2005:9.2). The figure below indicates the growing need for energy projected from 1970 to 2025:

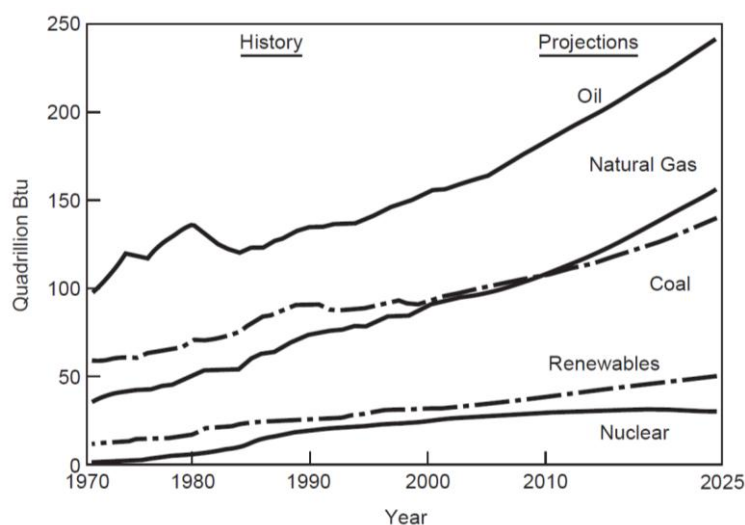


FIGURE 2.2: PROJECTION OF THE GROWING NEED FOR ENERGY
(Kitto & Stultz, 2005:9.1)

In the world and more specifically South Africa, most electrical power is generated from coal (Kitto & Stultz, 2005:14.2; Li *et al.*, 2008:1370; van der Merwe, 2014:1). This is due to the abundance of coal in the world (Kurose *et al.*, 2009:144), especially lignite and sub-bituminous coal types. Coal is relatively cheap compared to other fossil fuels (gas and oil) and is very flexible in different applications (da Silva & Krautz, 2014:430; Kim *et al.*, 2014:212; Kitto & Stultz, 2005:9.1). Coal is the second most used fuel in the world (after oil), producing 23% of the world's energy (Kitto & Stultz, 2005:9.1) and there are still coal reserves for another 200 years (Prassas, 1998:32).

2.1.1 COAL QUALITY AND CHARACTERISTICS

Since the composition of coal comprises different substances, it is a heterogeneous substance and coal is classified by rank. The rank specifies the characteristics and history of coal (Kitto & Stultz, 2005:9.5). The rank of coal is determined by a system that uses fixed C, volatile matter and heating value as ranking standards. The fixed C and volatile matter are parameters primarily used for coal ranking (Kitto & Stultz, 2005:9.5).

One of these analyses is proximate analysis. This is determined according to a procedure in a laboratory, which determines the moisture, volatile matter, fixed C and ash remaining after complete combustion (Kitto & Stultz, 2005:9.5; Rajoo, 2010:18).

The other analysis used is an ultimate analysis. Ultimate analysis or elemental analysis provides values for the content of N, O, C, ash and surface and inherent moisture. This analysis is obtained with an ultimate analyser, which determines these gravimetric amounts (Kitto & Stultz, 2005:9.8; Rajoo, 2010:18).

Both these procedures can be on an as-received, air-dried, or dry basis. The as-received coal for analysis still contains the total moisture (surface and inherent) (Kitto & Stultz, 2005:9.5; Rajoo, 2010:19) and is usually what power stations receive and burn; air-dried analysis excludes the surface moisture but still contains the inherent moisture and a dry-basis analysis has no moisture content (Kitto & Stultz, 2005:9.7; Rajoo, 2010:19).

As mentioned previously, coal contains inherent moisture and surface moisture. Inherent moisture is naturally deposited inside the coal. Surface moisture is located on the outside of the coal particle and is obtained externally (Kitto & Stultz, 2005:9.7). The surface moisture can be removed by air-drying the coal, but the inherent moisture is more difficult to remove. It can only be done after the coal has been ground to a fineness of <300 μm and then heated to vaporise the remaining moisture (BS ISO 17246:2010).

There has always been a dispute about the CV of coal. It has a huge effect on thermal efficiency owing to its obtainable analysis accuracy of ± 0.25 MJ/kg (Storm, 1998:A-56). The heating value indicates the maximum amount of energy available in a substance and is measured in MJ/kg (Kitto & Stultz, 2005:9.8). Two different methods of determining the heating value need to be taken into account. One is the GCVv and the other is the NCVp (Kitto & Stultz, 2005:9.8; Obert, 1973:91). The GCVv is determined by using a bomb calorimeter where the latent energy of the moisture is recovered over time (Obert, 1973:91; Rajoo, 2010:18; Storm, 1998:A-58). Water vapour is formed during the combustion of the H_2 in the coal (Kitto & Stultz, 2005:10.10).

The bomb calorimeter is an artificially ignited closed system process, therefore at a constant volume, where coal is ignited and the energy is measured (Kitto & Stultz, 2005:10.7; Obert, 1973:91; Storm, 1998:A-57). The bomb calorimeter is used most often in industry for the analysis of solid fuels such as coal, because of the quick results and low cost (Kitto & Stultz, 2005:10.7).

The NCVp can only be measured by a flow calorimeter, which functions at constant pressure (Kitto & Stultz, 2005:10.9). It is basically a water-cooled furnace (Kitto & Stultz, 2005:10.9). The fuel and air combust inside the combustion chamber where the products are cooled and the water temperature is measured (Obert, 1973:91). The heat is therefore measured at constant pressure and not at constant volume, as with a bomb calorimeter (Kitto & Stultz, 2005:10.9). The vapour of the moisture does not condense in the furnace to allow the latent energy to be included (Kitto & Stultz, 2005:10.10; Obert, 1973:92; Storm, 1998:A-58). The bomb calorimeter, however, poses a problem since the GCVv varies for different bomb calorimeters (Storm, 1988; A-57).

In the bomb calorimeter, the ignition and combustion take place differently from that in a burner since it is ignited artificially by an electric spark, whereas ignition in a burner results from sustained exothermic heat transfer. Combustion also takes place under different atmospheric conditions from those in a burner. In the bomb calorimeter the combustion occurs under 3 – 5 MPa pressure in pure O₂, as opposed to air under ambient pressure in a burner. Furthermore, the bomb calorimeter is calibrated with a benzoic tablet that has a CV value of almost double that of the coal in South Africa, making the measurement less accurate (Storm, 1998:A-58). Since fuels are mostly burned in a constant pressure environment and no latent heat is recovered, the NCVp value is the correct value to use for burner, furnace and boiler efficiency calculations (Kitto & Stultz, 2005:10.9; Storm, 1998:A-58). The flow calorimeter at constant pressure is mostly used to determine the heating value for gaseous and liquid fuels (Obert, 1973:91). Although the GCVv of solid fuels can be determined with a bomb calorimeter, the NCVp for solid fuels is not as easily determined (Kitto & Stultz, 2005:10.10). The Dulong formula is often used to check calorimeters and is reasonably accurate (Kitto & Stultz, 2005:10.10).

Several types of coal can be used for heat generation, namely lignite, sub-bituminous, bituminous, peat, and anthracite. Peat is the first product when coal is formed. It has a low heating value and very high moisture content. Lignite is the lowest ranking coal because it has a low heating value and high moisture content. However, it also has a high volatile content, which allows the coal to combust adequately. Sub-bituminous coal has a fairly high moisture content, is high in volatile content, low in ash content and has of a relatively high heating value. Bituminous coal is used most often in coal-fired power stations to generate electricity. It possesses a higher heating value due to its high C content, has lower moisture, but lacks in volatile matter, as opposed to the sub-bituminous and lignite coals. The top-ranked coal is anthracite, because of its very high C content. However, it has a very low volatile content, causing it to combust slowly (Kitto & Stultz, 2005:9.6).

2.1.2 COMBUSTION OF GASEOUS AND SOLID FUELS

The flammability of PC has been researched for more than half a century. It includes research of coal in terms of burning velocity, energy release and propagation limit concentrations for different types of coal (Slezak *et al.*, 1985:251). The definition of combustion is the chemical reaction of O₂ with the combustible elements of a fuel. The main combustible elements in most fossil fuels are sulphur, C and H. Good combustion entails all of the heat energy of the fuel being released with minimum loss in combustion deficiencies and excess air (Kitto & Stultz, 2005:10.1). To ensure a good combination, the three Ts need to be considered, namely: temperature - high enough for ignition purposes; turbulence - to ensure proper mixing of O₂ and fuel; time - to complete the process (Kitto & Stultz, 2005:10.1).

Two types of combustion of solid fuel occurs. The first is **homogeneous** or gas-phased combustion, which includes the chemical reaction of different gas elements. During this chemical reaction, heat is released, which forces elements to combine to form products (Kitto & Stultz, 2005:6.7). The second is **heterogeneous** combustion, which is a reaction that undergoes two phases and is a very broad field (Yang, 1993:97).

Coal undergoes this type of reaction, undertakes physical processes such as devolatilisation, coal drying, and chars oxidation. While entering the combustion zone, the coal particle is heated and moisture is vaporised (Cloke *et al.*, 1997:1266; Kitto & Stultz, 2005:6.8). After this, devolatilisation occurs, which ignites the gaseous fuel released from the coal and produces a charred particle. The charred particle consists of ash and char residue. After this, the char consisting of C is also burned and only the ash is left (Kim *et al.*, 2014:213; Kitto & Stultz, 2005:6.8). **Bituminous coal's** combustion type changes from heterogeneous to homogeneous, depending on the coal particle size (Kim *et al.*, 2014:213). The above can be seen in Figure 2.3.

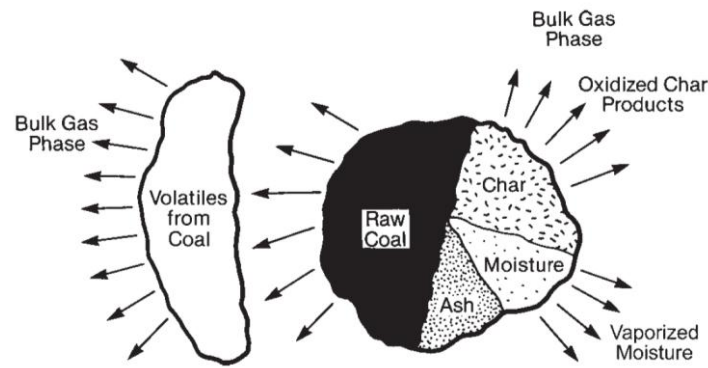


FIGURE 2.3: DIAGRAMMATIC ILLUSTRATION OF COAL COMBUSTION
(Kitto & Stultz, 2005:14.3)

The **fundamental physical laws** on which combustion calculations are based are the following:

- Conservation of matter - Matter cannot be created or destroyed. There must be a balance of matter entering and exiting the combustion process.
- Conservation of energy - Energy cannot be created or destroyed and the entering energy must be equal to the exiting energy.
- Ideal gas law - An ideal gas's volume is directly proportional to its temperature and indirectly proportional to its pressure.
- Law of combining weights - All elements combine in weight relationships.
- Avogadro's law - Equal volumes for different gases at the same temperature and pressure have the same number of molecules.
- Dalton's law - The total pressure of a mixture for different gases is equal to the total of partial pressures of each individual substance in the volume of the mixture.
- Amagat's law: The total volume of a mixture of different gases is equal to the total of each individual's volume in the same pressure and temperature as the mixture (Kitto & Stultz, 2005:10.3).

Air consists of 20.946% O, 78.102% N, 0.916% argon, and 0.033% CO₂ on a volumetric basis. Gravimetrically the composition is as follows - O, 23.14% and N, 76.86% with the rest being negligible (Kitto & Stultz, 2005:10.4; Worgas, 2011:8). **Theoretical air** (stoichiometric air) is the minimum amount of air needed for complete combustion of fuel (Kitto & Stultz, 2005:14.7; Worgas, 2011:5). This means that oxidation occurs of C to form CO₂, H₂ to form H₂O (water) and sulphur to form SO₂. The amount of other products that form is very small and therefore insignificant (Kitto & Stultz, 2005:10.6).

Sub-stoichiometric combustion occurs when too little air is provided. This is undesired with coal, since it leads to incomplete combustion or flameout (Kitto & Stultz, 2005:10.16). In practice, however, **excess air** is supplied to ensure proper combustion to ensure adequate mixing (Kitto & Stultz, 2005:10.15; Worgas, 2011:6). The quantity of excess air should be minimised because it is at a lower temperature than the combustion, therefore cooling the flame down. The amount of excess air needed varies for different fuels, it is 20% for natural gas (Worgas, 2011:9) and 13-20% for PC (Kitto & Stultz, 2005:10.15). In large power station furnaces, the amount of excess air is such that the remnant O₂ in flue gas after combustion ranges between 1% and 2% at full load at the exit of the boiler pressure parts (Kitto & Stultz, 2005:10.15).

A stoichiometric A:F is also depicted by the symbol $\lambda = 1$. A value of $\lambda < 1$ indicates a sub-stoichiometric A:F and a value of $\lambda > 1$ indicates that excess air is supplied to the combustion. The flame profile will also subsequently change, as can be seen in Figure 2.4 below (Worgas, 2011:17)

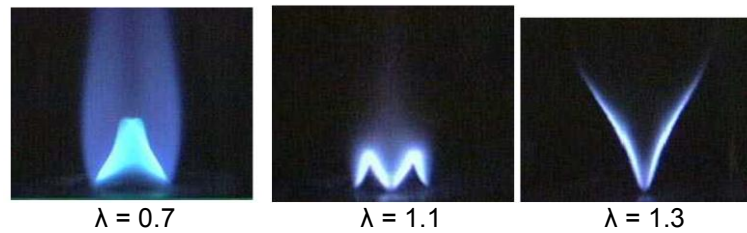


FIGURE 2.4: GAS FLAME PROFILES WITH DIFFERENT AMOUNTS OF EXCESS AIR
(Worgas, 2011:17)

Since coal is composed of different substances, the ignition temperature will vary greatly. The starting point in designing the burner etc. is with the **combustion calculations** for the specific coal quality. These calculations establish the amount of combustion air for the coal quantity and quality (Kitto & Stultz, 2005:10.16).

A certain fuel needs a certain **temperature to ignite**. These ignition temperatures will differ for fuels because there are other variables such as pressure, velocity, catalytic materials, ignition source and air-fuel mixture uniformity that influence it. The effect of higher pressure will lower the ignition temperature, whereas moisture in the combustion air will cause the ignition temperature to rise. The ignition temperature of coal is usually assumed to be the ignition temperature of the fixed C content because the volatile content is not ignited before this temperature is reached (Kitto & Stultz, 2005:10.10). The ignition temperatures of different fuels and the elements contained in coal are listed below:

TABLE 2.1: VARIOUS IGNITION TEMPERATURES OF FUELS AND ELEMENTS IN COAL

(Kitto & Stultz, 2005:10.11)

Ignition Temperatures of Fuels in Air
(Approximate Values or Ranges
at Atmospheric Pressure)

Combustible	Formula	Temperature, F
Sulfur	S	470
Charcoal	C	650
Fixed carbon (bituminous coal)	C	765
Fixed carbon (semi-anthracite)	C	870
Fixed carbon (anthracite)	C	840 to 1115
Acetylene	C_2H_2	580 to 825
Ethane	C_2H_6	880 to 1165
Ethylene	C_2H_4	900 to 1020
Hydrogen	H_2	1065 to 1095
Methane	CH_4	1170 to 1380
Carbon monoxide	CO	1130 to 1215
Kerosene	—	490 to 560
Gasoline	—	500 to 800

Other studies show that coal in general will ignite at 700°C (Li *et al.*, 2009:1222).

2.2 PARAMETERS AFFECTING COMBUSTION:

The physical form of coal burning used most often is PC in an open side furnace. The **size of coal particles'** surface area controls the combustion rate of coal; smaller particles promote increased

burnout (Kitto & Stultz, 2005:14.2; Storm, 1998:4-4; Xiumin *et al.*, 2000:3). The PF fineness distribution of coal is thus an important determining factor and can be plotted on a Rosin-Rammler graph (Figure 2.5). The PF, which is plotted on the Rosin-Rammler graph, is sampled iso-kinetically from the PF transport pipes. The coal is then dried and inserted into a shaker with different sieve sizes which range from 300 μm to 75 μm . The x-axis represents the size of a particle and the y-axis represents the percentage of the sample passing through each sieve gravimetrically (Storm, 1998:Appendix 51).

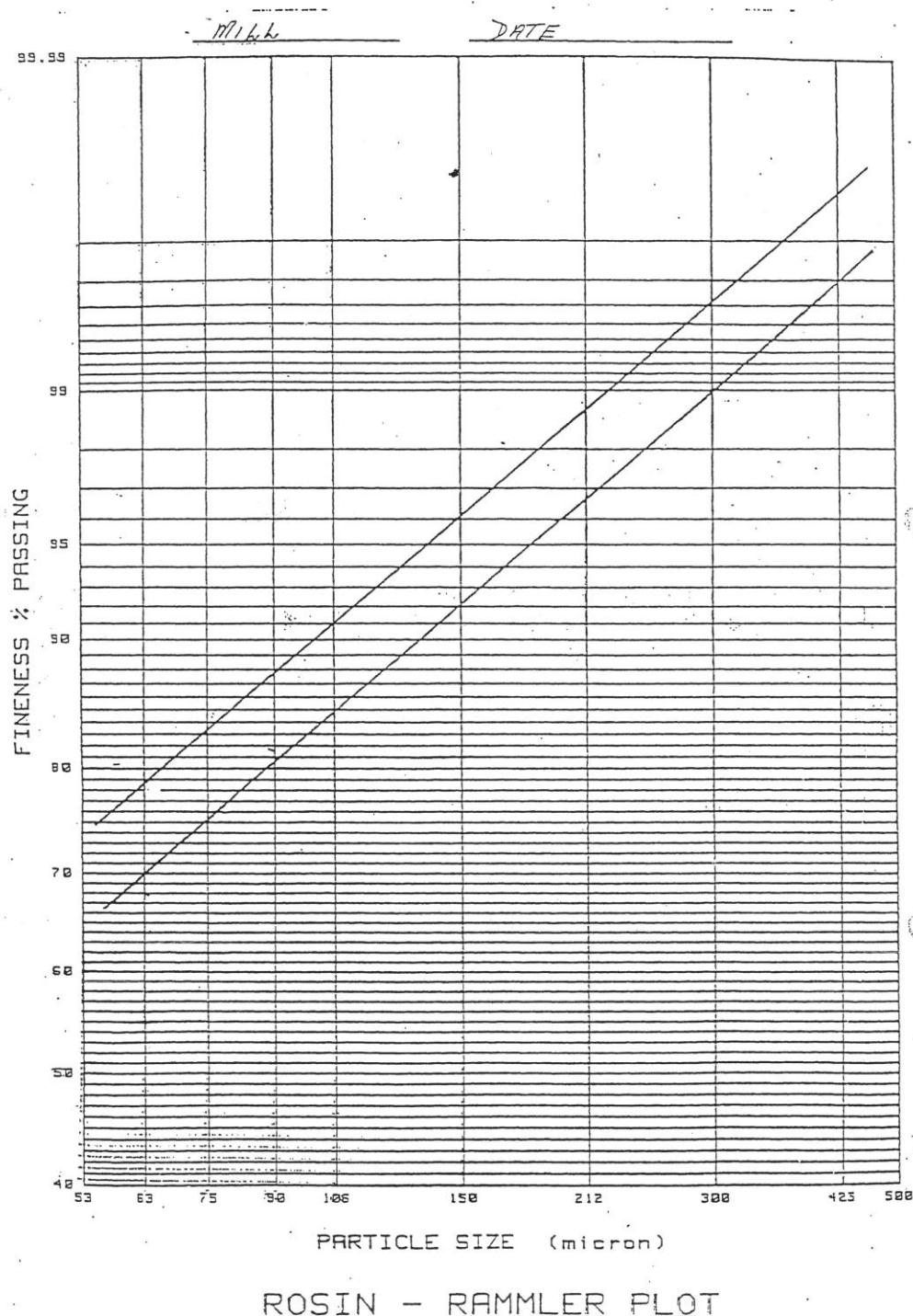


FIGURE 2.5: STANDARD TEMPLATE OF A ROSIN-RAMMLER PLOT FOR PF FINENESS CRITERIA
(Storm, 1998:Appendix 51)

As a general guideline, the Rosin-Rammler criterion for the combustion of bituminous coal requires that a fineness distribution of >70% passing through a 75 μm sieve, and <1% remaining on top of a 300 μm sieve and the intermediate sieve sizes' percentages lying on the straight line interconnecting those. That concerns iso-kinetic sampled PF, also to evaluate mill effectiveness.

Complete combustion of PF particles can occur within one to two seconds if the particle has a nominal size of <150 μm (Kitto & Stultz, 2005:14.2). A **char particle's** combustion rate depends on the particle size, composition and porosity, temperature zone and O_2 partial pressure (Cloke *et al.*, 1997:1267; Kitto & Stultz, 2005:14.3). Volatile matter of coal is a crucial parameter for flame stability, ignition, and char burnout (Howard *et al.*, 1966:20; Kitto & Stultz, 2005:14.3; Messerle *et al.*, 2014:294). The devolatilisation process depends on the residence time in the combustion zone, the composition of the coal and the temperature to which the coal is exposed (Kitto & Stultz, 2005:14.3).

Moisture also poses a problem to ignition. Although most of the surface moisture is evaporated, it is still carried along with the PC to the burner. This and the inherent moisture left inside the coal absorbs heat to vaporise and be super-heated. Furthermore, when the temperature is high enough the water molecules dissociate and absorb more heat (Kitto & Stultz, 2005:14.3).

Ash also interferes with combustion because of interference with respect to heat transfer and absorption. Ash also causes slag, which can foul the furnace (Kitto & Stultz, 2005:14.4). Heat transfer via elements in contact with the flame front also cools down the flame (Worgas, 2011:19).

There are analyses of coal designed to forecast certain aspects of combustion behaviour. As mentioned before, **proximate analysis**, **ultimate analysis**, and **heating value** are some of these analyses, but further tests are available, such as **drop tube furnace** tests, **thermography**, free swelling index and **petrographic** examination (Kitto & Stultz, 2005:14.4; Rajoo, 2010:9). These extended tests provide information on char reactivity, ignition, evaporation of volatiles and burnout behaviour (Kitto & Stultz, 2005:14.4).

Burnout time of a coal particle is one of the most important parameters in terms of combustion response. Once again, burnout time is decreased with a decrease in particle size and is increased with low volatile matter (Makino & Law, 2009:2068). In an environment of 1300 K the burnout times of particles with a size between 40 μm and 160 μm is between 0.1 and 2.3 seconds (Makino & Law, 2009:2072). It takes about 0.02 seconds for a temperature of 1100°C to be reached in the core of a 100 μm particle and 0.001 seconds in a 30 μm particle (Howard *et al.*, 1966:C37). The drop tube furnace (DTF) is a laboratory-scale furnace, which mimics the conditions of an oxidising environment, thermal environment, and residence time inside a PC furnace. The ignition and combustion performance is evaluated by small representative samples of coal passed through the furnace (Rajoo, 2010:20).

The maximum theoretical temperature that a fuel's products are able to reach after combustion is called the **adiabatic flame temperature**. This is determined by using the fuel's heat release and by applying a stoichiometric air-fuel ratio (because excess air will absorb heat). By using the adiabatic enthalpy of the flue gas, the adiabatic flame temperature can be determined. This temperature can never exist in practice because the combustion is not instantaneous and because some heat is absorbed (Kitto & Stultz, 2005:10.11). The adiabatic flame temperature of coal is 1975 K (Draper *et al.*, 2013:2780; Slezak *et al.*, 1985:259). The more realistic **temperatures inside a furnace** can reach up to 1650°C (Kitto & Stultz, 2005:14.5; Messerle *et al.*, 2014:299) although some sources indicate that furnace temperatures are between 650°C and 1400°C (Chen *et al.*, 2011:719; Costa *et al.*, 1997:286; Li *et al.*, 2008:1379).

Swirl is a very important parameter and affects various aspects such as:

- The mixing of air and fuel, flame length and drawing the flame to the centreline of the combustion chamber to prevent damage to the walls by means of a resulting vortex (Khanafer & Aithal, 2011:5030; Li *et al.*, 2007:125).
- Providing flame stability as well as increasing residence time and enhancing ignition speed (Chen *et al.*, 2011:722; da Silva & Krautz, 2014:433; Nettleton, 2004:256).
- Swirl also influences the mixing ability of air and fuel and therefore the resulting combustion temperature (Khanafer & Aithal, 2011:5036; Xue *et al.*, 2009:3564).

- The greater the tangential velocity and/or the swirl angle, the higher the swirl and the larger the recirculation zone (Gu *et al.*, 2005:2096; Khanafer & Aithal, 2011:5036; Storm, 1998:4-24).
- A longer residence time inside the furnace improves combustion (Kitto & Stultz, 2005:14.5).
- **Longer residence time** reduces unburned fuel. Residence times normally vary from 1 to 3 seconds (Kitto & Stultz, 2005:14.19; Rajoo, 2010:36), but depending on the swirl, others are between 0.01 and 0.04 seconds (Gu *et al.*, 2005:2099).
- If the swirl is too excessive, the centrifugal force can divert particles away from the recirculation zone (Chen *et al.*, 2011:722; Gu *et al.*, 2005:2097; Orfanoudakis *et al.*, 2005:2).
- With the correct swirl, particles larger than 100 μm can spend more time in the recirculation zone (Chen *et al.*, 2011:722; Orfanoudakis *et al.*, 2005:2).
- Excessive swirl can influence flame stability in a negative way (Nettleton, 2004:256).
- By increasing the amount of swirl, up to a limit, there is a decrease in unburned C and vice versa (Xue *et al.*, 2009:3561).
- The swirl intensity is represented by a swirl number (S). The recirculation zone starts forming with swirl numbers of at least 0.65. A 30% increased recirculation zone represents a 60% increase of S (Orfanoudakis *et al.*, 2005:2). The higher the swirl number, the greater the swirl, and the higher the intensity of the recirculation zone, as can be seen in Figure 2.6.

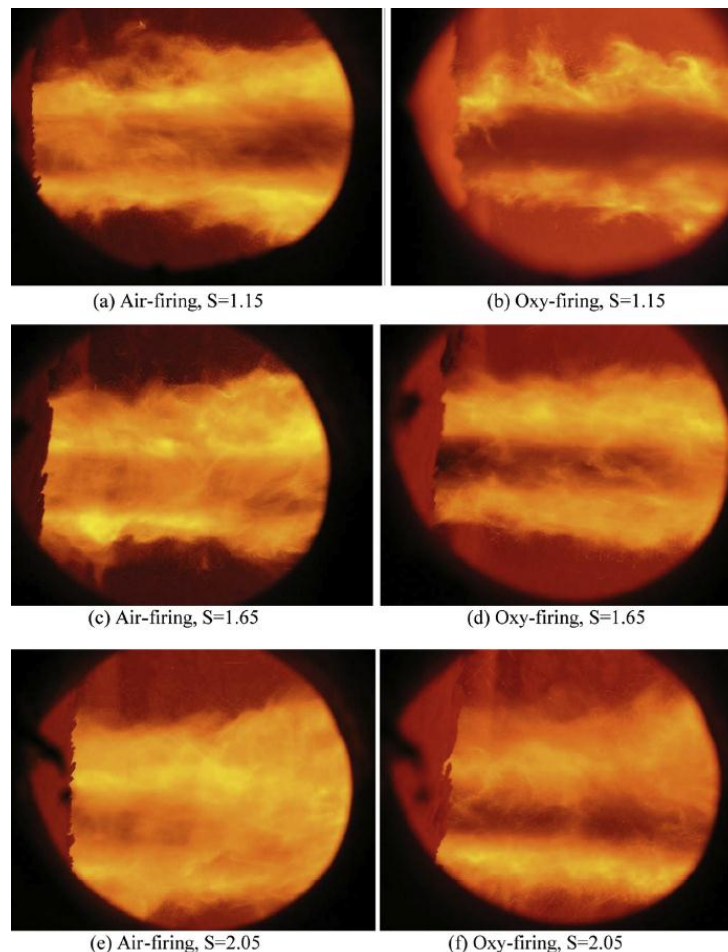


FIGURE 2.6: EFFECTS OF INCREASED SWIRL ON FLAME PROFILE
(da Silva & Krautz, 2014:434)

2.3 BURNER CHARACTERISTICS

A **burner** is a device mixing a fuel and the correct amount of air to promote favourable combustion, such that the resulting exothermic chemical reaction between the fuel and an oxidiser can release heat in a controlled manner (Worgas, 2011:4). To design a burner the A:F, modulation range, impact of the chemical reactions and factors affecting the temperature, such as furnace cooling etc., need to be known (Worgas, 2011:12).

The point of departure for good combustion is the burner design. Its function is to mix fuel and air to enable combustion (Khanafer & Aithal, 2011:5030). It regulates the supplied A:F distribution, dictates sequential furnace design and controls combustion (Kitto & Stultz, 2005:14.6). An effective burner is intended to render combustion as efficient as possible and to produce a small amount of nitrous oxide (NO_x) (Khanafer & Aithal, 2011:5030). One of the advantages of PC burners is their versatility, since they can handle a range of coal qualities. It is the method used most often to burn PC. The conventional coal-firing set-up places the burners in the lower section of the furnace. Another arrangement is to place the burners on opposing walls, or in the corners, named tangentially fired burners (Kitto & Stultz, 2005:14.4; Zhou *et al.*, 2009:5376).

The functioning of a burner is as follows: Primary air (PA) is used to transport the PC to the burner mouth and for coal classification. The PA is preheated to about 66°C to vaporise the moisture on the surface of the coal particle and to pre-heat the coal before combustion. After the PA has been mixed with the PC, the PC is then again mixed with another air quantity called secondary air (SA). This SA is supplied to ensure the right amount of air so that combustion can take place (Kitto & Stultz, 2005:14.2). SA is usually pre-heated to 316 °C (Kitto & Stultz, 2005:14.4; Li *et al.*, 2007:129) and is supplied by forced draft fans to wind boxes that enclose the burners. Sometimes the SA is staggered in stages for NO_x reduction. The PC/PA mixture and SA are mixed at the burner throat where ignition can commence. Combustion continues as the gases and unburned fuel moves towards the furnace centre (Kitto & Stultz, 2005:14.4). The burnout of the fuel depends on the coal's composition, size distribution, temperature exposure, air fuel mixing and the amount of excess air. Thereafter the products of combustion exit at the convection pass of the furnace (Kitto & Stultz, 2005:14.5).

The above-mentioned method of firing PC is for using coal with an average amount of volatile matter. When coal with low **volatile matter** content is fired, adjustments have to be made. As stated before, low volatile matter poses difficulties for ignition, burnout, and flame stability (Kitto & Stultz, 2005:14.5; Li *et al.*, 2007:123). Because of these problems, higher thermal furnaces are required, owing to the increase in flame length. Down-shot furnaces, where the burners are tilted downwards, can relieve these problems by producing a downward-directed flame and then the hot gases flow upward thereafter. As depicted below (Figure 2.7), the fuel that underwent combustion recirculates the hot gases to the burner, via the recirculating zone, which assists in ignition of the unburned fuel at the burner mouth (Kitto & Stultz, 2005:14.5).

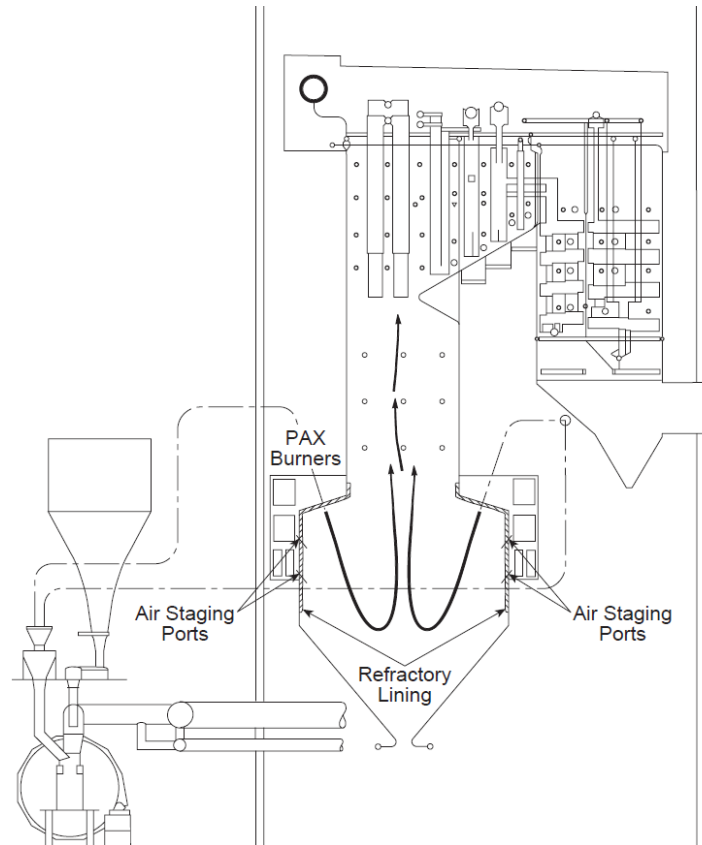


FIGURE 2.7: A FURNACE WITH DOWNWARD-TILTED BURNERS

(Kitto & Stultz, 2005:14.5)

When dealing with **high-moisture coal**, hot gases are taken from the upper furnace to dry and prepare the coal because pre-heated air is inadequate to do so. The temperature of these gases is in the region of 1000°C and is of more use for drying coal than pre-heating air. The hot gases are low in O_2 content, which decreases the danger of explosions inside the grinding mill (Kitto & Stultz, 2005:14.5).

Burners are divided into two groups: pre-mix and throat mix. Pre-mix burners combine the PC, PA and SA prior to the combustion chamber. Throat-mix burners, as mentioned before, combine the SA at the throat of the burner just before the furnace and are the most common type of burner. The SA is supplied in a certain way to force proper mixing, which will create a stable flame and improve ignition of the fuel. The mixing of the air and fuel affects the stability of the flame, products or emissions produced and the form of the flame (Kitto & Stultz, 2005:14.6).

The **A:F** of the PA can vary greatly depending on the type of coal and mill settings. The PA has a direct impact on combustion and emissions. When increasing the PA, the SA must automatically decrease because of an overall fixed A:F for good combustion and because essentially the total air supplied to the boiler is the sum of the PA and SA. The ratio of SA:PA can also differ, where a greater SA:PA contributes to better aerodynamic control over air fuel mixing and burner performance. Lower SA:PA can cause a less stable flame (Kitto & Stultz, 2005:14.7).

The velocity at which the PC exits the burner also has an impact on where the flame is situated. The higher the velocity, the further the ignition point is from the burner. Too high velocities can cause flame blowout and flame instability. The SA is also preheated by regenerative air heaters, which transfer the heat from hot flue gases to the air (Kitto & Stultz, 2005:14.7).

Flame velocity:

Burning velocity contains very important information for reactivity, diffusivity, and exothermicity. The data gathered is important for validating chemical kinetic mechanisms and modelling of turbulent combustion (Katre & Bhele, 2013:33).

Regarding flame velocity, three types occur. **Burning velocity** signifies the rate at which the flame front ignites the unburned fuel behind it (Pitsch, 2014:9), **flame speed** represents the velocity of the flame compared to a fixed point and the difference between flame speed and burning velocity is the expansion velocity (Huzayyin *et al.*, 2008:39). The burning velocity for propane is 0.455 m/s and for liquid petroleum gas (LPG) 0.432 m/s (Huzayyin *et al.*, 2008:54). Coal with a 50 μm fineness and 36% volatile matter content has a broad maximum burning velocity of 0.37 m/s at an equivalence ratio (ϕ) of 4 and a minimum computed burning velocity of 0.09 m/s at an equivalence ratio of 8 (Slezak *et al.*, 1985:251). The ϕ is defined as the actual ratio of fuel-air to ideal or stoichiometric fuel-air ratio (Pulkrabek, 2004:64)

In South Africa, one of the greatest problems experienced on power plants is **PF settling**. PF settling creates a pressure imbalance in the burner pipes that worsens PF distribution and consequently load-loss (van der Merwe, 2014:9). Although the FFFR imply that the minimum velocity to avoid PF settling is 18 m/s, power stations in South Africa operate in the region of 20 – 29 m/s to prevent settling (van der Merwe, 2014:54). Other sources, such as Kitto and Stultz (2005:14.6) and Nettleton (2004:253), indicate that the PF/PA stream must travel at a minimum of 15 m/s or 20 m/s to prevent particles from settling in the burner pipes. A severe case of PF settling is shown (Figure 2.8).



FIGURE 2.8: PF SETTLING IN A BURNER PIPE
(van der Merwe, 2014:10)

To create proper **mixing** between air and fuel, numerous factors must be taken into account. One is staging the SA supplied to the furnace and creating an internal recirculation zone. Mixing is crucial in combustion. In burners where the fuel and air streams are concentric, entrainment occurs owing to jet expansion. This type of burner is usually implemented in corner-fired furnaces. Mixing is at a maximum during full loads in power stations because of high air and fuel flow rates and velocities. Mixing taking place in the burner can be created by using the PF/PA, the SA or both of these supplies. The PF/PA supplied is often mixed by swirl generators, deflectors, or bluff bodies. The deflectors or impellers are implemented at the burner end, which scatters the PF/PA mixture, causing it to mix with the SA. Thus, a shorter flame length may be obtained during this type of mixing. Bluff bodies are placed end to end or inside the burner exit, which forces the stream to accelerate on the upstream of the body and recirculate on the downstream. The recirculation creates an increase in residence time and more mixing. The PF/PA stream can also be injected by multiple jets, which will increase the surface area of the fuel jets (Kitto & Stultz, 2005:14.8).

2.3.1 CIRCULAR THROAT BURNERS

In **circular throat burners**, the SA supply is mostly used to create swirl. At the upper end of the burner throat, **swirl** generators are implemented to create rotating motion of the SA. The SA now has a tangential, radial, and axial component when it exits the burner throat. Because of this, axial and radial pressure gradients are formed before the burner throat with the lowest pressure located in the centreline of the throat. The low pressure increases as the amount of swirl increases. The low pressure draws the flow inwards, forces it to move backwards, and allows for a recirculation pattern. This type of swirling promotes mixing and therefore flame stability. Burner throat velocity directly

influences the mixing capability and is important when designing a burner. However, when the burner velocity increases, there is a drop in pressure in the burner and the better the mixing, the higher the NO_x emissions (Kitto & Stultz, 2005:14.8). One of the earliest burners that used swirl to combust PC was the **circular burner**. This burner was very successful, was in use for more than six decades, and was capable of burning different types of coal. The burner consists of a nozzle through which the PC/PA moves to the burner exit. At the exit of the nozzle, an impeller is placed to scatter the PC/PA mixture into the SA. The SA is equipped with interlinked vanes that are organised in a circular pattern between two plates, Figure 2.9. During no-load conditions, the vanes are closed; during ignition, the vanes are partly open, and during normal operation, the vanes are fully open. These vanes affect the swirl element as well as the amount of SA to the furnace. The more the swirl is decreased, the more the vanes are opened, whereas the quantity of SA is increased and vice versa. A well-shaped flame is created by an adequate amount of swirl, while an extreme amount of swirl can produce a flat flame against the firing wall and create local slag formation. Insufficient swirl can affect the flame stability in a negatively. The circular burner can operate with natural gas, oil, and PC (Kitto & Stultz, 2005:14.9).

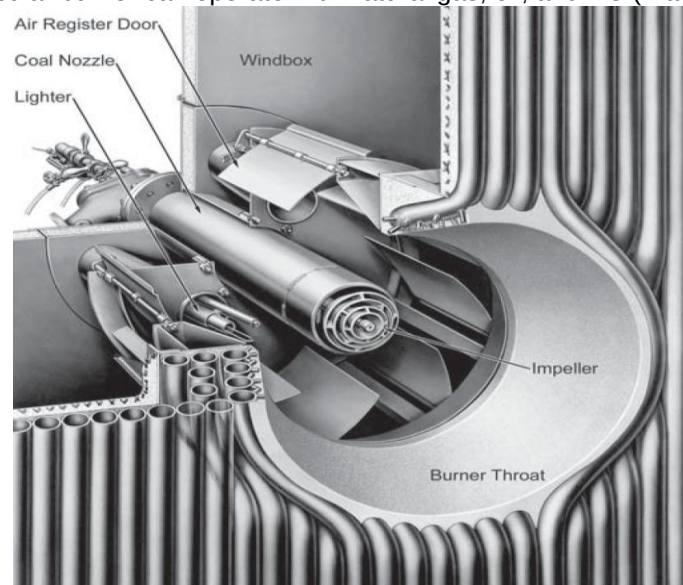


FIGURE 2.9: CIRCULAR BURNER WITH SWIRL VANES
(Kitto & Stultz, 2005:14.6)

2.3.2 CELL BURNERS

The **cell burner** uses two to three circular burners that are placed above one another, which act as a single unit (Figure 2.10). The cell burner is very efficient but creates very high NO_x levels and is also unreliable, (Kitto & Stultz, 2005:14.9).

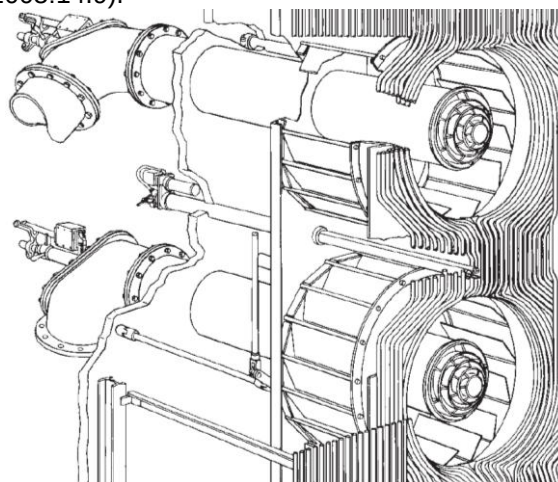


FIGURE 2.10: CELL BURNER
(Kitto & Stultz, 2005:14.9)

2.3.3 BABCOCK S-TYPE BURNER

The circular burner was replaced by the **S-type burner**. The main difference was that the SA control system was mechanically improved, where the quantity of flow and that of swirl are controlled separately. The amount of flow is controlled by a sliding air damper, which moves backwards or forwards in the burner barrel. The burner barrel consists of spin vanes controlling the amount of swirl. The problem of either more swirl or less flow and vice versa does not exist with the S-type burner, which yields better combustion efficiency as well as mechanical reliability (Kitto & Stultz, 2005:14.9).

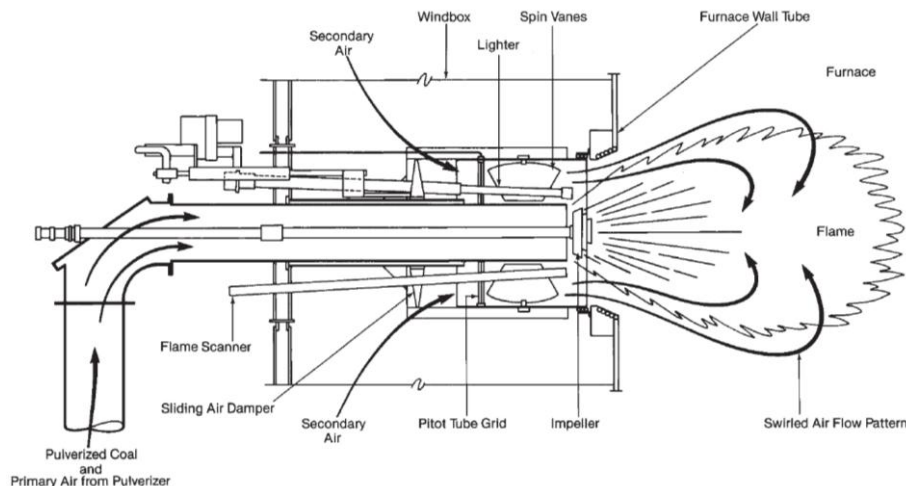


FIGURE 2.11: S-TYPE BURNER

(Kitto & Stultz, 2005:14.10)

2.3.4 LOW NO_x BURNERS

Almost 75% of the **NO_x** that is formed during combustion comes from the N₂ inside the fuel (fuel NO_x), while the remainder emanates from the N₂ inside the air supplied for combustion (thermal NO_x). Thus, the best way to limit NO_x formation is limiting fuel NO_x formation. During devolatilisation and char burnout, fuel NO_x is formed by the oxidation of the inherent N₂ in the fuel. Usually coal contains 0.5 to 2% of N₂. High amounts of O₂ and resulting temperatures promote the formation of NO_x during the devolatilisation process. Coal with high volatile and low fixed C content is most likely to form NO_x. Lowering the availability of O₂ during the devolatilisation process can be effective in reducing NO_x formation. The O₂ is then added later so that combustion efficiency remains as high as possible (Kitto & Stultz, 2005:14.10).

- **The first option** to limit the O₂ available during devolatilisation is the following: The burner can supply all the required air, but not all of it is mixed in the flame, causing only a fraction of the air to mix with the fuel during devolatilisation. The remaining air is then gradually mixed later in the furnace to ensure good combustion. This technique shows a 60% decrease in NO_x formation. The lower NO_x formation does, however, come at a cost, sacrificing good mixing of fuel and air so that the flame envelope is larger than fast-mixing burners. By sacrificing mixing rate, the amount of unburned fuel is increased. The unburned fuel can be dealt with by improved burner aerodynamic control, increased PC fineness, and improved preparation of fuel (Kitto & Stultz, 2005:14.10).
- **The second option is** to reduce the availability of O₂ to the PC during devolatilisation. The burner does not supply all of the air and adds it in stages somewhere else in the furnace. This technique is called air staging. In some cases, a reduction of 10% airflow in the burner is satisfactory for NO_x formation control. By adding a sub-stoichiometric airflow, a decrease in NO_x formation will occur, but this usually increases the amount of unburned fuel (Kitto & Stultz, 2005:14.10). The quantity of excess air that is left over is then added via staging ports and can be as much as 25% of the total air. The staged air must be controlled carefully because staged air in high quantities can create NO_x reformation and slagging in deep staging systems (Kitto & Stultz, 2005:14.11).

- **The third option** to decrease NO_x formation is re-burning of fuel or fuel staging. This method eliminates the NO_x that is formed. As with the staging of air, this method introduces fuel in stages. Most of the fuel is burned at an almost stoichiometric air-fuel ratio but then the rest of the fuel is introduced with only a partial amount of air to establish a reducing zone before combustion is complete. Hydrocarbon radicals are formed in the reduction zone which absorb the O₂ from the NO_x and only N₂ remains. After this, the rest of the air is added to ensure good combustion. These systems contain separate burners or jets for the fuel supply and NO_x nozzles for the remaining air supply. This method can be implemented for coal burners but special methods need to be applied to reduce the amount of unburned fuel. Natural gas is more suitable for this method, since it has the ability to form more hydrocarbon radicals. The fuel-staging systems for PC are not as cost-effective as other low-NO_x systems have proven to be. New fuel-staging technology has reduced NO_x formation by 70% by taking a small portion of fuel to create hydrocarbon radicals and introducing it to the combustion zone to absorb the NO_x formed from earlier combustion (Kitto & Stultz, 2005:14.11).
- **The dual register burner (DRB)** was one of the first low NO_x burners developed by the Babcock and Wilcox Company. It consisted of two air zones, each being controlled by a separate register, which was placed around a coal nozzle. The inner air zone was used to swirl a portion of the SA extremely high for flame stability and ignition purposes. The remaining SA was introduced steadily to the flame through the outer air zone using less extreme swirl. The coal nozzle was shaped in the form of a venturi to scatter the PC/PA mixture and therefore no deflection was required. The burner could maintain a stable flame with a fuel-rich centre, and air was steadily mixed further down the furnace, Figure 2.12. This burner decreased NO_x formation by 60% (Kitto & Stultz, 2005:14.11; Li *et al.*, 2008:1372).

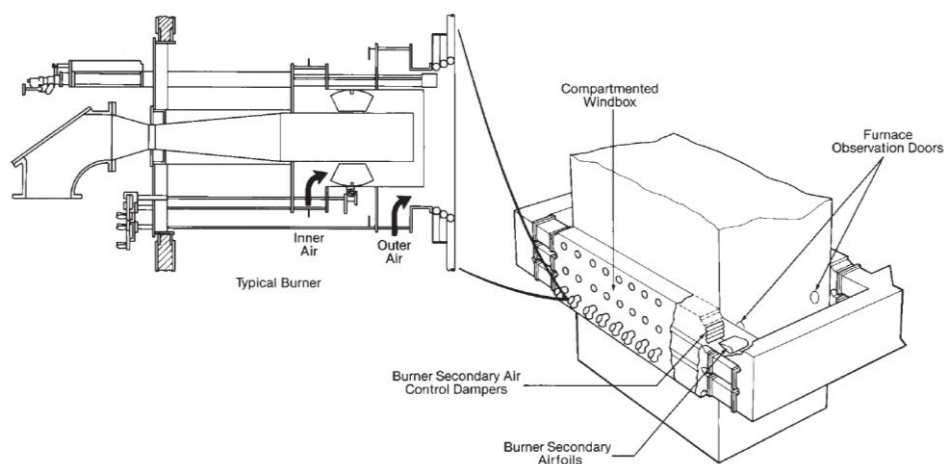


FIGURE 2.12: THE DUAL REGISTER BURNER

(Kitto & Stultz, 2005:14.11)

- The replacement of the DRB was the DRB XCL that incorporated air staging (implemented in the original DRB) and fuel staging. The **DRB XCL** is based on the S-type burner mentioned before. The quantity of air is controlled by a sliding air damper. A Pitot tube mechanism is placed in the burner barrel to determine and control the amount of SA flow by sliding the air damper. Low NO_x burner systems require well-balanced distribution of air and fuel. Further, down the Pitot tube mechanism the inner and outer air zones are situated. High swirl is created in the inner air zones for flame stabilisation and ignition purposes. The outer air zone consists of two stages of vanes, where the upper stage is fixed to improve exterior distribution of air inside the burner and the downstream stage has adjustable vanes to create swirl for appropriate mixing in the flame. The burner's PF/PA nozzle consists of an impeller for good distribution. This greatly improves flame stability while reducing flame length and lowering NO_x emissions. The DBR XCL decreases NO_x emissions by 70% and can fire coal, natural gas, and fuel oil, Figure 2.14 (Kitto & Stultz, 2005:14.12).

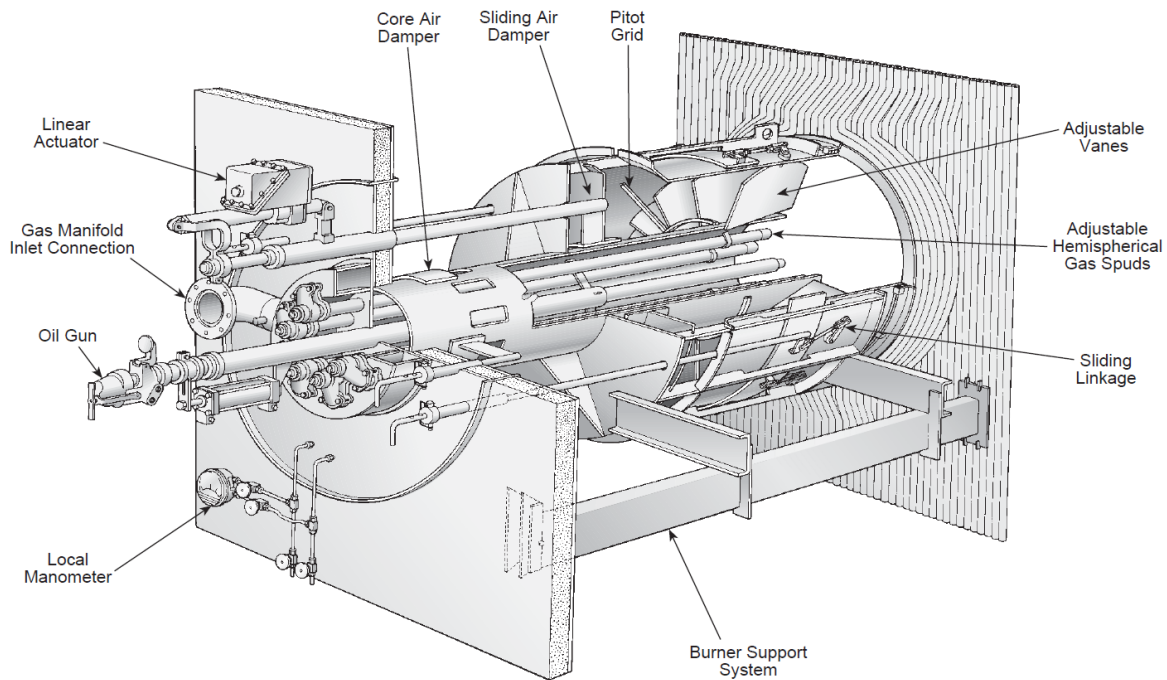
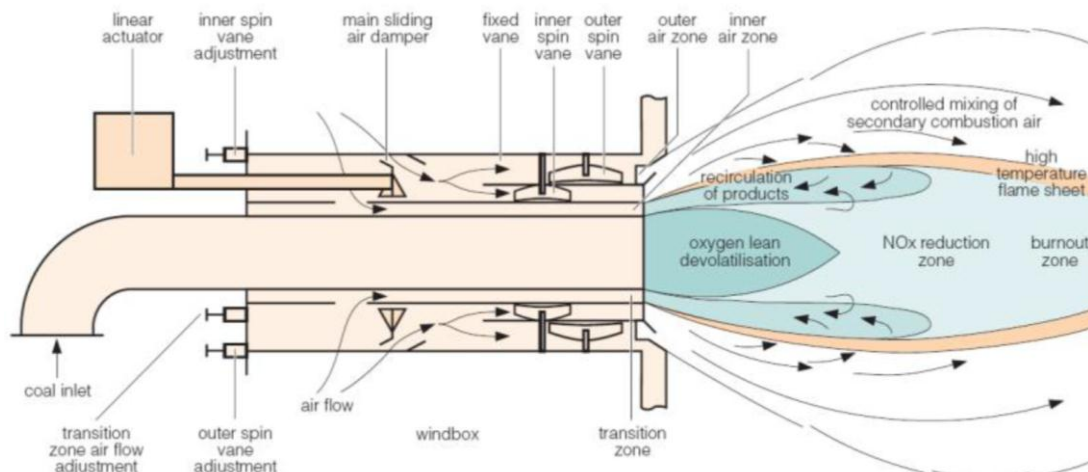


FIGURE 2.13: THE DUAL REGISTER BURNER XCL
(Kitto & Stultz, 2005:14.12)

- The follow-up model to the DBR XCL was the **DBR 4Z**. The DBR 4Z adds more emission control mechanisms in a mechanical configuration. This burner reduces NO_x and improves combustion by controlling the PA zone stoichiometry with fuel and air staging. It has been proved that adding another air zone to separate the coal nozzle from the principal air zones benefits the burner. The extra air zone is called the transition zone, which creates a barrier between the SA flow and the flame. The transition zone is designed to suck in hot combustion gases from the outer parts of the flame towards the centre of the flame. The NO_x formed in the outer parts of the flame in a high O₂ environment is reduced to other N₂ species by being drawn back to the core flame. The transition zone can also change the stoichiometry in the primary zone. Insufficient air permits sub-stoichiometry in the primary zone and can hinder combustion during the devolatilisation process. The transition zone can supply sufficient air to the primary zone. Once again, the SA introduced through the inner air zone creates extreme swirl for flame stability and the outer air zone permits air steadily to the flame for low NO_x formation. An 80% decrease in NO_x emissions is possible with the DBR 4Z burner, Figure 2.13 (Kitto & Stultz, 2005:14.13; Li *et al.*, 2008:1373).



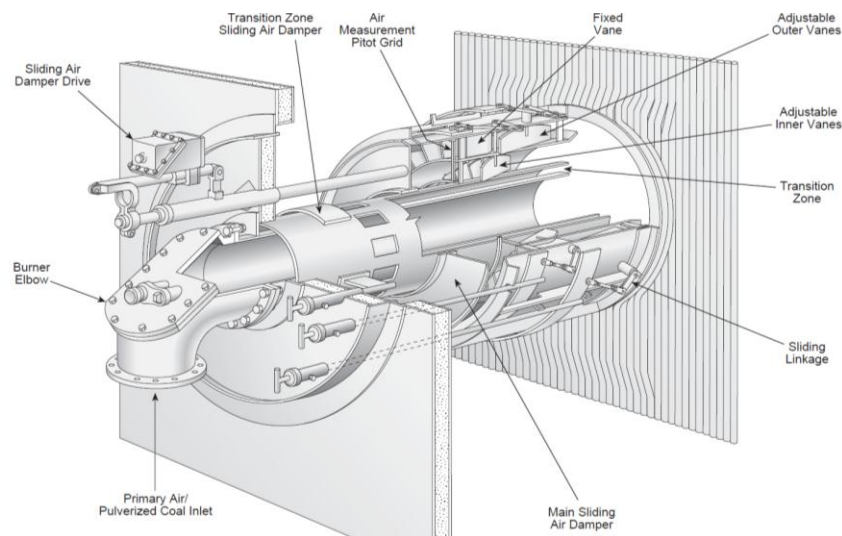


FIGURE 2.14: THE DUAL REGISTER BURNER 4Z
(Kitto & Stultz, 2005:14.13,14.14; van der Merwe, 2014:7)

- With **low NO_x cell burners**, all of the PC is supplied to one throat of the burner with some of the SA as well. The remaining SA is then supplied to the other throat, which is essentially a NO_x port. A decrease of 50% NO_x formation can be achieved while still sustaining high combustion efficiency (Kitto & Stultz, 2005:14.14).
- **Corner-fired burners** are used globally because they have low NO_x emissions, can fire various coals, and exhibit good flame stability. There are however, negatives such as slagging in the burner and corrosion on the water-cooled wall, as well as heat and temperature imbalances in the horizontal flue gas pass. The deviation of temperatures inside the horizontal flue gas pass is due to the intense vortex caused by the swirl that the tangential burners create (Zhou *et al.*, 2009:5375). In corner-fired burners, the over-firing of air have proven to be useful for NO_x control (Kitto & Stultz, 2005:14.16). In these burners, coal and air are fed to the furnace through wind boxes positioned in the corners of the furnace at different levels. In the furnace, the wind box nozzles create an abstract firing circle by channelling the wind box nozzles at a certain angle from the diagonal, Figure 2.15. The firing angles and momentum of the coal and air streams create a circular flow flame in the centre of the furnace (Kitto & Stultz, 2005:14.17; Zhou *et al.*, 2009:5376). In this system, the corner-fired wind boxes act as igniters and the furnace acts as a huge burner where the swirling motion stabilises the flame. The fireball can be raised or lowered, depending on the need. The lowered NO_x emissions are obtained by air staging technology (Kitto & Stultz, 2005:14.17).

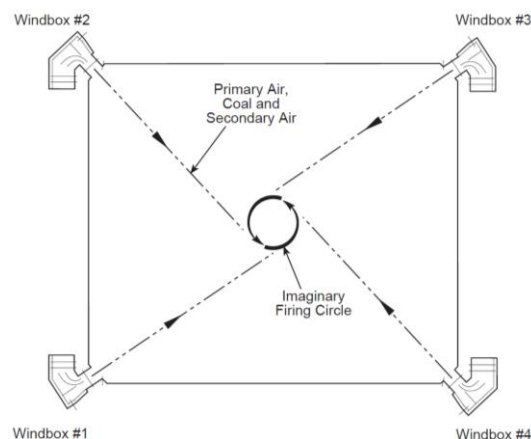


FIGURE 2.15: CORNER-FIRED BURNERS
(Kitto & Stultz, 2005:14.17)

2.3.5 OTHER CONSIDERATIONS REGARDING COMBUSTION

Unburned fuel can be caused by coal reactivity, coal fineness, O_2 availability, air/fuel imbalances, residence time, and gaseous losses. One of the most effective ways to reduce unburned fuel is to reduce the size of coarse particles (Kitto & Stultz, 2005:14.19). The fineness of coal particles should be as stated by the Rosin-Rammler guideline.

The quantity of O_2 supplied depends on the O_2 content of the coal, the temperature and time of exposure. By increasing the excess air, the O_2 increases and decreases the unburned fuel content but at the cost of heat efficiency and higher NO_x emissions (Kitto & Stultz, 2005:14.19).

Carbon monoxide (CO) is an indication of incomplete combustion (Kitto & Stultz, 2005:14.19).

Igniters using natural gas or liquid fuel are used to ignite the PF/PA mixtures as these are first supplied to the burner and for flame stability. The igniters are operated with a capacity of 10% of the actual coal burner (Howard *et al.*, 1966:C27; Kitto & Stultz, 2005:14.20; Messerle *et al.*, 2014:294). Igniters are also used to heat up the combustion chamber for spontaneous combustion of coal and initiate the air heaters.

In Russia, fuel oil consumption for ignition and flame support is approximately 5 million tons per year. In Kazakhstan, the amount is estimated at 1 million tons per year (Messerle *et al.*, 2014:295).

2.3.6 UNCONVENTIONAL AND NEW BURNER TECHNOLOGY

- **The cyclone furnace** was developed to combust coal with a low melting point for its ash that would otherwise enter the super heaters of conventional furnaces and create slagging. This furnace requires less fuel preparation and operating cost, is smaller, and produces 15%-30% fly ash compared to 80% for conventional PC burners. The coal and a portion of air are introduced to the front of the cyclone by a uniquely designed burner. A swirling motion is created in the main barrel by SA entering the upper barrel tangentially. The re-entering throat allows the combustion gases to exit. The ash creates molten slag, which is drained through a slag tap after creating a layer in the cyclone barrel. The cyclone furnace can operate with large coal particles in comparison to conventional PC burners. Where a nominal particle size of 70% passing through a 75 μm mesh is acceptable in conventional PC burners, the cyclone furnace can operate with coal particles with 95% passing through a 4.75 mm mesh (Kitto & Stultz, 2005:15.1). Coal particles of this size are too large for complete combustion in a conventional burner, and would pass through rendering excessive unburned C. Thus, a high residence time for the particles is required for complete combustion. The residence time is obtained by creating a molten slag layer on the wall, capturing the large particles while the smaller particles and volatiles combust and provide the necessary heat for the large particles to burn completely. The molten slag layer is caused by very high heat and turbulent conditions created by the combustion air, which is inserted tangentially at a high velocity. This tangentially inserted air's centrifugal force throws the large particles and molten ash to the inside of the cyclone surface. The tangential air creates a recirculation zone in the core of the cyclone, as a vortex. The combustion gases flow from the recirculation zone to the vortex, where they exit the furnace. The residence time in the recirculation zone is affected by the velocity of the tangential SA air and the size of the re-entrant throat. Because of the high heat release rate and low heat absorption rate the temperatures are very high and therefore create very high NO_x emissions (Kitto & Stultz, 2005:15.2). Erosion and corrosion are the two most serious maintenance factors (Kitto & Stultz, 2005:15.10).

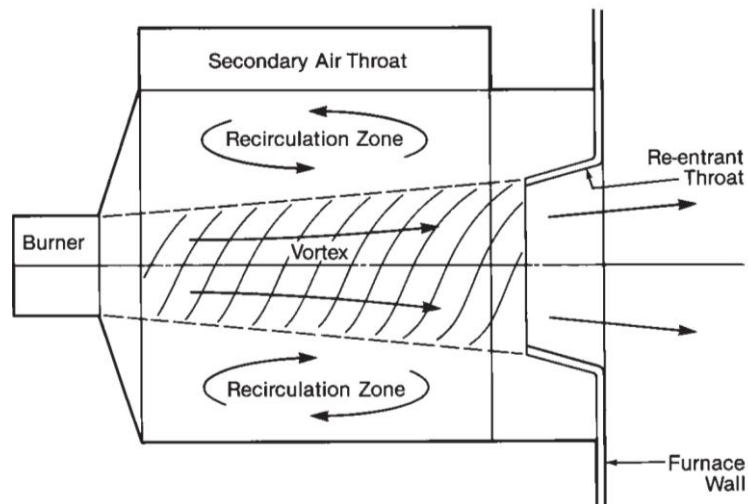
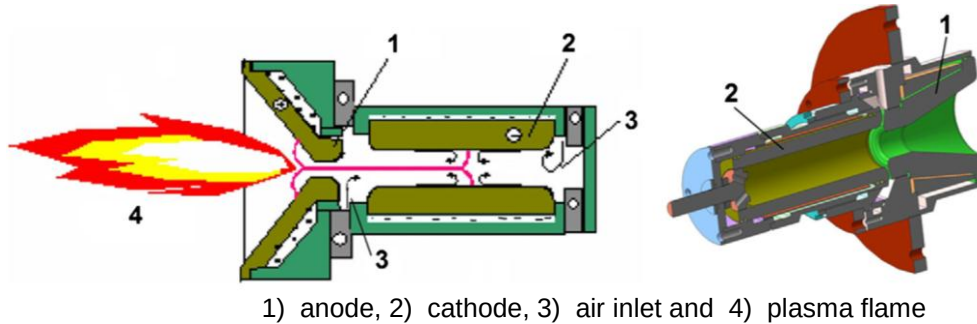
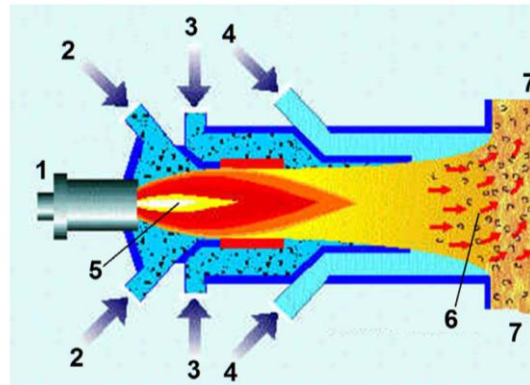


FIGURE 2.16: CYCLONE FURNACE

(Kitto & Stultz, 2005:15.2)

- The radial biased swirl coal burner** consists of a PA tube with a constantly changing cross-section and an impact ring. Inside the PA tube, the PC and air mixture is divided into two streams, an inner PC-rich annular jet and an outer PC-lean jet, so that biased firing is created. The impact ring further divides both streams where most of the rich PC is situated on the outside of the PA pipe. The rich PF/PA mixture flows through the flaring PA to increase the recirculation zone and enhance the ignition capability of the burner. The inner and outer SA nozzles comprise of integral-movable axial spinning swirl generators with widening areas. These swirl generators cause proper mixing, initial combustion, and a recirculation zone. Low-ranked coal can be burned in this burner because of the burner's versatility. It also consists of over-fire air and air staging (Xue *et al.*, 2009:3559).
- The **collision block type fuel-rich/lean burner** was designed to combust low volatile coal with lower NO_x emissions. It is used in tangential corner-fired burners. The collision block or rich/lean fuel separator is placed in the fuel line to create a rich and a lean fuel/air stream. The fuel rich stream is injected into the flame via a nozzle and is a good stabiliser because of its high concentration of fuel. The fuel-lean stream is introduced through a nozzle between the water wall and the fuel-rich nozzle. The combustion that takes place in sub-stoichiometric conditions lowers NO_x emissions (Zhou & Cen *et al.*, 2007:718).
- Plasma burner technology** was developed to introduce better ignition and combustion of solid fuels and to decrease NO_x emissions. The fact that fuel oil is mostly used for flame support and ignition of solid fuels means it is very expensive and requires high maintenance. This support is especially required when low-ranking coal is fired (Messerle *et al.*, 2014:294, 295). Therefore, the plasma burner technology comes in very handy to replace the fuel oil (Messerle *et al.*, 2014:295). A portion of the coal is fed to the plasma fuel system where it receives high energy from the plasma flame (4000°C – 6000°C), which causes gasification and incomplete oxidisation of the coal. Because of the lack of O, the charred coal is oxidised to CO. At completion of the process, a mixture of combustible gases and oxidised char is acquired at a temperature of 1300 K. The mixture then travels to the furnace, where it is ignited when interacting with the SA. The particles also disintegrate and decrease in size when introduced to the high-temperature plasma flame. This speeds up the devolatilisation and oxidation process (Messerle *et al.*, 2014:295). Ignition occurs within 2-3 seconds (Messerle *et al.*, 2014:296). The unburned C is decreased by 75% and NO_x emissions by 50%. The lower NO_x emissions are credited to the fact that the N₂ contained in the fuel forms molecules and not NO_x because of the lack of O₂. The only NO_x formed originates from the combustion of air (Messerle *et al.*, 2014:297). The plasma burner technology also improves on combustion efficiency (Messerle *et al.*, 2014:299).

**FIGURE 2.17: PLASMA BURNER**(Messerle *et al.*, 2014:295)

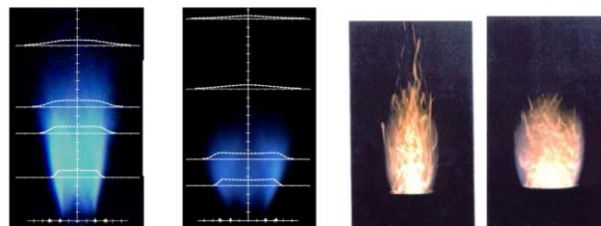
1) plasmatron, 2) primary air/coal mixture (stage 1), 3) primary air/coal (stage 2), 4) secondary air, 5) plasma flame, 6) plasma activated PC flame and 7) furnace

FIGURE 2.18: PLASMA BURNER(Messerle *et al.*, 2014:295)

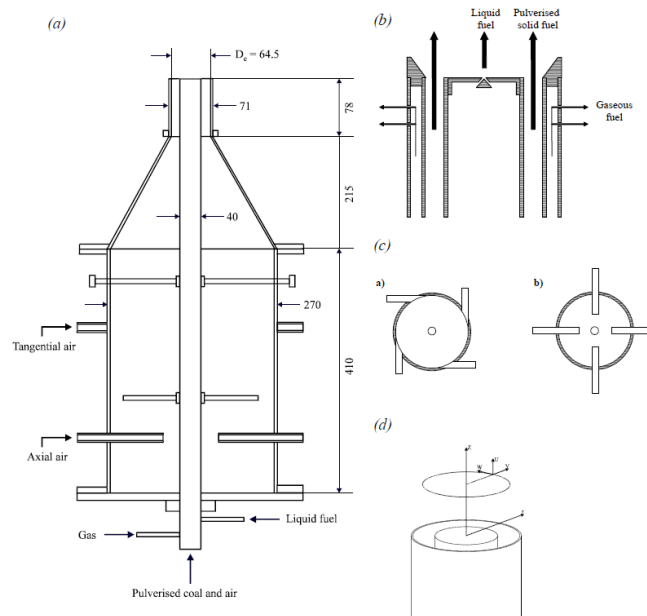
2.4 SCALE BURNERS

2.4.1 100 kW SCALE BURNER

A scale model of a 10 MW industrial burner has been manufactured at the Mechanical Engineering Department, Laboratory for Steam Boilers, Turbines & Thermal Plants, Greece. This model is 100 kW in thermal rating and has the ability to combust gaseous fuels, liquid fuels and solid fuels (Orfanoudakis *et al.*, 2005:2). It is not easy to scale down dimensions when developing a scale model of a larger burner because not all physical and chemical processes scale down linearly and equally. It is difficult to develop a scaled-down burner because the residence time is decreased; causing less combustion gas to be released and that creates flame instability. It is not possible to sustain a constant flame for solid fuels and gaseous fuel is needed for support burning, Figure 2.19 (Orfanoudakis *et al.*, 2005:3).

**FIGURE 2.19: GAS AND PF FLAME PROFILES WITH AND WITHOUT SWIRL**(Orfanoudakis *et al.*, 2005:18)

The most prominent problem is the residence time, which was as low as 20 ms in this scaled-down burner. This is not enough time for devolatilisation to take place. The burner was equipped with a cylindrical SA pipe and a central fuel pipe with a quarl at the exit. The fuel pipe contains three coaxial tubes where the inner tube provides liquid fuel, the middle tube provides PC, and the outer one provides gaseous fuel. The amount of swirl can be varied by changing the ratio of axial and tangential velocities, keeping the total flow rate of air constant (Figure 2.20) (Orfanoudakis *et al.*, 2005:5).



a) enlarged view of fuel gun, b) horizontal cross-section of burner, c) (i) tangential and (ii) axial inlets, d) coordinate system and velocity components

FIGURE 2.20: SCHEMATIC ILLUSTRATION OF THE 100 kW SCALE BURNER
(Orfanoudakis *et al.*, 2005:18)

2.4.2 500 kW SCALE BURNER

Another **scale burner** has been developed with a 500 kW thermal rating. This burner only operates with O_2 (Vega *et al.*, 2013:1), which changes flame characteristics, heat transfer and emissions (da Silva & Krautz, 2014:431). It is equipped with three coaxial pipes providing coal and O , Figure 2.21. The coal is transported by PA at a low temperature to avoid ignition, though the other two pipes operate at higher temperatures. This burner is operated with an excess air quantity to render 3-5% O_2 content in the flue gas. It also has a flue gas recirculation ratio of 60 – 70% (Vega *et al.*, 2013:1). The effect of different chamber lengths and diameters has been tested, as well as the quarl angle. The length of the combustion chamber affects the residence time; a 6 m long combustion chamber leads to complete combustion. The diameter of the combustion chamber influences the ignition, flame stability, and CO concentration. With a larger diameter, the ignition point is closer to the burner, and residence time is increased, CO emissions are decreased and a higher temperature is obtained, Figure 2.22. A 1500 mm diameter combustion chamber was chosen to be the optimum. The quarl angle greatly affected the flame profile, ignition, and flame stability, where the larger angle of 50° was found to affect these parameters positively (Vega *et al.*, 2013:1).

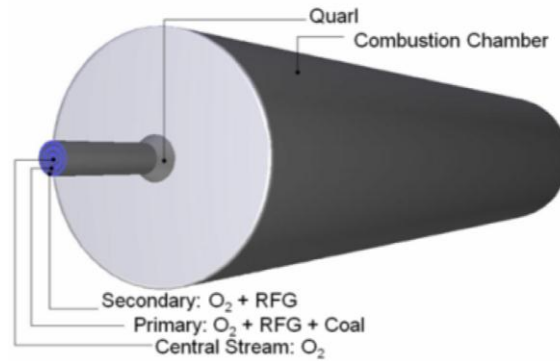


FIGURE 2.21: SCHEMATIC ILLUSTRATION OF THE 500kW SCALE BURNER
(Vega *et al.*, 2013:1)

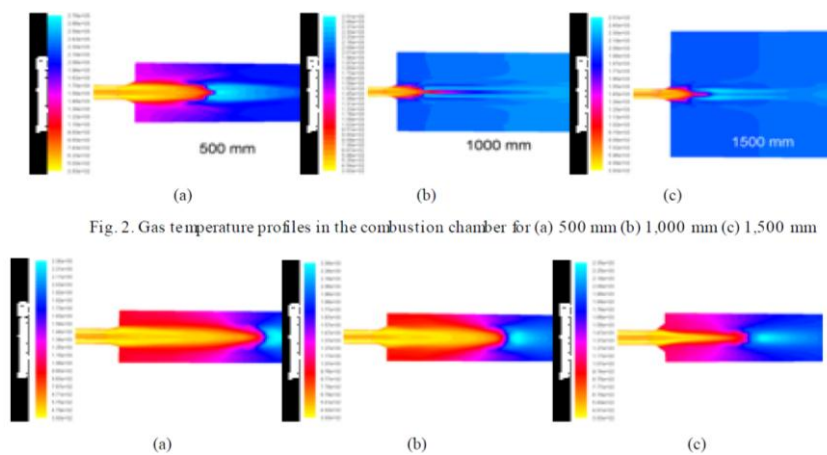


FIGURE 2.22: TEMPERATURE PROFILES AT DIFFERENT QUARL ANGLES AND COMBUSTION DIAMETERS
(Vega *et al.*, 2013:2,3)

2.4.3 ESKOM 1 MW PILOT SCALE TEST BURNER

South Africa's 1 MW pilot scale combustion test facility (PSCTF) is located at the Eskom research facility, Research Testing and Development. Before the coal that has to be tested is inserted, diesel is used as light-up fuel and igniter. The PSCTF is designed to use residence time scaling, which states that the coal entering the PSCTF will be exposed to the same temperature and residence time as in a full-scale power plant (Rajoo, 2010:23). The furnace consists of a single down-fired variable swirl burner, which is flexible to swirl numbers between 0 and 2 and has over-fire ports for low NO_x testing, Figure 2.23 (Rajoo, 2010:24).

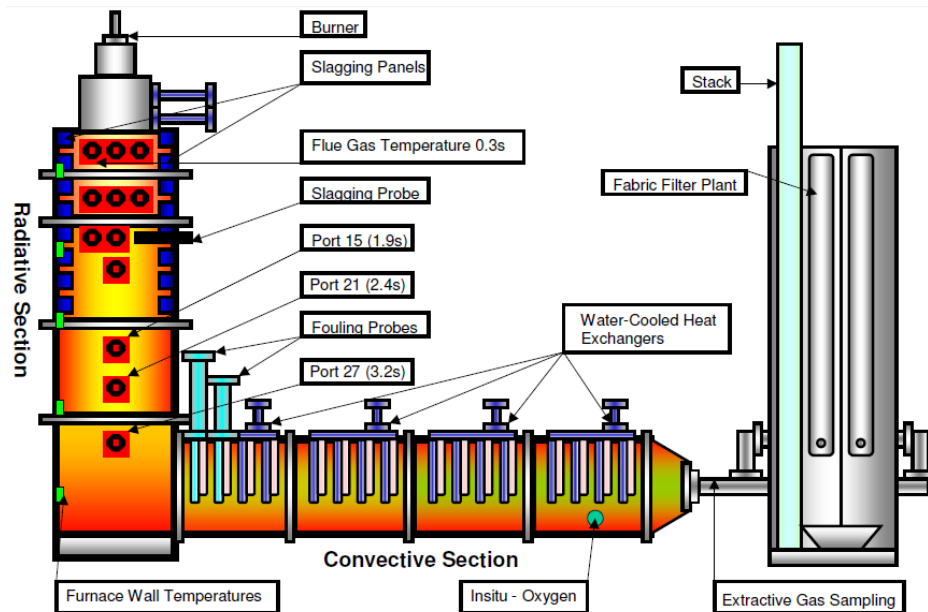


FIGURE 2.23: ILLUSTRATION OF 1MW PILOT SCALE TEST BURNER
(Rajoo, 2010:23)

2.5 COMPUTATIONAL FLUID DYNAMICS:

The advancement of computer technology has created the opportunity for engineers to deal with large-scale problems in more detail (Kitto & Stultz, 2005:6.1; Munson *et al.*, 2010:701; Sayma, 2009:8). Numerical models used in computer technology have emerged as an important field in engineering. Numerical methods are used if an analytical method does not exist or when the analytical method needs to be calculated repeatedly. Physical and mathematical models are created to present an actual apparatus. A mathematical model has two approaches. The first one is to model the behaviour of a system. Examples of these are network flow models and heat transfer. The second is to model the fundamental physics of a system to determine the behaviour. Examples of these are CFD and chemical reaction models (Kitto & Stultz, 2005:6.1; Sayma, 2009:8).

Discretisation is a process to develop a set of algebraic equations replacing the partial differential equations on which fluid dynamics are based. The three techniques used most often are the infinite difference method, the finite element or volume method and the boundary element method (Ashgriz & Mostaghimi, 2002:9,11,14,15; Munson *et al.*, 2010:701).

Once the transport equations have been discretised, they must be solved throughout the domain at separate points. This means that the separate points must be specified and relationships identified. These points with connections between them then create a pattern that looks like a grid or mesh. Mesh generation is very important where the following factors play a big role:

- The mesh must represent the geometry being modelled.
- Enough detail should be used across the domain.
- The total size and quality of the mesh influences the accuracy.

The units in a mesh are called control volumes or cells and envelope the mesh without overlapping, Figure 2.24. The types of cells are triangular, quadrilateral, tetrahedron, hexahedron, pyramid, and prism, Figure 2.25. The mesh may consist of a single cell type or multiple cell types. The two types of meshes are structured and unstructured. The structured mesh places the cells in an orderly fashion, making it easy to obtain the neighbouring cells' information. It is furthermore easy and fast to generate. This type of mesh has difficulty with complex geometry. The unstructured mesh is sufficient for complex geometries owing to its flexibility, but obtaining the neighbouring cells' information is more difficult. The capability of different types of cells also increases the accuracy of the mesh (Ashgriz & Mostaghimi, 2002:44,46,47; Kitto & Stultz, 2005:6.9,6.10; Sayma, 2009:35).

Combustion systems consist of turbulent and reacting flow conditions. It is not yet possible to design a detailed CFD model of such turbulent conditions taking place in power stations and therefore a simplified model needs to be used. To simplify the model, the fundamental equations must be time-averaged to eliminate fluctuations of turbulence and to create a separate turbulence model to account for the turbulent influences on the flow. The fundamental equations can then be solved for the average quantities. Otherwise, a large eddy simulation (LES) can be used by using a turbulence model for the small-scale fluctuations, whereas the large-scale turbulent fluctuations can be directly solved. The LES is a huge step forward in turbulence modelling, but a great amount of computer power is necessary (Kitto & Stultz, 2005:6.5; Kurose *et al.*, 2009:144; Warzecha & Boguslawski, 2014:733).

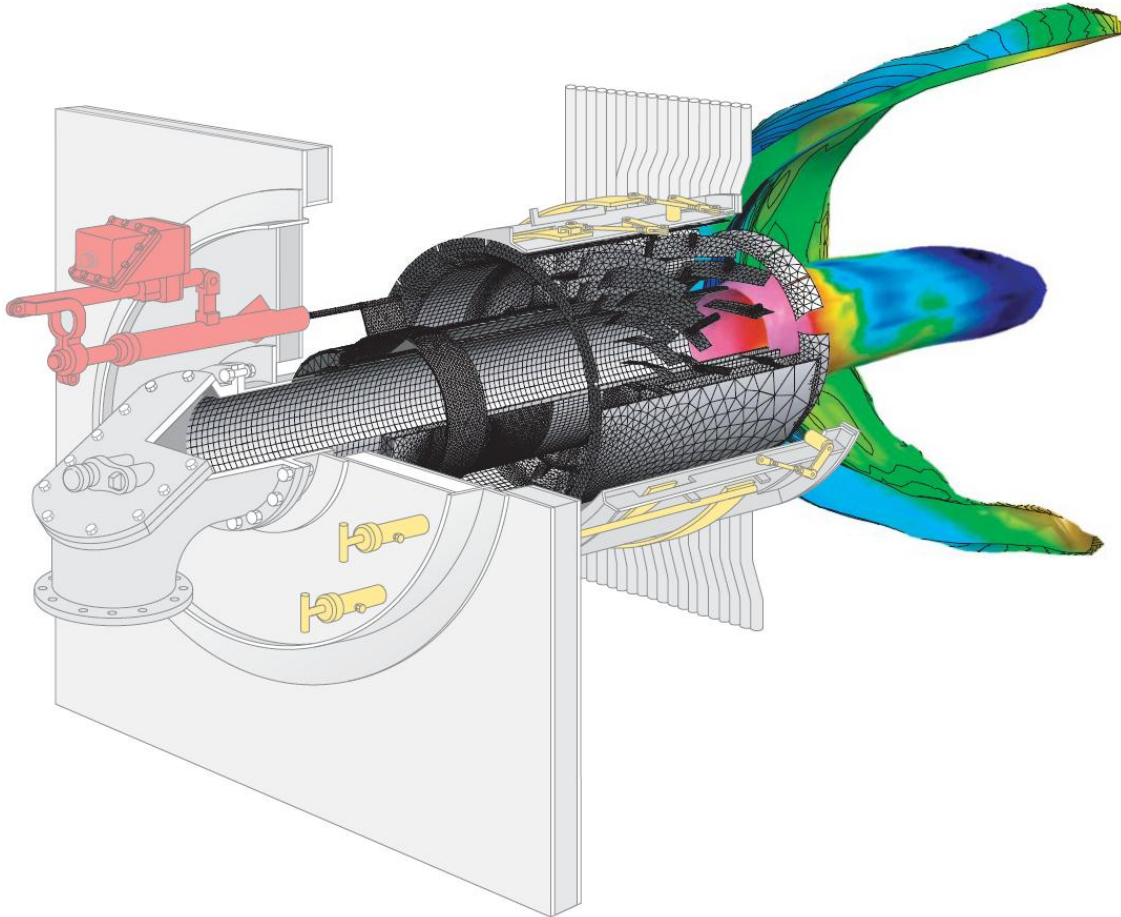


FIGURE 2.24: CFD OF A BURNER SHOWING MESH GENERATION AND FLOW ZONES
(Kitto & Stultz, 2005:6.26)

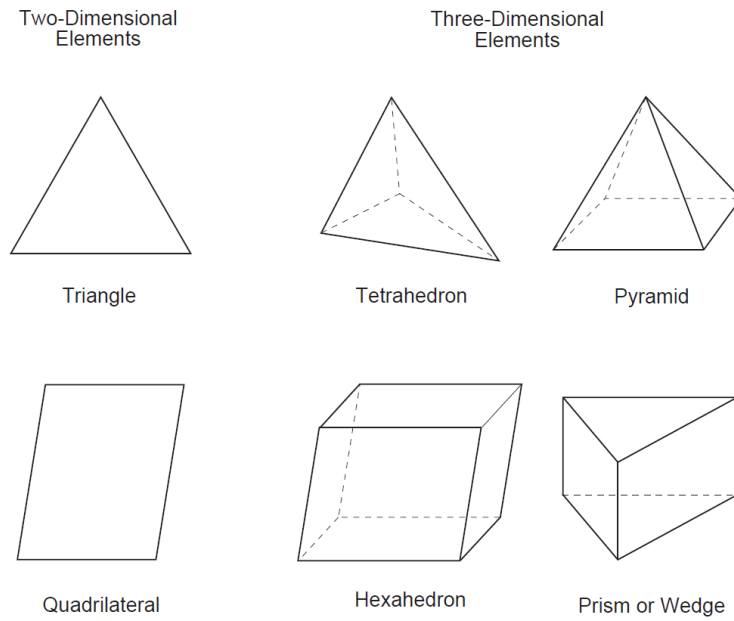


FIGURE 2.25: CELLS USED FOR MESH GENERATION
(Kitto & Stultz, 2005:6.10)

3. CONCEPT DESIGN OF THE RESEARCH FACILITY

3.1 CONSIDERATIONS IN BURNER DESIGN

The following section systematically explains the sequential order of functionality and the considerations taken into account in designing this small-scale micro coal burner.

To conduct the tests mentioned in Section 1.1 eventually, it is very important that the micro coal burner should operate and replicate combustion in the same manner as that of a full-scale burner in a power station. Because of the size restriction and cost, the coal burner will be scaled down to a practical minimum. The size of the burner and heat released must not influence the tests that will be determined in the future, provided the functioning is dimensionally correct. As with burners on power stations, this micro coal burner will use PA for transportation of PF, SA to provide enough air for complete combustion, an igniter for starting combustion and a combustion chamber at constant atmospheric pressure (Kitto & Stultz, 2005:14.2).

The first step is to determine the size of the burner in terms of rating and physical dimensions. The amount of heat release in kW that the burner can produce affects its physical dimensions. The micro coal burner has to be as small as possible. This will be determined by the smallest amount of PC that can be fed to the burner at a constant flow rate (independent variable). A constant flow rate is necessary for a stable flame for a required period to enable a representative mass-energy balance.

In another research project, it was determined that the lowest mass flow of PC ($\dot{m}_{\text{COAL}}$) that a small-scale feeder could deliver is 1.6 g/s. The micro burner is required to conduct tests on various types of coal, therefore the A:F will differ for each type of coal. Furthermore, this burner will be difficult to adjust to different mass flow rates of air ($\dot{m}_{\text{AIR}}$), because of the small scale. The cost of manufacturing of adjustable swirl generators and the burning velocity that must be kept constant. By using the A:Fs for various types of coal and a fixed $\dot{m}_{\text{AIR}}$, the mass flow of coal ($\dot{m}_{\text{COAL}}$) must thus be adjusted accordingly. This means that the lowest $\dot{m}_{\text{COAL}}$ will be used for the highest coal quality and vice versa, for constant heat release. Based on the above criteria, it was determined that a burner of 40 kW will be designed.

The A:Fs were determined by using 3.5% excess O_2 in the flue gas, which corresponds to 20% excess air at the burner outlet to ensure complete combustion, although it might cool down the flame by a fraction (Kitto & Stultz, 2005:10.15). To ensure safety the A:Fs inside the PF/PA and primary core air (PCA)/gas igniter tubes will be calculated to be outside the upper and lower explosion limits (UEL and LEL respectively) of coal and gas so that they comply with the FFFR (Eskom, 2012:35).

Furthermore, the combustion of coal and gas was investigated and the essential values were calculated for three different types of coal by using the fundamental laws of combustion. These laws are:

- Conservation of matter
- Conservation of energy
- Ideal gas law
- Law of combining weights
- Avogadro's law
- Dalton's law
- Amagat's law

(Kitto & Stultz, 2005:10.3).

The burning velocity mentioned earlier is another parameter that is very important, since it determines at what velocity the fuel and air must be inserted into the various tubes, as well as the combustion chamber so that a stable flame can be sustained. The burning velocity and the mass flow of both air and fuel will then be used to determine the combustion chamber and tube diameters. The burning velocity for gas is about 0.4 m/s. The burning velocity for coal, ranging from 0.09 m/s – 0.37 m/s,

depends on the size and composition of the coal particle. The 0.37 m/s value is a broad maximum and was obtained for a 50 μm particle with 36% volatile matter (Huzayyin *et al.*, 2007:54; Slezak *et al.*, 1985:251).

South African power stations use coal that varies from low grade to high grade. The low-grade coal is more difficult to ignite. According to Li *et al.* (2009:1222) coal does not ignite easily in an environment below 700°C and thus an igniter is necessary. An LPG gas burner will be used as igniter. Although LPG has a flame temperature of 1970°C (ref), the combustion chamber will be raised to a temperature of 700°C or higher. This will ensure coal combustion if the coal does not pass through the gas flame directly. The gas burner will form part of the micro coal burner. LPG gas will be used, since it has a high flame temperature and high NCVp and is commercially available. According to Kitto & Stultz (2005:14.20), igniters on power stations operate at 10% or higher of the actual coal burner heat rate. The gas burner for the micro coal burner will therefore be 6 kW in thermal capacity, which is 15% of the coal burner. This is higher than 10% because in power stations, the furnace consists of multiple burners and igniters, which heat up a furnace faster than a 10% gas burner will do. The gas burner can also not provide excessive thermal energy, since the combustion core air provided for the gas forms part of the PA. The core air, PA and SA then comprise the combustion air for the coal. Also, according to Kitto & Stultz (2005:14.7), the higher the SA:PA is, the better the burner performance.

Since combustion and airflow are unpredictable, this gas burner will also be used as an experimental apparatus for the coal burner. To achieve this, the gas burner will operate in the same way as the coal burner does.

The manner in which the air and fuel are inserted into the burner plays an important role. It was decided to swirl the air, PF and gas. Swirl increases the mixing of fuel and air, reduces flame length and creates a recirculation zone. These factors provide better combustion, higher furnace temperature, higher residence time, shorter ignition time, and better flame stability and draw the flame to the centreline of the furnace. A shorter flame length requires a shorter combustion chamber and will save cost. The high temperature will create more NO_x formations, but will decrease the amount of unburned fuel and CO emissions. Low NO_x will not be taken into account, since it is more important to have complete combustion with low unburned fuel and CO emissions (Kitto & Stultz, 2005:14.19) in determining the NCVp in the future. Normally with burners, decreasing the NO_x emissions will increase the CO emissions and unburned C (Chen *et al.*, 2010:722). Since it is difficult to maintain the coal flame without gas support (Orfanoudakis *et al.*, 2005:3), the recirculation zone must increase the residence time of coal particles. The burnout time for coal particles varies between 0.1 and 2.8 seconds (Makino & Chung, 2009:2072), thus an increased residence time will greatly assist to achieve complete combustion. Since finer PC promotes better combustion (Xiumin *et al.*, 2000:3), the PC will be ground to < 212 μm . The fact that the recirculation or vortex zone draws the flame to the centreline of the furnace means the temperature on the walls of the furnace will be lower, avoiding temperature excursions. To ensure adequate burnout time, the combustion chamber must be long enough and have an adequate diameter, since these factors also affect combustion efficiency (Vega *et al.*, 2013:1).

In full-scale burner pipes, it is recommended that the velocity of PF/PA must be a minimum of 18 m/s to avoid PF settling (Eskom, 2012:66). The PF/PA velocity in the tube of the micro burner will be designed to be as high as possible, dimensionally scaled down. Because of geometry influences, this will not necessarily be equal or higher than the recommended minimum velocity. This will still be adequate because the PF in power stations travel long horizontal distances before it enters the furnace. The PF in the micro coal burner will not have a long horizontal distance to travel because the PF is unlikely to settle even at lower velocities. Therefore, no need to comply to the FFFR.

CFD were used to understand the flow dynamics and design of the micro coal burner better. CFD was, however, not dealt with in extensive detail, but only used as a superficial guideline.

3.2 CONCEPT DESIGN

The schematic illustration below (Figure 3.1) shows the anticipated layout of the micro coal burner.

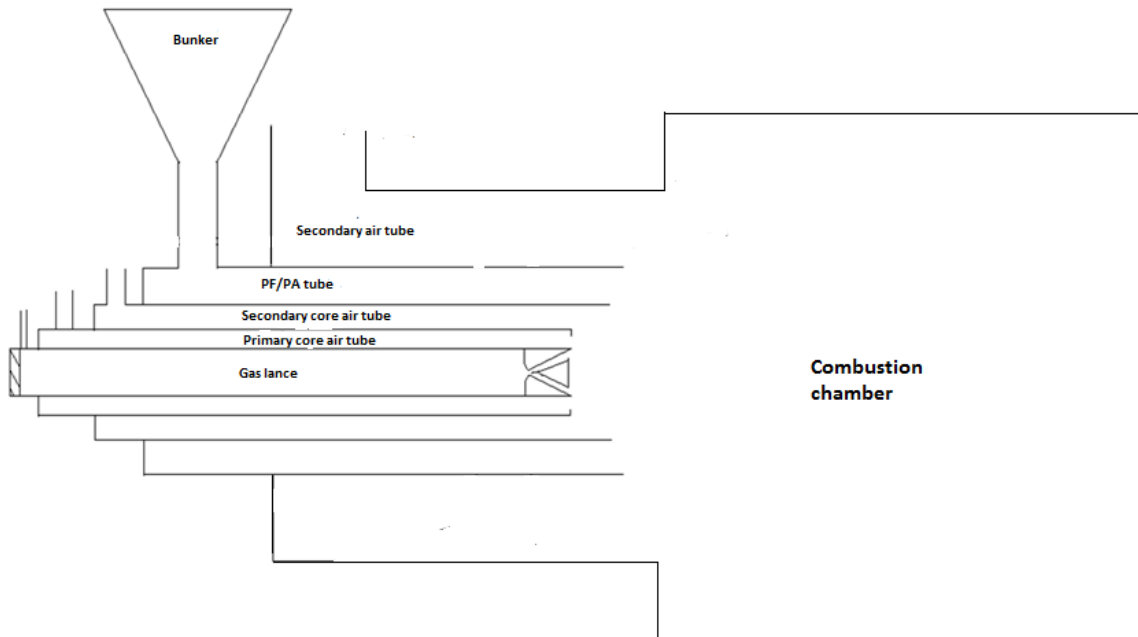


FIGURE 3.1: INITIAL SCHEMATIC REPRESENTATION OF MICRO COAL BURNER CONCEPT DESIGN

Discussion of the layout:

The LP gas used in the gas lance as primary igniter has an NCVp of 46.190 MJ/kg and a flame temperature of 1970 °C. The core air, which is the air supply for the combustion of the gas, will be divided into two portions. Firstly, the PCA is the smaller quantity to only swirl and mix with the gas partially. The quantity thereof is less than that given for the LEL for propane. Secondly, the secondary core air (SCA) will supply sufficient excess air to create a combustible A:F for the gas.

After the feeder, the PF will mix with the PA and be transported to the burner mouth. Similarly, the PA will transport the PF to the burner mouth, also with a safe sub-stoichiometric A:F inside the PF/PA tube to prevent upstream combustion. This is to comply with the FFFR Regulations (Eskom, 2012:35). Since the A:F is too low in the PF/PA tube for ignition and sustained exothermic combustion, the SA tube provides the excess air and cooling for those tubes.

3.3 COMBUSTION CALCULATIONS

In this section, the combustion calculations for coal and gas is explained. The outcome from this section will be to determine the A:F_s for the predetermined $\dot{m}_{\text{COAL}}$ (independent variables) of core, PA and SA for three different types of coal (low, intermediate and high grades).

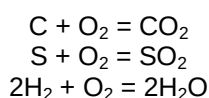
Because of the different types of coal that will be burnt, the rating of the burner will be close to 40 kW for the range of coal qualities. As mentioned before, to generate this quantity of energy, a certain amount of coal needs to be supplied. The $\dot{m}_{\text{COAL}}$ is calculated with the energy ($\dot{Q}_{\text{COAL}}$, 40 kW) and the CV (MJ/kg) of the type of coal used. The minimum, nominal, and maximum CVs (14 MJ/kg, 20 MJ/kg, and 26 MJ/kg respectively) will be used for the burner range. Three known coal analyses from Eskom power stations as close possible to those mentioned will be used in the calculations. The CVs and elemental analysis of Lethabo, Kendal, and Duvha power stations are used. From this, the A:F_s and the $\dot{m}_{\text{AIR}}$ need to be determined for the best possible combustion. The $\dot{m}_{\text{AIR}}$, however, will be determined by the applicable A:F determined for each type of coal by balancing the chemical equations for combustion.

3.3.1 IDEAL STOICHIOMETRIC AIR-TO-FUEL RATIO

Three different types of A:F_s exist, sub-stoichiometric, stoichiometric and hyper-stoichiometric. Stoichiometric suggests that the theoretical exact amount of air is supplied for ideal combustion to ensure that there is no unburnt C or volatiles and there is no excess air (O₂) remaining in the flue gas. Since this is theoretical, there is also no N₂ reacting to form NO_x. A sub-stoichiometric state suggests too little air is supplied for complete combustion and therefore there might be unburnt C and CO in the flue gas. Finally, hyper-stoichiometric means excess air (O₂) is supplied, producing less unburnt C, etc. and having excess air (and O₂) in the flue gas. In the case of coal, it is important to achieve complete combustion, therefore a hyper-stoichiometric A:F is supplied.

The balancing of the chemical equation for combustion is explained below by using a nominal type of coal as example.

Firstly, the ideal combustion equations with a stoichiometric A:F is solved. For each of the elements taking part in combustion, the equations need to be balanced, as in the equations below:



Then, assigning the atomic mass to each of the elements in the equations above and dividing each one by the atomic mass of C, S and H₂ respectively, the volumetric/mole quantity (kg/kmole) is obtained for each element in the equation. This shows the volumetric amount of O₂ needed for stoichiometric combustion (since mole fraction = volume fraction, according to Amagat-LeDuc's law). The elemental analysis below provides the gravimetric mass (kg) values for N, O, C, ash, and moisture contained in the coal as a percentage.

TABLE 3.1: ELEMENTAL ANALYSIS FOR NOMINAL COAL

COAL QUALITY					
GRAVIMETRIC %	SYMBOL	UNITS	AS RECEIVED	AIR DRIED	DRY BASIS
ELEMENTAL ANALYSIS					
Nitrogen	N	%	1.151	1.180	1.227
Oxygen (by difference)	O	%	7.166	7.350	7.640
Carbon TOTAL	C _{TOTAL}	%	53.021	54.380	56.525
Ash	A	%	28.465	29.195	30.347
Sulphur	S	%	0.878	0.900	0.936
Hydrogen	H	%	3.120	3.200	3.326
Surface Moisture	M _S	%	2.500	0.000	0.000
Inherent Moisture	M _I	%	3.700	3.795	0.000
Total moisture	M _T	%	6.200	3.795	0.000
Total		%	100.000	100.000	100.000

(See Appendix A.1 for the proximate and ultimate (elemental) analyses of nominal, high and low grade coal used for this burner design.)

These values are used to determine the mass in gram needed for combustion per 100 g of coal. The equations are as follows:

TABLE 3.2: STOICHIOMETRIC COMBUSTION EQUATIONS FOR NOMINAL COAL

(mole)	C	+	O ₂	=	CO ₂
(g/mole)	12.011		31.999		44.010
(g)	1.000		2.664		3.664

(g)	53.021		141.253		194.274
(mole)	S	+	O ₂	=	SO ₂
(g/mole)	32.060		31.999		64.059
(g)	1.000		0.998		1.998
(g)	0.878		0.876		1.753
(mole)	2H ₂	+	O ₂	=	2H ₂ O
(g/mole)	4.032		31.999		36.030
(g)	1.000		7.937		8.937
(g)	3.120		24.763		27.883

However, the elemental analysis shows that some O₂ is already contained in the coal and the total O₂ needed for combustion per 100 kg of coal is calculated considering this:

$$141.253 + 0.876 + 24.763 - 7.640 = 159.252 \text{ g O}_2 / 100 \text{ g coal.}$$

The coal is not burnt in pure O₂ but in air, so to convert the amount of O₂ to the amount of air required, it is divided by 23.19422% (the gravimetric fraction of O₂ in air). Thus, the total amount of air amounts to 688.647 g air/100 g coal. The stoichiometric amount of air is subsequently calculated for the actual $\dot{m}_{\text{COAL}}$ (1.949 g/s) determined earlier, which amounts to 13.422 g_{AIR}/s. Thus, the stoichiometric A:F needed for this type of coal is 6.886:1.

3.3.2 ACTUAL STOICHIOMETRIC AIR-TO-FUEL RATIO

In reality not only C, S, and H₂ take part in combustion. There is also CO in the flue gas and fractions of the C that do not fully combust. Therefore, excess air is added to restrict incomplete combustion to the minimum. In addition, because of the high temperature expected in this combustion (+ 1350 °C), the N₂ in coal (fuel NO_x) and in air (thermal NO_x) also take part and form N₂O, but not add heat as C, S and H₂ do.

To determine the amount of air, the same procedure as with stoichiometric combustion is used, except that the combustion of C is staged into initial formation of CO, then the CO (not all) undergoes further combustion to CO₂. In addition, a minor portion of the N₂ present reacts with O₂ to form NO_x (where the greater portion was found to be N₂O). A value of 1.15% unburnt C in ash, which is typical for this coal, is used. The combustion chemical equations for the actual case are therefore as follows:

TABLE 3.3: ACTUAL STOICHIOMETRIC COMBUSTION EQUATIONS FOR NOMINAL COAL

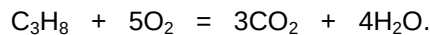
(mole)	2C	+	O ₂	=	2CO
(g/mole)	24.022		31.999		56.021
(g)	1.000		1.332		2.332
(g)	52.693		70.191		122.884
(mole)	2CO	+	O ₂	=	2CO ₂
(g/mole)	56.021		31.999		88.020
(g)	1.000		0.571		1.571
(g)	122.884		70.191		193.074
(g)	122.805		70.146		192.951
(mole)	S	+	O ₂	=	SO ₂

(g/mole)	32.060		31.999		64.059
(g)	1.000		0.998		1.998
(g)	0.878		0.876		1.753
(mole)	2H ₂	+	O ₂	=	2H ₂ O
(g/mole)	4.032		31.999		36.030
(g)	1.000		7.937		8.937
(g)	3.120		24.763		27.883
(mole)	2N ₂	+	O ₂	=	2N ₂ O
(g/mole)	56.027		31.999		88.026
(g)	1.000		0.571		1.571
(g)	0.81880		0.468		1.286

The amounts of CO and N₂O are determined from a flue gas analysis. For this exercise, the chemical equation quantities were solved to produce typical values of 100 ppm CO and 1000 ppm N₂O in the flue gas, on a volumetric dry basis (the value detected by most gas analysers). Following the same procedure as for the stoichiometric case above, the actual stoichiometric amount of air for the same $\dot{m}_{\text{COAL}}$ (1.949 g/s) amounts to 13.384 g_{AIR}/s, which is slightly different from that of the ideal case (13.422 g_{AIR}/s).

3.3.3 GAS IGNITER

Since this micro coal burner is ignited with gas, the $\dot{m}_{\text{GAS}}$ and corresponding $\dot{m}_{\text{AIR}}$ used by the gas must also be determined and included in the air supply for the combustion of the coal. The choice for the igniter gas is LPG, in this case propane. It is readily available and its properties are well known. The NCVp for this LPG (propane, C₃H₈) is 46.19 MJ/kg (Pulkrabek, 444). It also has a high flame temperature and a relatively high CV. The balanced chemical equation for the stoichiometric combustion of propane is:



Similar to the above (for coal), the stoichiometric O₂ for propane amounts to 3.628 g O₂/g C₃H₈ and the corresponding air is 15.643 g_{AIR}/g C₃H₈, which is the stoichiometric A:F. To achieve the previously determined 6 kW thermal rating for the gas igniter (see 3.1 page 3-2) the propane must be supplied at 0.13 g/s. This mass flow rate at the CV of 46.19 MJ/kg will produce the required 6 kW heat-rate.

For 0.13 g/s of propane and a stoichiometric A:F of 15.643 kg_{AIR}/kg C₃H₈, the air supply for the combustion of propane is 2.032 g_{AIR}/s. The LEL of propane is 6.57 g_{AIR}/g C₃H₈ and the upper UEL is 29.72 g_{AIR}/g C₃H₈ (Razus, 2004:776). This means that propane cannot ignite or combust with an A:F outside those limits. To comply with safety regulations, the propane will be transported (and pre-mixed with air), by primary core air, in a quantity less than the LEL in a primary core air tube to the burner mouth. It is common practice at power stations to have PA quantities even 2 – 2.5 times less than the LEL, for coal, fuel oil and gas.

Therefore, the A:F for primary core air amounts to 2.78, which renders 0.361 g/s. The SCA will join the propane/primary core air mixture only at the burner mouth, to avoid upstream combustion. The quantity of SCA is such that the total amount of air for propane combustion is the above stoichiometric quantity, since most gaseous fuels (and some liquid fuels) combust at an optimum close to stoichiometric A:F in practice as well.

3.3.4 ACTUAL HYPER-STOICHIOMETRIC AIR-TO-FUEL RATIO

For the actual case, the amount of excess air for this burner to ascertain optimum combustion is calculated to produce 3.5% O₂ in the flue gas. The FFFR suggests a minimum of 2% (at full load) and a maximum of 9% (at minimum load) (Eskom, 2012:35). Similar to the above, for this case (nominal grade coal), the actual hyper-stoichiometric A:F is calculated as 8.217 g_{AIR} : g_{COAL}, which equates to 19.663% excess air and therefore 16.016 g_{AIR}/s is to be supplied.

To calculate the corresponding hyper-stoichiometric air quantities for the minimum and maximum CV coal's A:F's the same procedure is followed as above. For the resulting calculated flue gas analyses thereof, see Appendix (Section B.1 and B.2, pages 9-4 to 9-9), containing tables for stoichiometric and hyper-stoichiometric airflows, for low, nominal and high coal grades, and gravimetric wet as well as volumetric dry analyses for each.

In conclusion, the following values were obtained for the three different types of coal for A:F, $\dot{m}_{COAL}$ and $\dot{m}_{AIR}$:

TABLE 3.4: SUMMARY OF HYPER-STOICHIOMETRIC COMBUSTION CALCULATIONS

	Minimum	Nominal	Maximum
A:F	5.963	8.217	9.453
$\dot{m}_{COAL}$ [g/s]	2.671	1.949	1.697
$\dot{m}_{AIR}$ [g/s]	15.927	16.014	16.040

These values are calculated for the combustion of coal only, but the quantity needed for the gas is added, since the gas will eventually be shut down, but the core air for the gas will still flow and contribute to excess air.

3.4 DIMENSIONS OF THE MICRO COAL BURNER

In this section, the dimensions for the coal and gas burner tubes was determined.

3.4.1 BURNING VELOCITY AND DIMENSIONS OF THE BURNER

Once the mass flow of air and coal has been determined, the dimensions can be calculated. One of the most important factors to take into account is the burning velocity. Burning velocity determines the velocity at which the air and fuel supply enters the combustion chamber. If the velocity is too high, there can be a flame blowout. If the velocity is too low the flame can burn backwards, which can cause combustion inside the PF/PA or gas tubes. Thus, when the velocity of the incoming fluids are the same as the flame velocity, a stable flame can be sustained, which is also safe. As mentioned earlier, the flame velocity of gas is ± 0.4 m/s. The flame velocity for PF is a lot more complicated than gas, but can be averaged to 0.37 m/s (Huzayyin *et al.*, 2007:54; Slezak *et al.*, 1985:251). The volatile matter in coal is gaseous and ignites almost immediately. PF particles vary in size and composition and therefore the time for combustion fluctuates between particles. The burning velocity for PF lies between 0.09 m/s and 0.37 m/s. Hence, the diameter of the tubes will be calculated corresponding to a flame velocity of 0.4 m/s.

The gas burner was designed to be 15% of the capacity of the coal burner, which is 6 kW. This will be sufficient in order to raise the temperature of the environment to 700°C or higher for the PF to ignite. The calculations for the gas burner's dimensions are given below:

A few initial input values need to be explained:

The A:F ratio in the PCA and SCA tubes was determined by using the upper and lower explosion limits (UEL and LEL) of gas (Razus, 2004:776). This means that it will be impossible to burn LPG gas outside these A:F's. The LEL is used to guarantee that there cannot be gas burning inside the PCA tube. The LEL's gravimetric A:F is 6.5701, but an A:F of 2.7794 will be used to ensure safety, as previously mentioned.

The temperature of gas when it enters the burner is assumed to be atmospheric temperature, although it is a lot cooler when it expands on exiting the pressurised container.

Air heaters will pre-heat air for the PCA, SCA and PF/PA to 70 °C and the SA to 275 °C. It assists the PF not to form bridges in the bunker and cause interlocking, because most of the surface moisture will evaporate. The calculations that influence the coal burner are made when the air heaters are already fully active, because the higher temperature affects the density of the combustion air and hence the velocity. However, during start-up, when the air heaters are not yet heated, these values will differ. The calculations were programmed in Microsoft Excel.

3.4.2 GAS BURNER DIMENSION CALCULATIONS

The sequential steps to determine the parameters of the gas burner were:

- The mass flow of gas ($\dot{m}_{\text{GAS}}$) was determined in the same manner as $\dot{m}_{\text{COAL}}$ in Section 3.2.
- Thereafter the volume flow of the gas ($\dot{V}_{\text{GAS}}$) was calculated when it left the gas lance and entered the PCA tube which, was then at atmospheric pressure.
- This enabled the value of the gas velocity (C_{GAS}) to be calculated and then the area and diameter of the gas nozzle and lance were determined.
- Repeating the previous steps for the core air provided the values of $\dot{m}_{\text{PCA}}$, $\dot{V}_{\text{PCA}}$, C_{PCA} , area and diameter for the nozzles that allowed the air to enter the PCA tube.
- When the gas and the PCA were mixed, the total volume flow thereof was used to calculate the diameter of the tube to enable a velocity of 0.4 m/s.
- The same procedure was followed to determine the $\dot{m}_{\text{SCA}}$, $\dot{V}_{\text{SCA}}$, C_{SCA} , area and diameter of the nozzles for the SCA tube.
- The diameter of the SCA tube was then calculated by taking the total of the gas, PCA, and SCA into account, to enable 0.4 m/s.

The following formulae were used to determine these values:

$$\dot{Q} = CV * \dot{m}$$

$$P\dot{V} = \dot{m}RT$$

$$\dot{m} = \rho AC$$

$$C = \sqrt{\frac{2\Delta P * 1000}{\rho}} * Cd$$

where,

P = pressure [kPa]

$\dot{V}$ = volume flow [m^3/s]

$\dot{m}$ = mass flow [kg/s]

R = specific gas constant [$\text{kJ}/\text{kg}\cdot\text{K}$]

T = temperature [K]

ρ = density [kg/m^3]

A = [m^2]

C = [m/s]

Cd = [-] coefficient of discharge

TABLE 3.5: RESULTS FOR THE GAS BURNER WITH AIR HEATERS COLD

GAS BURNER CORE AIR & GAS @ AMBIENT TEMPERATURE: 25 [°C]					
VOL. FLOW PROANE	0.090	[l/s]	C ACTUAL GAS NOZZLE	130.658	[m/s]
VOL. FLOW STOICH. CORE AIR	2.022	[l/s]	C ACTUAL PRIM. CORE AIR NOZZLE	98.207	[m/s]

VOL. FLOW _{PRIMARY CORE AIR}	0.359	[l/s]	C _{ACTUAL SEC. CORE AIR NOZZLE}	113.631	[m/s]
VOL. FLOW _{SECONDARY CORE AIR}	1.662	[l/s]	TARGET MASS FLOW _{PROPANE}	0.130	[g/s]
VOL. FLOW _{PRIM. CORE AIR & GAS}	0.449	[l/s]	TARGET MASS FLOW _{PRIM. CORE AIR}	0.361	[g/s]
VOL. FLOW _{TOTAL CORE AIR & GAS}	2.111	[l/s]	TARGET MASS FLOW _{SEC. CORE AIR}	1.671	[g/s]
Ø ID _{PRIMARY CORE TUBE}	37.800	[mm]	STD. PIPE Ø OD _{PRIMARY CORE TUBE}	42.8 x 2.5	[mm]
Ø ID _{TOTAL CORE TUBE}	81.977	[mm]	STD. PIPE Ø ID _{TOTAL CORE TUBE}	88.9 x 3.5	[mm]

TABLE 3.6: RESULTS FOR THE GAS BURNER WITH AIR HEATERS HOT

COAL BURNER CORE AIR @ PRIMARY TEMPERATURE: 70 [°C]					
VOL. FLOW _{STOICH. CORE AIR}	2.166	[l/s]	C _{ACTUAL PRIM. CORE AIR NOZZLE}	105.374	[m/s]
VOL. FLOW _{PRIMARY CORE AIR}	0.385	[l/s]	C _{ACTUAL SEC. CORE AIR NOZZLE}	121.923	[m/s]
VOL. FLOW _{SECONDARY CORE AIR}	1.781	[l/s]	MASS FLOW _{PRIM. CORE AIR}	0.336	[g/s]
			MASS FLOW _{SEC. CORE AIR}	1.557	[g/s]
C _{AXIAL PRIM CORE AIR TUBE}	0.343	[m/s]	MASS FLOW _{TOTAL CORE AIR}	1.894	[g/s]
C _{AXIAL SEC CORE AIR TUBE}	0.464	[m/s]	Ø ID _{PRIMARY CORE AIR TUBE}	37.8	[mm]
C _{AXIAL TOTAL CORE AIR TUBE}	0.410	[m/s]	Ø ID _{TOTAL CORE AIR TUBE}	81.977	[mm]

It can be seen that the $\dot{m}_{AIR}$ is higher when the air heaters are cold, as opposed to when they are heated. This higher $\dot{m}_{AIR}$ is the value when the gas burner is started cold. When the air heaters are heated to operating temperature, the $\dot{m}_{AIR}$ decreases, since the density is decreased while the cross-sectional area remains constant. The lower $\dot{m}_{AIR}$ will be used for the coal burner dimensions, since the air heaters will then be active for the duration of combustion.

3.4.3 COAL BURNER DIMENSIONS AND CALCULATIONS

After the gas burner has raised the temperature to ± 700 °C the PF will be fed to the burner. The gas burner must still be active for some time for it to ignite. The total $\dot{m}_{AIR}$ for the PF will be the total of the PCA, SCA, PA, and SA. A safe $\dot{m}_{AIR}$ of 3.355 g/s will be used in the PF/PA tube. This is 25% of the total stoichiometric air required. The results obtained, which are similar to those of the gas, are as follow:

TABLE 3.7: RESULTS FOR THE COAL BURNER: PF/PA TUBE

COAL BURNER PRIMARY AIR @ TEMPERATURE: 70 [°C]					
VOL. FLOW _{PF NOMINAL}	0.001448	[l/s]	C _{ACTUAL PF/PA NOZZLE}	116.740	[m/s]
VOL. FLOW _{STOICH. CORE AIR}	2.166	[l/s]			
VOL. FLOW _{PRIMARY AIR}	3.837	[l/s]	C _{ACTUAL PA RESULTANT}	116.740	[m/s]
VOL. FLOW _{PRIMARY AIR & PF}	3.839	[l/s]	TARGET MASS FLOW _{PA}	3.355	[g/s]
VOL. FLOW _{CORE AIR, PA & PF}	6.004	[l/s]			
Ø ID _{ANNULAR PF/PA TUBE}	140.760	[mm]	STD. PIPE Ø OD _{ANNULAR PF/PA TUBE}	152.4 x 4.5	[mm]
Ø ID _{TOTAL PF/PA & CORE TUBE}	136.493	[mm]	STD. PIPE Ø ID _{TOTAL PF/PA TUBE}	139.7 x 3	[mm]

TABLE 3.8: RESULTS FOR THE COAL BURNER: SA TUBE

COAL BURNER SECONDARY AIR @ TEMPERATURE: 275 [°C]					
VOL. FLOW _{PF NOMINAL}	0.001448	[l/s]	C _{ACTUAL SA NOZZLE}	165.432	[m/s]
VOL. FLOW _{STOICH. CORE AIR}	2.166	[l/s]			
VOL. FLOW _{PRIMARY AIR}	3.837	[l/s]	C _{ACTUAL SA RESULTANT}	165.433	[m/s]
VOL. FLOW _{SECONDARY AIR}	18.233	[l/s]	TARGET MASS FLOW _{SA}	10.604	[g/s]
VOL. FLOW _{CORE AIR, PA, PF & SA}	24.237	[l/s]			
Ø ID _{ANNULAR SA TUBE}	275.841	[mm]	STD. PIPE Ø OD _{ANNULAR SA TUBE}	280 x 2.5	[mm]
Ø ID _{TOTAL SA PF/PA & CORE TUBE}	274.231	[mm]	STD. PIPE Ø ID _{TOTAL SA TUBE}	280 x 2.5	[mm]

TABLE 3.9: AIR MASS FLOW IN THE TUBES

AIR MASS FLOW [g/s]				
STOICHIOMETRIC	TOTAL	CORE AIR	PA	SA
13.420	16.040	1.894	3.355	10.791
13.420	16.014	1.894	3.355	10.766
13.420	15.927	1.894	3.355	10.678
83.667%	100.000%	11.807%	20.917%	67.277%
83.802%	100.000%	11.826%	20.950%	67.224%
84.258%	100.000%	11.890%	21.064%	67.045%
1.195	9.453	1.116	1.977	6.360
1.193	8.217	0.972	1.722	5.524
1.187	5.963	0.709	1.256	3.998

TABLE 3.10: RESULTS OF HEAT AND MASS FLOW CALCULATIONS

HEAT & MASS FLOW		
Q _{IN BURNER MAX CV}	39.912	[kW _{TH}]
Q _{IN BURNER NOMINAL CV}	40.000	[kW _{TH}]
Q _{IN BURNER MIN CV}	38.064	[kW _{TH}]
Q _{IN GAS BURNER}	6.000	[kW _{TH}]
MASS FLOW _{PROPANE}	0.130	[g/s]
MASS FLOW _{STOICH. CORE AIR}	2.032	[g/s]
MASS FLOW _{PRIMARY CORE AIR}	0.361	[g/s]
MASS FLOW _{SECONDARY CORE AIR}	1.671	[g/s]

The diameter of the combustion chamber is calculated at 0.5 m. This diameter was determined to ensure that the flue gas would move at 0.4 m/s after combustion, towards the exit of the combustion chamber. When using the maximum burnout time of 2.8 seconds (Makino & Chung, 2009:2072) for coal particles and the burning velocity of 0.4 m/s for coal, a minimum length of 1.2 m for the combustion chamber is determined. If some of the particles do not take part in the swirl burning zone

(which will be discussed later in more detail), the length of the combustion chamber, together with an induced vortex, will ensure more complete combustion, which also correlates with the results of Vega et al. (2013:1).

4. GAS BURNER EXPERIMENT

As previously mentioned, the gas burner will be used as an igniter for the coal as well as an experiment, since it represents similar combustion and airflow characteristics as the coal. Using the previously determined dimensions, the following design was done:

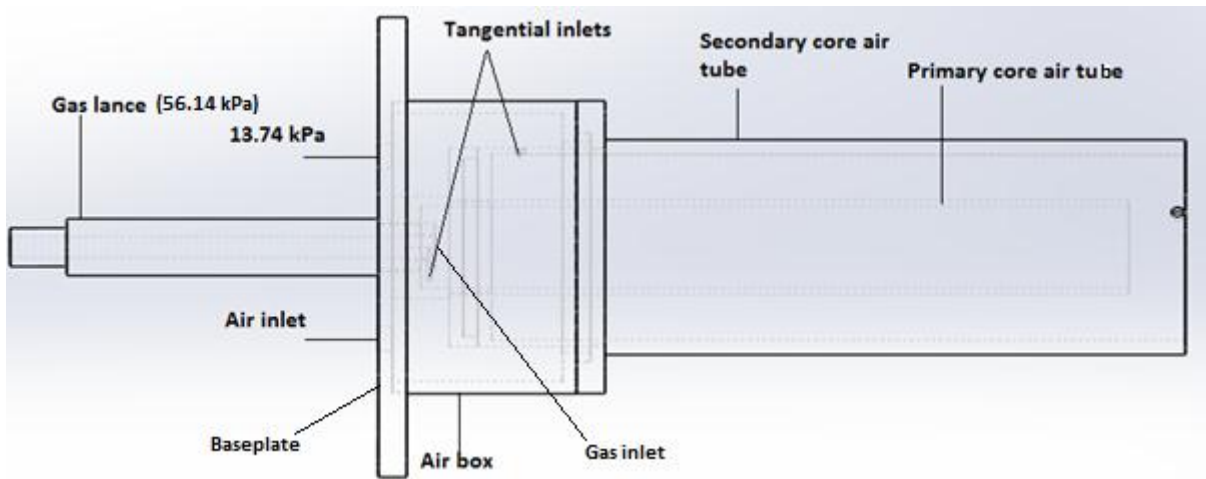


FIGURE 4.1: SIDE VIEW OF GAS BURNER GEOMETRY

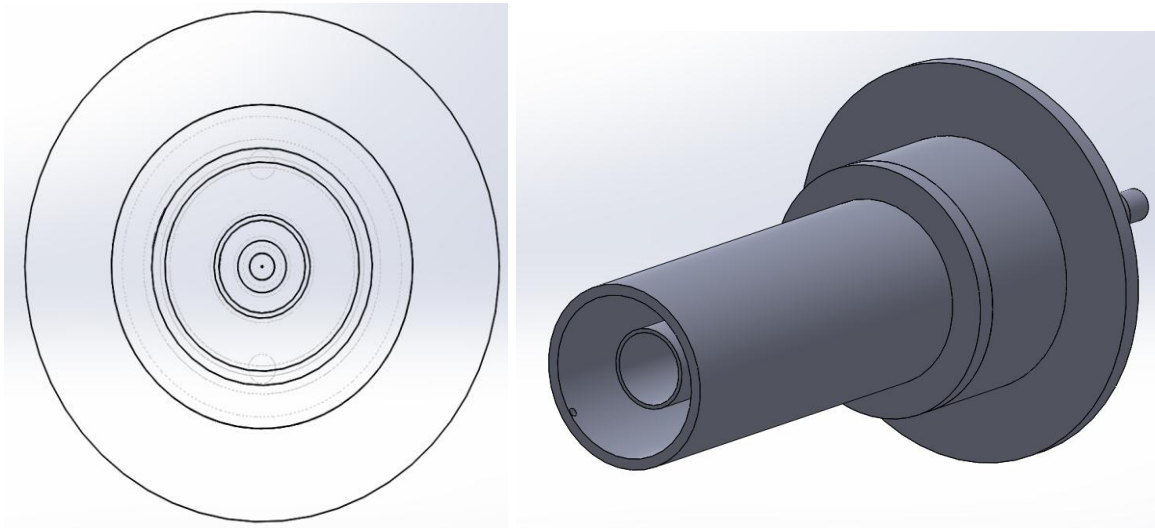


FIGURE 4.2: FRONTAL AND 3-D VIEW OF GAS BURNER MODEL

The gas burner consists of a gas lance, gas nozzle, baseplate, airbox, PCA and SCA tubes. (See machine drawing assembly in Appendix D, page 9-10). Each one of these will now be discussed:

- **Gas lance:** The gas lance is used to transport the gas from the gas pressure bottle to the gas nozzle via a flexible gas hose. The gas lance's inner diameter is $\varnothing$ 10 mm, which is much larger than the gas nozzle, to ensure the pressure measured is the static or total (stagnation) pressure, which does not have a significant dynamic pressure component. The pressure measured at the gas lance is 56.14 kPa (gauge), as in Figure 4.1 above. This pressure delivers a mass flow of 0.13 g/s through the nozzle.
- **Gas nozzle:** The gas nozzle has a $\varnothing$ 0.7 mm diameter hole to ensure that, with a gauge pressure of 56.14 kPa and a C_d factor of 0.628 (determined with CFD), the mass flow of gas is 0.13 g/s. The gas nozzle is also manufactured from brass, which will act as a flame arrestor to prevent combustion inside the gas lance.
- **Air box:** This is a concentric tube that encapsulates the PCA and SCA tubes and admits air into the said via nozzles of $\varnothing$ 2 mm and $\varnothing$ 4 mm respectively. With C_d factors of 0.64 and 0.74 respectively (determined with CFD) and applying a pre-calculated pressure of 13.74 kPa (gauge) inside the air box, the airflows of the previously calculated 0.361 g/s and 1.671 g/s are admitted respectively. The air is supplied by a compressor and is connected to the air inlet of the air box. As with the gas lance, the area inside the air box is much larger than the air supply nozzle areas of the PCA and SCA tubes and therefore the total pressure can be measured.
- **Base plate:** Used as anchor point for the burner. The air box is mounted on the baseplate.
- **PCA tube:** The PCA tube has a tangential air supply hole of $\varnothing$ 2 mm in diameter (mentioned above), which creates swirl inside the tube.
- **SCA tube:** The SCA tube also has a tangential air supply nozzle of $\varnothing$ 4 mm, which creates swirl. The pressure inside the air box delivers the required mass flow. The SCA tube is longer than the PCA tube to create a 45° angle between the two exits, longitudinally. This will cause proper mixing of the gas, PCA and SCA.

4.1 GAS BURNER TEST RESULTS

The following figures show the gas burner in operation:



FIGURE 4.3: SUB-STOICHIOMETRIC A:F COMBUSTION CONDITIONS

When igniting the gas, the amount of air is low to ensure a fuel-rich mixture that can easily be ignited. Once combustion is under way, it can be seen in the figure above, that the flame has blue and yellow parts. The yellow part represents incomplete combustion. The blue part of the flame indicates combustion that is more complete. The flame also has a swirling shape that shows that the tangential air supplies create swirl.

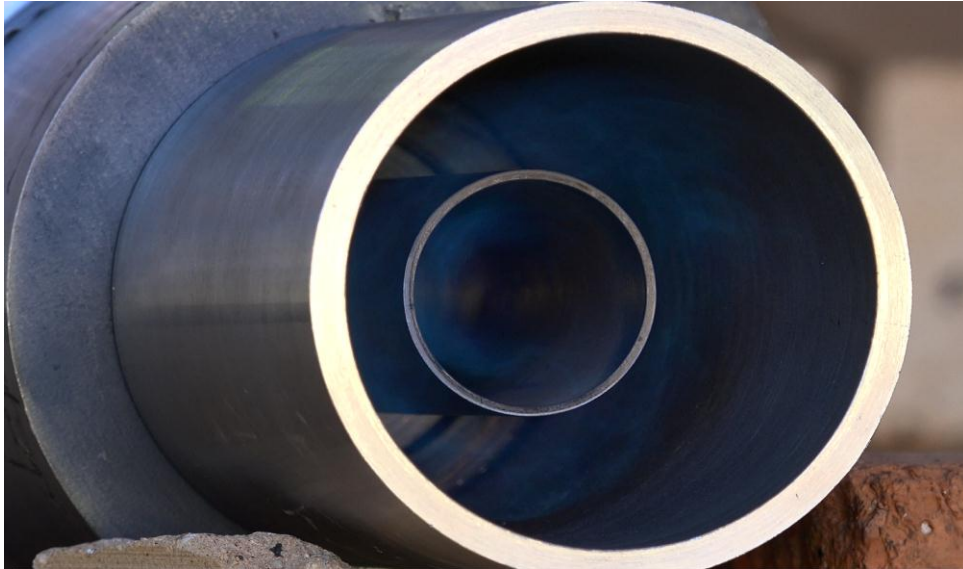


FIGURE 4.4: STOICHIOMETRIC A:F COMBUSTION CONDITIONS

During combustion of a stoichiometric A:F, it can be seen that a blue flame forms throughout. Most of the gas is burnt, since airflow and mixing are adequate. The proper mixing is due to swirl caused by the tangential air entry into the PCA and SCA tubes. The swirl generated by the tangential holes and the adequate amount of air provided confirm good combustion, which can be seen in the flame pattern and colour.

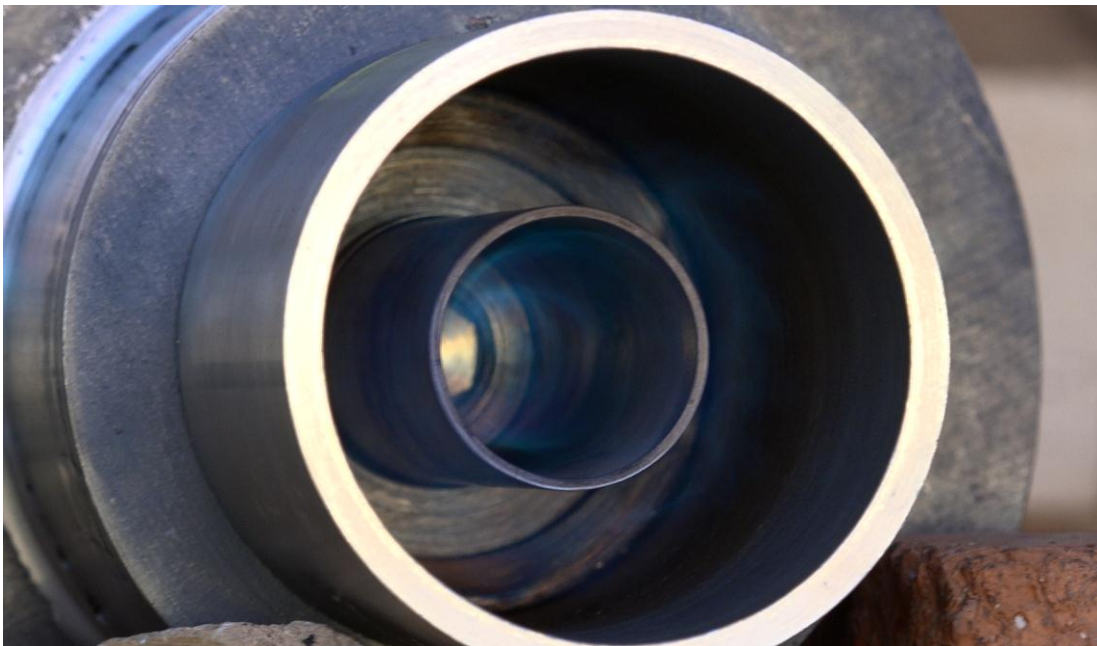


FIGURE 4.5: HYPER-STOICHIOMETRIC A:F COMBUSTION CONDITIONS

However, when increasing the air mass flow to a stoichiometric A:F, it also increases the velocity through the air supply nozzles. This increase in velocity in the tangential air supply seems to create a vortex or recirculation zone (causing a low pressure in the centre of each tube). This vortex sucks air back into the centre of the PCA tube from the atmosphere and causes a combustible A:F. Therefore, the gas burns inside the PCA tube, as can be seen in Figure 4.4 above. The flame also burns inside the SCA tube, indicating that a suction of gas into the SCA tube occurs. This is an unwanted condition, because it does not satisfy the theory of providing enough air inside the PCA tube for swirl, but insufficient air for combustion. This does not comply with the safety feature implemented to avoid burn-back either.

Increasing the amount of air further will cause a hyper-stoichiometric A:F. As can be seen in Figure 4.5 above, there is still good combustion because there are no yellow flames. This high amount of air provided will again increase the velocity. Although complete combustion for the gas occurs at close to stoichiometric conditions, the excess air supplied is to test the flame velocity of 0.4 m/s and to see when a flameout occurs. However, this does not happen, because the vortex created sucks more air in than air moving out. The recirculation zone seems to be even greater than with the stoichiometric combustion. The higher velocity of the air supply and the resulting vortex induce a greater recirculation zone. This correlates with the findings of Khanafer & Aithal (2011:5036). It can also be observed that most of the gas is burnt inside rather than outside the PCA tube, which emphasises the safety and reliability concern, which is due to incipient metal temperature excursions.

4.1.1 METAL TEMPERATURE EXCURSIONS

From the video timer, after 2 min and 20 seconds (Figure 4.6 below) it can be seen that the PCA tube started to melt and oxidise, because of the high flame temperature of propane (1970°C). The fact that the flame is on both the inner and outer sides of the PCA tube restricts cooling. The SCA tube does not suffer the same consequences since it has cooling on the outside provided by the atmosphere.



FIGURE 4.6: MELTING POINT AND OXIDATION OF PCA TUBE

4.1.2 CFD ANALYSIS OF THE TEST GAS BURNER

To investigate the problems caused by the intense vortex, a separate CFD (Star CCM+) study was done by the study leader to investigate the flow in more detail. The test burner geometry, pressures,

and flows served as input parameters. This study confirmed the behaviour of the burner, as seen in figure 4.7:

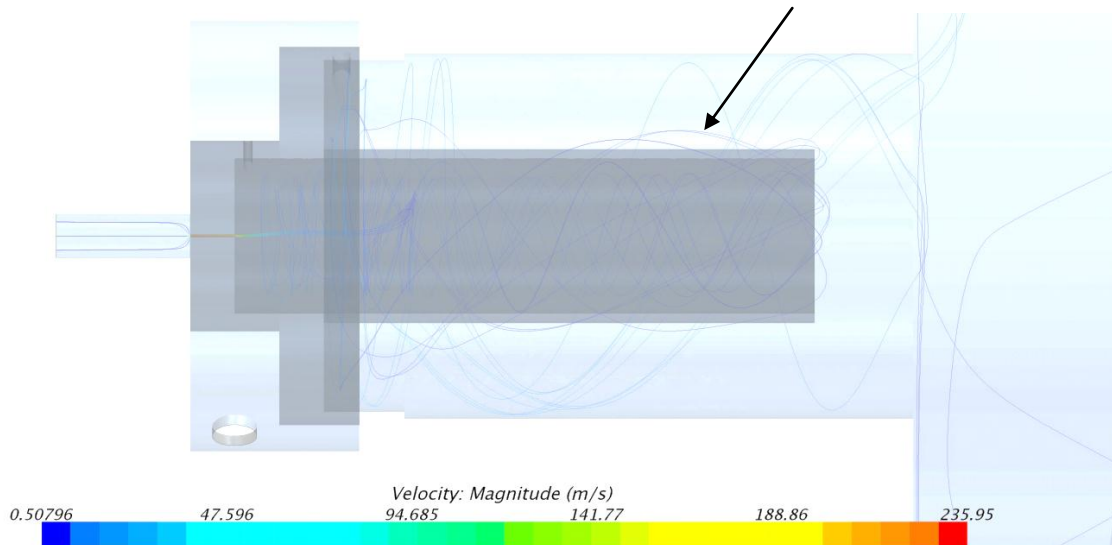


FIGURE 4.7: CFD ANALYSIS (1) - VELOCITY AND FLOW PATH OF GAS PARTICLES IN THE TEST GAS BURNER

This representation of the gas burner embodies only a few gas particles but is still under the influence of the PCA and SCA. The volume to the right of the gas burner represents the combustion chamber at atmospheric conditions. In the figure above, it can be seen that when the gas enters the PCA tube it starts to swirl. The gas moves towards the combustion chamber, but because of the vortex created inside the PCA, the gas is drawn back in towards the gas lance. This backward and forward movement continues three to four times before the gas exits the PCA tube.

The arrow indicates the gas entering the SCA tube while maintaining its swirl. This is due to the vortex created inside the SCA tube. The gas is drawn towards the back of the SCA tube and then exits towards the combustion chamber. This confirms the speculation about the vortex, caused by the tangentially placed air supply nozzles.

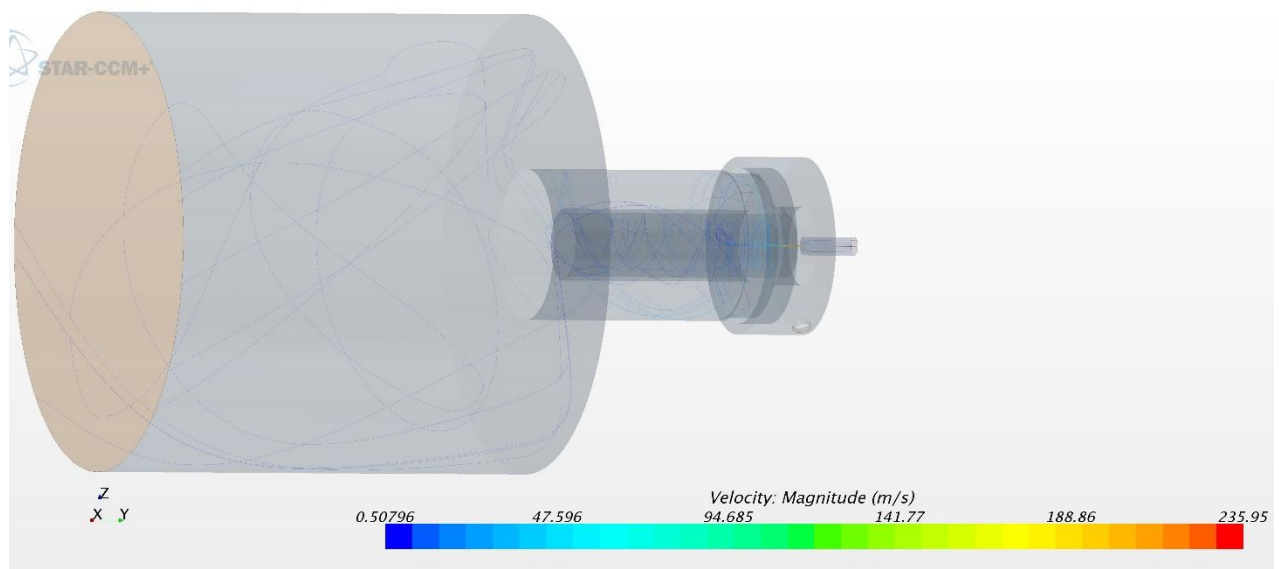


FIGURE 4.8: CFD ANALYSIS (2) - VELOCITY OF GAS PARTICLES INSIDE THE TEST GAS BURNER AND COMBUSTION CHAMBER

Figure 4.9 below indicates the vortices forming in response to the induced swirl.

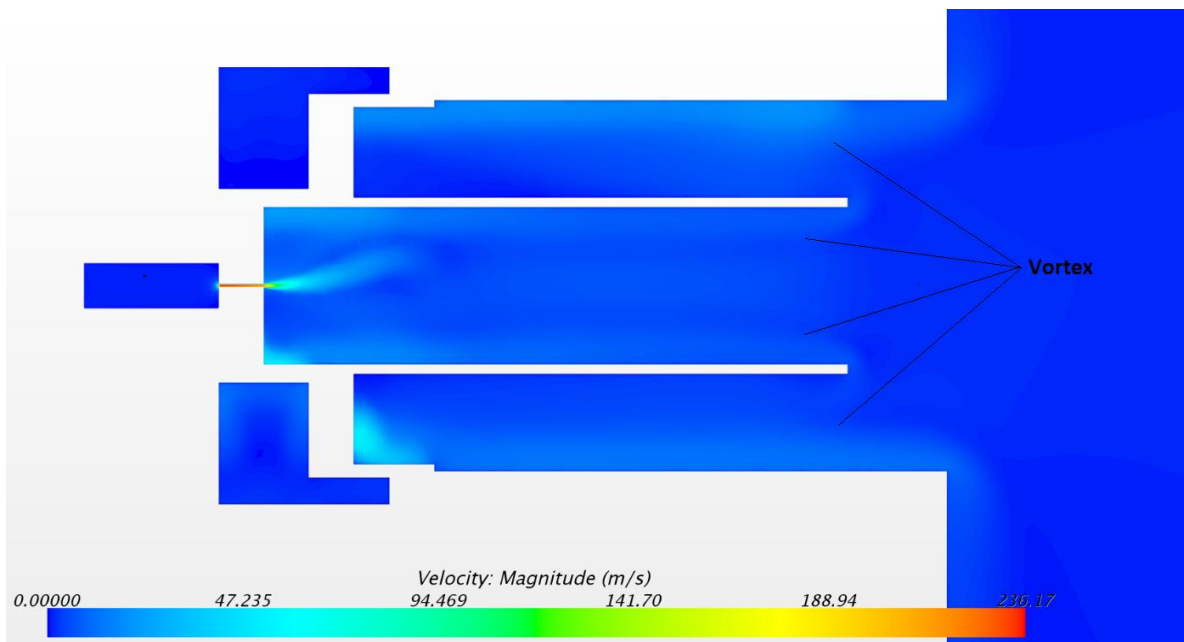


FIGURE 4.9: CFD ANALYSIS (3) - VELOCITY OF GAS PARTICLES INSIDE THE TEST GAS BURNER

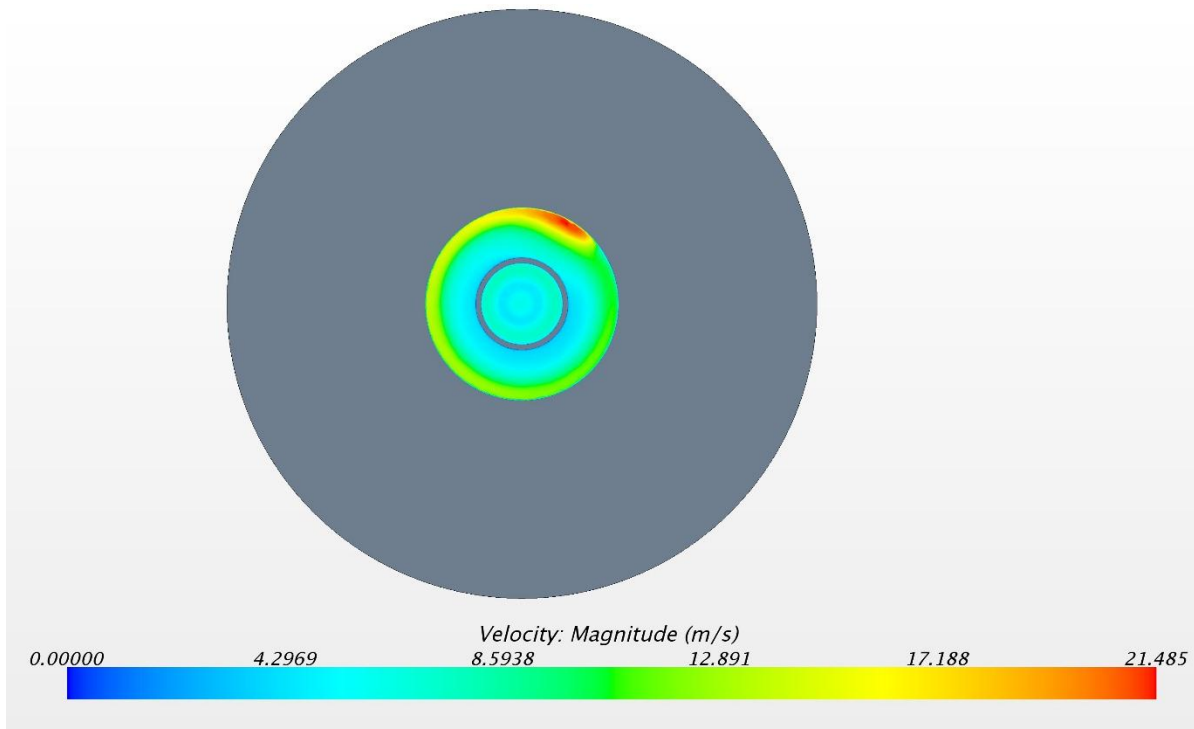


FIGURE 4.10: CFD ANALYSIS (4) – FRONTAL VIEW OF VELOCITY OF GAS PARTICLES INSIDE GAS BURNER AND DISTRIBUTION

Figure 4.10 above is also an indication of the mass distribution inside the gas burner. The red colour indicates the concentration of the gas. This is an indication that the gas is not distributed

homogenously throughout the cross-sectional surface, but is concentrated more on the outer diameter of the SCA tube.

4.1.3 DISCUSSION OF THE TEST GAS BURNER RESULTS

This test gas burner's initial design shows positive and negative results:

- Positives: The fact that a blue flame was obtained indicates good combustion. Good combustion indicates that the correct A:F mixture was supplied to the burner and therefore confirms the previous calculations. The good combustion is also due to proper mixing of air and gas, which shows that the swirl has a large impact. The swirl effect also ensures a short flame length. The flame is flaring out to the outer radius of the chamber, which will cause the entering PF to be injected into the gas flame for ignition.
- Negatives: The velocity through the PCA and SCA supply nozzles (105 m/s and 122 m/s respectively) is too high and creates too intense vortices. These vortices create suction, forcing the gas to burn in, towards the back of the tubes. The gas flame burning back into the tubes creates metal temperature excursions, safety, and reliability problems. The flame is not well distributed throughout the pipe and is seen as a hot spot on the CFD in Figure 4.10.
- From the above, a conclusion can be drawn: in full-scale burners at power stations the fact that adequate swirl rarely occurs is a problem. Because of the dimensional scaling down of this burner, the swirl is too severe.

4.2 CORRECTIVE ACTION

The most prominent deficiency in the operation of this test burner is the severe vortex. A traditional remedy will be implemented and evaluated.

4.2.1 VORTEX BREAKER

On a power station, the mills have classifiers inducing a cyclone effect to remove PF particles too large for the burner. To break the resulting vortex, an inverted cone is installed in a position where it will prevent large particles being sucked up the burner pipe, causing them to fall down through gravity into a return pipe back to the mill. A vortex breaker will thus be installed in the PCA tube mouth and tested.

The vortex breaker is installed in front of the PCA tube, has the same diameter as the PCA tube's inner diameter, but protrudes to render the same circumferential area at the exit. Figure 4.11 shows the vortex breaker used on the gas burner.

Figures 4.12, 4.13 and 4.14 indicate how the gas burner operates with the vortex breaker installed in the PCA tube end, for sub-stoichiometric, stoichiometric and hyper-stoichiometric A:F combustion conditions respectively. The arrows in the figures indicate that the gas flame is not drawn into the PCA tube as much as before. This tends to prove that the vortex breaker prevents air being drawn in because of the vortex formed. The vortex breaker does not affect the combustion of the gas, as indicated by the blue flame. The circle in Figure 4.13 below shows how much the flame is concentrated in one section and not distributed as desired. This refers back to the CFD's "hot spot" shown previously.

However, in Figure 4.14 it can be seen that in the SCA tube, two layers of air are forming. The outer gas moves outwards and the inner gas moves inwards. The inner layer transports the gas towards the inside of the SCA tube and moves out into the outer layer. This is still unsatisfactorily, since the eventual overheating of the SCA tube is visible.

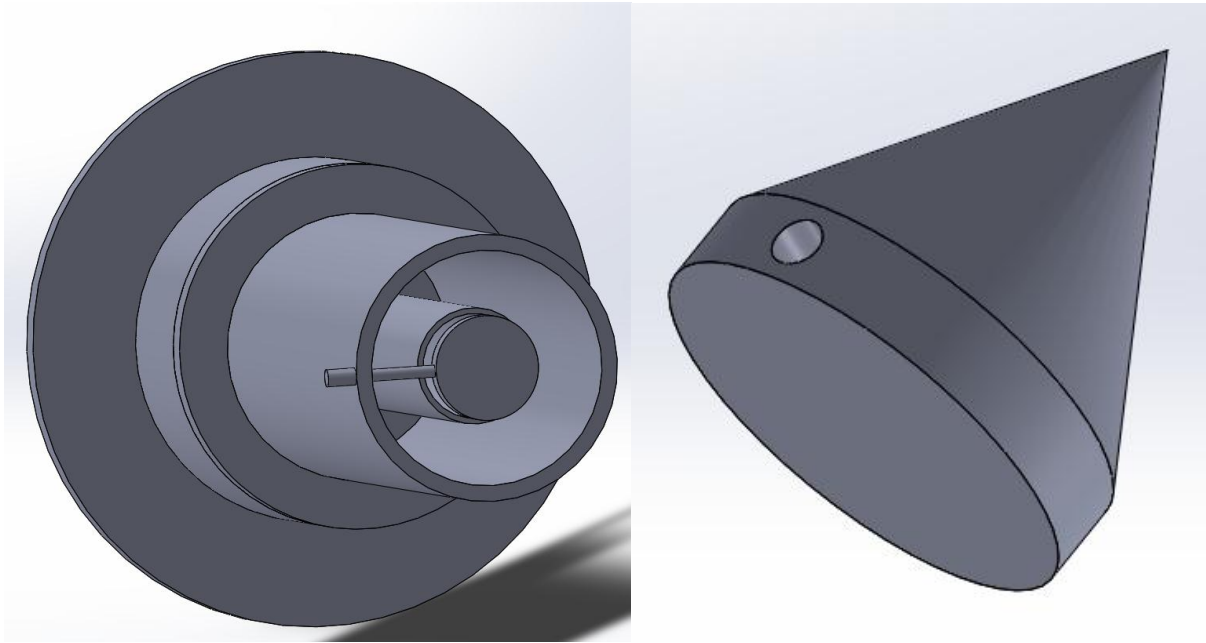


FIGURE 4.11: MODEL OF THE VORTEX BREAKER USED IN TEST GAS BURNER



FIGURE 4.12: STOICHIOMETRIC COMBUSTION CONDITIONS OF GAS BURNER WITH VORTEX BREAKER



FIGURE 4.13: COMBUSTION OF GAS BURNER WITH VORTEX BREAKER - FLAME DISTRIBUTION

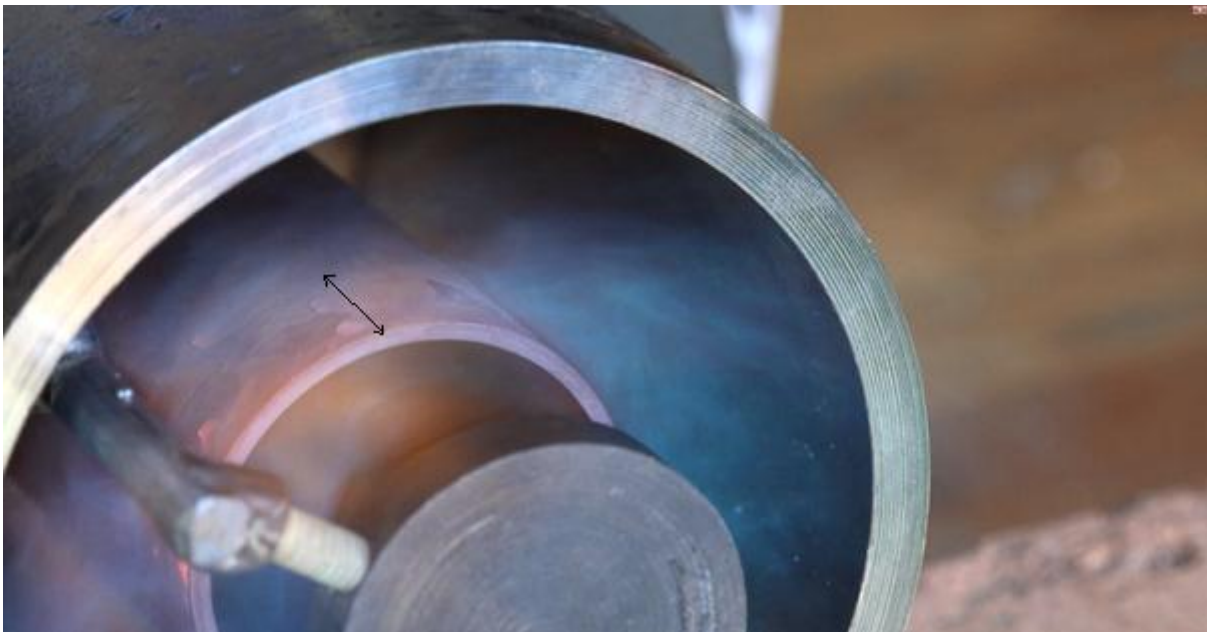


FIGURE 4.14: COMBUSTION OF GAS BURNER WITH VORTEX BREAKER - FLAME DEPTH AND PCA TUBE OVERHEATING

4.2.2 VORTEX BREAKER FOR BOTH PCA AND SCA TUBES

A possible solution can be to use the vortex breaker and diverging tubes to make the aperture as small as possible. This may force the air out, breaking the inner layer. The only layer required must be the air moving outward and therefore no air should be sucked in. The aperture for the PCA, SCA tubes, and vortex breaker is calculated such that the circumferential area remains the same as the annular area, to accommodate a 0.4 m/s velocity for the mixture. The aperture between the PCA and SCA tubes was smaller than the above-mentioned criteria to break the inner layer. To correct the problem of distribution, four tangentially placed air supply holes for the PCA and SCA tubes were installed.

The design is therefore as in Figure 4.15. To eliminate unnecessary cost, time, and manufacturing, a superficial CFD analysis was performed by using the SolidWorks flow simulator.

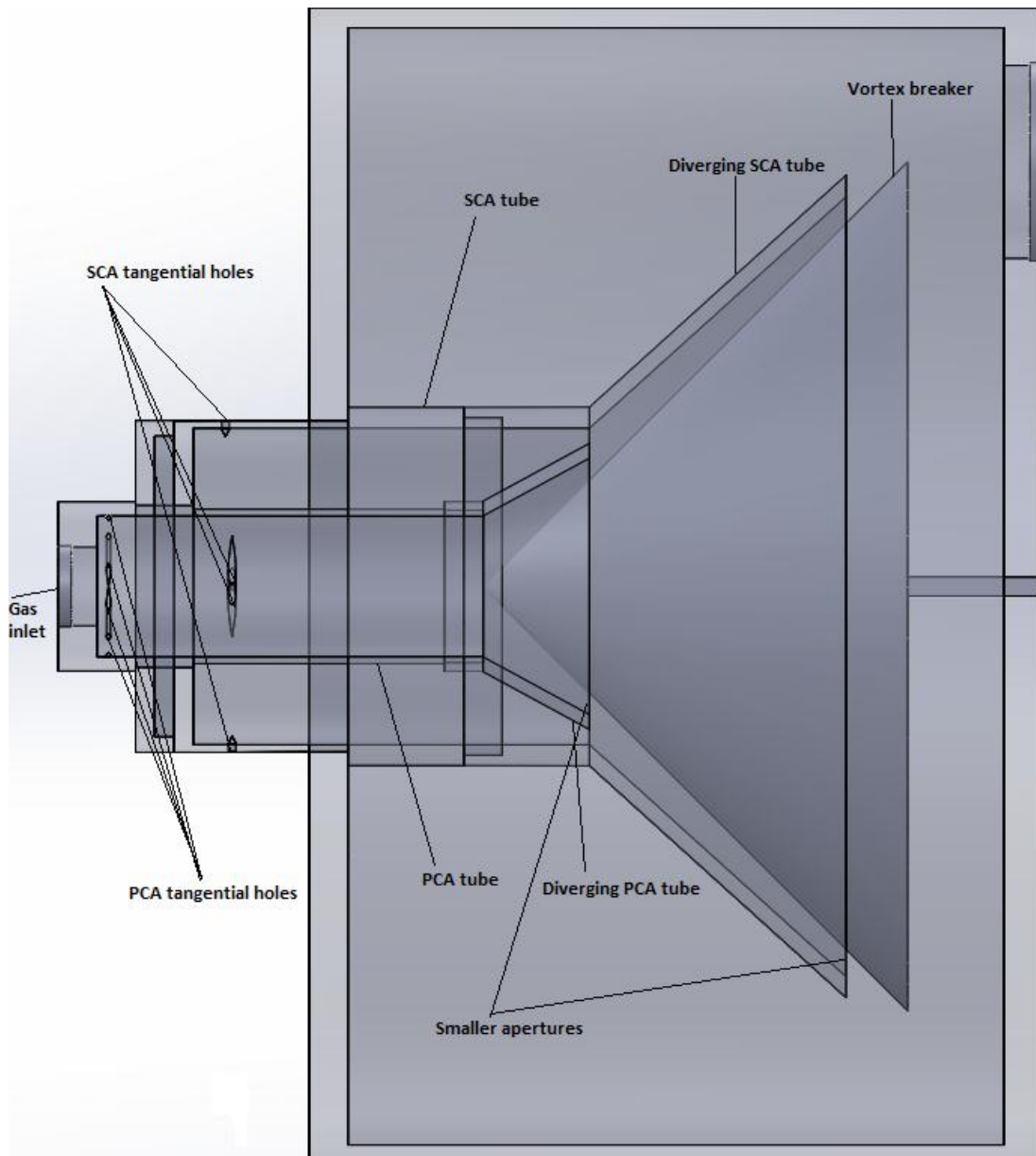


FIGURE 4.15: CAD MODEL OF GAS BURNER WITH LARGE VORTEX BREAKER AND DIVERGING TUBES

4.2.3 SOLIDWORKS CFD ANALYSIS OF DIVERGING TUBES

The CFD analysis of the flow inside the gas burner is discussed systematically to show and explain more accurately what happens.

The boundary conditions now follows:

Inlet conditions:

- Gas: Temperature: 293.2 K
Mass flow: 0.00013 kg/s
Pressure: 85 kPa
Boundary layer: Turbulent
- PCA: Temperature: 343.2 K
Mass flow: 0.000361 kg/s
Pressure: 85 kPa
Boundary layer: Turbulent
- SCA: Temperature: 343.2 K
Mass flow: 0.001671 kg/s
Pressure: 85 kPa
Boundary layer: Turbulent

Outlet conditions:

- Pressure: 85 kPa
- Mass flow:

Turbulence intensity: 2%

Turbulence length: 0.0052 m

Turbulent energy: 1 J/kg

Turbulent dissipation: 1 W/kg

Mesh:

- Type of cell: Rectangular parallelepipeds
- Number of cells: 52106
- Figure 4.16 shows the path taken by one gas particle under the influence of both the flows in the PCA and SCA tubes. It shows how the particle moves backwards and forwards, recirculating in the PCA tube before it exits to the SCA. The small area marked on the figure is designed in the hope of breaking the suction created by the SCA tube and no gas or atmospheric air is drawn in.

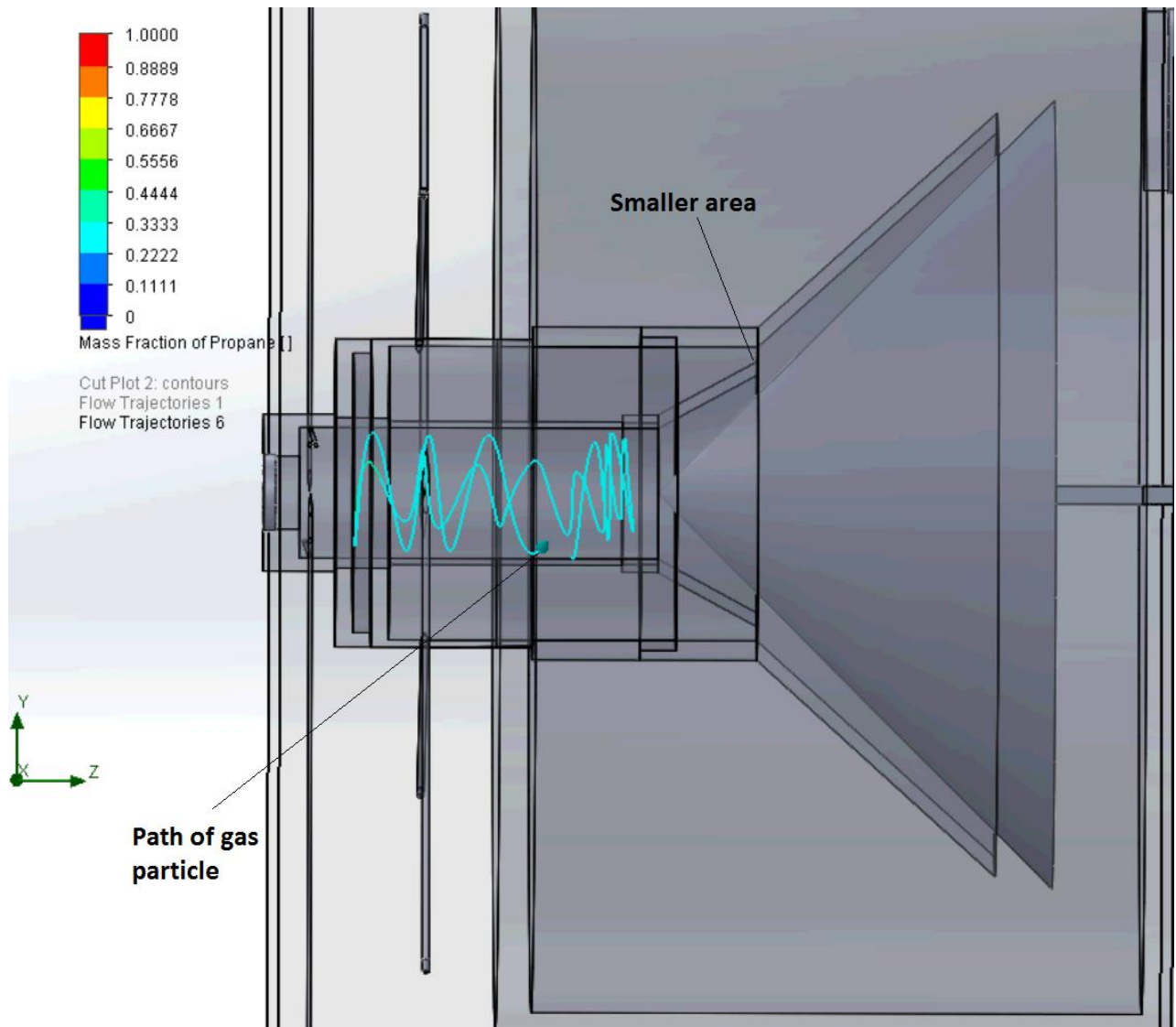


FIGURE 4.16: SOLIDWORKS CFD ANALYSIS (1) - GAS BURNER WITH LARGE VORTEX BREAKER AND DIVERGING TUBES

- After the gas particle has re-circulated five times inside the PCA tube, it exits and enters the SCA tube. It appears as if the particle will exit the SCA tube without being drawn in backwards. However, in Figure 4.17 the particle shows a turning point where it moves backwards again towards the back of the SCA tube.

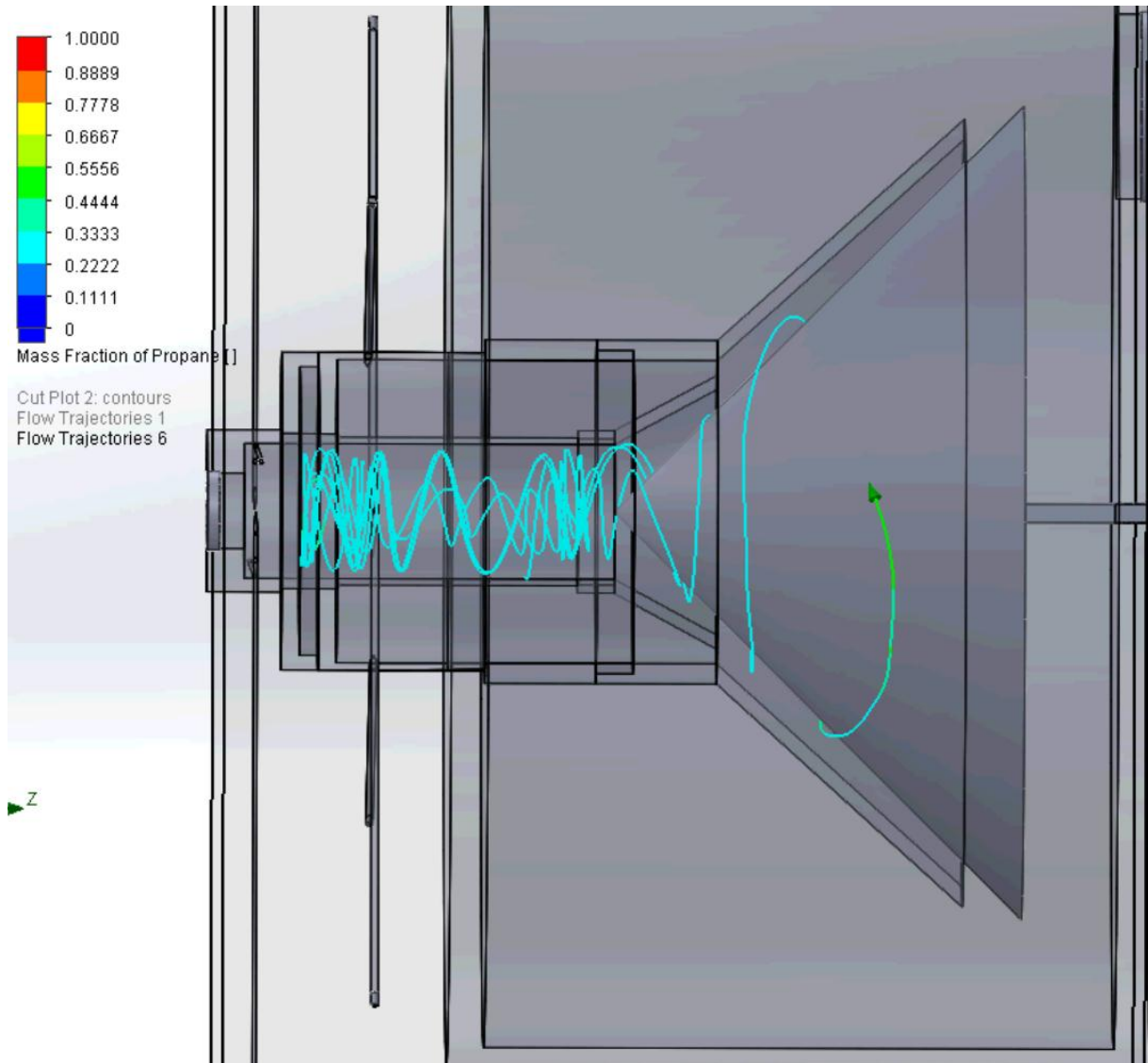


FIGURE 4.17: CFD ANALYSIS (2) - GAS BURNER WITH LARGE VORTEX BREAKER AND DIVERGING TUBES

- The point marked in Figure 4.18 shows the point where the gas particle is drawn back into the SCA tube. This indicates that even though the annular area is very small, the vortex still creates two layers. The gas particle moves backwards on the inner radius of the SCA tube. This means that the gas has the opportunity to combust and draw the flame back into the SCA tube, although the vortex breaker prevents atmospheric air from being drawn inwards.
- Thereafter the gas circulates once inside the SCA tube and then exits the gas burner through the outer layer, as shown in Figure 4.19.

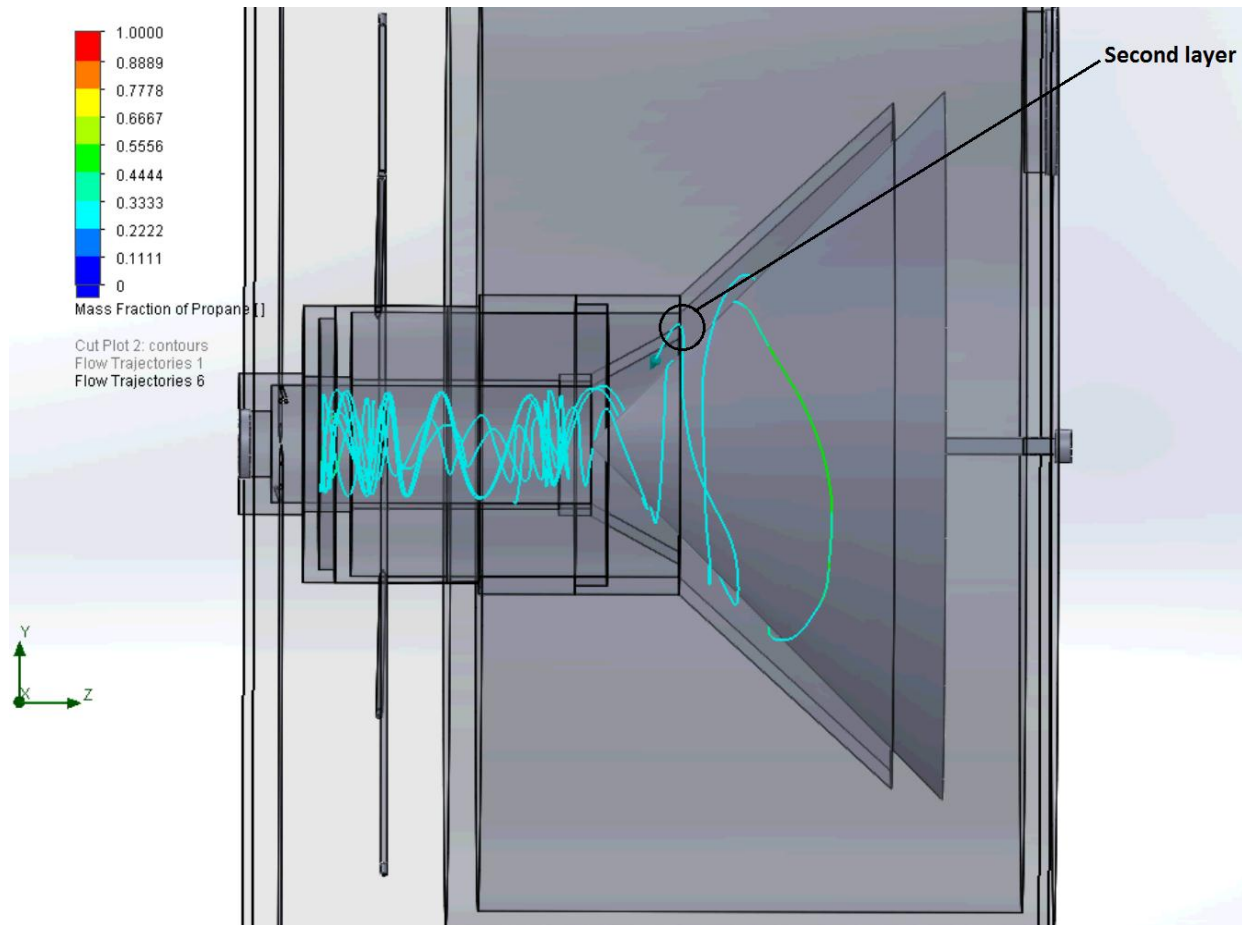


FIGURE 4.18: SOLIDWORKS CFD ANALYSIS (3) - GAS BURNER WITH LARGE VORTEX BREAKER AND DIVERGING TUBES

4.3 CONCLUSION OF GAS BURNER EXPERIMENT WITH TANGENTIAL AIR SUPPLIES

The tangentially placed air supply nozzles induces very good swirl. However, the vortices it creates are too severe to eliminate. The swirl ensures very good mixing, high residence time, high temperatures and a short flame length, but the intense swirl does not provide good distribution and draws the flame inside the tubes, causing metal temperature excursions. The only way to create swirl with tangential holes is to place the different tubes concentrically within each other. The vortex breaker prevents air being drawn in from the atmosphere, but the concentric SCA tube sucks in the gas. The gas is able to burn since it has heat (due to the conduction through the tubes from the flame) and the necessary amount of air.

The burning back of the flame defies the purpose of the gas burner, which will not ignite the coal if most of the gas burns inside the tubes. It also causes metal temperature excursions and therefore causes the metal to melt after a short period.

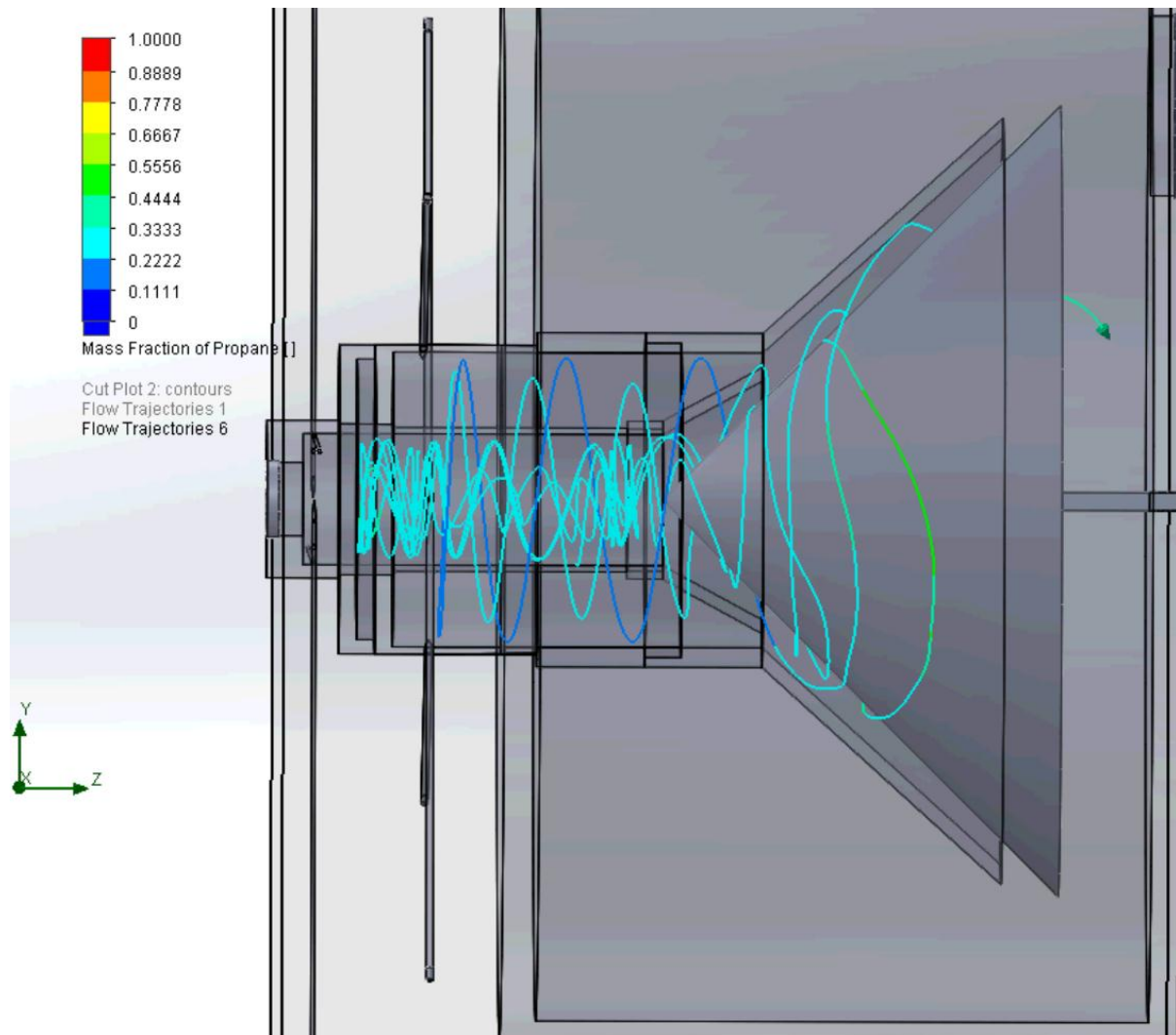


FIGURE 4.19: CFD ANALYSIS (4) - GAS BURNER WITH LARGE VORTEX BREAKER AND DIVERGING TUBES

The gas burner experiment also proves that the rest of the micro coal burner cannot operate in the same manner as the gas burner by using tangentially placed air supply holes with such high entry velocities. This gas burner operates in the same manner as the cyclone furnace in Section 2.3.4. If the coal burner would also operate in this manner, the heat will be tremendous and slag will form. This will interfere with the heat transfer to the water cooling system and it will be impossible to measure the heat accurately. Thus, this burner philosophy will not support a functioning micro burner with the future purpose of conversion into the measurement of NCVp.

5. FINAL DESIGN

Although the tangentially placed air supply nozzles were easier to manufacture, they created other problems that cannot be overcome. In spite of the problems created by the tangentially placed holes, there are still calculations and results that can be applied. The combustion and dimension calculations are still valid and can be used, but the manner in which the air will be introduced into the burner will differ. The swirl and vortex aspects can still be used and will be explained later. To still create swirl for proper mixing, swirl generators must be designed. The swirl generators cannot generate as much swirl as the tangential nozzles, but the swirl angles can be controlled much more easily. The swirl generators can be designed to have multiple jets, thus distribution of the air and fuel in the combustion chamber will be improved. These and other aspects will be determined in the following section.

5.1 PHILOSOPHY

A new philosophy of swirl was devised. It is not currently used on any power stations described in the literature. The swirl generators are rings with holes at certain angles to create swirl. The rings are assembled concentrically inside the tubes in which the swirl must be created. The amount of swirl is calculated to ensure proper mixing, a recirculation zone, and an adequate residence time. The amount of swirl should also comply with the fact that none of the gas, PCA, SCA, PF/PA or SA flow should have **different velocities and move away from one another**. This implies that when the different supplies enter the combustion chamber through its swirl generator, each must have the same **axial velocity component, as well as angular velocity**. This will allow the air, gas and especially the coal to stay in contact with one another for good combustion. Thus, the angles of each swirl generator are dependent upon one another. The functioning of these swirl generators' relative velocities will now be explained:

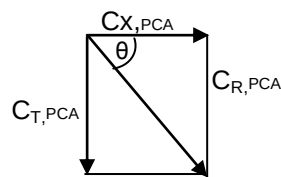


FIGURE 5.1: VELOCITY VECTOR DIAGRAM OF PCA SWIRL GENERATOR

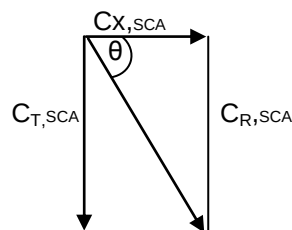


FIGURE 5.2: VELOCITY VECTOR DIAGRAM OF SCA SWIRL GENERATOR

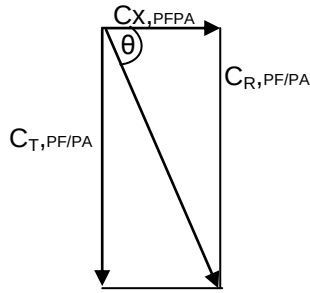


FIGURE 5.3: VELOCITY VECTOR DIAGRAM OF PF/PA SWIRL GENERATOR

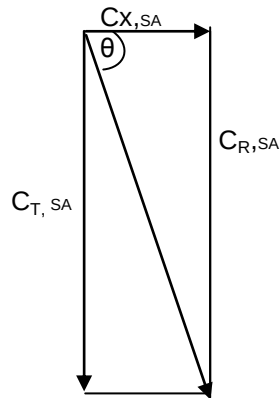


FIGURE 5.4: VELOCITY VECTOR DIAGRAM OF SA SWIRL GENERATOR

where,

C_X = axial velocity

C_T = tangential velocity

C_R = resulting velocity

Θ = swirl angle

The figures above show the relative velocity triangles of the different swirl generators for the PCA, SCA, PF/PA and SA tubes. As mentioned before, all axial velocities must be equal, so therefore:

$$C_{X,PCA} = C_{X,SCA} = C_{X,PF/PA} = C_{X,SA}.$$

Once the angle and axial velocity for one swirl generator have been chosen, its related C_R and C_T values can be calculated. However, to determine the rest of the velocities of the other swirl generators with the same angular velocity, the following relation holds:

$$\omega = \frac{C_{T,PCA}}{r_{PCA}} = \frac{C_{T,SCA}}{r_{SCA}} = \frac{C_{T,PF/PA}}{r_{PF/PA}} = \frac{C_{T,SA}}{r_{SA}}$$

where,

ω = angular velocity

r = pitch radius of the swirl hole.

5.2 GEOMETRY OF THE SWIRL GENERATORS

With this relation, the remaining velocities and angles can be determined. Note: the outlets will be the holes placed on the rings' pitch circle; the inlet holes will have to be placed off the pitch circle.

However, the above-mentioned criteria are dependent upon the geometry of the swirl generator. The geometry of the swirl generator (which will be discussed in more detail later) is limited to the possible size, number of holes and realistically available angles. When the numbers, angles and size of the holes are determined, the C_R velocity can be calculated. This C_R value must correlate with the C_R value determined earlier by means of the relative velocity triangles. Therefore, this is an iterative process. Subsequently, by using the angle, amount, and size of holes for the swirl generator, the geometry was considered. When addressing the geometry, the following was avoided:

- Overlapping of hole inlet and outlet, Figure 5.5:

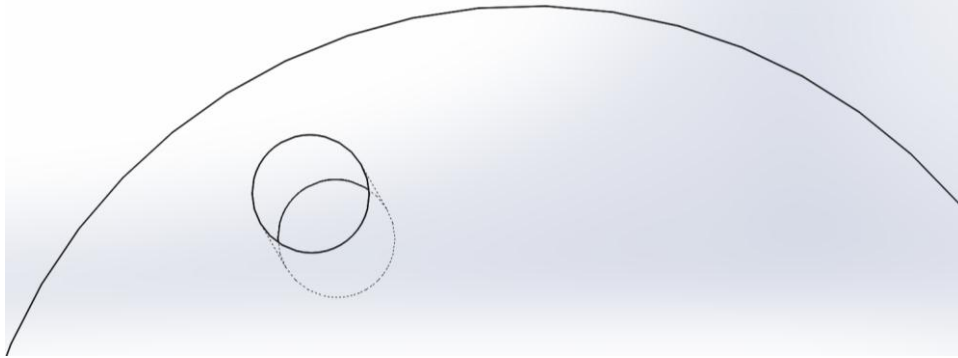


FIGURE 5.5: OVERLAPPING OF INLET AND OUTLET HOLES

Since the holes are at an angle, the inlet and outlet of the hole might overlap, which defies the purpose, because part of the flow might pass through the hole at no angle. This can be solved by increasing the angle, increasing the axial thickness of the swirl generator or reducing the diameter of the hole.

- Overlapping of outlet holes, Figure 5.6:

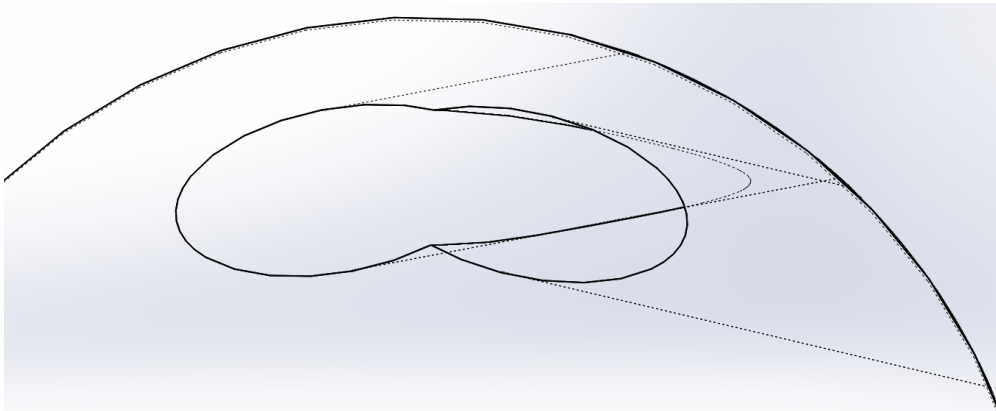


FIGURE 5.6: OVERLAPPING OF OUTLET HOLES

This is caused by outlet holes, which are too close to each other, and overlap because of the elliptic shape the angle creates. This can be solved by reducing the holes' diameters, creating fewer holes or adjusting the angle of the holes.

- Inlet holes crossing the outside diameter Figure 5.7



FIGURE 5.7: INLET HOLES CROSSING THE OUTSIDE DIAMETER

The problem of a hole crossing the outside diameter can be solved by reducing the thickness of the swirl generator or reducing the angle of the hole.

Initially, the maximum number, size and swirl angle of the holes need to be calculated to ensure the axial velocity is as close as possible to 0.4 m/s and that it can be manufactured. The geometric method to ensure the swirl generators apply to these three avoiding factors mentioned above will subsequently be explained:

- Determine the outside and inside diameter of the swirl generator:
 - The applicable swirl generator, for example the SCA, will be inserted between the PCA and SCA tubes. Thus, it has an outside diameter equal to the SCA tube's inside diameter and an inside diameter equal to the outside diameter of the PCA tube.
- Determine the pitch diameter of the outlet hole:
 - The centre of the hole outlet will be placed on the centre line between the outside and inside diameter of the swirl generator.
- Choose a tentative angle and determine the maximum ellipse radius:
 - This is because of the angle of the hole, the inlet and outlet of the hole do not form a circle but an ellipse on the surface. The ellipse has different radii because of its profile. Its maximum radius can be calculated by using the angle of the hole. The minimum radius of the ellipse is equal to the radius of the hole.
- Choose a thickness of the swirl generator tentatively.
- Determine the horizontal distance between the inlet and outlet of the hole in Figure 5.8:

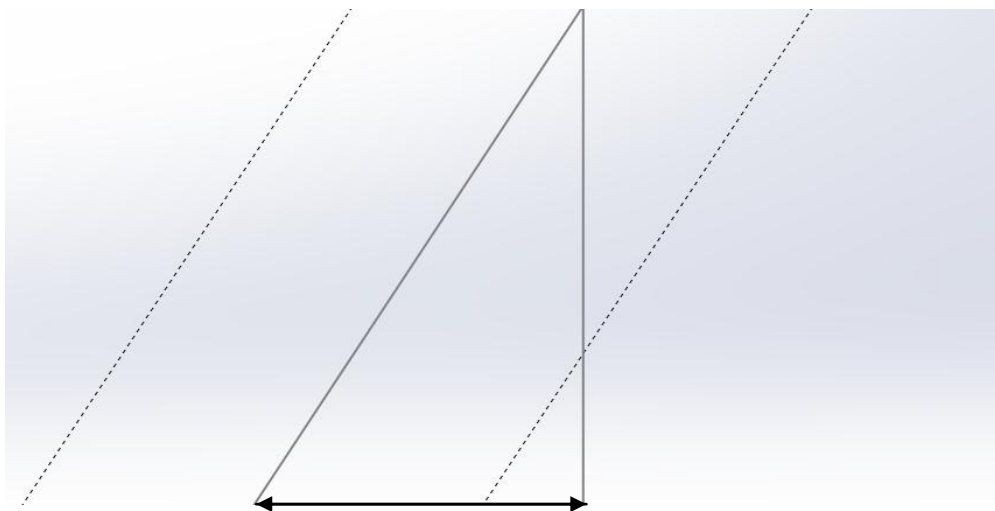


FIGURE 5.8: INLET AND OUTLET HOLE CENTRE DISTANCES

- Determine the pitch diameter of the inlet hole, Figure 5.9:
 - This will be calculated by using the horizontal distance between the inlet and outlet of the hole, as well as the thickness of the swirl generator.

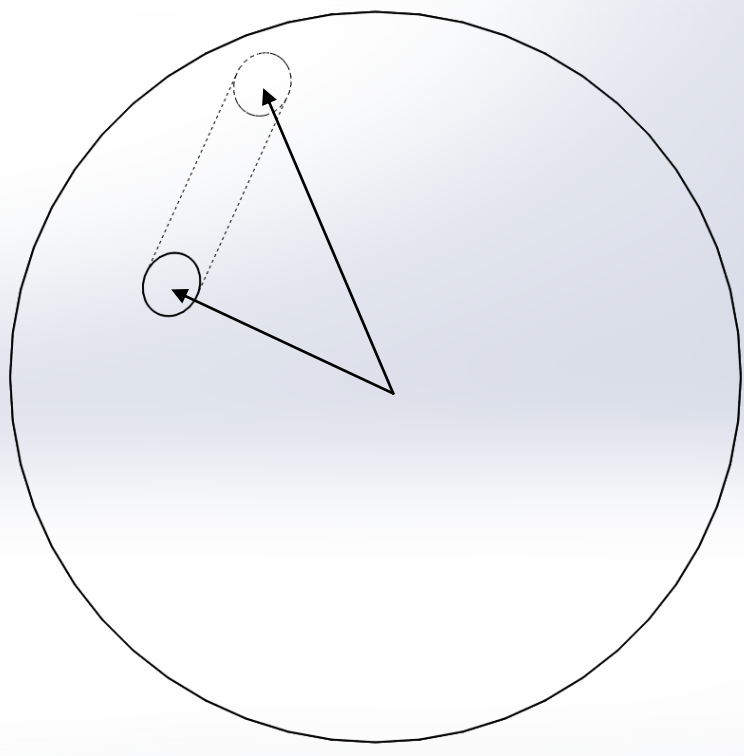


FIGURE 5.9: PITCH DIAMETER OF INLET HOLE

- Determine the arc length between the outlet holes' centre, Figure 5.10.

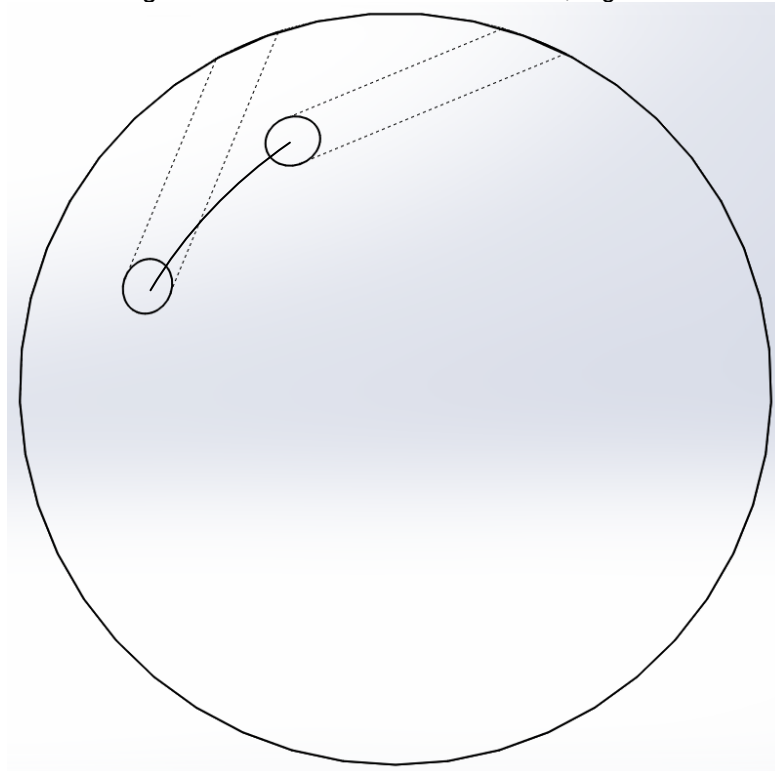


FIGURE 5.10: ARC LENGTH BETWEEN OUTLET HOLES

Determine if the inlet hole crosses the outside diameter of the swirl generator, Figure 5.11. The figure shows the inlet and outlet of the hole, where the inlet is situated closest to the outside diameter of the swirl generator. The procedure to determine if the inlet hole will cross the outside diameter or not is explained:

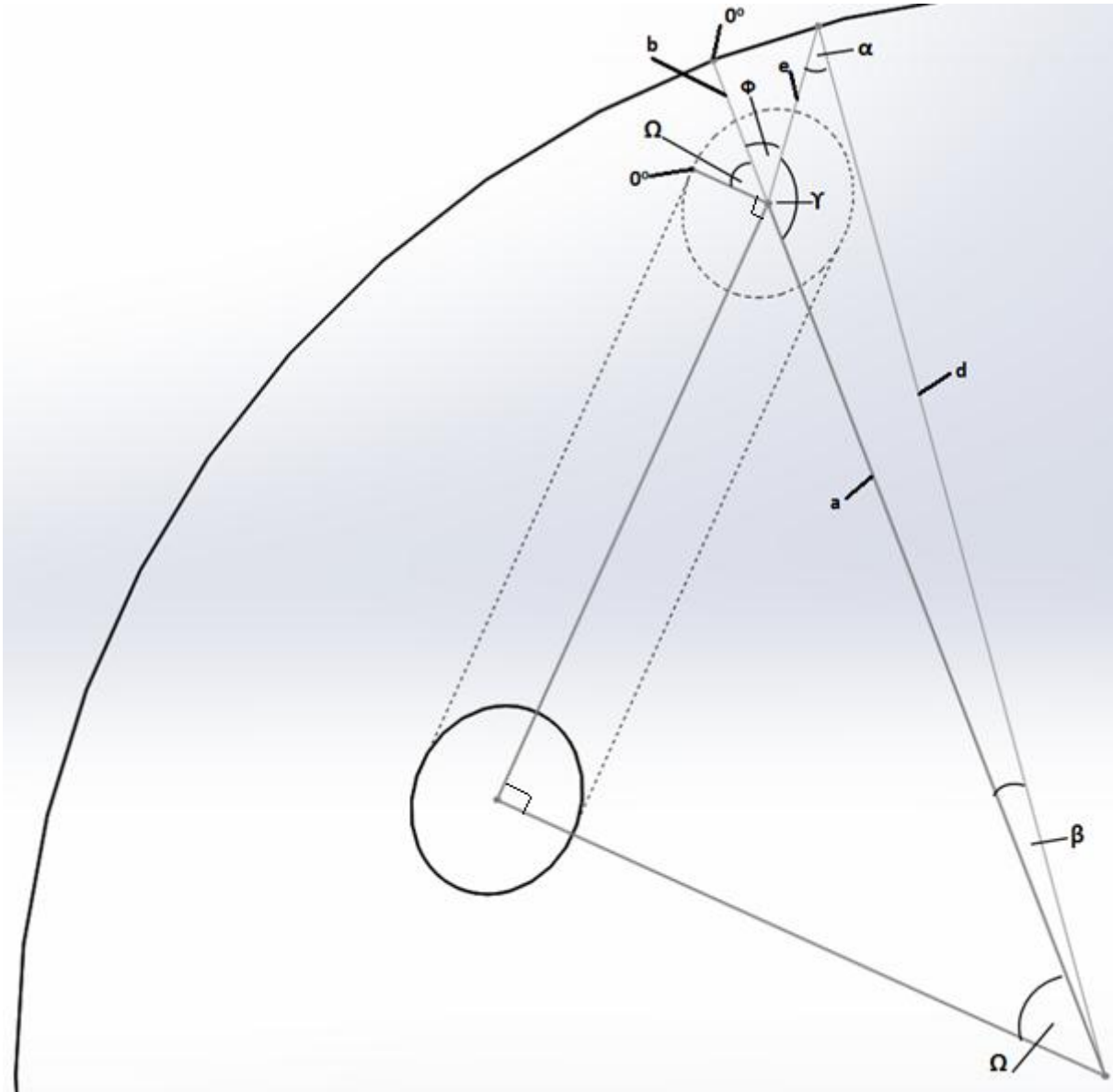


FIGURE 5.11: GEOMETRY METHOD TO DETERMINE IF THE INLET HOLE IS CROSSING THE OUTSIDE DIAMETER

The outlet and inlet of the hole have an ellipse profile on the surface, because of the angle. The radius of the ellipse will differ between a minimum and a maximum. The pitch diameter of the inlet hole defines where it is situated. At this specific point, the distance (e) to the outside diameter can be determined, dependent on the angle (Φ). By using the distance (e) to the outside diameter and the radii of the ellipse, it can be verified if the inlet hole crosses the outside diameter. The geometry shown in the figure above helps in determining this distance and radii. To make the iterative process easier, the distance (e) from the centre of the inlet hole to the outside diameter, and the radii of the ellipse, must be dependent on one another to see at which angles the inlet hole will cross the outside

diameter. The distance line (e) and the radius line will have no angle between them when determining if the inlet crosses the outside diameter. This is done by subtracting the applicable radius from the distance (e). One reference angle 0° , represents the minimum distance (e) since it is situated on the radius line of the swirl generator. The other 0° reference line represents the minimum radius of the ellipse.

The symbols in Figure 5.11 represents the following:

- e: Distance to outside diameter
- 0° : These two lines will act as references to the angles Φ and Ω
- Ω : Angle between the two reference lines.
- Φ : The angle between reference line and e
- d: Equalled to the radius of swirl generator
- a: Pitch radius of inlet hole centre
- b: Subtracting d and a
- r: radius of ellipse
- f: Max. ellipse radius
- g: Min. ellipse radius

To determine the radius of the ellipse the following formula is used:

$$\frac{1}{r^2} = \frac{1}{2} * \left[\left(\frac{1}{g^2} + \frac{1}{f^2} \right) - \left(\frac{1}{g^2} - \frac{1}{f^2} \right) * (\cos 2\Omega) \right].$$

The difference between e and r must be greater than zero to avoid the hole from crossing the outside diameter.

Furthermore, the difference between the maximum ellipse diameter and the distance between the inlet and outlet hole centres determines whether the inlet and outlet will overlap.

The difference in the arc length between the outlet holes and the maximum ellipse diameter determines whether the outlet holes will overlap.

By following this iterative procedure, the maximum number of holes was calculated at their respective angles and was manufactured.

5.3 SWIRL GENERATOR VELOCITIES

5.3.1 MAXIMUM AREA FILLED BY THE HOLES

The maximum area the fluid can flow through has been calculated and the following results have been obtained:

TABLE 5.1: MINIMUM AXIAL VELOCITIES AND PITCH RADIUS FOR MAXIMUM FLOW AREA

$C_{X \text{ PRIM CORE AIR}}$	1.300	[m/s]	$\emptyset_{\text{PRIM CORE AIR}}$	21.000	[mm]
$C_{X \text{ SEC CORE AIR}}$	1.300	[m/s]	$\emptyset_{\text{SEC CORE AIR}}$	63.000	[mm]
$C_{X \text{ PF/PA}}$	1.300	[m/s]	$\emptyset_{\text{PF/PA}}$	114.500	[mm]
$C_{X \text{ SA}}$	1.300	[m/s]	$\emptyset_{\text{SA}}$	216.000	[mm]

TABLE 5.2: MAXIMUM VELOCITIES FOR FLOW AREA CALCULATED FROM VELOCITY TRIANGLES

TUBE	ω	N	C_T	ANGLE Θ		C_R
	rad/s	r.p.m.	[m/s]	$[\ ^\circ]$	[Radians]	[m/s]
PRIM CORE	15.407	147.123	0.162	7.093	0.124	1.310

SEC CORE	15.407	147.123	0.485	20.471	0.357	1.388
PF/PA	15.407	147.123	0.882	34.156	0.596	1.571
SA	15.407	147.123	1.664	52.000	0.908	2.112

TABLE 5.3: SWIRL HOLES AND RESULTANT VELOCITIES FOR MAXIMUM FLOW AREA CALCULATED FROM GEOMETRY

SWIRL HOLES	swirl holes	n	TOTAL SWIRLER	VOL FLOW	C _R
Ø [mm]	r [mm]	[-]	AREA [mm ²]	[l/s]	[m/s]
6.800	3.4	8	290.534	0.385	1.325
11.000	5.5	14	1330.464	1.781	1.339
13.500	6.75	17	2433.360	3.837	1.577
28.000	14	14	8620.530	18.233	2.115

TABLE 5.4: MAXIMUM SWIRL AREA GEOMETRY FOR PCA AND SCA

<u>Primary core air</u>		<u>Secondary core air</u>	
OR	18.500	OR	41.000
IR	2.500	IR	22.000
Pitch radius of outlet holes	10.500	Pitch radius of outlet holes	31.500
Max ellipse radius (f)	3.426	Max ellipse radius (f)	5.871
Thickness	72.000	Thickness	35.000
Inlet, outlet holes distance	8.960	Inlet, outlet holes distance	13.066
Possible distance between inlet and outlet	15.232	Possible distance between inlet and outlet	26.244
Arc length between outlet holes	8.247	Arc length between outlet holes	14.137
Pitch radius of inlet holes	13.803	Pitch radius of inlet holes	34.102
Check: inlet crosses OD	1.282	Check: inlet crosses OD	1.080
Check: inlet, outlet overlap	2.107	Check: inlet, outlet overlap	1.325
Check: outlet holes overlap	1.394	Check: outlet holes overlap	2.396
Check: holes fit inside annular area	9.200	Check: holes fit inside annular area	8.000

TABLE 5.5: MAXIMUM SWIRL AREA GEOMETRY FOR PF/PA AND SA

<u>PF/PA</u>		<u>Secondary air</u>	
OR	68.000	OR	141.000
IR	46.500	IR	75.000
Pitch radius of outlet holes	57.250	Pitch radius of outlet holes	108.000
Max ellipse radius (f)	8.157	Max ellipse radius (f)	22.740
Thickness	25.000	Thickness	37.000
Inlet, outlet holes distance	16.962	Inlet, outlet holes distance	47.358
Possible distance between inlet and outlet	36.694	Possible distance between inlet and outlet	90.648
Arc length between outlet holes	21.160	Arc length between outlet holes	48.470
Pitch radius of inlet holes	59.710	Pitch radius of inlet holes	117.927
Check: inlet crosses OD	0.235	Check: inlet crosses OD	1.421
Check: inlet, outlet overlap	0.648	Check: inlet, outlet overlap	1.878
Check: outlet holes overlap	4.846	Check: outlet holes overlap	2.991
Check: holes fit inside annular area	8.000	Check: holes fit inside annular area	38.000

The tables above show that the lowest possible axial velocity that could be achieved through the holes of swirl generators was 1.3 m/s. They also show that the angular velocities of the swirl generators are the same. This means that all the fluids will move together when exiting the holes.

The C_R values, when calculated in the two different ways, are almost equal, only varying on the second decimal. The geometry calculations also state that the swirl generators can be manufactured, since there are no negative values in the “check” tables.

5.3.2 SWIRL GENERATORS FOR LOWEST AXIAL VELOCITY

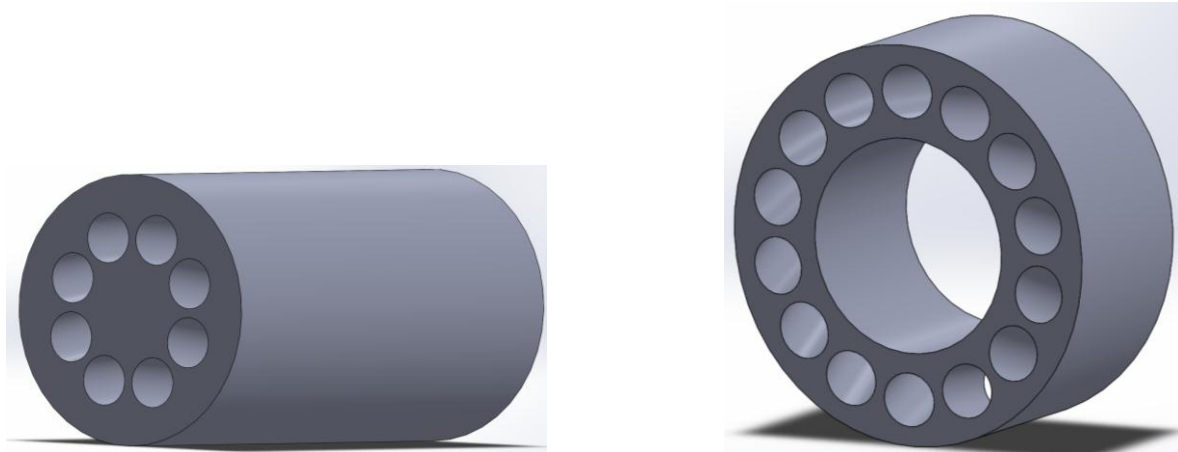


FIGURE 5.12: PCA AND SCA SWIRL GENERATORS FOR LOWEST AXIAL VELOCITY

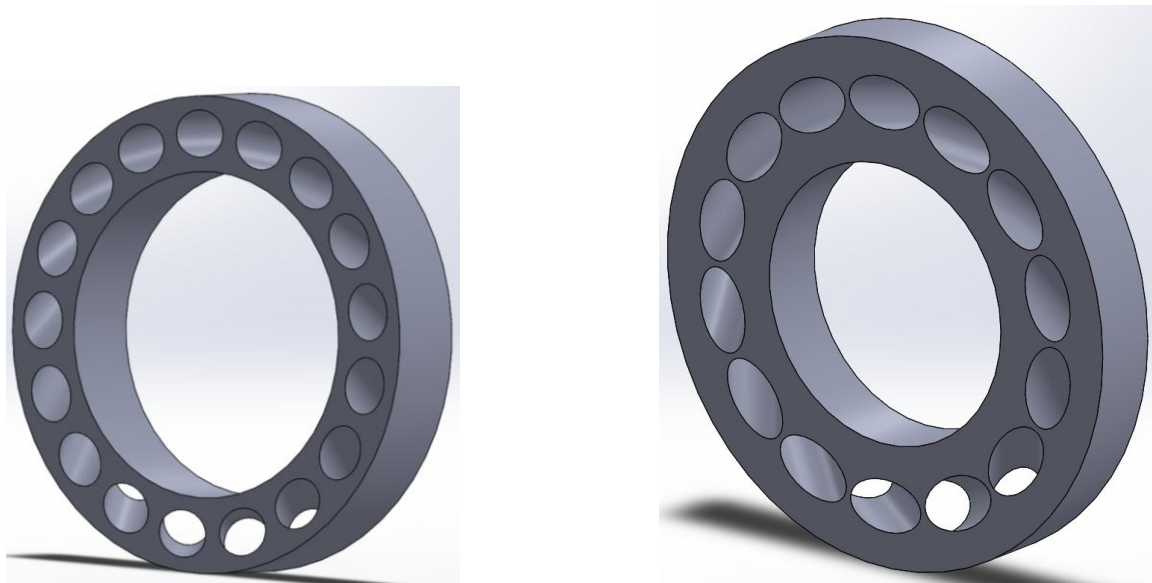


FIGURE 5.13: PF/PA AND SA SWIRL GENERATORS FOR LOWEST AXIAL VELOCITY

The figures above show the swirl angles designed to accommodate the maximum possible flow area where all the swirl generators are dependent upon one another. The swirl generators will be positioned on the same plane in front of the PCA, SCA, PF/PA, and SA tubes. This will prevent air, gas, and PF from being drawn into the tubes by the recirculation zone because there is only an outward flow out of the swirl generators that enters the combustion chamber to control the angle, angular and axial velocity. The swirl generators will be simulated to see what the effects are on the flow. The set-up of the micro coal burner that will be used in the CFD model will look as follows, Figure 5.14:

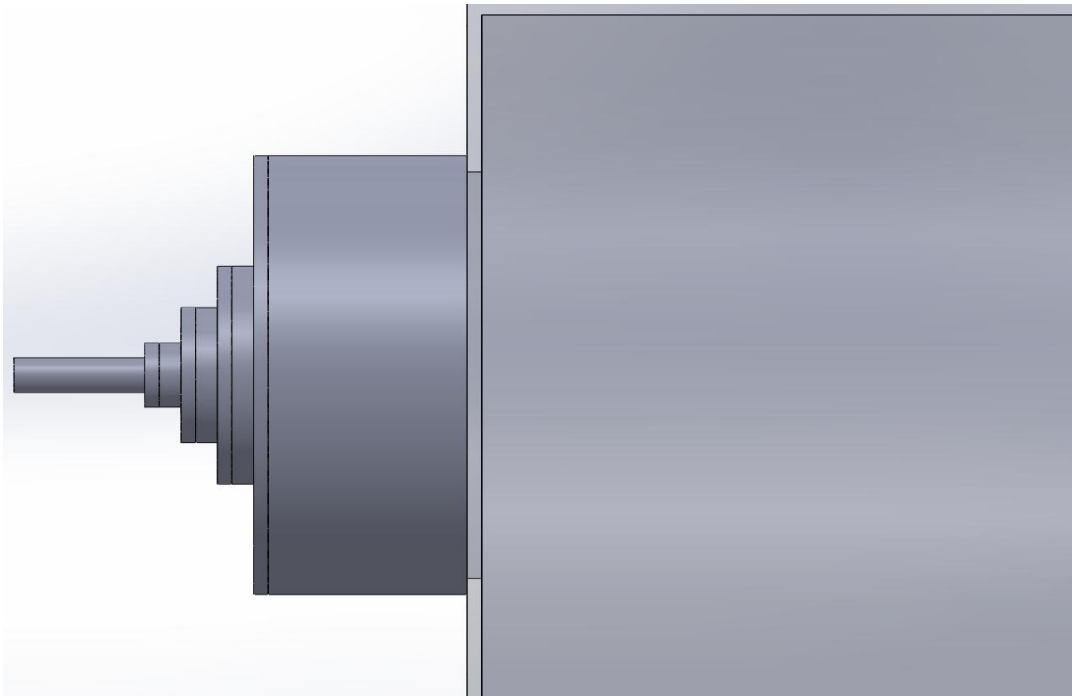


FIGURE 5.14: MICRO COAL BURNER ASSEMBLY (1)

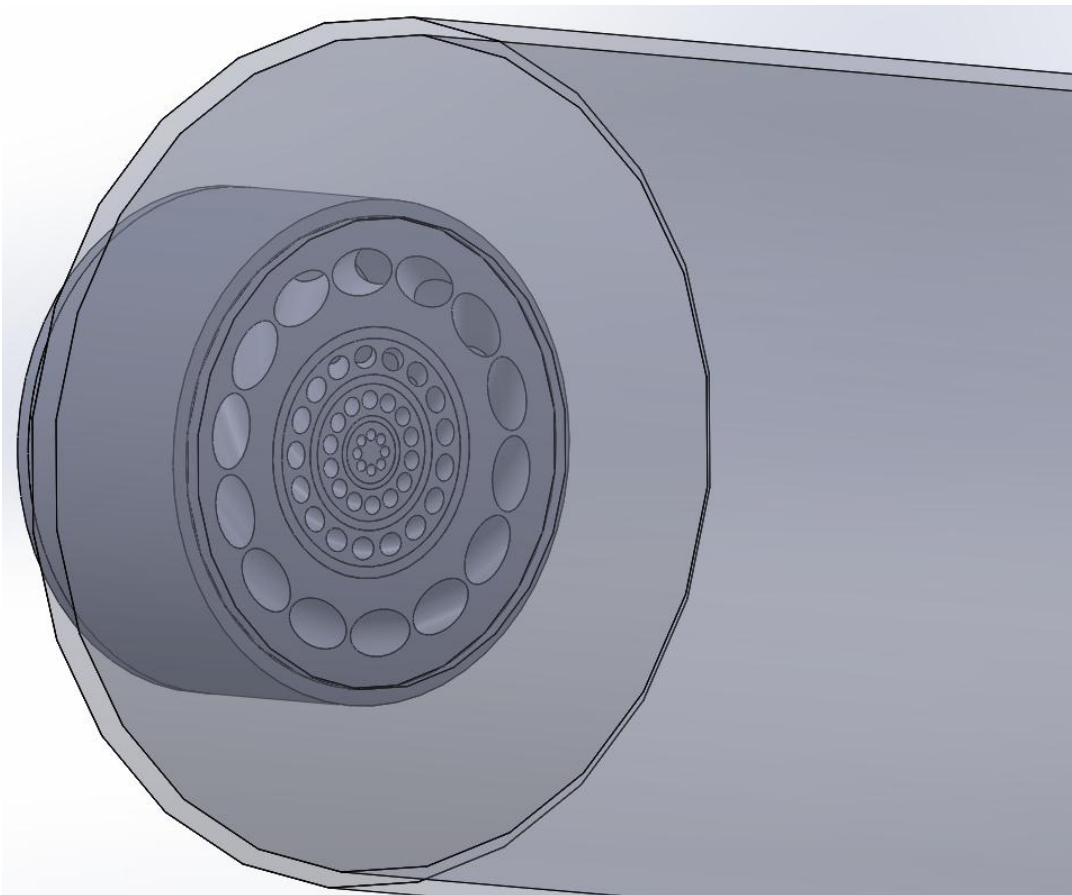


FIGURE 5.15: MICRO COAL BURNER ASSEMBLY (2)

The gas will mix with the PCA before it enters the swirl generator.

In the SolidWorks simulation, the applicable mass flows and temperatures were applied to the different tubes. The following boundary conditions remain the same for the different flow simulations done of the different swirl generators designs due to the simplistic purpose of the CFD:

Inlet conditions:

- Gas: Temperature: 293.2 K
Mass flow: 0.00013 kg/s
Pressure: 85 kPa
Boundary layer: Turbulent
- PCA: Temperature: 343.2 K
Mass flow: 0.000361 kg/s
Pressure: 85 kPa
Boundary layer: Turbulent
- SCA: Temperature: 343.2 K
Mass flow: 0.001671 kg/s
Pressure: 85 kPa
Boundary layer: Turbulent
- PF/PA: Temperature: 343.2 K
Mass flow: 0.003355 kg/s
Pressure: 85 kPa
Boundary layer: Turbulent
- SA: Temperature: 548.2 K
Mass flow: 0.010644 kg/s
Pressure: 85 kPa
Boundary layer: Turbulent

Outlet conditions:

- Pressure: 85 kPa
- Mass flow: 0.016117 kg/s

Turbulence intensity: 2%

Turbulence length: 0.0052 m

Turbulent energy: 1 J/kg

Turbulent dissipation: 1 W/kg

Mesh:

- Type of cells: Rectangular parallelepipeds
- Number of cells: 215016

At first, the mass flows were inserted as if the distribution was 100% throughout the tube. The outlet pressure of the combustion chamber was set to atmospheric pressure. The combustion chamber was set to 4 m to assure the flow had settled and only moved out of the combustion chamber.

The following criteria need to be adhered to if the swirl generators are to be effective:

- The fluids need to mix properly.
- A vortex or recirculation zone must be created in the combustion chamber.
 - A vortex ensures better mixing, higher residence time and draws the fuel and air to the centreline of the combustion chamber. If further combustion takes place in the centre, the flame's contact with the wall will reduce which will cause less heat loss and wall damage.

The flow simulation does not account for combustion or two-phase flow. It simulates the fluids as they enter the combustion chamber at constant temperature. The following simulation model shows the flow caused by the new swirl generators with maximum flow area:

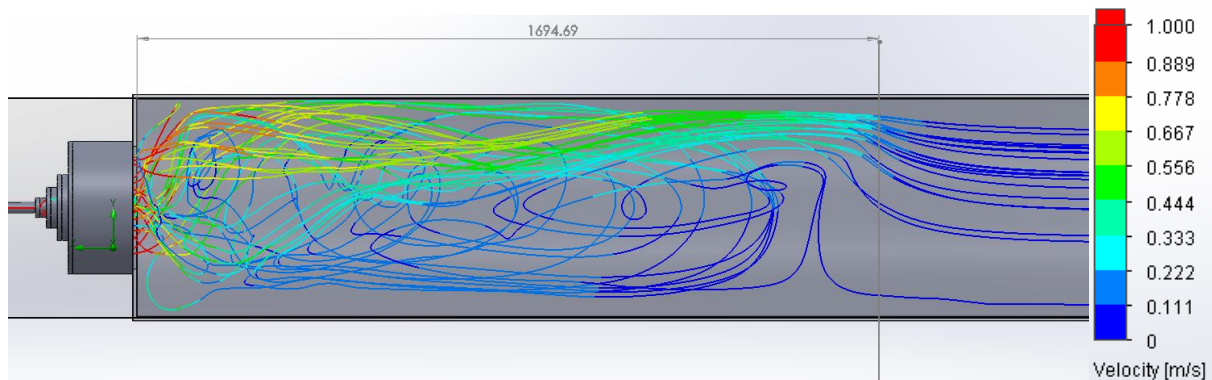


FIGURE 5.16: SOLIDWORKS CFD ANALYSIS (1) - FLOW FOR MICRO COAL BURNER WITH MINIMUM VELOCITY SWIRL GENERATORS

Figure 5.16 shows the flow of the fluids inside the burner for 20 particles of gas, PCA, SCA, PFPA and SA each. There are turbulent flow and recirculation on the side plane instead of the front plane where the swirl generators were intended to create the swirl. The flow seems to be concentrated in the upper part of the combustion chamber. The distance it takes for the flow to settle is almost 1.7 m.

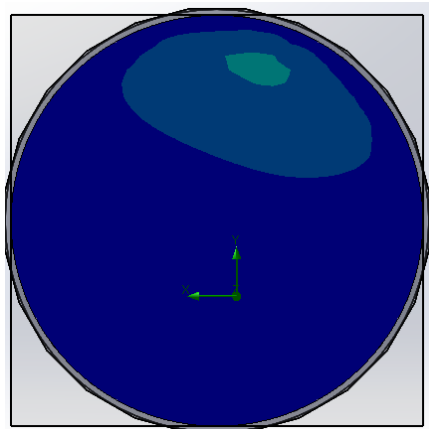


FIGURE 5.17: SOLIDWORKS CFD ANALYSIS (2) - VELOCITY DISTRIBUTION OF MICRO COAL BURNER WITH MINIMUM VELOCITY SWIRL GENERATORS, FRONTAL VIEW

Figure 5.17 shows the distribution of the fluids inside the combustion chamber at a plane 1.7m downstream. The current set of swirl generators does not meet the necessary requirements because of improper mixing, the recirculation zone being in the wrong plane and inadequate distance for the flow to stabilise. Thus, adjustments were required.

5.4 FINAL DESIGN OF SWIRL GENERATORS

After several iterations, it was discovered that the swirl generators for an axial velocity of 1.3 m/s did not create a vortex or ensure proper mixing because the axial velocity of 1.3 m/s was too low. When the flow exits the swirl generators at an axial velocity of 1.3 m/s, it slows down after a certain time inside the combustion chamber (because of the increase in area) to 0.15 m/s. This is because the combustion chamber was designed to force the flow to move at 0.4 m/s after combustion since the

volume flow increase. The fact that it slows down so much suggests that the flow cannot travel to the side of the circular combustion chamber and start a circular motion.

Therefore, swirl generators were designed to create higher axial velocity so that the flow will not slow down as much and will still travel to the side of the combustion chamber to create swirl. The results obtained were as follows:

TABLE 5.6: AXIAL VELOCITY AND PITCH RADIUS FOR FINAL DESIGN FLOW AREA OF SWIRL GENERATORS

$C_{X \text{ PRIM CORE AIR}}$	3.000	[m/s]	$\varnothing_{\text{PRIM CORE AIR}}$	21.000	[mm]
$C_{X \text{ SEC CORE AIR}}$	3.000	[m/s]	$\varnothing_{\text{SEC CORE AIR}}$	63.000	[mm]
$C_{X \text{ PF/PA}}$	3.000	[m/s]	$\varnothing_{\text{PF/PA}}$	114.500	[mm]
$C_{X \text{ SA}}$	3.000	[m/s]	$\varnothing_{\text{SA}}$	216.000	[mm]

TABLE 5.7: AXIAL VELOCITIES FOR FINAL DESIGN FLOW AREA CALCULATED FROM VELOCITY TRIANGLES

TUBE	ω	N	C_T	ANGLE θ		C_R
	rad/s	r.p.m.	[m/s]	[$^\circ$]	[Radians]	[m/s]
PRIM CORE	48.112	459.440	0.505	9.559	0.167	3.042
SEC CORE	48.112	459.440	1.516	26.802	0.468	3.361
PF/PA	48.112	459.440	2.754	42.556	0.743	4.073
SA	48.112	459.440	5.196	60.000	1.047	6.000

TABLE 5.8: SWIRL HOLES AND RESULTANT VELOCITIES FOR FINAL DESIGN FLOW AREA CALCULATED FROM GEOMETRY

SWIRL HOLES	swirl holes	n	TOTAL SWIRLER	VOL FLOW	C_R
$\varnothing$ [mm]	r [mm]	[-]	AREA [mm ²]	[l/s]	[m/s]
4.500	2.25	8	127.235	0.385	3.025
7.200	3.6	13	529.296	1.781	3.365
10.000	5	12	942.478	3.837	4.071
20.700	10.35	9	3028.817	18.233	6.020

TABLE 5.9: FINAL DESIGN SWIRL AREA GEOMETRY FOR PCA AND SCA

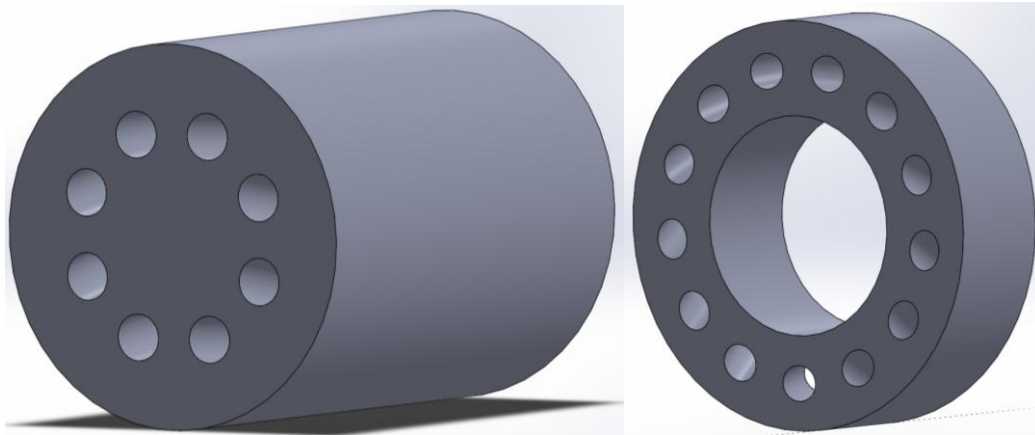
Primary core air		Secondary core air	
OR	18.500	OR	41.000
IR	2.500	IR	22.000
Pitch radius of outlet holes	10.500	Pitch radius of outlet holes	31.500
Max ellipse radius (f)	2.282	Max ellipse radius (f)	4.033
Thickness	50.000	Thickness	25.000
Inlet, outlet holes distance	8.420	Inlet, outlet holes distance	12.630
Possible distance between inlet and outlet	15.232	Possible distance between inlet and outlet	26.244
Arc length between outlet holes	8.247	Arc length between outlet holes	15.225

Pitch radius of inlet holes	13.459	Pitch radius of inlet holes	33.938
Check: inlet crosses OD	2.772	Check: inlet crosses OD	3.090
Check: inlet, outlet overlap	3.856	Check: inlet, outlet overlap	4.563
Check: outlet holes overlap	3.683	Check: outlet holes overlap	7.158
Check: holes fit inside annular area	11.500	Check: holes fit inside annular area	11.800

TABLE 5.10: MAXIMUM SWIRL AREA GEOMETRY FOR PF/PA AND SA

<u>PF/PA</u>		<u>Secondary air</u>	
OR	68.000	OR	141.000
IR	46.500	IR	75.000
Pitch radius of outlet holes	57.250	Pitch radius of outlet holes	108.000
Max ellipse radius (f)	6.788	Max ellipse radius (f)	20.700
Thickness	20.000	Thickness	30.000
Inlet, outlet holes distance	18.363	Inlet, outlet holes distance	51.961
Possible distance between inlet and outlet	36.694	Possible distance between inlet and outlet	90.648
Arc length between outlet holes	29.976	Arc length between outlet holes	75.398
Pitch radius of inlet holes	60.123	Pitch radius of inlet holes	119.850
Check: inlet crosses OD	1.239	Check: inlet crosses OD	1.883
Check: inlet, outlet overlap	4.787	Check: inlet, outlet overlap	10.561
Check: outlet holes overlap	16.400	Check: outlet holes overlap	33.998
Check: holes fit inside annular area	11.500	Check: holes fit inside annular area	45.300

The tables above show that an axial velocity of 3 m/s is to be chosen as optimal value for the swirl generators. As previously, all the swirl generators' axial and angular velocities are equal. The C_R values also correlate well.

**FIGURE 5.18: FINAL SWIRL GENERATORS - PCA AND SCA**

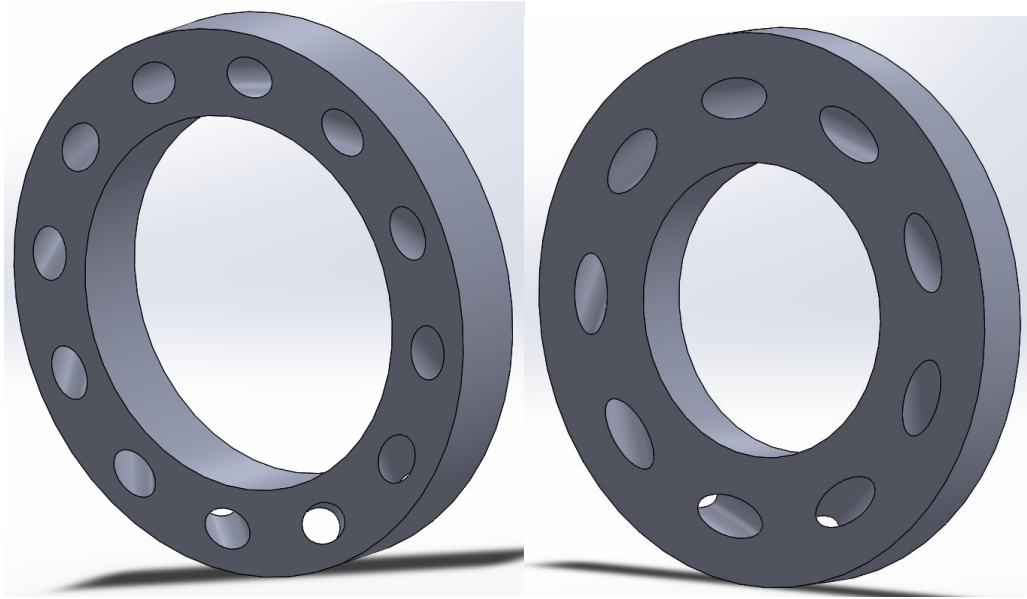


FIGURE 5.19: FINAL SWIRL GENERATORS – PF/PA AND SA

A SolidWorks CFD simulation was done with these swirl generators and the results are given below:

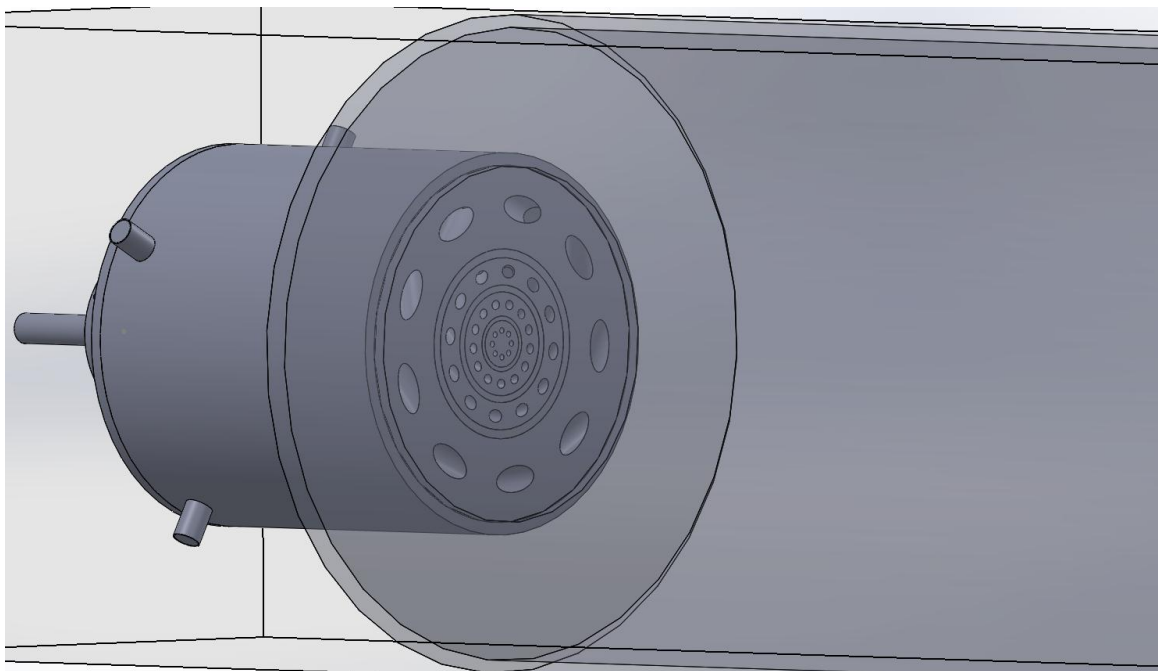


FIGURE 5.20: FINAL COAL BURNER SET-UP (1)



FIGURE 5.21: FINAL COAL BURNER SET-UP (2)

This setup simulates the means by which the actual burner will operate. The difference between this setup and the previous one is the swirl generator angles, the four banjo inlets for each tube (instead of volute/tangential), the SCA tube's increased outer diameter is and the diverging exit of the combustion chamber.

These changes are explained as follows:

- The number and size of holes for the different swirl generators were changed. As before, the swirl generators is placed in front of the tubes in the same plane. The four banjo inlets for the tubes and their size experimentally proved to provide the best distribution inside the tube before it enters the swirl generators. This is very important because if the distribution is poor, more fluid will move through certain holes and less through others. The velocity will not be equal through the holes, defying the design criteria, and influencing the mixing capabilities and recirculation zone.
- Because of PF settling initiated by too low velocities, the annular area in which the PF/PA flows was calculated to a minimum and made equal to the total area of the PF/PA swirl generator holes.
- The converging end of the combustion chamber was implemented to account for the hole through which the flue gas will exit. If only a hole is situated at the end of the combustion chamber, it creates recirculation of flow at the exit, on the side plane, because not all of the flow can exit simultaneously. When implementing the converging end, all the flue gas exits and does not recirculate on the side plane. The combustion chamber is also shorter and is taken to 2 m.

The arrows in Figure 5.22 show the distribution through the swirl generators' holes caused by only one banjo inlet per tube.

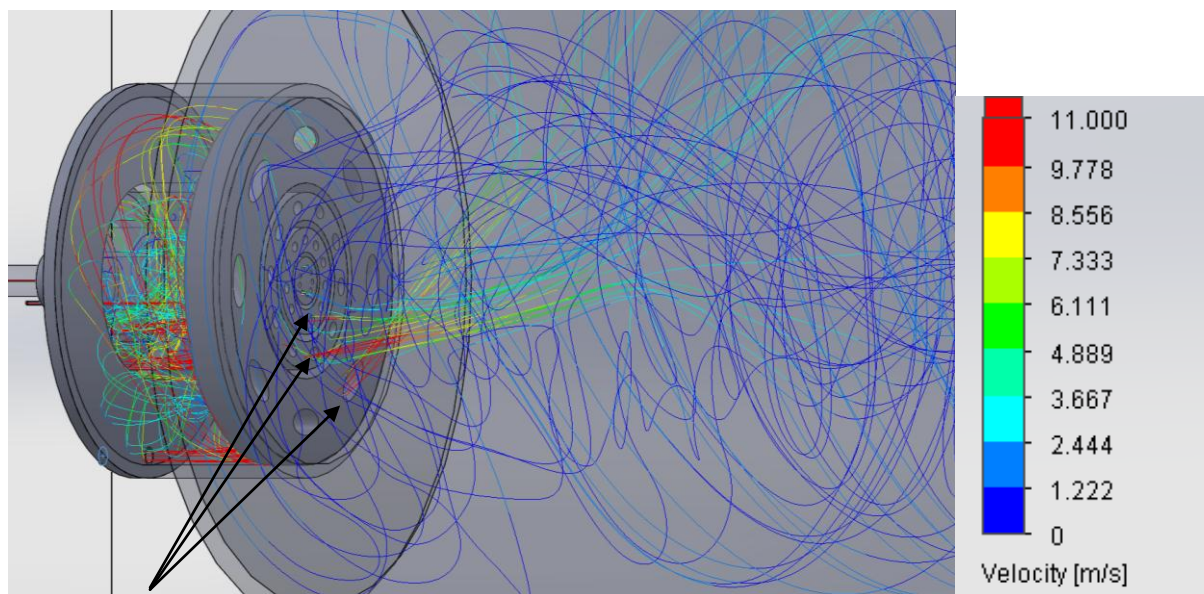


FIGURE 5.22: SOLIDWORKS CFD ANALYSIS (1) - DISTRIBUTION THROUGH SWIRL GENERATOR HOLES

In Figure 5.23, the arrow indicates swirl motion on the side plane.

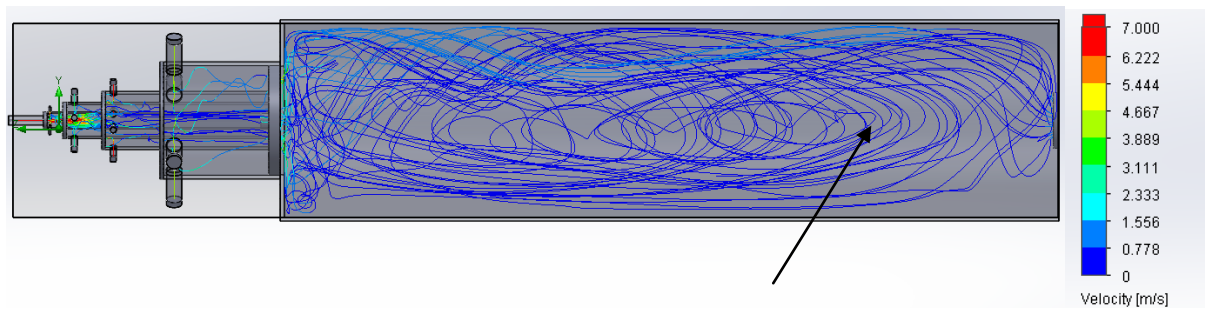


FIGURE 5.23: SOLIDWORKS CFD ANALYSIS (2) - FLOW SIMULATION OF COMBUSTION CHAMBER HOLE EXIT

The flow analysis of the actual burner is displayed in the figures below. Figure 5.24 shows a representation of one gas particle, but under the influence of the PCA, SCA, PF/PA, and the SA. The gas particle recirculates twice before exiting.

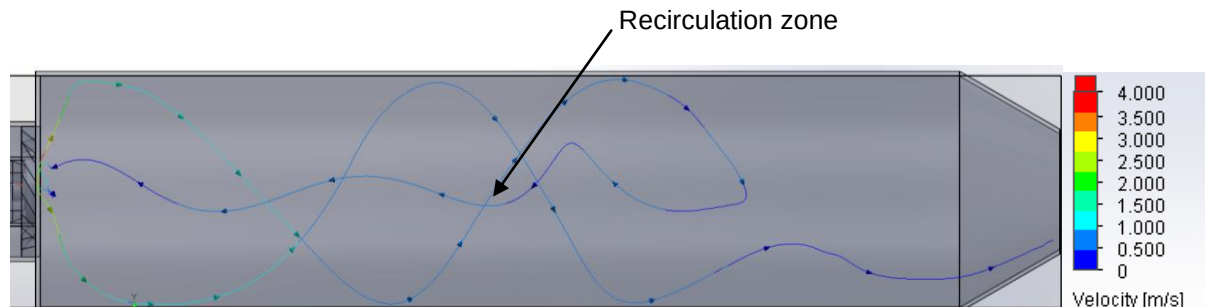


FIGURE 5.24: SOLIDWORKS CFD ANALYSIS (3) - FLOW SIMULATION SHOWING 1 GAS PARTICLE

A representation of one PCA particle per inlet is shown in Figure 5.25. Two of the four PCA particles recirculate once before exiting.

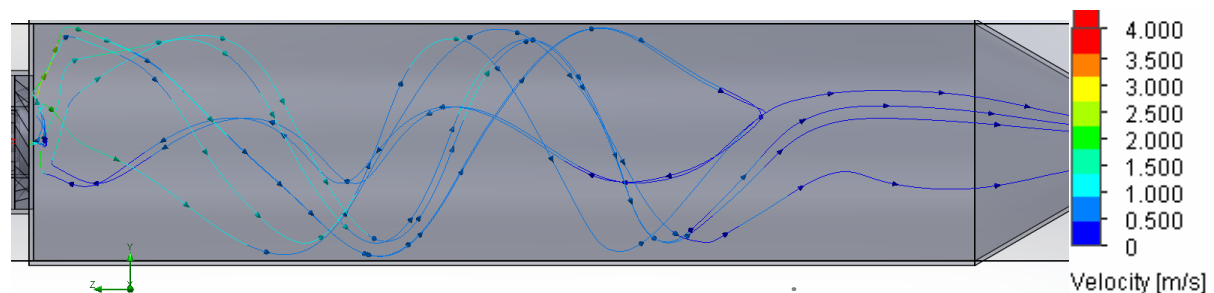


FIGURE 5.25: CFD ANALYSIS (4) - FLOW SIMULATION SHOWING 1 PCA PARTICLE FOR EACH INLET

The velocity representation against the face of the PCA swirl generator is shown in Figure 5.26. The figure shows that the velocity through the holes, varies from 3 m/s to 4.5 m/s. This correlates well with the calculations. It also shows good distribution through each hole.

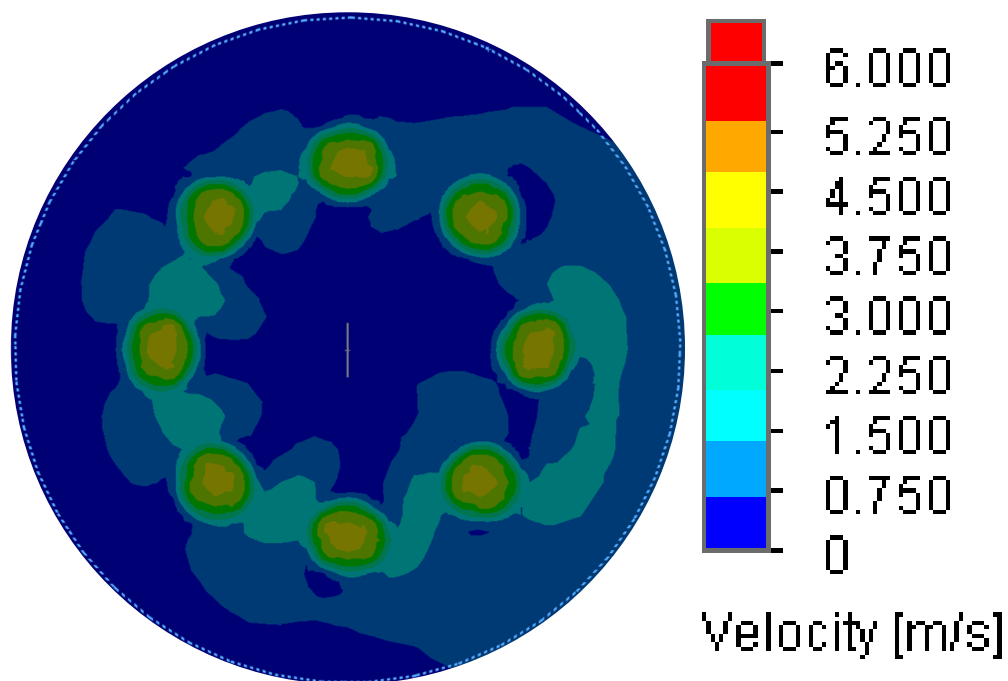


FIGURE 5.26: SOLIDWORKS CFD ANALYSIS (5) - FLOW SIMULATION SHOWING VELOCITY DISTRIBUTION THROUGH PCA SWIRL GENERATOR

A representation of one SCA particle per inlet is shown in Figure 5.27. Two of the four particles recirculate twice before exiting.

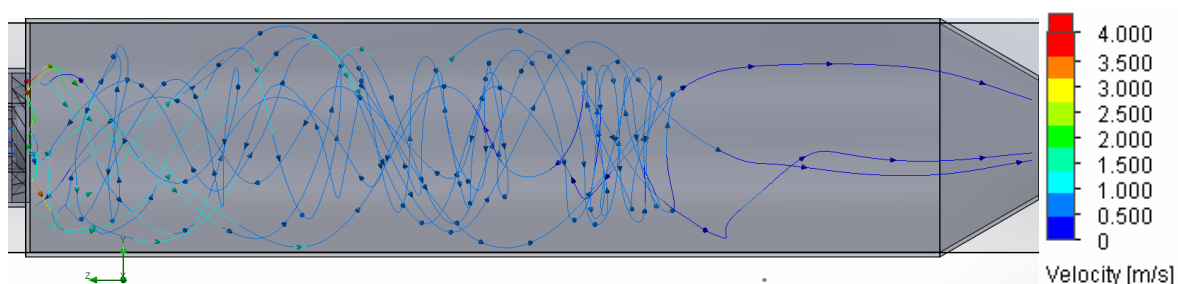


FIGURE 5.27: SOLIDWORKS CFD ANALYSIS (6) - FLOW SIMULATION SHOWING 1 SCA PARTICLE FOR EACH INLET

A velocity representation against the face of the SCA swirl generator is shown in Figure 5.28. The figure shows that the velocity through the holes is 4.125 m/s, which correlates well with the calculated values.

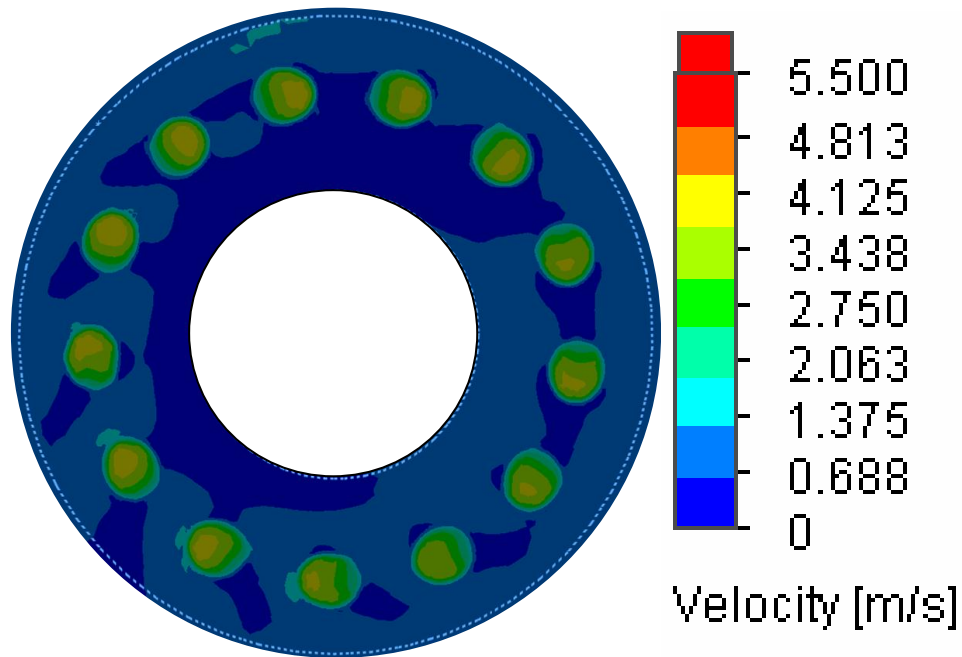


FIGURE 5.28: SOLIDWORKS CFD ANALYSIS (7) - FLOW SIMULATION SHOWING VELOCITY DISTRIBUTION THROUGH SCA SWIRL GENERATOR

A representation of one PF/PA particle per inlet is shown in Figure 5.29. Two of the four particles recirculate five times before exiting.

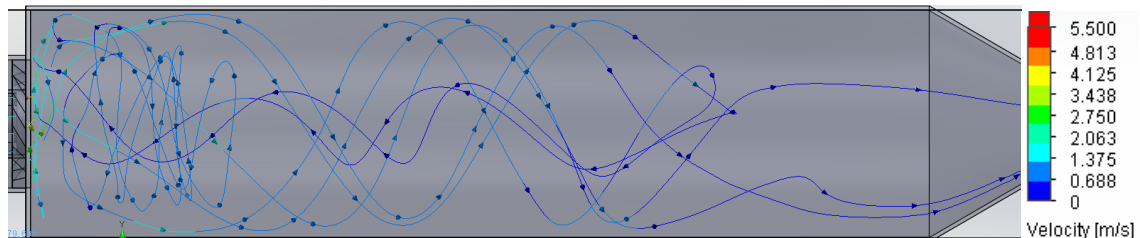


FIGURE 5.29: SOLIDWORKS CFD ANALYSIS (8) - FLOW SIMULATION SHOWING 1 PF/PA PARTICLE FOR EACH INLET

The velocity distribution against the face of the PF/PA swirl generator is shown in Figure 5.30. The figure shows that most of the holes' velocity is at 4 m/s and compares well with the calculations made.

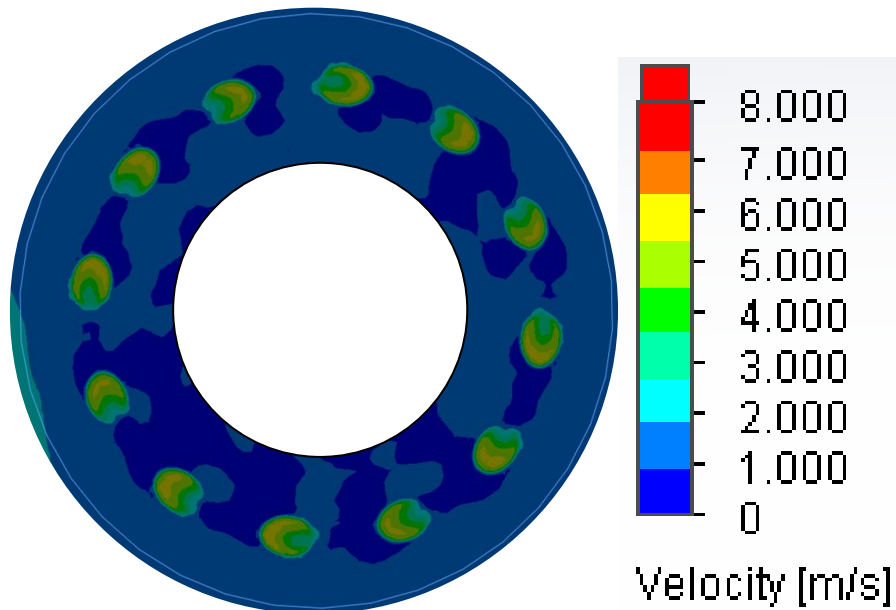


FIGURE 5.30: SOLIDWORKS CFD ANALYSIS (9) - FLOW SIMULATION SHOWING VELOCITY DISTRIBUTION THROUGH PF/PA SWIRL GENERATOR

The representation of the velocity distribution inside the PF/PA tube is shown in Figure 5.31, which indicates whether the PF will fall out or settle.

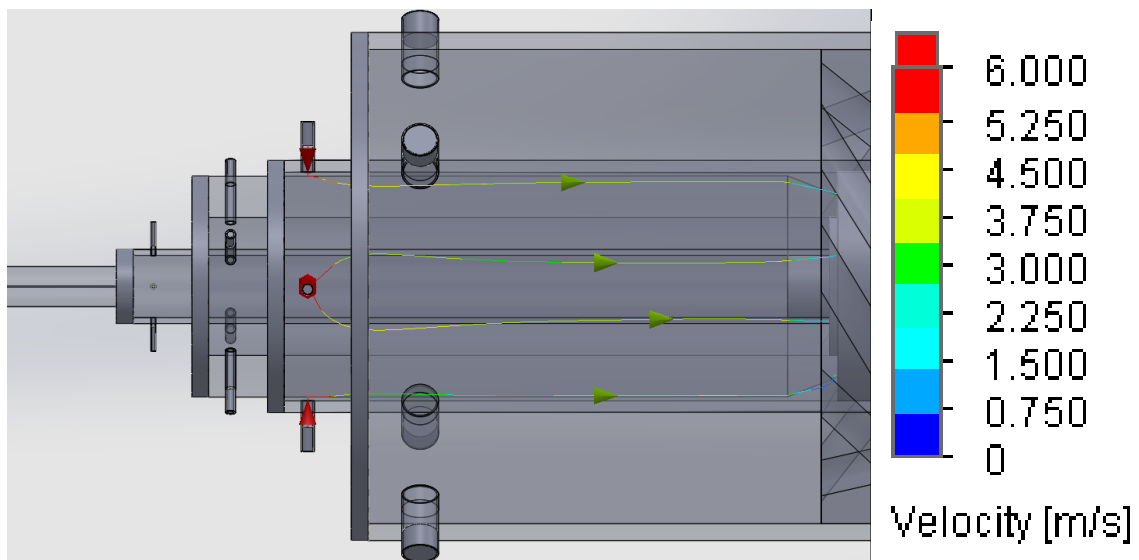


FIGURE 5.31: SOLIDWORKS CFD ANALYSIS (10) - FLOW SIMULATION SHOWING 1 PF/PA PARTICLE'S VELOCITY FOR EACH INLET

The representation of one SA particle per inlet is shown in Figure 5.32. All four particles recirculate once, and some as many as three times.

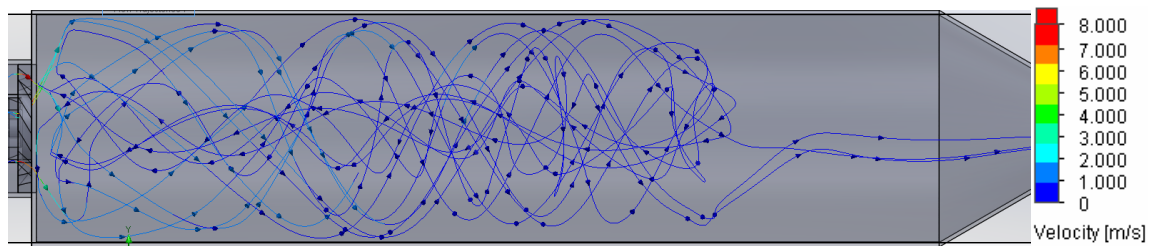


FIGURE 5.32: SOLIDWORKS CFD ANALYSIS (11) - FLOW SIMULATION SHOWING 1 SA PARTICLE FOR EACH INLET

The velocity through the holes is mostly at 6 m/s and relates well to the calculations made in Table 5.8 (Figure 5.33).

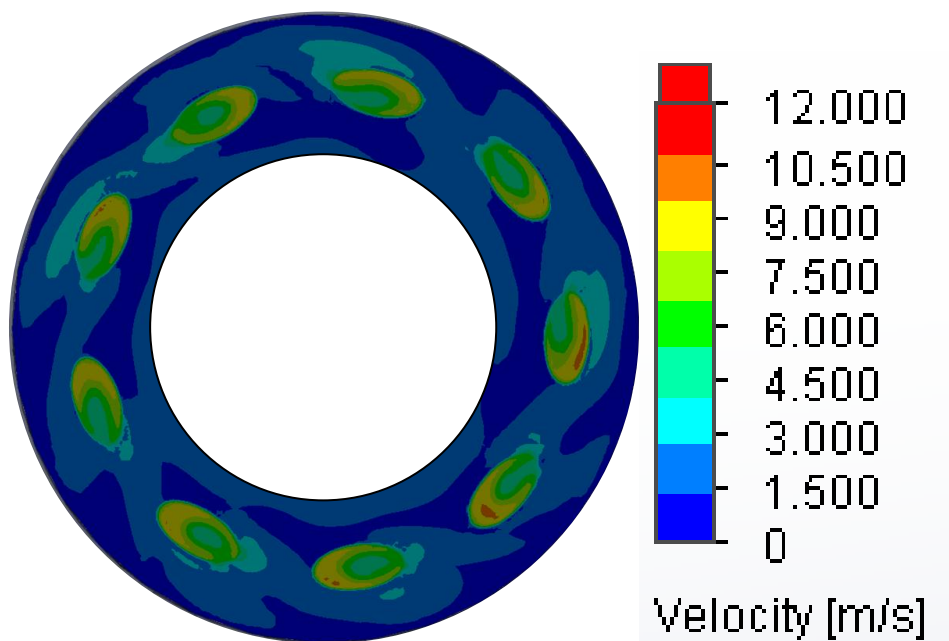


FIGURE 5.33: SOLIDWORKS CFD ANALYSIS (12) - FLOW SIMULATION SHOWING THE VELOCITY DISTRIBUTION THROUGH THE SA SWIRL GENERATOR

The representation of one particle each for gas, PCA, SCA, PF/PA and SA per inlet is shown in Figure 5.34.

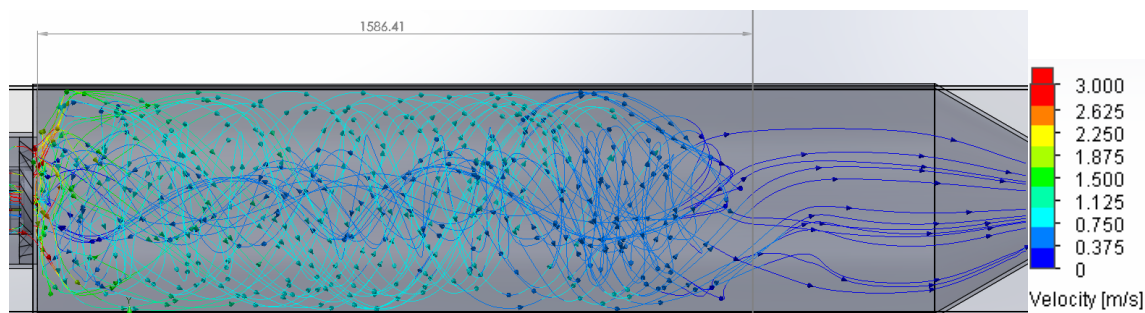


FIGURE 5.34: SOLIDWORKS CFD ANALYSIS (13) - FLOW SIMULATION SHOWING 1 GAS, PCA, SCA, PF/PA AND SA PARTICLE FOR EACH INLET

A representation of velocity distribution at the end of the combustion chamber is shown in Figure 5.35.

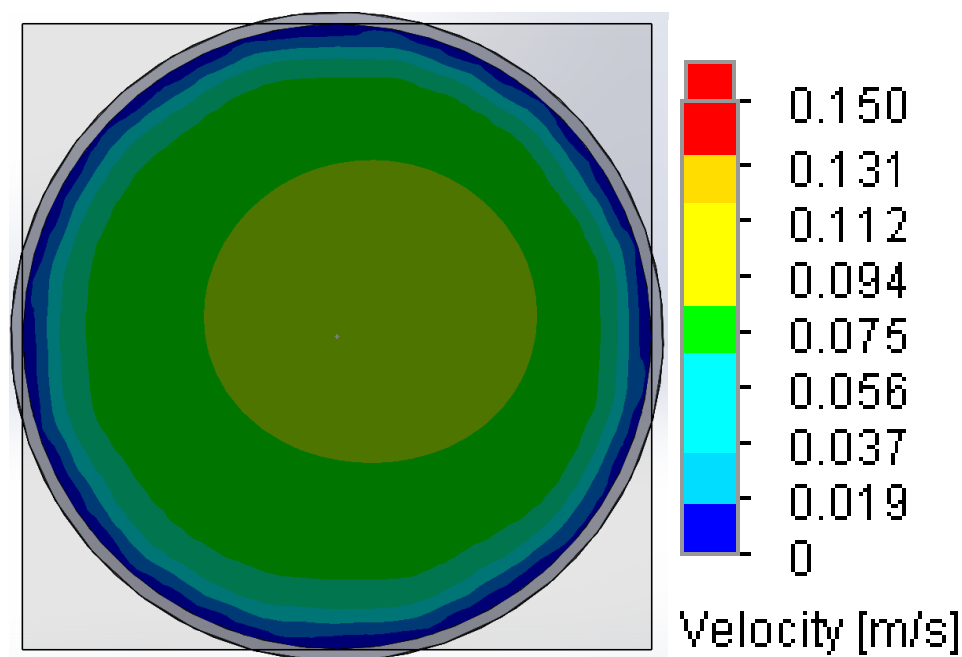


FIGURE 5.35: SOLIDWORKS CFD ANALYSIS (14) - FLOW SIMULATION SHOWING DISTRIBUTION OF FLOW AT EXIT OF COMBUSTION CHAMBER

A representation of the pressure distribution in the micro burner. The orange part shows the lower pressure the vortex forms and where the recirculation zone is formed, Figure 5.36.



FIGURE 5.36: SOLIDWORKS CFD ANALYSIS (15) - FLOW SIMULATION SHOWING DISTRIBUTION OF PRESSURE IN THE COMBUSTION CHAMBER

5.5 CONCLUSION

The flow inside the combustion chamber shows that proper mixing takes place because of the path that the different particles follow. This is accomplished by increasing the velocity further as the fluids exit the swirl generators. The fluid now has the momentum needed to reach the combustion chamber wall and start to circulate. The circulation creates a vortex and the recirculation zone.

The vortex creates a low-pressure zone towards the centre, which draws the air and fuel to the centre of the combustion chamber. This suction is good for two reasons. One is that heat is taken away from the wall of the combustion chamber. The second is that, it allows the air and fuel to have a longer retention time in the combustion chamber and flame. This is very good, since it ensures that most of the coal can experience higher burnt-out, because some particles take up to 2.8 seconds for complete combustion. This design is not subject to the same problems created by the vortex, since the swirl generators block the centre of the tubes.

Figures 5.26, 5.28, 5.30 and 5.33 indicate that the four banjo inlets confirm a satisfactory distribution through the swirl generators' holes. Figure 5.31 shows that the velocity inside the PF/PA tube is 4 m/s, which is equal to the velocity through the swirl generator holes, which should be adequate to avoid PF settling.

Figure 5.34 shows that there is very good distribution before the flow exits the combustion chamber. Furthermore, the converging end specifies that there is no recirculation in the side plane, as shown in Figure 5.23. The flow also starts to stabilise after 1586.41 mm, which means a combustion chamber of 2 m in length should be adequate for complete combustion taking place, which will expand the fluids by a factor of about three, because of the temperature increase.

Finally, this design theoretically satisfies all the set criteria for good combustion, for example: adequate A:F ratio, proper mixing and recirculation that will increase residence time, etc. It also complies with safety features such as: low A:F inside the PCA and PF/PA tubes and no reverse flow in the tubes, thus also reducing damage to the walls by creating the vortex inside the tubes.

6. RESULTS

6.1 MANUFACTURING PROCESS

In order to produce the burner, it has to be manufactured to a certain standard to operate in the correct manner. This standard requires the tolerance of each component to be according to the dimensions of the design. The tolerance must be in the range of $\pm 100 \mu\text{m}$ to the dimensions of the design.

Different manufacturing methods were used to produce this micro coal burner. These methods are milling for the tubes, baseplates and screw threads, welding to mount the tubes onto the baseplates and boring for the banjo inlets. A five-axis CNC milling machine was used to create the swirl generators' holes because of the angle each swirl generator hole must have.



FIGURE 6.1: MANUFACTURING OF A SWIRL GENERATOR USING THE FIVE-AXIS CNC MILLING MACHINE

6.2 PRODUCT AFTER MANUFACTURING

After the different methods of manufacturing had been performed and these components had been assembled, the micro coal burner looked as follows:



FIGURE 6.2: ASSEMBLED MICRO COAL BURNER (1)



FIGURE 6.3: ASSEMBLED MICRO COAL BURNER (2)

6.3 COAL BURNER SET-UP

As mentioned previously, the burner only mixes the air and fuel in a manner suitable for appropriate combustion but, some extra components were still needed. These are:

- **Pulverised coal feeder:** This feeder provides the correct amount of coal, which is transported to the burner by the PA.
- **Combustion chamber:** The combustion chamber is mounted on the baseplate of the burner. It allows for proper mixing of fuel and air, since the circular chamber maintains the swirling motion initiated by the swirl generators. The combustion chamber is not yet water-cooled and this will be implemented later.
- **Transport pipe system:** To ensure that each banjo inlet is supplied with an equal amount of air or fuel for each tube, a pipe network is implemented. At first, the total flow for a specific tube is given through one single pipe, and then this pipe is connected to a T-piece that divides the flow equally into two. The two airflows are then again divided by using another two T-pieces, resulting in four equal flows, which are connected to the banjo inlets. The pipe diameters are selected such that the restricting diameter will be at the banjo inlets. This is to ensure that the pressure drop that occurs in the pipe network before the flow enters the different tubes is as small as possible. The PF/PA pipe network is the only exception to the previously mentioned criteria, since high-velocity flow through the pipes is needed to avoid PF settling.
- **Fan and compressors:** The fan used can supply an amount of 200 l/s free air delivery. This air volume flow is more than sufficient since the required amount is 24.237 l/s. However, since the maximum amount of pressure the fan can deliver is only 5 kPa, the fan is not able to supply all the air, because of pipe friction. The fan delivers 25 l/s at the banjo inlets because of pipe friction, which is still adequate. Because of the high pressure inside the PF/PA and core air transporting pipes, two compressors in parallel are used. The compressors can supply sufficient air and pressure via the pipe network. The fan is used to supply only the SA, since the compressors cannot deliver sufficient volume flow for all air supply needs.

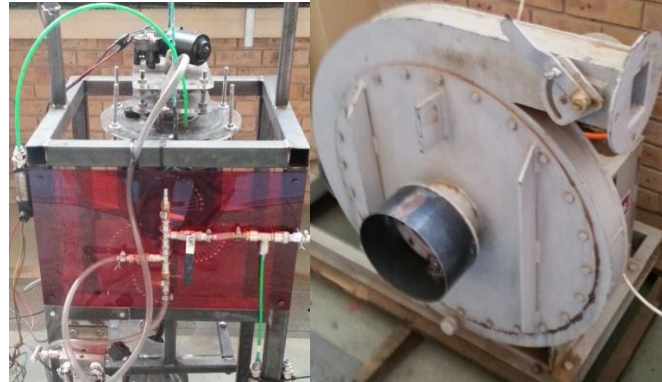


FIGURE 6.4: COAL FEEDER AND FAN

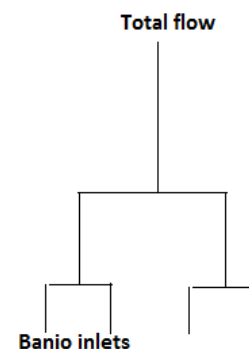


FIGURE 6.5: COMBUSTION CHAMBER AND PIPE NETWORK



FIGURE 6.6: MICRO COAL BURNER SET-UP

The air heaters will not be implemented yet because they are still in the development phase in other projects. Because of this, the mass flow of air provided will increase to 24.633 g/s as opposed to 15.88 g/s of heated air, since the volume flow will stay the same. The effect of swirl is dependent on the velocity through the swirl generators' holes and the velocity is dependent on the volume flow and not the mass flow. By sustaining the volume flow and increasing the mass flow, the mixing component (which is most important for this exercise) can be retained while the increased mass will only cool down the flame a little because of too much excess air.

6.4 INSTRUMENTATION

Since all the objectives (except to enable the micro coal burner to burn) have been met, only the volume flow of air and mass flow of coal needs to be measured at this point. The volume flow of air is measured with a rotameter, which is calibrated appropriately. The volume flow is correlated by using an air velocity anemometer and the area through which the separate flows pass. The mass flow of coal is measured by using a load cell and the time interval. The mass flow of PF and time are used and have been programmed and calibrated to produce a value in g/s.

6.5 AIRFLOW TESTS

After the set-up had been completed, tests were conducted on each of the PCA, SCA, PA and SA to determine whether the burner passed the required airflows.

Before connecting the burner to the combustion chamber, the correct volume flow was measured before it entered the burner by using the rotameters. The volume flow was also measured at the exit of the combustion chamber by using the anemometer and the different pipes' areas. After the correct amounts of air for each tube had been established, the air and coal distribution through the swirl generators' holes was evaluated by using an anemometer. The flows were well distributed among the swirl holes and no PF settled inside the transport pipes or PF/PA tube. This is a good correlation with the CFD done earlier. However, when the combustion chamber was connected, the airflow was difficult to determine at the exit, since air was being pulled inwards as well as flowing outwards. This was due to the vortex created by the swirl generators, indicating that there would be sufficient swirl and proper mixing should take place. However the air that was pulled inwards did not correlate with the CFD, which indicated that there was only air exiting.

6.6 COMBUSTION TESTS

Since the flame can only be seen from an axial direction at the outlet of the combustion chamber, combustion tests were performed with and without the combustion chamber. This was to see how the gas and coal reacted to the different airflows through the different swirl generators.

6.6.1 COMBUSTION WITHOUT THE COMBUSTION CHAMBER

At first, only the gas was ignited, without any air support, thus burning in the atmosphere. Then the PCA and SCA were supplied, followed by the PA and SA. Finally, the PF was supplied once the gas flame had stabilised. The figure below shows side and frontal views of the burner to observe the micro coal burner when the combustion chamber was not connected.

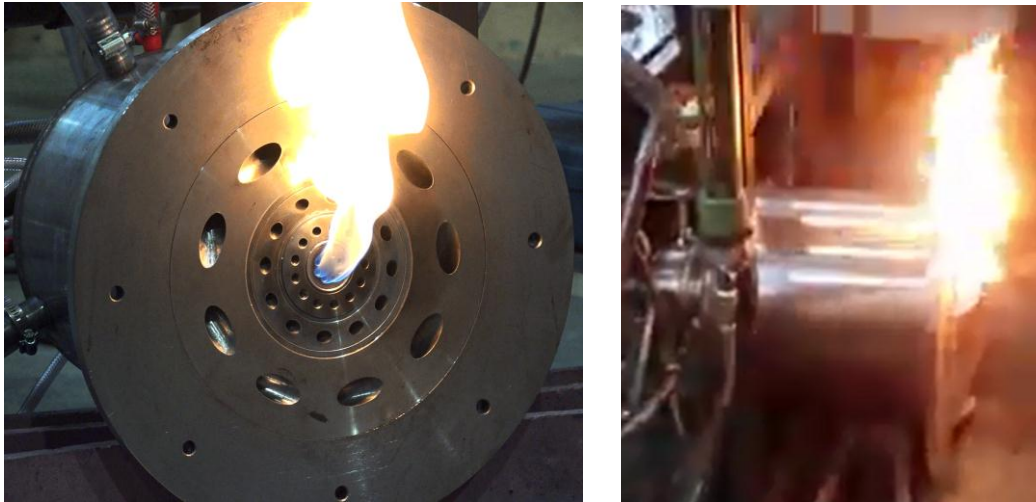


FIGURE 6.7: GAS FLAME WITH NO AIR SUPPORT

The figures above show how the gas burns when it is ignited using only the air in the atmosphere. The gas flame is ignited without air support from the swirl generators, since the high velocity of air supplied forces ignition flameouts. It can be noted how long and yellow the flame is which indicates incomplete combustion. Combustion only takes place once the gas/air mixture exits the PCA tube, proving that the A:F and velocity are safe (below the LEL).

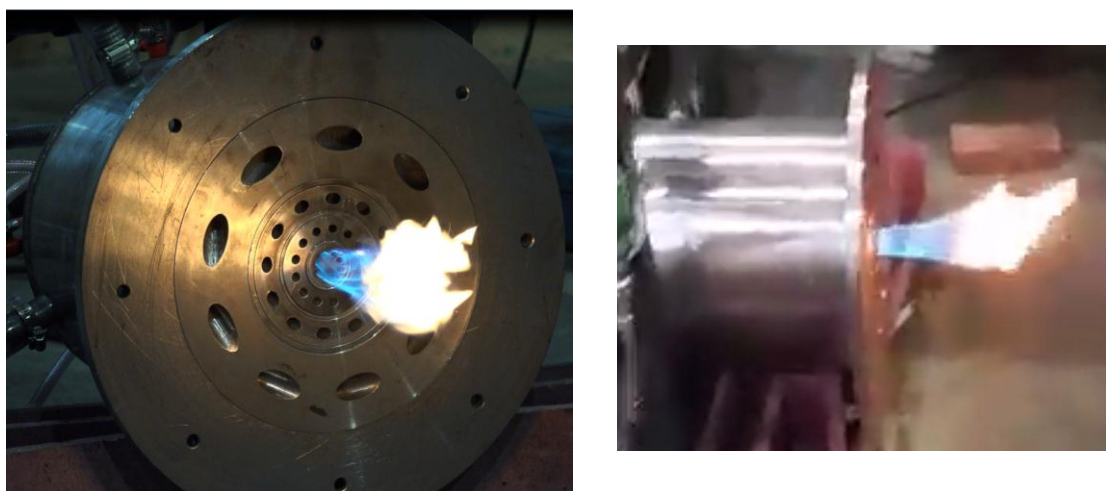


FIGURE 6.8: GAS FLAME WITH PCA AND SCA SUPPORT

The PCA and SCA were subsequently added to the gas flame in the figures above. When the PCA and SCA were added, the flame profile and colour changed immediately. It can be seen that the flame has a more blueish colour than before, indicating better combustion. There is however still a yellow part to the flame, representing the SCA not mixing properly with the gas. It can be seen that the PCA swirl generator acts as multiple little gas burners, each creating swirl. The swirl caused by the PCA and SCA swirl generators creates a shorter flame length. It cannot be seen clearly in the figures above, but ignition occurs 2-3 mm away from the swirl generator face. This indicates that the air velocity is higher than the flame velocity of the gas inside the PCA tube and back burn cannot take place.

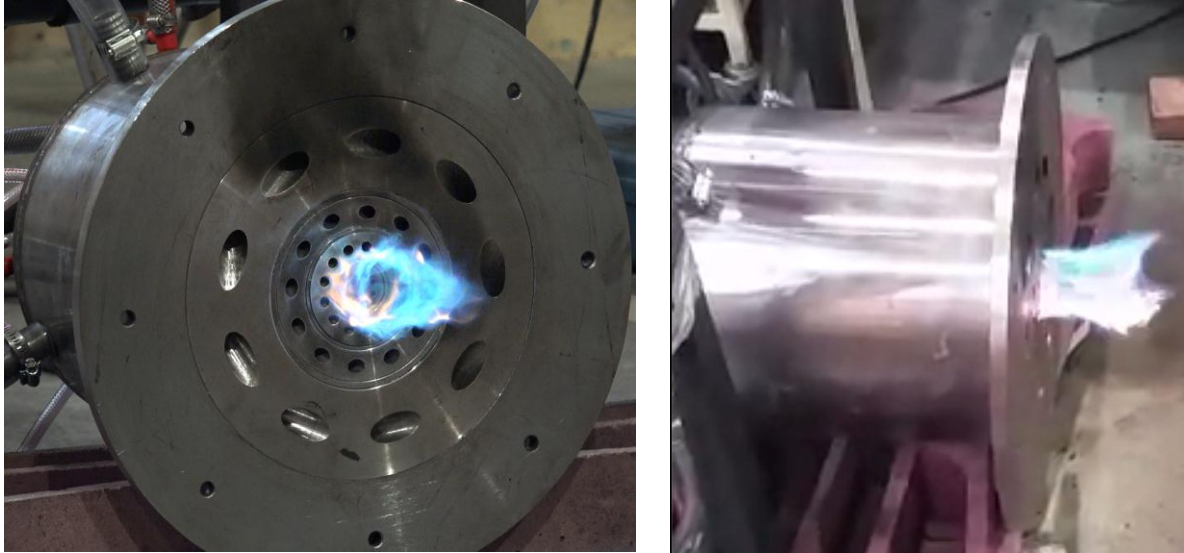


FIGURE 6.9: GAS FLAME WITH PCA, SCA AND PA SUPPORT

Figure 6.9 above show the gas flame when PA is also added. Once again, the flame immediately changes in profile and colour once the PA is added. The flame is now almost completely blue, indicating almost complete combustion. This is because there is a greater vortex and better mixing of air and gas due to the higher amount of swirl. The flame length is now even shorter than before, showing more air being recirculated.

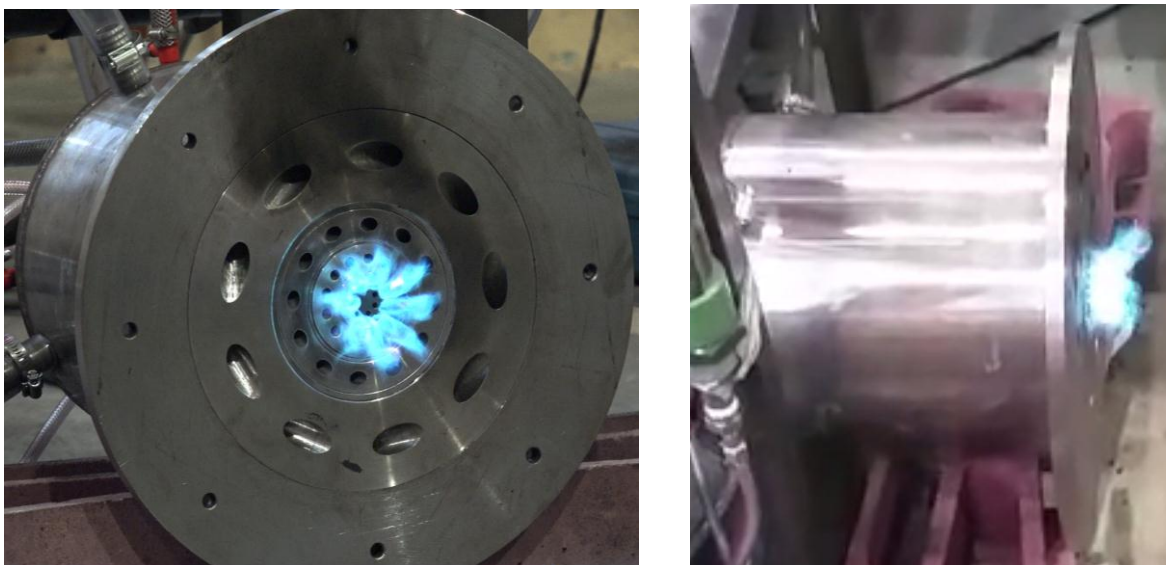


FIGURE 6.10: GAS FLAME WITH PCA, SCA, PA AND SA SUPPORT

When adding the SA in the figures above, the colour of the flame becomes blue without any yellow. Complete combustion takes place, since there is sufficient air, swirl and mixing. The flame becomes very short, indicating an even greater vortex than before.

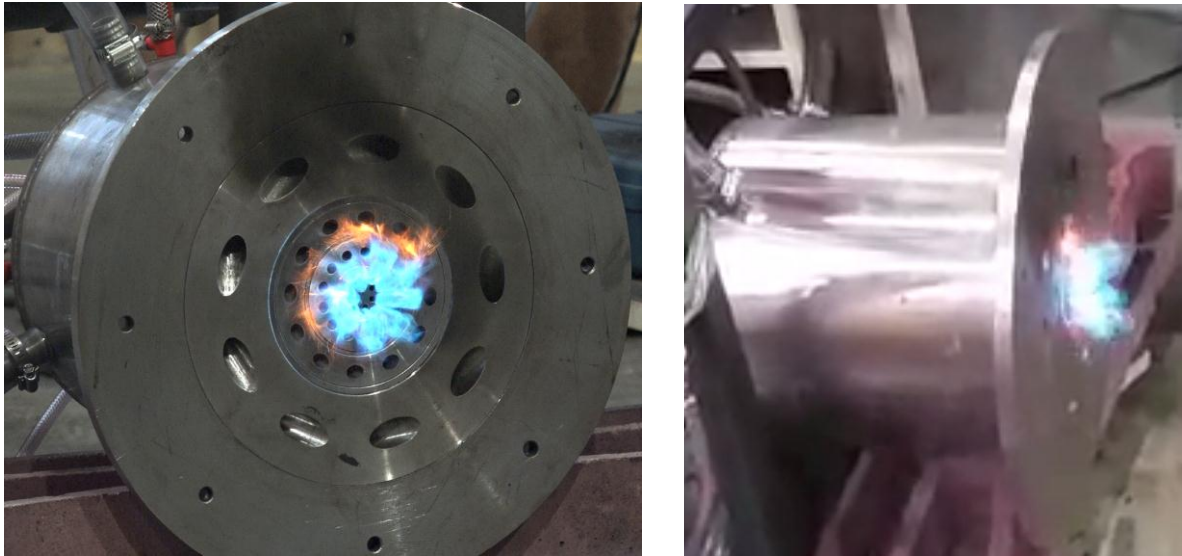


FIGURE 6.11: IGNITION OF PF

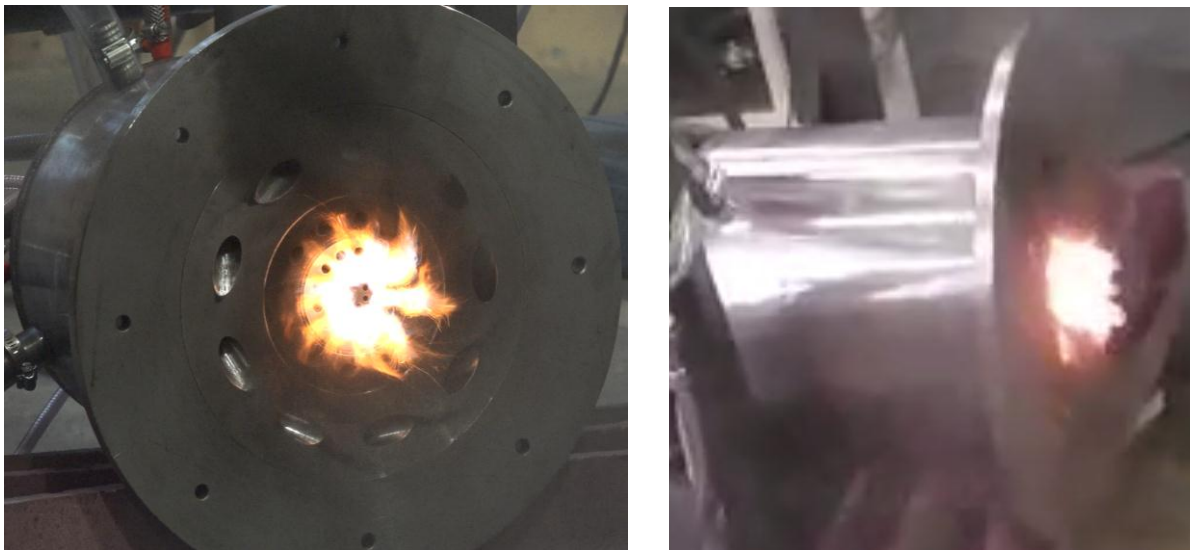


FIGURE 6.12: FLAME DUE TO COMBUSTION OF PF AND GAS

As can be seen in Figure 6.11, the gas starts to ignite the PF as the PF is being fed into the flame. The PF flame is the orange part on the outside of the blue gas flame. Figure 6.12 represents the ignited PF flame and gas flame. Unfortunately, there is a lot of PF that does not burn; the flame can also not be sustained when turning the gas supply off. The PF flame is slightly more radially outward than the gas flame was and combusts near the swirl generator face and between the holes. The PF is able to burn because it gets sucked into the vortex that is created.

It was expected that the PF would not sustain a flame without the combustion chamber, since the combustion chamber creates more swirl and recirculation because of its circular wall. It is quite interesting to see a vortex being created without a combustion chamber in each of the scenarios above, since the airflows exiting the swirl generator holes only have a linear movement. It is expected that the vortex will only be enhanced once the combustion chamber is attached. It is, however, a good sign that swirl and a vortex are still being created and that combustion is taking place without the combustion chamber.

6.6.2 COMBUSTION WITH THE COMBUSTION CHAMBER ATTACHED

The same procedure was followed as previously, but with the combustion chamber attached. Since the combustion chamber has only one axial opening, only a frontal view was possible.



FIGURE 6.13: GAS FLAME WITH NO AIR SUPPORT

As the gas is ignited in the figure above, no core air support is supplied. The gas flame has the same tendencies of colour and profile as before (without the combustion chamber attached). The gas is able to burn since there is still air inside the combustion chamber. The gas will not continue to burn if all the O_2 has been depleted.

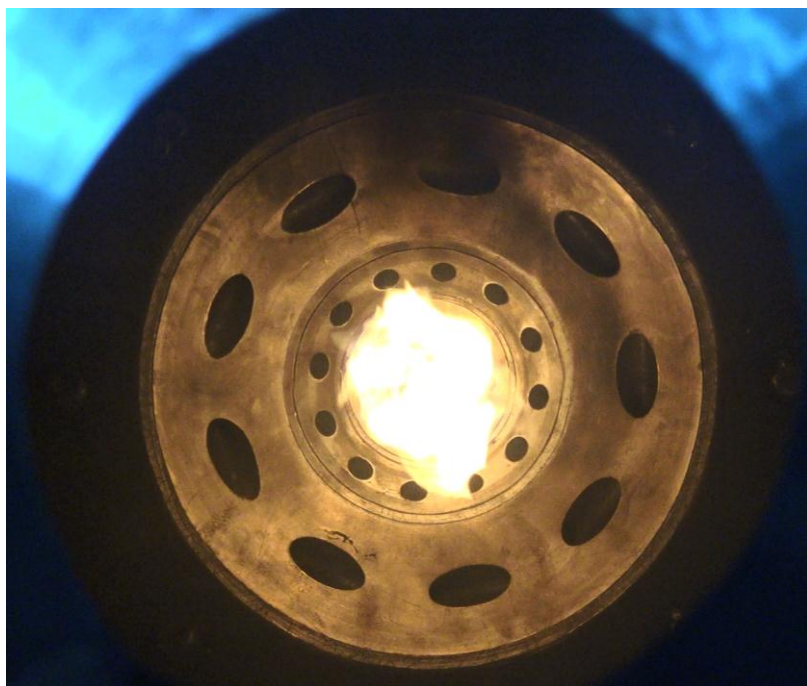


FIGURE 6.14: GAS FLAME WITH PCA AND SCA SUPPORT

By evaluating Figure.6.13, the characteristics of the gas flame are the same as before, without the combustion chamber. The combustion chamber has no effect on the amount of swirl, since the space is too large for the core air to have an effect.

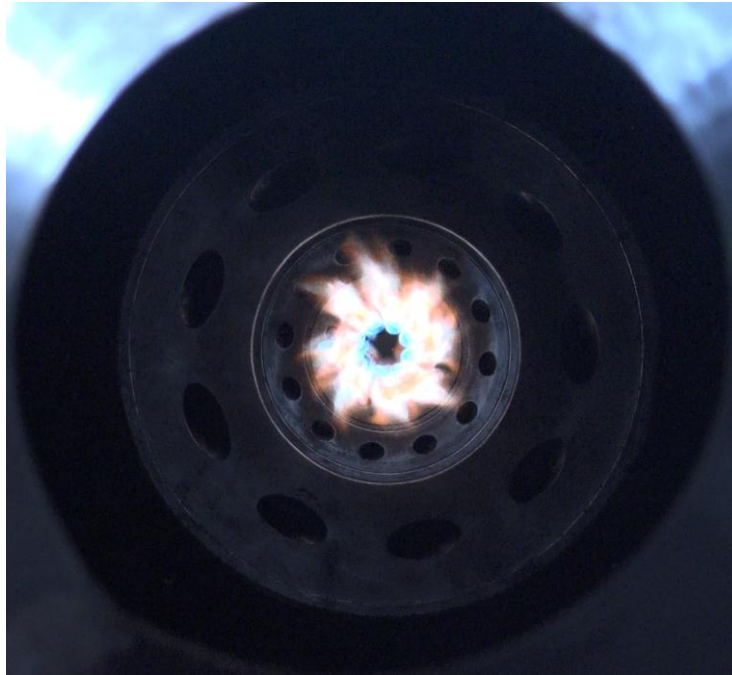


FIGURE 6.15: GAS FLAME WITH PCA, SCA AND PA SUPPORT

The gas flame is now different from before when evaluating Figure.6.15. It now has a similar shape as the gas flame without a combustion chamber and with SA supplied. The shape suggests a larger vortex forming, which proves that the combustion chamber has a large influence on the swirl. The colour of the flame indicates better combustion. The incomplete combustion cannot be because of too little swirl, but owing to too little air, suggesting that not all of the air is recirculating into the flame.

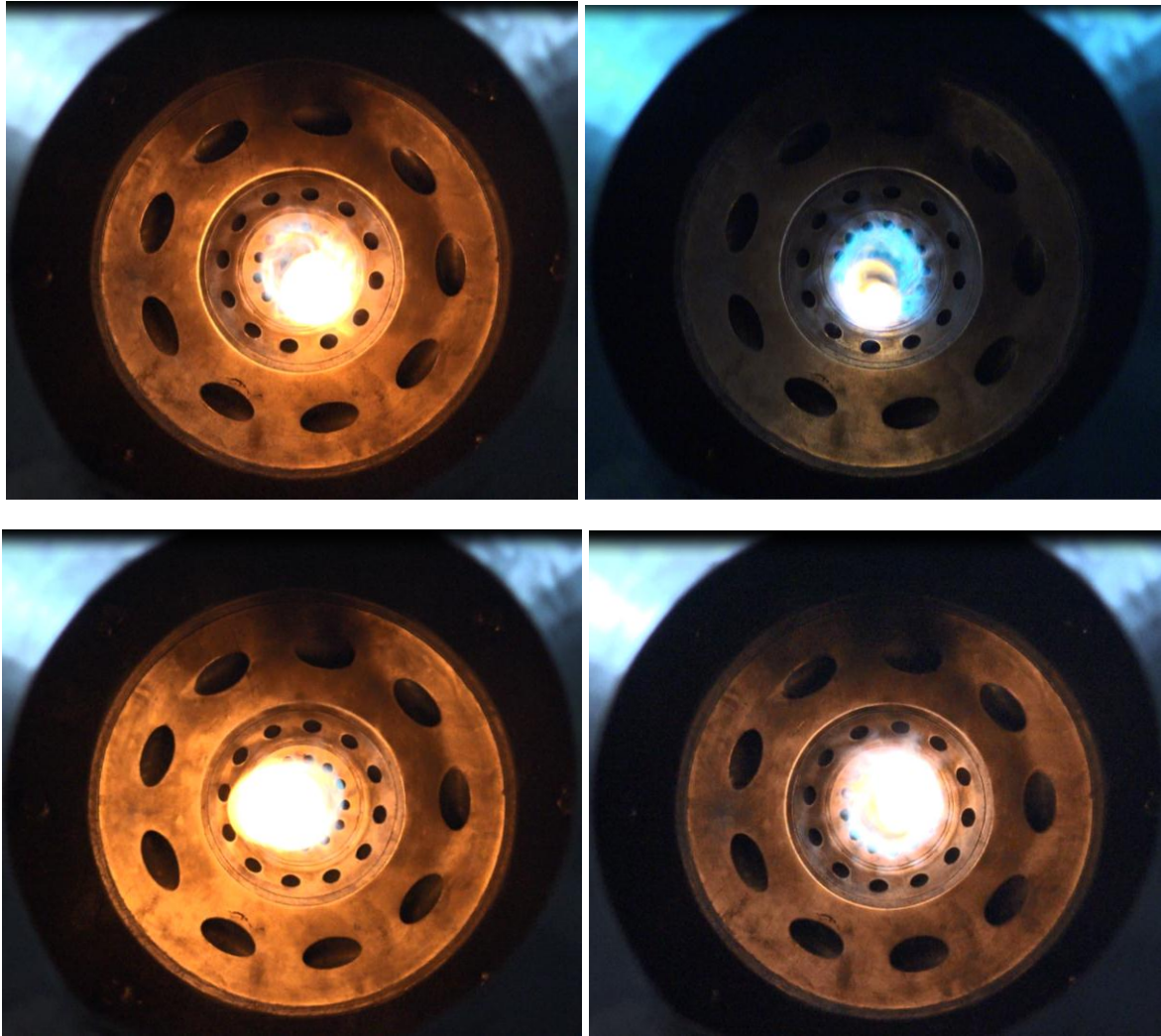


FIGURE 6.17: GAS FLAME WITH PCA, SCA, PA AND SA SUPPORT RESPECTIVELY

The figures above show combustion of the gas with the SA included. The flame colour and characteristics are completely different from before. Without the combustion chamber, the gas flame flared open more and there was complete combustion (since the flame was blue). With the combustion chamber attached, the colour of the flame changed to yellow and blue and the shape seemed to differ. The gas flame flared open to the face of the swirl generators, but also formed a fireball further down to the exit. The part of the flame that flared open had a blue colour and the part of the fireball had a yellow colour. Though the flame length was difficult to determine because of the limited angle, it seemed to be about 20 cm in length.

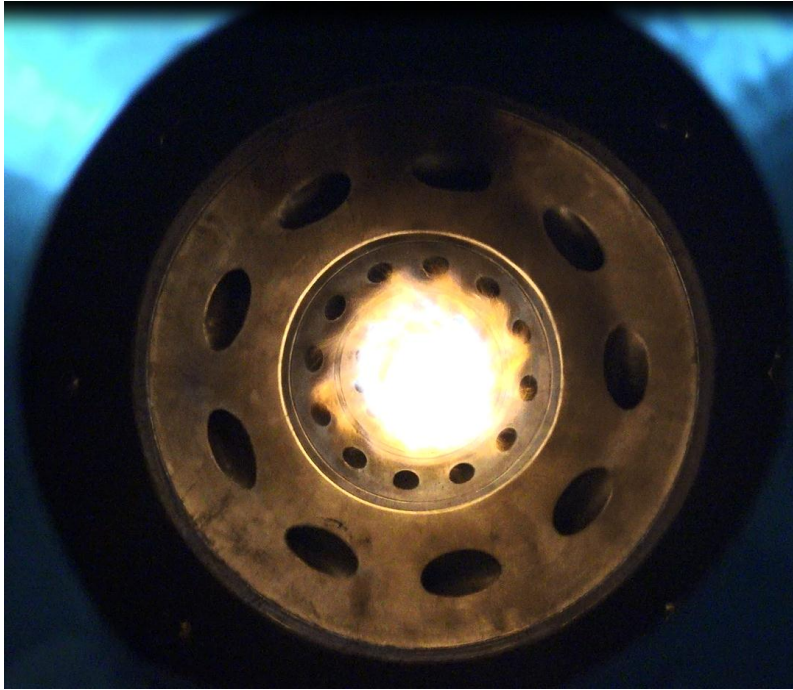


FIGURE 6.18: IGNITION OF PF



FIGURE 6.19: FLAME CONSISTING OF THE COMBUSTION OF PF AND GAS

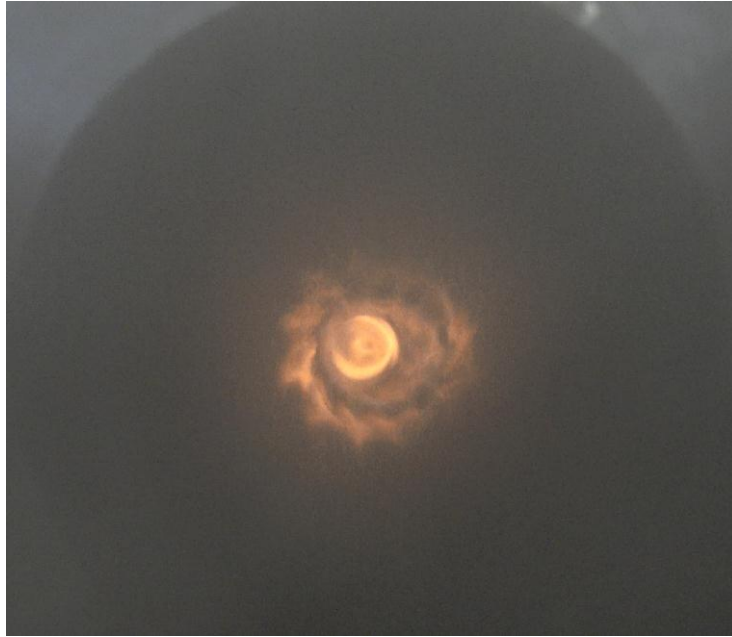


FIGURE 6.20: UNBURNED PF

Figure. 6.18 shows how the gas flame ignites the PF. The colour turns yellow and the fireball grows larger. Figure. 6.19 represents a flame consisting of gas and PF. The fireball is bright yellow, larger than before and combusts against the face of the swirl generators.

Although the flame consists of some PF, most of the PF does not take part in combustion. This can be seen in the unclear image of Figure. 6.20, which is actually unburned PF dust obscuring the view (low visibility).

The amount of SA was also varied to see whether the PF responded to more or less air, but the results remained the same, with only a portion of the PF burning.

7. CONCLUSION AND RECOMMENDATIONS FOR THE FUTURE

The functionality of the burner in practice correlated well with the design and intended outcome. The research reached all the objectives for this stage of the project (except one eventual requirement: burning the PF without gas support, see 7.2). Regarding this, the following should be borne in mind:

- The preheating of air and PF was not available at this stage of the project. The regenerative air heaters (another project) are still in the development phase. The design requires the PCA, SCA, and PF/PA to be preheated to 70 °C, and the SA to 275 °C. In these tests, all the air supplied was at ± 30 °C. This makes a significant difference to the ignition properties of the PF. Unnecessary heat energy is drawn from the flame to heat the air up to the combustion temperature, leaving less initial heat for ignition and combustion of the PF. Testimony to this, is found in the behaviour of full-scale power station burners during cold start-up, when the air attenuators are open.
- The swirl profile for these experiments was achieved by the correct volume flow for the velocities and swirl angles, but at a resulting higher mass flow, which cooled the flame down and did not produce the intended A:F.
- Because of logistic constraints, the PF samples received from power stations via Eskom RTD were very old (some dating to 2013 and even 2010). Although in sealed bags, this PF must have undergone a great extent of devolatilisation. The role of the volatiles is mostly to ignite the fixed C, which then combusts and delivers the major component of the heat. The volatiles also ignite quickly and do not require a relatively lengthy retention time as the fixed C does.

There are, however, many positive results emanating from this research thus far:

7.1 POSITIVE RESULTS

7.1.1 CFD ACCURACY

- The CFD done on the design was accurate compared to the practical behaviour.
- The flow through the swirl generators was distributed evenly, as the CFD predicted it would be.
- There was no PF settling inside the tubes, which means the assumption to move the flow at 4 m/s was correct and adequate.
- The CFD showed that there was only air flowing outwards at the exit of the combustion chamber and in practice there was air moving inwards. This problem can be solved when the extraction fan, heat exchangers, and filter bags are mounted downstream to the combustion chamber.
- When the gas was burning with and without the combustion chamber, the gas flame had a certain pattern. Although the patterns were different with and without the combustion chamber for the separate flows, correlations with the CFD can still be made.
- The figure below represents a good correlation between the actual gas flame and the CFD. The angle (indicated in the figure below by the arrows) at which the gas is burning, is approximately the same as the angle at which the gas particles exit the burner in the CFD. It can be assumed that the angle is the same with and without the combustion chamber, since the patterns of the flames are the same.

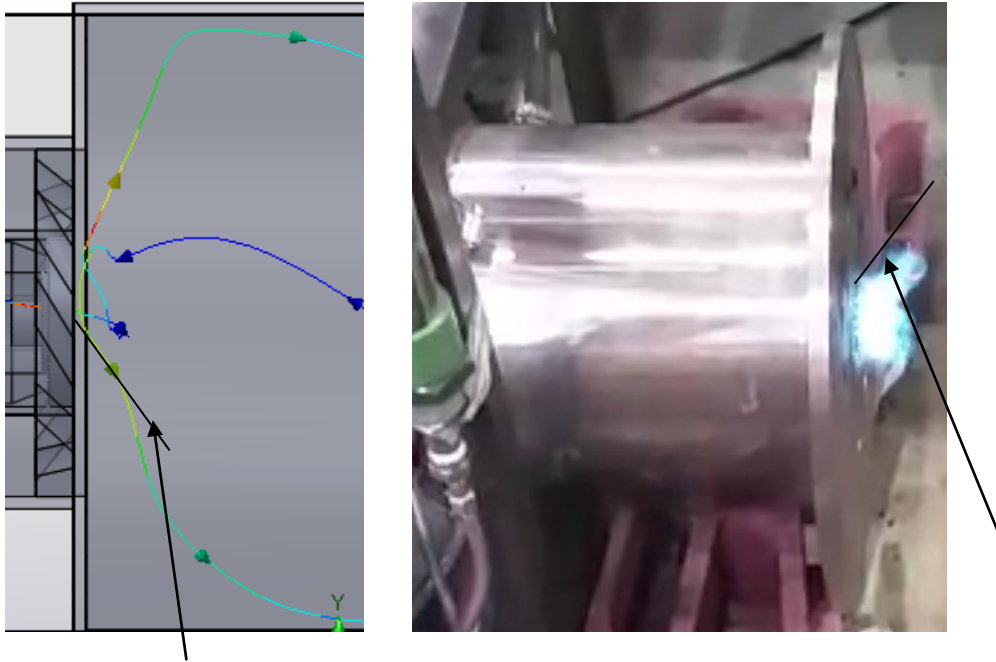


FIGURE 7.1: RESULT OF THE GAS FLAME COMPARED TO CFD (1)

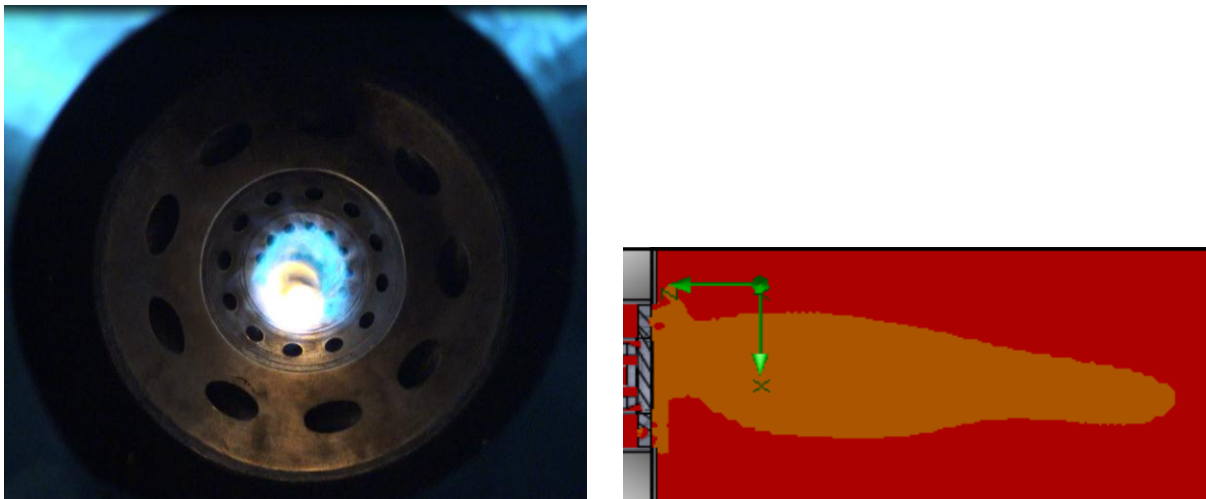


FIGURE 7.2: RESULT OF THE GAS FLAME COMPARED TO CFD (2)

- The actual gas flame with the combustion chamber and SA supplied has a similar pattern to that of the pressure-oriented CFD.

The statements above prove that the accuracy of the CFD done on the design was adequate.

7.1.2 SAFETY

During the combustion tests, it was observed that the flame was situated 2-3 mm from the swirl generators' face. This proved that the A:F inside the tubes that transport the fuel was safe. Although the recirculation zone forces combustion against the swirl generator face, the velocity at which the gas and air exit the swirl generator is much higher than the burning velocity of gas, preventing back burning and combustion inside the tubes. No metal temperature excursions occurred; even though the gas flame was close to the swirl generator face, the tubes remained cool.

7.1.3 COMBUSTION OF GAS

Overall, good combustion of gas was achieved with and without the combustion chamber. However, the fact that the gas did not combust very well when only the PCA and SCA were supplied, indicates that not enough air was fed directly into the flame. This can be explained by looking at the CFD.

Considering the case of adding the combustion chamber, the gas, PCA and SCA are minute compared to the SA. They cannot create a swirl and recirculation zone since they cannot reach the combustion chamber wall to start a circular motion. That is why they burn the same with and without the combustion chamber.

By adding the PA with the combustion chamber, combustion is enhanced. The PCA, SCA and PA are probably enough to reach the combustion chamber wall and create enough swirl for a recirculation zone. The combustion chamber has a clear impact on the flame profile in this scenario. The fact that the gas flame is flared open, means it overlaps with the PF/PA swirl generator, enabling some of the PF to make contact with the flame.

When providing the SA, the gas flame shape changes. When comparing the gas burner experiment with the micro coal burner, it can be assumed that the inner flows are drawn to the path of the outer swirling flows. The gas burner experiment showed that the PCA and gas took the path towards the SCA. The micro coal burner shows that the gas, PCA, SCA and PA follow the path of the SA. When doing so, only a portion of the gas can combust initially, because it exits the flame, leaving the remaining gas to combust when it recirculates back into the vortex zone. This results in incomplete combustion in the fireball, which shows the yellow part.

7.1.4 IGNITION AND COMBUSTION OF PF

- This experiment succeeded in igniting and combusting PF in a single micro burner, without the flame support of adjacent and opposite burners, as in full-scale boilers.
- The only negative finding was that not all the PF supplied ignited. Some of the PF was flung outwards behind the gas flame to the larger diameter combustion chamber wall (see 7.3 for a solution).

7.2 NEGATIVE RESULTS

As stated above, only a portion of the PF burned when it was introduced to the burner. The PF flame could also not be sustained without gas support. The reason for all the PF not burning will subsequently be explained:

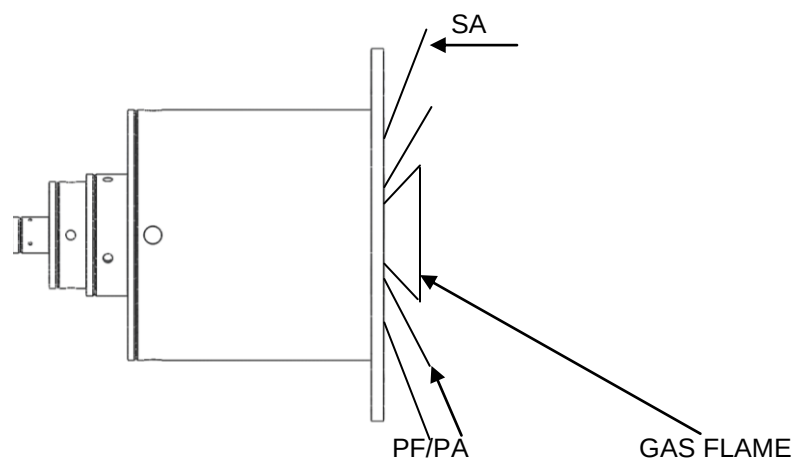


FIGURE 7.3: DIRECTION OF THE DIFFERENT FLOWS

The figure above will aid in the explanation of this negative result.

- The PF was influenced by the same factors as the gas. The PF also now follows the path of the SA, which forces most of the PF out of the flame, as in the figure above. Only a portion of the PF can burn, some of it when it makes contact with the flame, and some of it when it recirculates. The PF flame coincided with the gas flame.
- The assumption that the recirculation will be enough to force all the PF and sufficient air into the flame and that all the PF will move through the flame was negatively influenced by the cold air and A:F, as explained previously. The PF and air making contact are insufficient to ensure combustion.
- It can also be stated that the heat zone inside the combustion chamber was not large enough to ignite the PF. This was established by measuring the temperature on the outside of the combustion chamber. The temperature did not even reach 100 °C after 10 minutes of burning the gas, making it impossible to ignite the PF spontaneously (the spontaneous combustion temperature of most coals are 600 – 700 °C). Despite this, the portion of PF exposed to the gas flame did ignite and combust, which is positive.

7.3 POSSIBLE SOLUTION AND FUTURE WORK

Although not all the PF did combust as expected, a solution can be offered. The major problems to overcome are the heat zone and the mixing of air and fuel. The amount of swirl and volume flow of air are correct for combustion, but are not introduced in the correct place.

To solve the heat zone problem, a tube with the same diameter as the PF/PA tube can be fitted after the PF/PA swirl generator (Figures 7.4 and 7.5). This will force the gas flame and PF to be contained and channelled together in one tube. This will contain the PF that swirled outwards behind the gas flame. As could be seen from the results, the gas flame was flared open enough to overlap the PF/PA swirl generator diameter. If some PF does not enter the gas flame, the gas flame will create the appropriate heat zone for spontaneous combustion in this smaller contained volume. To avoid melting of the PF/PA tube, the SA on the outside will provide cooling. This assumption is based on the gas burner experiment, where only the PCA tube experienced metal temperature excursions but not the SCA tube, because of the atmosphere cooling the SCA tube.

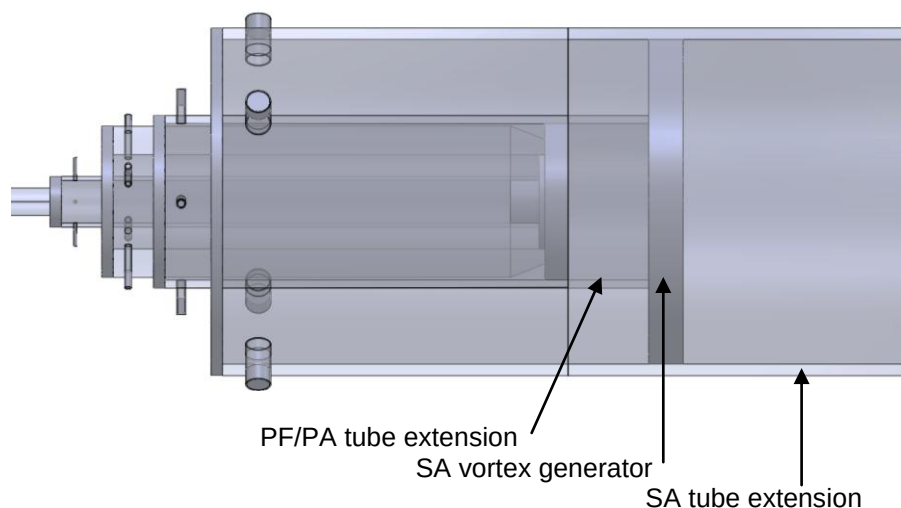


FIGURE 7.4: FUTURE DESIGN OF MICRO COAL BURNER (1)

If the PF experiences adequate heat provided by the gas flame and the extended PF/PA tube, it will not combust yet, since there is not enough air (far below stoichiometric A:F). The PF will, however, carry the heat it absorbed for a short distance and time and start to de-volatilise. Thus, the PF/PA tube must not be too long. Thereafter, the PF needs sufficient air to burn. When the PF exits the PF/PA tube, it enters the SA tube. By assuming that the gas, PCA, SCA and PF/PA are forced outward to the path of the SA, an extension of the SA tube will be fitted after the SA swirl generator. This extension

will channel the hot PF and SA into the SA tube and enable the PF to enter a smaller, contained volume. This mechanism is equivalent to the ignition and combustion process in full-scale burners. To avoid the problem of gas being sucked into the SCA tube, which occurred during the gas burner experiment, the SA swirl generator will be fitted in line with the end of the extended PF/PA tube. This will block the PF and gas from being sucked into the SA tube.



FIGURE 7.5: FUTURE DESIGN OF MICRO COAL BURNER (2)

Thereafter, it might be feasible to shorten the large combustion chamber, since most of the combusting occurred in the contained volumes. To validate this remedial action, a SolidWorks CFD was done on the new proposed geometry (Figures 7.6 to 7.9).

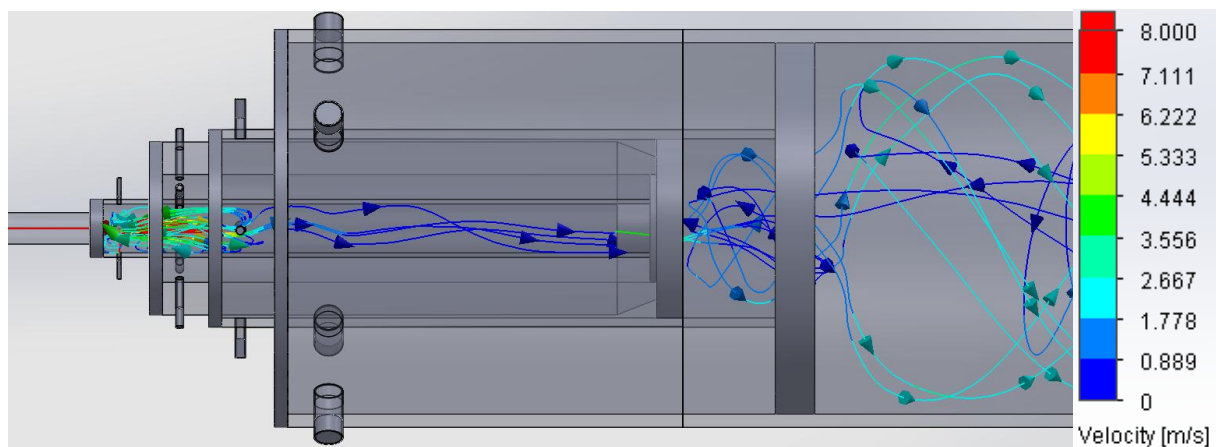


FIGURE 7.6: CFD ANALYSIS ON FUTURE DESIGN: FOUR GAS PARTICLES

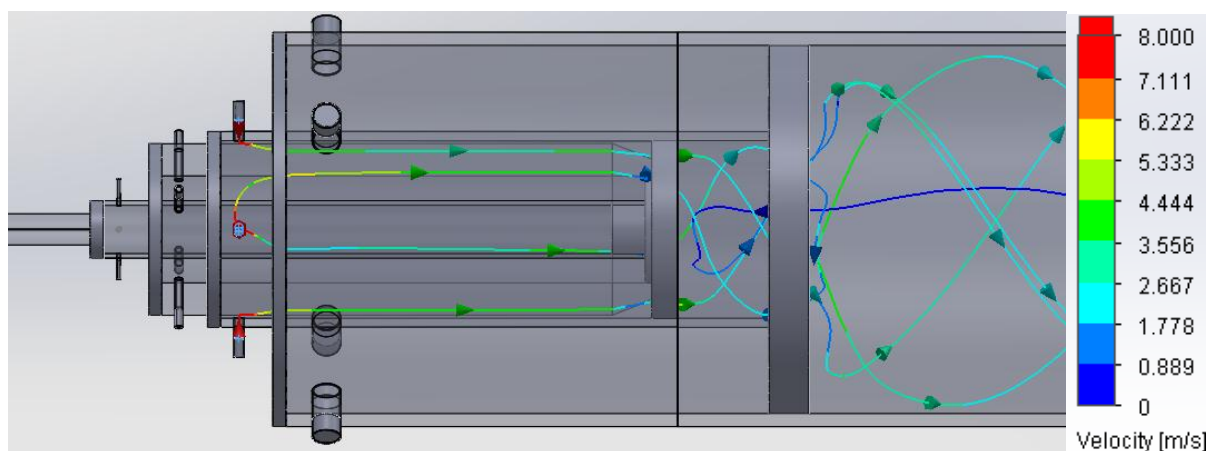


FIGURE 7.7: CFD ANALYSIS ON FUTURE DESIGN: ONE PA PARTICLE PER INLET

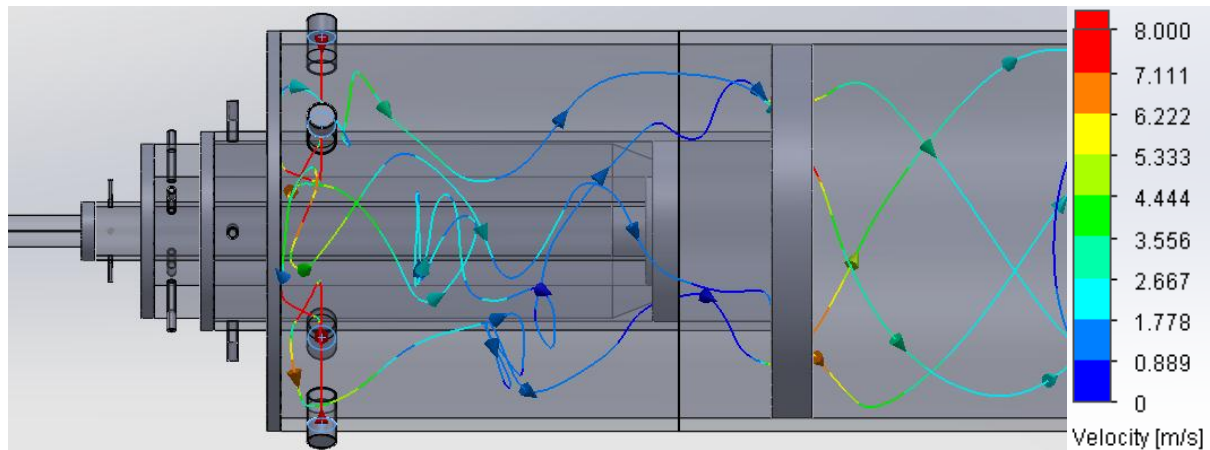


FIGURE 7.8: CFD ANALYSIS ON FUTURE DESIGN: ONE SA PARTICLE PER INLET

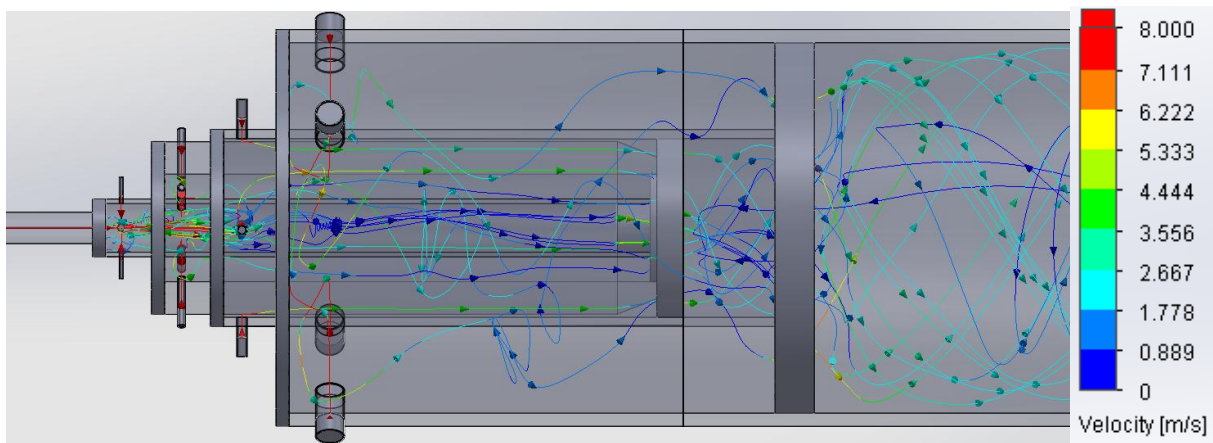


FIGURE 7.9: CFD ANALYSIS ON FUTURE DESIGN: ONE GAS, PCA, SCA, PA AND SA PARTICLE PER INLET

As can be seen in the figures above, the CFD analysis confirms the assumptions of the flow patterns. Figure 7.6 and 7.7 show the gas and PA following the same path inside the extended PF/PA tube, indicating that all the PF must enter the gas flame directly. Figure 7.7 and 7.8 show the PA and SA following the same path inside the SA extended tube. This is a good indication that the heat zone and mixing problems can be solved.

The above CFD-verified modifications will be validated in the next phase of this project. This will be an attempt to produce a micro-burner producing a self-sustaining PF flame, eventually without gas support. Thereafter it can be developed into a NCV_P coal analyser.

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9. APPENDICES

A. COAL ANALYSIS

A.1 NOMINAL GRADE COAL

TABLE A.1.1: PROXIMATE ANALYSIS NOMINAL GRADE COAL

COAL QUALITY					
GRAVIMETRIC %	SYMBOL	UNITS	AS RECEIVED	AIR DRIED	DRY BASIS
PROXIMATE ANALYSIS					
Carbon _{FIXED} (by difference)	C _{FIX}	%	43.300	44.410	46.162
Volatile matter	VM	%	22.035	22.600	23.491
Ash	Ash	%	28.465	29.195	30.347
Surface moisture	M _s	%	2.500	0.000	0.000
Inherent moisture	M _i	%	3.700	3.795	0.000
Total moisture	M _t	%	6.200	3.795	0.000
Gross calorific value	GCV _v	[MJ/kg]	20.524	21.050	21.880
Total		%	100.000	100.000	100.000

TABLE A.1.2: ELEMENTAL ANALYSIS NOMINAL GRADE COAL

COAL QUALITY					
GRAVIMETRIC %	SYMBOL	UNITS	AS RECEIVED	AIR DRIED	DRY BASIS
ELEMENTAL ANALYSIS					
Nitrogen	N	%	1.151	1.180	1.227
Oxygen (by difference)	O	%	7.166	7.350	7.640
Carbon _{TOTAL}	C _{TOTAL}	%	53.021	54.380	56.525
Ash	A	%	28.465	29.195	30.347
Sulphur	S	%	0.878	0.900	0.936
Hydrogen	H	%	3.120	3.200	3.326
Surface Moisture	M _s	%	2.500	0.000	0.000
Inherent Moisture	M _i	%	3.700	3.795	0.000
Total moisture	M _T	%	6.200	3.795	0.000
Total		%	100.000	100.000	100.000

A.2 HIGH-GRADE COAL

TABLE A.2.1: PROXIMATE ANALYSIS HIGH GRADE COAL

COAL QUALITY					
GRAVIMETRIC %	SYMBOL	UNITS	AS RECEIVED	AIR DRIED	DRY BASIS
PROXIMATE ANALYSIS					
Carbon _{FIXED} (by difference)	C _{FIX}	%	51.806	53.080	54.590
Volatile matter	VM	%	20.301	20.800	21.392
Ash	Ash	%	22.793	23.354	24.018
Surface moisture	M _s	%	2.400	0.000	0.000
Inherent moisture	M _i	%	2.700	2.766	0.000
Total moisture	M _t	%	5.100	2.766	0.000
Gross calorific value	GCV _v	[MJ/kg]	23.522	24.100	24.786
Total		%	100.000	100.000	100.000

TABLE A.2.2: ELEMENTAL ANALYSIS HIGH GRADE COAL

COAL QUALITY					
GRAVIMETRIC %	SYMBOL	UNITS	AS RECEIVED	AIR DRIED	DRY BASIS
ELEMENTAL ANALYSIS					
Nitrogen	N	%	1.405	1.440	1.481
Oxygen (by difference)	O	%	5.846	5.990	6.160
Carbon _{TOTAL}	C _{TOTAL}	%	60.941	62.440	64.216
Ash	A	%	22.793	23.354	24.018
Sulphur	S	%	0.595	0.610	0.627
Hydrogen	H	%	3.318	3.400	3.497
Surface Moisture	M _s	%	2.400	0.000	0.000
Inherent Moisture	M _i	%	2.700	2.766	0.000
Total moisture	M _T	%	5.100	2.766	0.000
Total		%	100.000	100.000	100.000

A.3 LOW-GRADE COAL

TABLE A.3.1: PROXIMATE ANALYSIS LOW GRADE COAL

COAL QUALITY					
GRAVIMETRIC %	SYMBOL	UNITS	AS RECEIVED	AIR DRIED	DRY BASIS
PROXIMATE ANALYSIS					
Carbon _{FIXED} (by difference)	C _{FIX}	%	32.968	34.370	35.691
Volatile matter	VM	%	17.074	17.800	18.484
Ash	Ash	%	42.329	44.129	45.825
Surface moisture	M _s	%	4.080	0.000	0.000
Inherent moisture	M _i	%	3.550	3.701	0.000
Total moisture	M _t	%	7.630	3.701	0.000
Gross calorific value	GCV _v	[MJ/kg]	14.251	14.857	15.428
Total		%	100.000	100.000	100.000

TABLE A.3.2: ELEMENTAL ANALYSIS LOW GRADE COAL

COAL QUALITY					
GRAVIMETRIC %	SYMBOL	UNITS	AS RECEIVED	AIR DRIED	DRY BASIS
ELEMENTAL ANALYSIS					
Nitrogen	N	%	0.873	0.910	0.945
Oxygen (by difference)	O	%	7.184	7.490	7.778
Carbon _{TOTAL}	C _{TOTAL}	%	39.078	40.740	42.306
Ash	A	%	42.329	44.129	45.825
Sulphur	S	%	0.499	0.520	0.540
Hydrogen	H	%	2.408	2.510	2.606
Surface Moisture	M _s	%	4.080	0.000	0.000
Inherent Moisture	M _i	%	3.550	3.701	0.000
Total moisture	M _T	%	7.630	3.701	0.000
Total		%	100.000	100.000	100.000

B. FLUE GAS ANALYSES OF STOICHIOMETRIC COMBUSTION

B.1 NOMINAL GRADE COAL

TABLE B.1.1: GRAVIMETRIC WET FLUE GAS ANALYSIS

GAS	[kg gas/100kg coal]	[kg/s]	% Gravimetric
CO ₂ (Ostwald CO ₂ ⁺⁺)	192.951	3.761	25.189
CO	0.078	0.002	0.010
SO ₂	1.753	0.034	0.229
NO _x	1.286	0.025	0.168
O ₂	0.000	0.000	0.000
H ₂ O from H ₂	27.883	0.543	3.640
H ₂ O from coal	6.200	0.121	0.809
H ₂ O from air	8.082	0.158	1.055
N ₂ from stoichiometric air	527.433	10.280	68.856
N ₂ from excess air	0.000	0.000	0.000
N ₂ from coal	0.332	0.006	0.043
Total Flue Gas	765.999	14.929	100.000
Total H ₂ O	42.165	0.822	5.505
Total N ₂	527.765	10.286	68.899

TABLE B.1.2: VOLUMETRIC DRY FLUE GAS ANALYSIS

GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric
CO ₂ (Ostwald CO ₂ ⁺⁺)	25.189	44.010	0.572	18.831
CO	0.010	28.010	0.000	0.012
SO ₂	0.229	64.059	0.004	0.118
NO _x	0.168	46.006	0.004	0.120
O ₂	0.000	31.999	0.000	0.000
H ₂ O from H ₂	0.000	18.015	0.000	0.000
H ₂ O from coal	0.000	18.015	0.000	0.000
H ₂ O from air	0.000	18.015	0.000	0.000
N ₂ from stoichiometric air	68.856	28.013	2.458	80.868
N ₂ from excess air	0.000	28.013	0.000	0.000
N ₂ from coal	0.043	28.013	0.002	0.051
Total Flue Gas	94.495	Molecular mass of Flue gas [kg/kmole]	3.039	100.000
Total H ₂ O	0.000		R _{FLUE GAS} [kJ/kg-K]	0.000
Total N ₂	68.899		0.25271	80.919

B.2 HIGH-GRADE COAL

TABLE B.2.1: GRAVIMETRIC WET FLUE GAS ANALYSIS

GAS	[kg gas/100 kg coal]	[kg/s]	% Gravimetric
CO ₂ (Ostwald CO ₂ ⁺⁺)	222.195	3.771	25.358
CO	0.090	0.002	0.010
SO ₂	1.190	0.020	0.136
NO _x	1.480	0.025	0.169
O ₂	0.000	0.000	0.000
H ₂ O from H ₂	29.657	0.503	3.385
H ₂ O from coal	5.100	0.087	0.582
H ₂ O from air	9.297	0.158	1.061
N ₂ from stoichiometric air	606.751	10.297	69.246
N ₂ from excess air	0.000	0.000	0.000
N ₂ from coal	0.464	0.008	0.053
Total Flue Gas	876.222	14.869	100.000
Total H ₂ O	44.054	0.748	5.028
Total N ₂	607.214	10.304	69.299

TABLE B.2.2: VOLUMETRIC DRY FLUE GAS ANALYSIS

GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric
CO ₂ (Ostwald CO ₂ ⁺⁺)	25.358	44.010	0.576	18.854
CO	0.010	28.010	0.000	0.012
SO ₂	0.136	64.059	0.002	0.069
NO _x	0.169	46.006	0.004	0.120
O ₂	0.000	31.999	0.000	0.000
H ₂ O from H ₂	0.000	18.015	0.000	0.000
H ₂ O from coal	0.000	18.015	0.000	0.000
H ₂ O from air	0.000	18.015	0.000	0.000
N ₂ from stoichiometric air	69.246	28.013	2.472	80.883
N ₂ from excess air	0.000	28.013	0.000	0.000
N ₂ from coal	0.053	28.013	0.002	0.062
Total Flue Gas	94.972	Molecular mass of Flue gas [kg/kmole]	3.056	100.000
Total H ₂ O	0.000		R _{FLUE GAS} [kJ/kg-K]	0.000
Total N ₂	69.299		0.25410	80.945

B.3 LOW-GRADE COAL

TABLE B.3.1: GRAVIMETRIC WET FLUE GAS ANALYSIS

GAS	[kg gas/100kg coal]	[kg/s]	% Gravimetric
CO ₂ (Ostwald CO ₂ ⁺⁺)	141.313	3.775	25.181
CO	0.057	0.002	0.010
SO ₂	0.997	0.027	0.178
NO _x	0.935	0.025	0.167
O ₂	0.000	0.000	0.000
H ₂ O from H ₂	21.517	0.575	3.834
H ₂ O from coal	7.630	0.204	1.360
H ₂ O from air	5.862	0.157	1.045
N ₂ from stoichiometric air	382.605	10.220	68.177
N ₂ from excess air	0.000	0.000	0.000
N ₂ from coal	0.278	0.007	0.050
Total Flue Gas	561.193	14.990	100.000
Total H ₂ O	35.009	0.935	6.238
Total N ₂	382.883	10.227	68.227

TABLE B.3.2: GRAVIMETRIC WET FLUE GAS ANALYSIS

GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric
CO ₂ (Ostwald CO ₂ ⁺⁺)	25.181	44.010	0.572	18.981
CO	0.010	28.010	0.000	0.012
SO ₂	0.178	64.059	0.003	0.092
NO _x	0.167	46.006	0.004	0.120
O ₂	0.000	31.999	0.000	0.000
H ₂ O from H ₂	0.000	18.015	0.000	0.000
H ₂ O from coal	0.000	18.015	0.000	0.000
H ₂ O from air	0.000	18.015	0.000	0.000
N ₂ from stoichiometric air	68.177	28.013	2.434	80.736
N ₂ from excess air	0.000	28.013	0.000	0.000
N ₂ from coal	0.050	28.013	0.002	0.059
Total Flue Gas	93.762	Molecular mass of Flue gas [kg/kmole]	3.014	100.000
Total H ₂ O	0.000		R _{FLUE GAS} [kJ/kg-K]	0.000
Total N ₂	68.227		0.25063	80.795

C. FLUE GAS ANALYSIS OF HYPER-STOICHIOMETRIC COMBUSTION

C.1 NOMINAL GRADE COAL

TABLE C.1.1: GRAVIMETRIC WET FLUE GAS ANALYSIS

GAS	[kg gas/100 kg coal]	[kg/s]	% Gravimetric
CO ₂ (Ostwald CO ₂ ⁺)	192.951	3.761	21.377
CO	0.078	0.002	0.009
SO ₂	1.753	0.034	0.194
NO _x	1.286	0.025	0.143
O ₂ (Ostwald O ₂ ⁺)	31.318	0.610	3.470
H ₂ O from H ₂	27.883	0.543	3.089
H ₂ O from coal	6.200	0.121	0.687
H ₂ O from air	9.671	0.188	1.071
N ₂ from stoichiometric air	527.433	10.280	58.434
N ₂ from excess air	103.708	2.021	11.490
N ₂ from coal	0.332	0.006	0.037
Total Flue Gas	902.615	17.592	100.000
Total H ₂ O	43.754	0.853	4.847
Total N ₂	631.473	12.307	69.960

TABLE C.1.2: VOLUMETRIC DRY FLUE GAS ANALYSIS

GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric
CO ₂ (Ostwald CO ₂ ⁺)	21.377	44.010	0.486	15.679
CO	0.009	28.010	0.000	0.0100000
SO ₂	0.194	64.059	0.003	0.0978815
NO _x	0.143	46.006	0.003	0.1000000
O ₂ (Ostwald O ₂ ⁺)	3.470	31.999	0.108	3.500
H ₂ O from H ₂	0.000	18.015	0.000	0.000
H ₂ O from coal	0.000	18.015	0.000	0.000
H ₂ O from air	0.000	18.015	0.000	0.000
N ₂ from stoichiometric air	58.434	28.013	2.086	67.332
N ₂ from excess air	11.490	28.013	0.410	13.239
N ₂ from coal	0.037	28.013	0.001	0.042
Total Flue Gas	95.153	Molecular mass of Flue gas [kg/kmole]	3.098	100.000
Total H ₂ O	0.000		R _{FLUE GAS} [kJ/kg-K]	0.000
Total N ₂	69.960		0.25758	80.613

C.2 HIGH-GRADE COAL

TABLE C.2.1: GRAVIMETRIC WET FLUE GAS ANALYSIS

GAS	[kg gas/100kg coal]	[kg/s]	% Gravimetric
CO ₂ (Ostwald CO ₂ ⁺)	222.195	3.771	21.502
CO	0.090	0.002	0.009
SO ₂	1.190	0.020	0.115
NO _x	1.480	0.025	0.143
O ₂ (Ostwald O ₂ ⁺)	36.024	0.611	3.486
H ₂ O from H ₂	29.657	0.503	2.870
H ₂ O from coal	5.100	0.087	0.494
H ₂ O from air	11.125	0.189	1.077
N ₂ from stoichiometric air	606.751	10.297	58.716
N ₂ from excess air	119.289	2.024	11.544
N ₂ from coal	0.464	0.008	0.045
Total Flue Gas	1033.363	17.536	100.000
Total H ₂ O	45.881	0.779	4.440
Total N ₂	726.504	12.329	70.305

TABLE C.2.2: VOLUMETRIC DRY FLUE GAS ANALYSIS

GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric
CO ₂ (Ostwald CO ₂ ⁺)	21.502	44.010	0.489	15.698
CO	0.009	28.010	0.000	0.0100000
SO ₂	0.115	64.059	0.002	0.0577384
NO _x	0.143	46.006	0.003	0.1000000
O ₂ (Ostwald O ₂ ⁺)	3.486	31.999	0.109	3.500
H ₂ O from H ₂	0.000	18.015	0.000	0.000
H ₂ O from coal	0.000	18.015	0.000	0.000
H ₂ O from air	0.000	18.015	0.000	0.000
N ₂ from stoichiometric air	58.716	28.013	2.096	67.343
N ₂ from excess air	11.544	28.013	0.412	13.240
N ₂ from coal	0.045	28.013	0.002	0.051
Total Flue Gas	95.560	Molecular mass of Flue gas [kg/kmole]	3.112	100.000
Total H ₂ O	0.000		R _{FLUE GAS} [kJ/kg-K]	0.000
Total N ₂	70.305		0.25878	80.634

C.3 LOW-GRADE COAL

TABLE C.3.1: GRAVIMETRIC WET FLUE GAS ANALYSIS

GAS	[kg gas/100kg coal]	[kg/s]	% Gravimetric
CO ₂ (Ostwald CO ₂ ⁺)	141.313	3.775	21.396
CO	0.057	0.002	0.009
SO ₂	0.997	0.027	0.151
NO _x	0.935	0.025	0.142
O ₂ (Ostwald O ₂ ⁺)	22.755	0.608	3.445
H ₂ O from H ₂	21.517	0.575	3.258
H ₂ O from coal	7.630	0.204	1.155
H ₂ O from air	7.017	0.187	1.062
N ₂ from stoichiometric air	382.605	10.220	57.931
N ₂ from excess air	75.351	2.013	11.409
N ₂ from coal	0.278	0.007	0.042
Total Flue Gas	660.454	17.641	100.000
Total H ₂ O	36.164	0.966	5.476
Total N ₂	458.234	12.240	69.382

TABLE C.3.2: VOLUMETRIC DRY FLUE GAS ANALYSIS

GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric
CO ₂ (Ostwald CO ₂ ⁺)	21.396	44.010	0.486	15.804
CO	0.009	28.010	0.000	0.0100000
SO ₂	0.151	64.059	0.002	0.0765730
NO _x	0.142	46.006	0.003	0.1000000
O ₂ (Ostwald O ₂ ⁺)	3.445	31.999	0.108	3.500
H ₂ O from H ₂	0.000	18.015	0.000	0.000
H ₂ O from coal	0.000	18.015	0.000	0.000
H ₂ O from air	0.000	18.015	0.000	0.000
N ₂ from stoichiometric air	57.931	28.013	2.068	67.222
N ₂ from excess air	11.409	28.013	0.407	13.239
N ₂ from coal	0.042	28.013	0.002	0.049
Total Flue Gas	94.524	Molecular mass of Flue gas [kg/kmole]	3.076	100.000
Total H ₂ O	0.000		R _{FLUE GAS} [kJ/kg-K]	0.000
Total N ₂	69.382		0.25578	80.510

D. MACHINE DRAWINGS

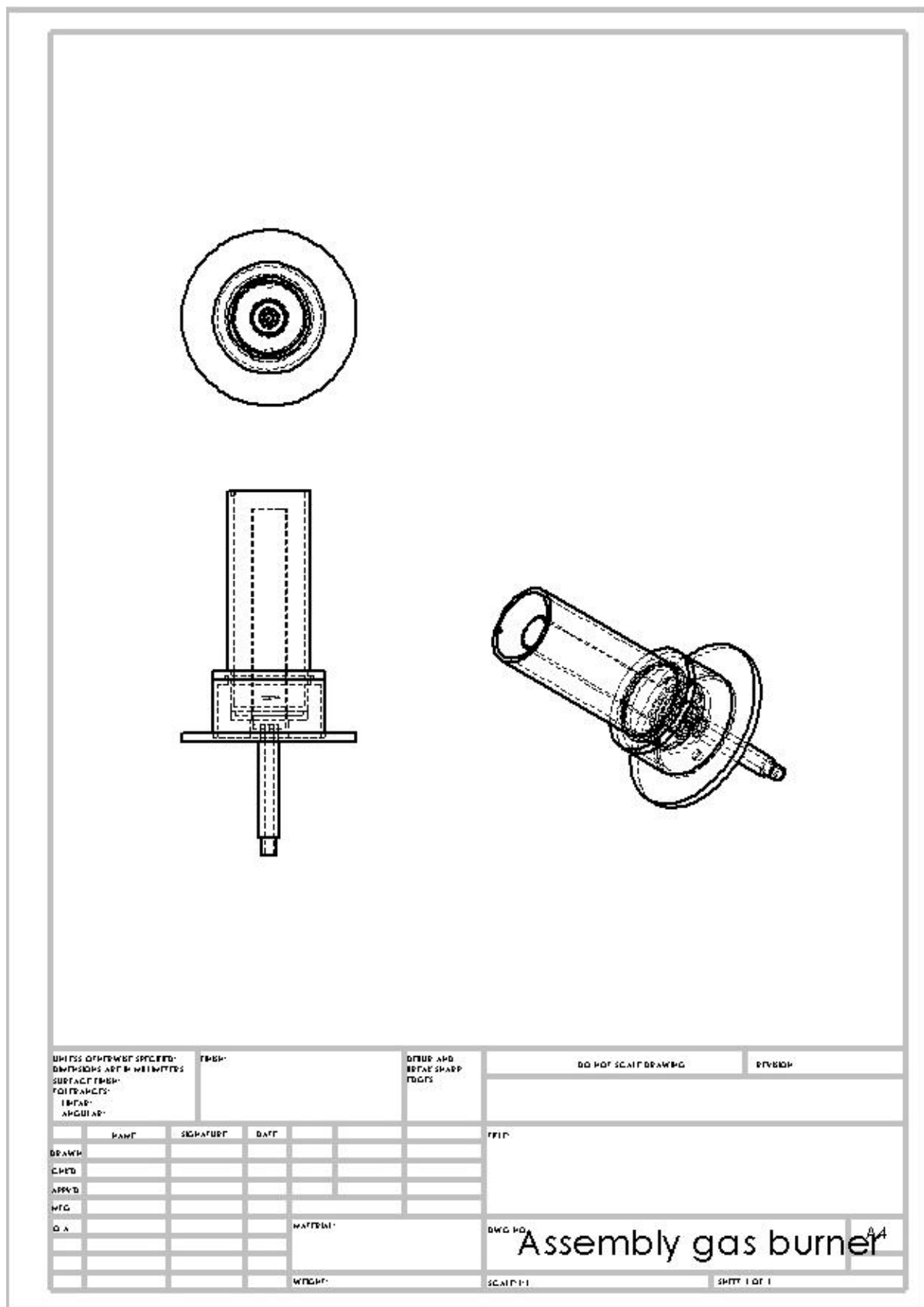


FIGURE D.1: ASSEMBLY DRAWING OF EXPERIMENTAL GAS BURNER

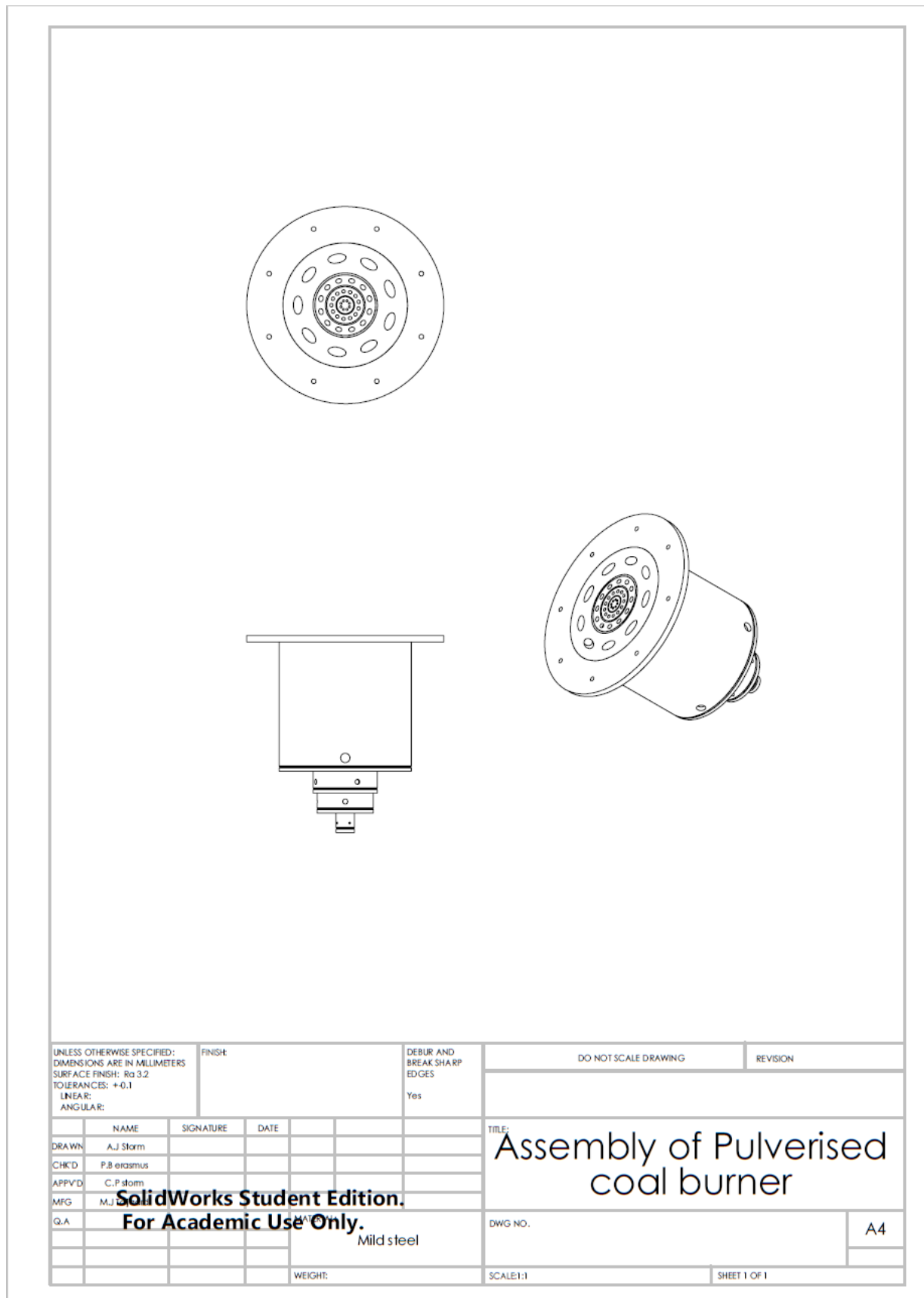


FIGURE D.2: ASSEMBLY DRAWING OF MICRO COAL BURNER

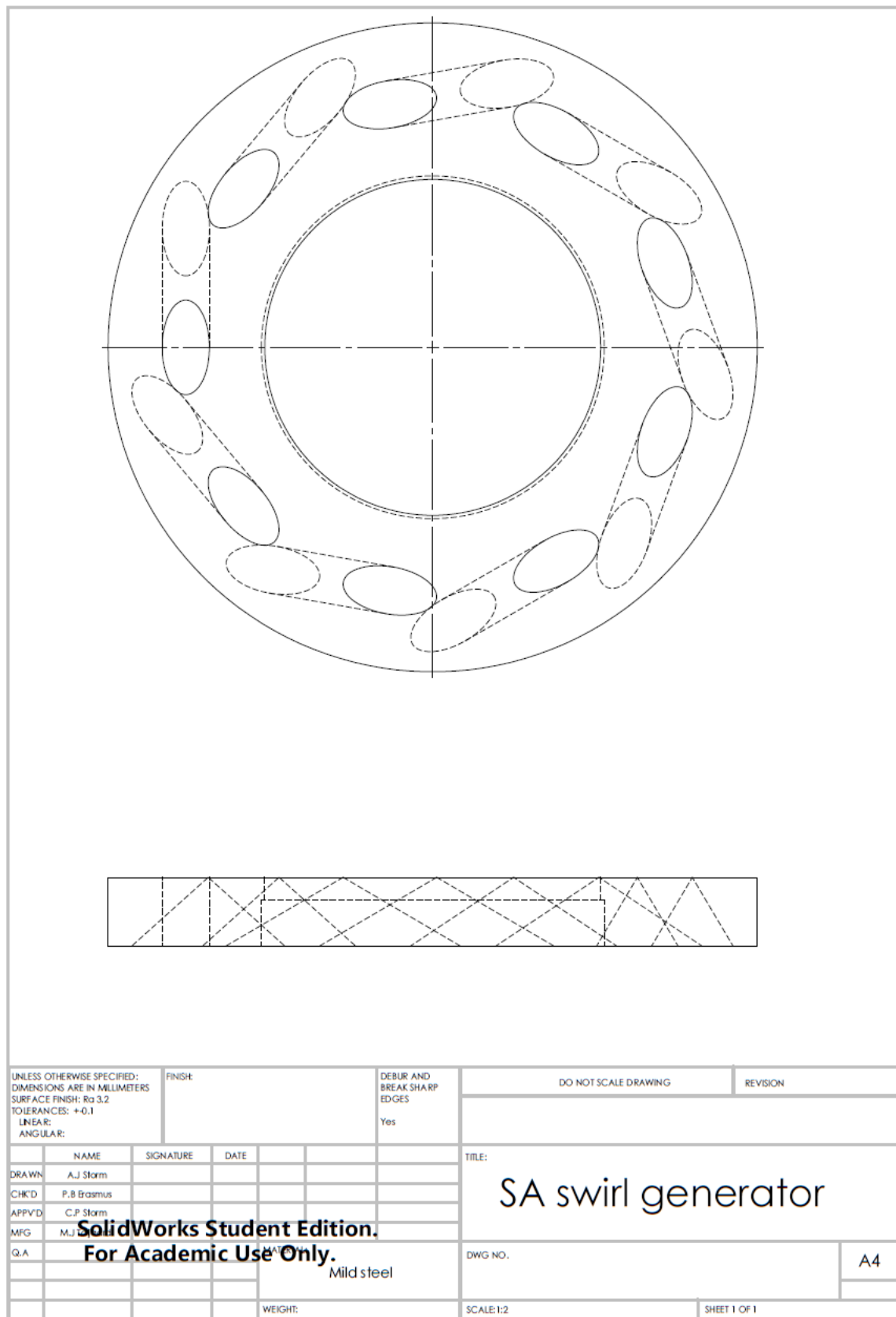


FIGURE D.3: MACHINE DRAWING OF SA SWIRL GENERATOR

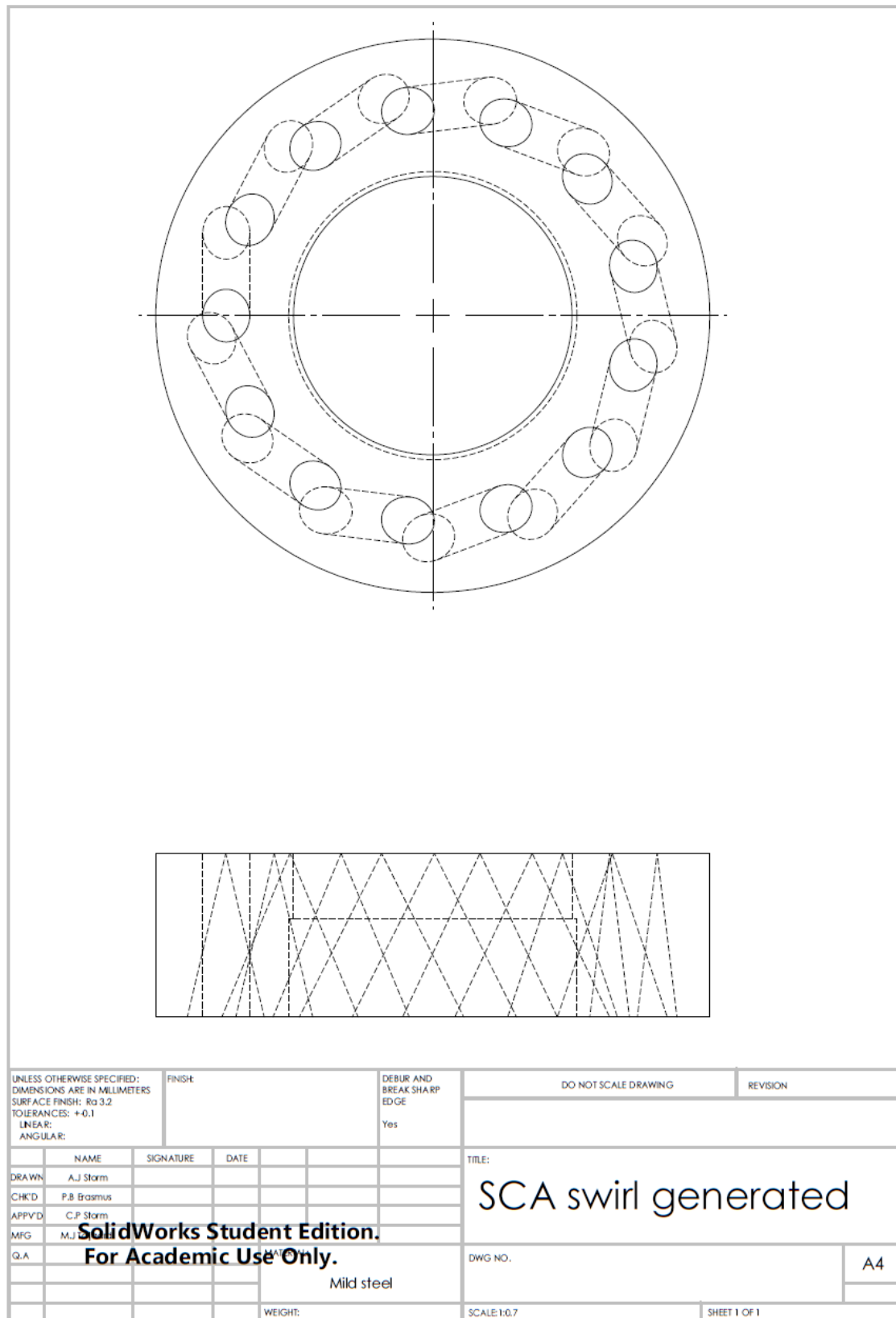


FIGURE D.4: MACHINE DRAWING OF SCA SWIRL GENERATOR

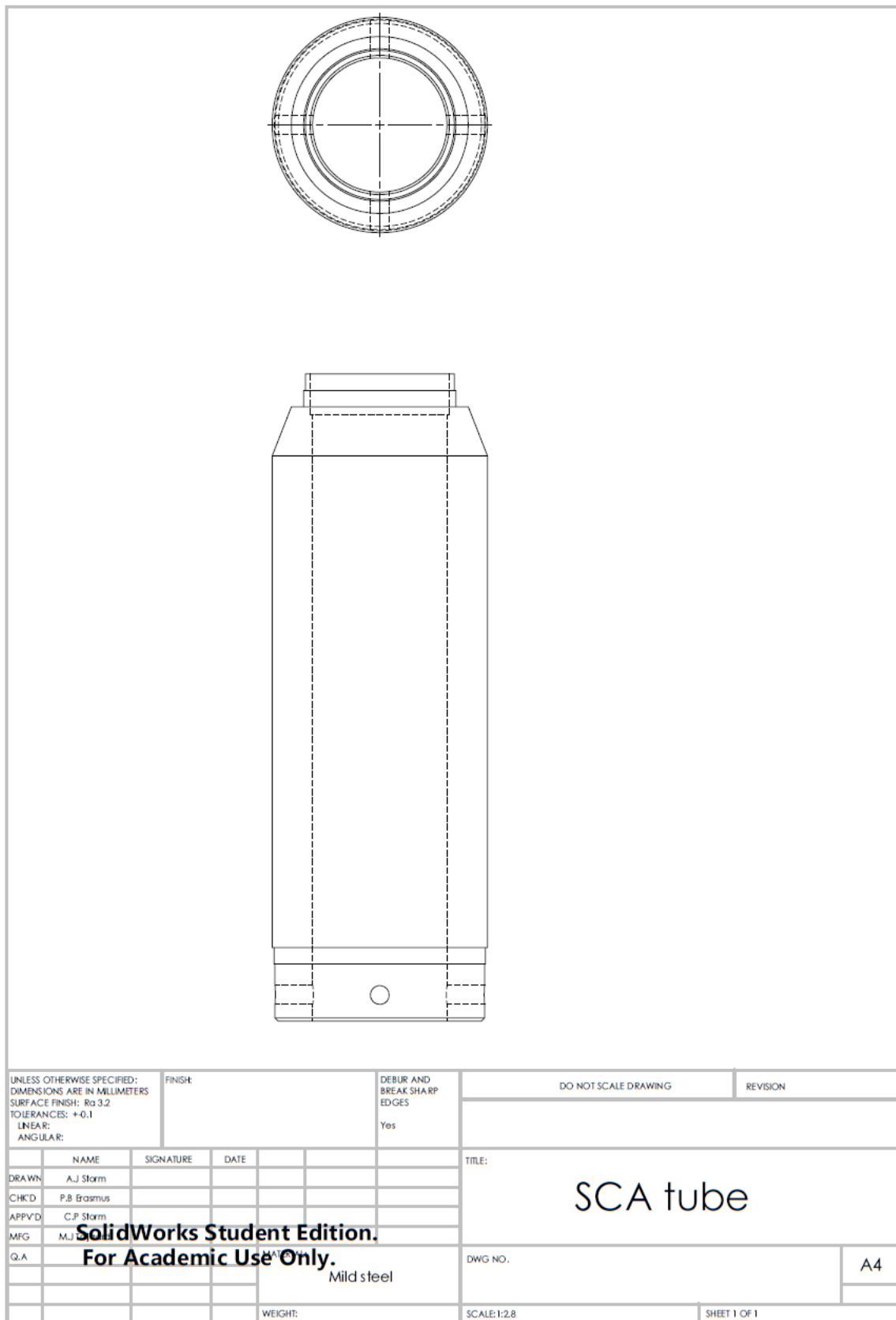


FIGURE D.5: MACHINE DRAWING OF SCA TUBE

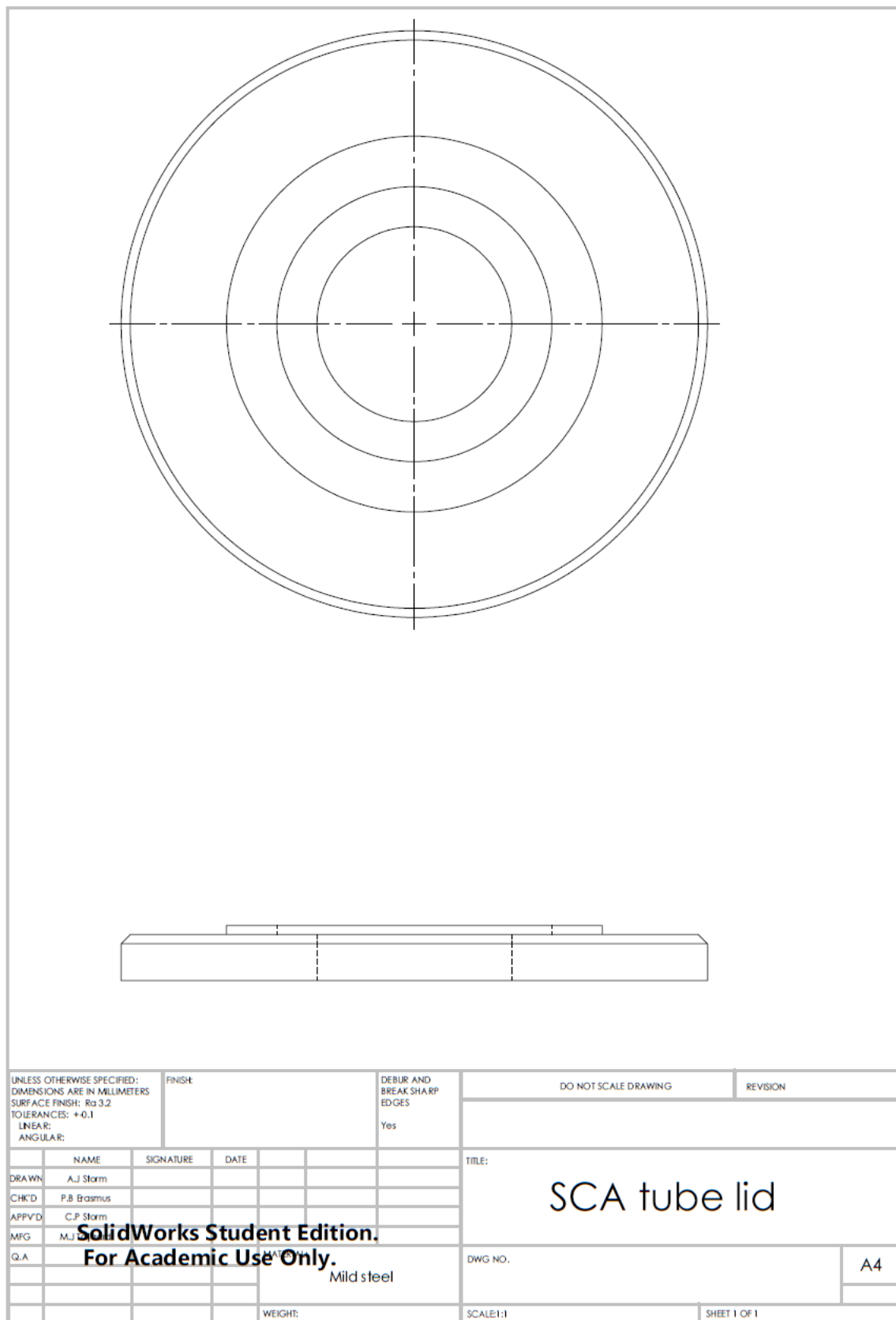
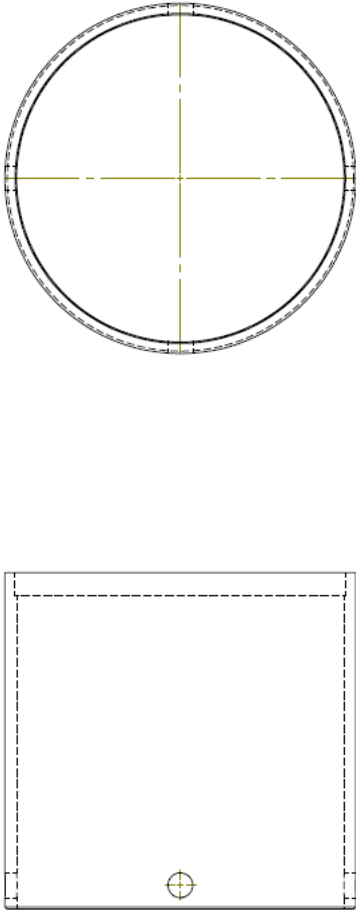


FIGURE D.6: MACHINE DRAWING OF SCA TUBE LID



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: Ra 3.2 TOLERANCES: +0.1 LINEAR: ANGULAR:		FINISH:		DEBUR AND BREAK SHARP EDGES Yes		DO NOT SCALE DRAWING		REVISION	
DRAWN: A.J. Storm		SIGNATURE:		DATE:		TITLE: SA tube			
CHK'D: P.B. Erasmus									
APP'VD: C.P. Storm									
MFG: M.J. ...									
Q.A:									
				Mild steel		DWG NO.		A4	
				WEIGHT:		SCALE: 1:5		SHEET 1 OF 1	

FIGURE D.7: MACHINE DRAWING OF SA TUBE

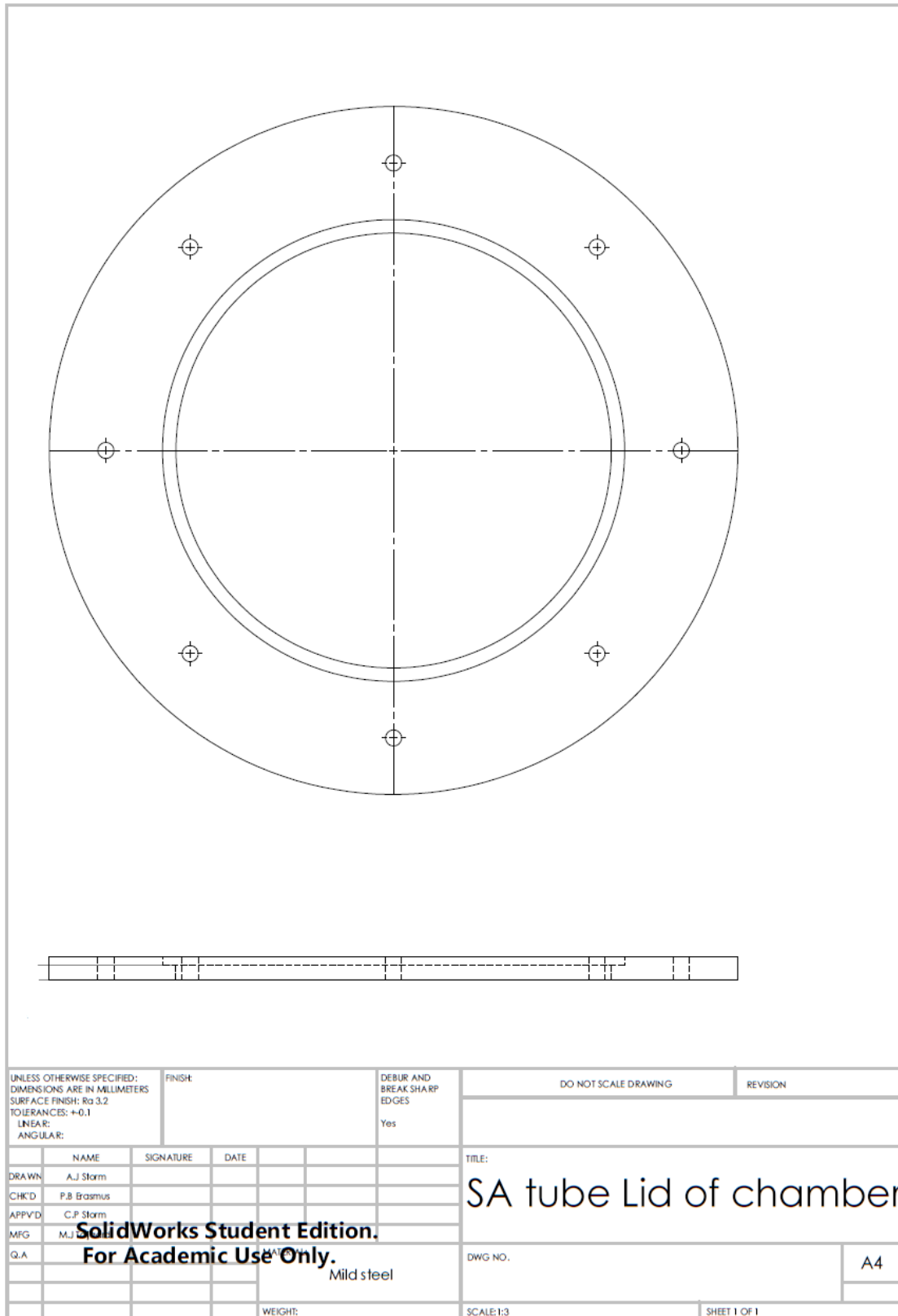


FIGURE D.8: MACHINE DRAWING OF SA TUBE LID OF CHAMBER

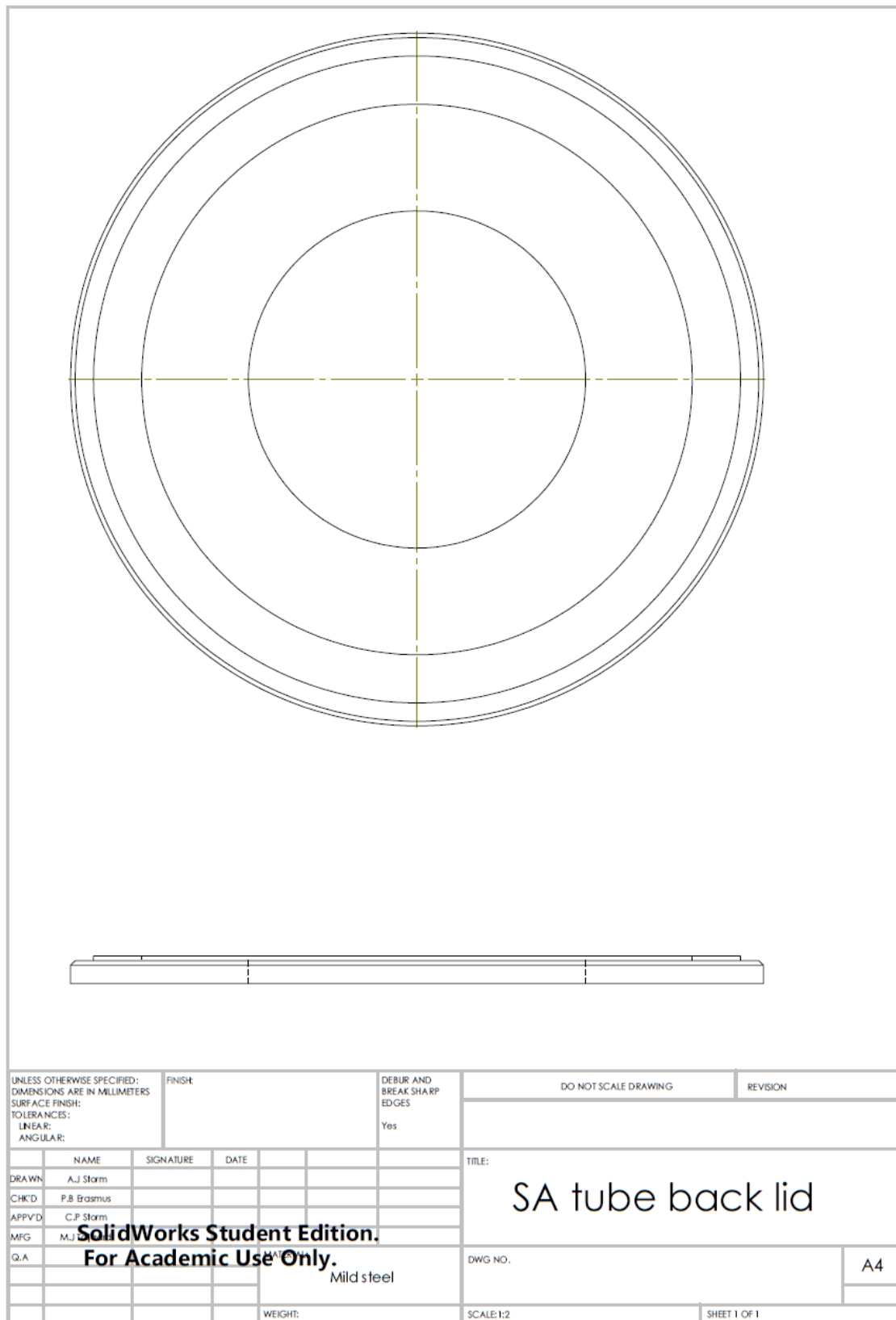
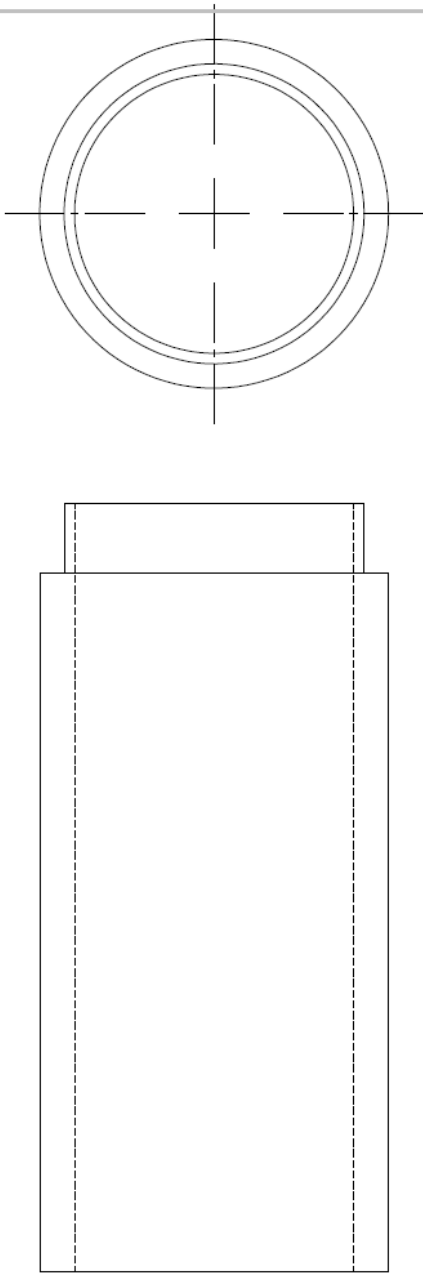


FIGURE D.9: MACHINE DRAWING OF SA TUBE BACK LID



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: Ra 3.2 TOLERANCES: +0.1 LINEAR: ANGULAR:		FINISH:		DEBUR AND BREAK SHARP EDGES Yes		DO NOT SCALE DRAWING		REVISION	
DRAWN: A.J. Storm		SIGNATURE		DATE		TITLE:			
CHK'D: P.B. Erasmus						SA inlet nozzle			
APPVD: C.P. Storm									
MFG: M.J.									
Q.A:									
				Mild steel		DWG NO.		A4	
				WEIGHT:		SCALE 2.5:1		SHEET 1 OF 1	

FIGURE D.10: MACHINE DRAWING OF SA INLET NOZZLE

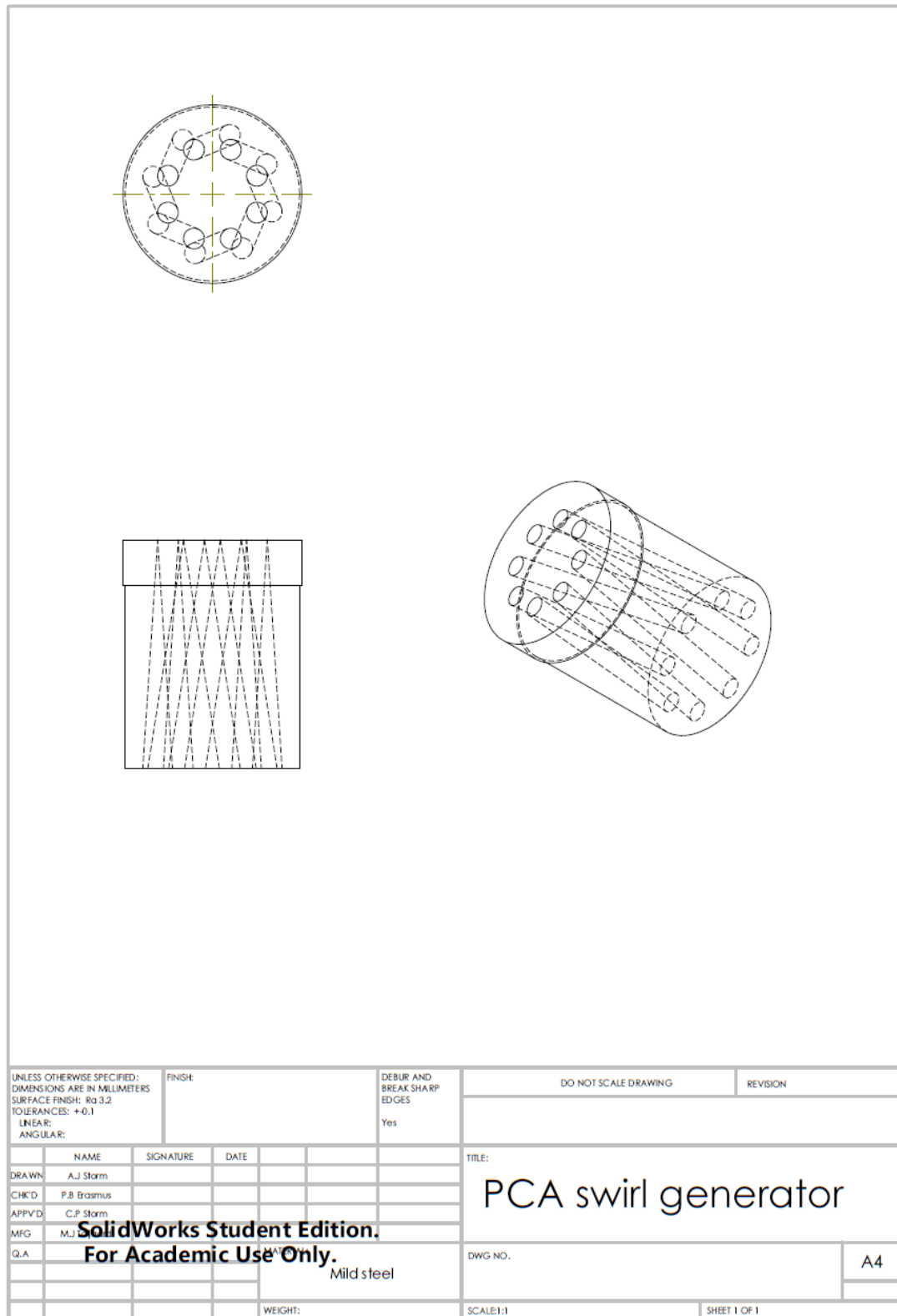


FIGURE D.11: MACHINE DRAWING OF PCA SWIRL GENERATOR

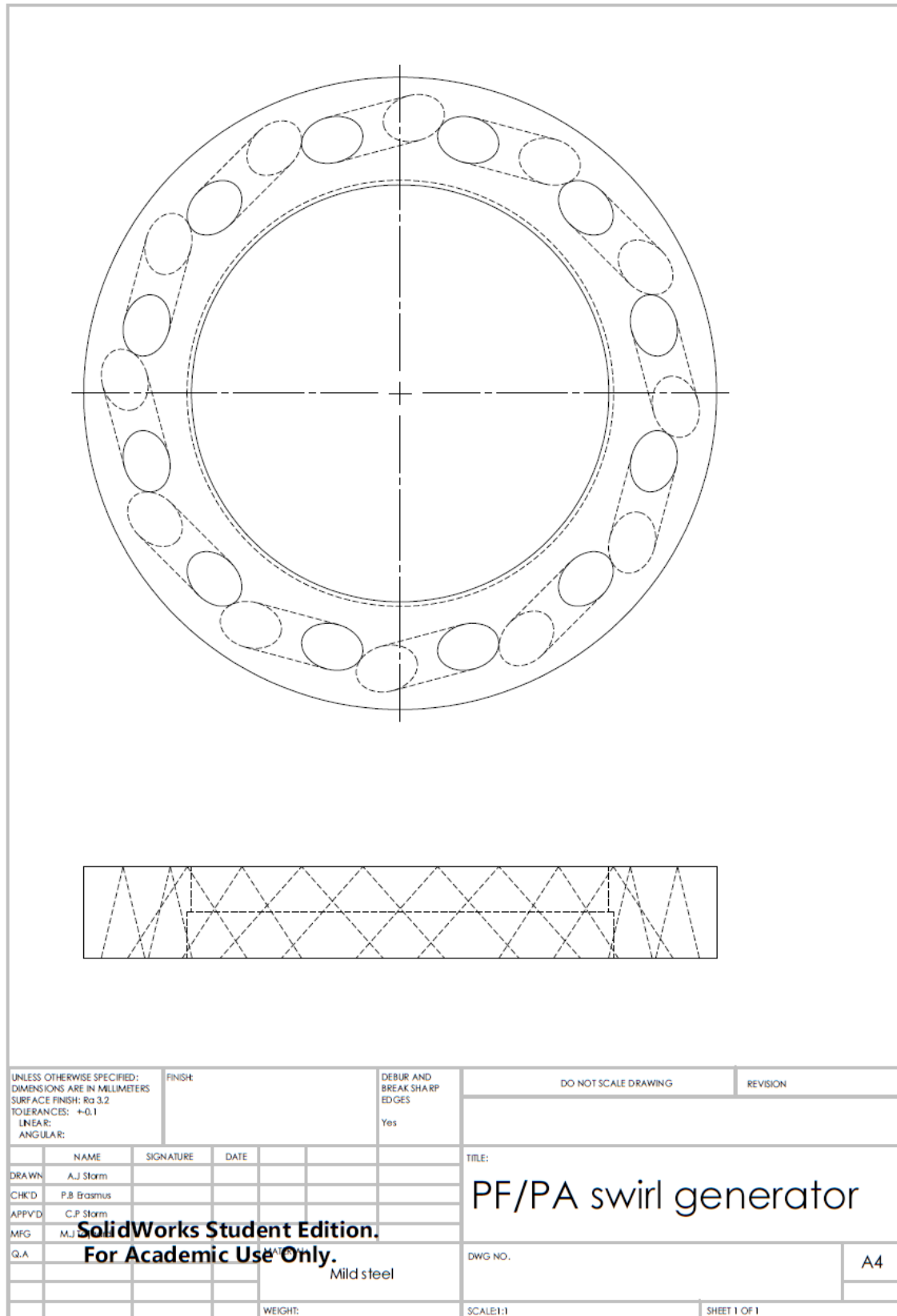


FIGURE D.12: MACHINE DRAWING OF PF/PA SWIRL GENERATOR

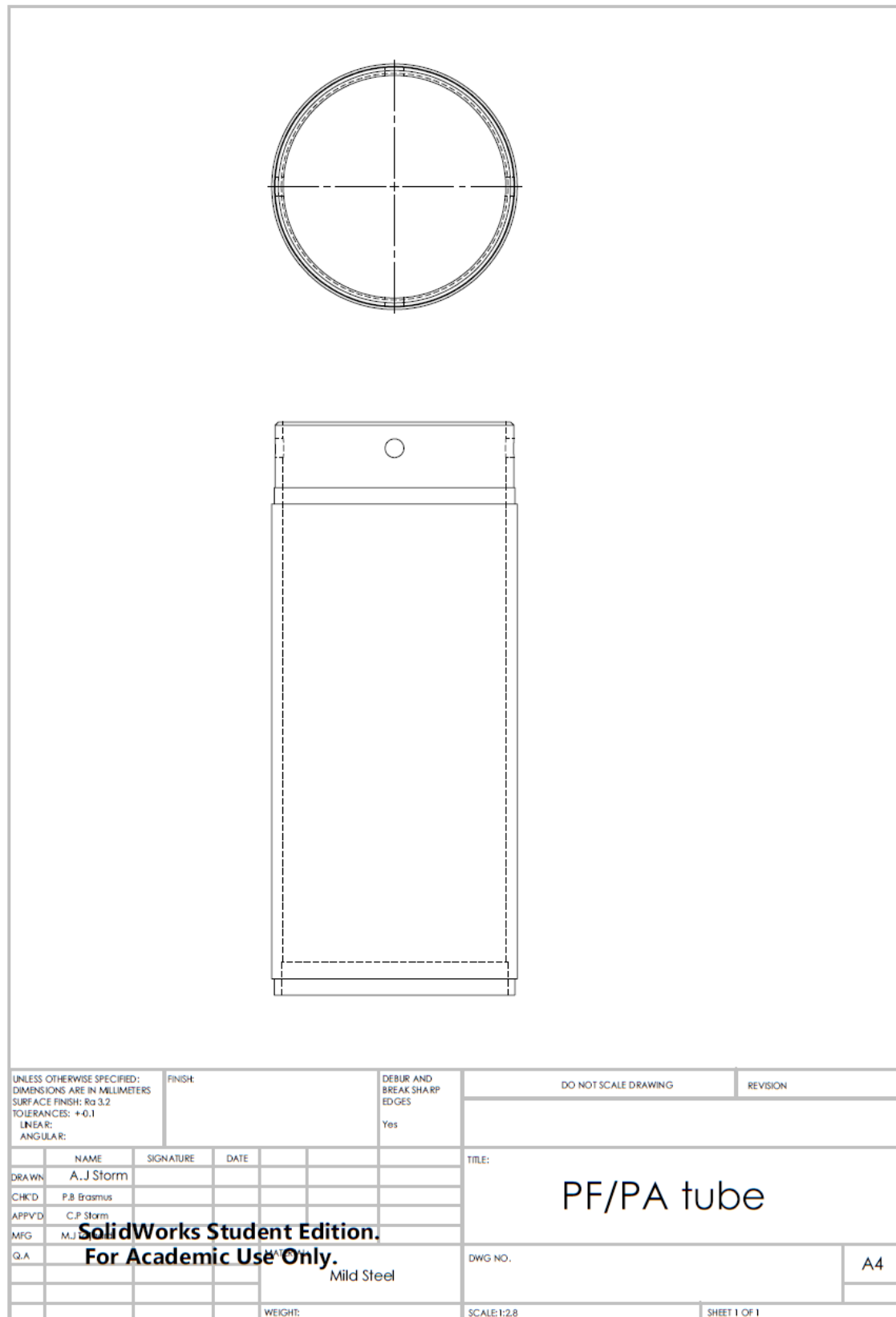


FIGURE D.13: MACHINE DRAWING OF PF/PA TUBE

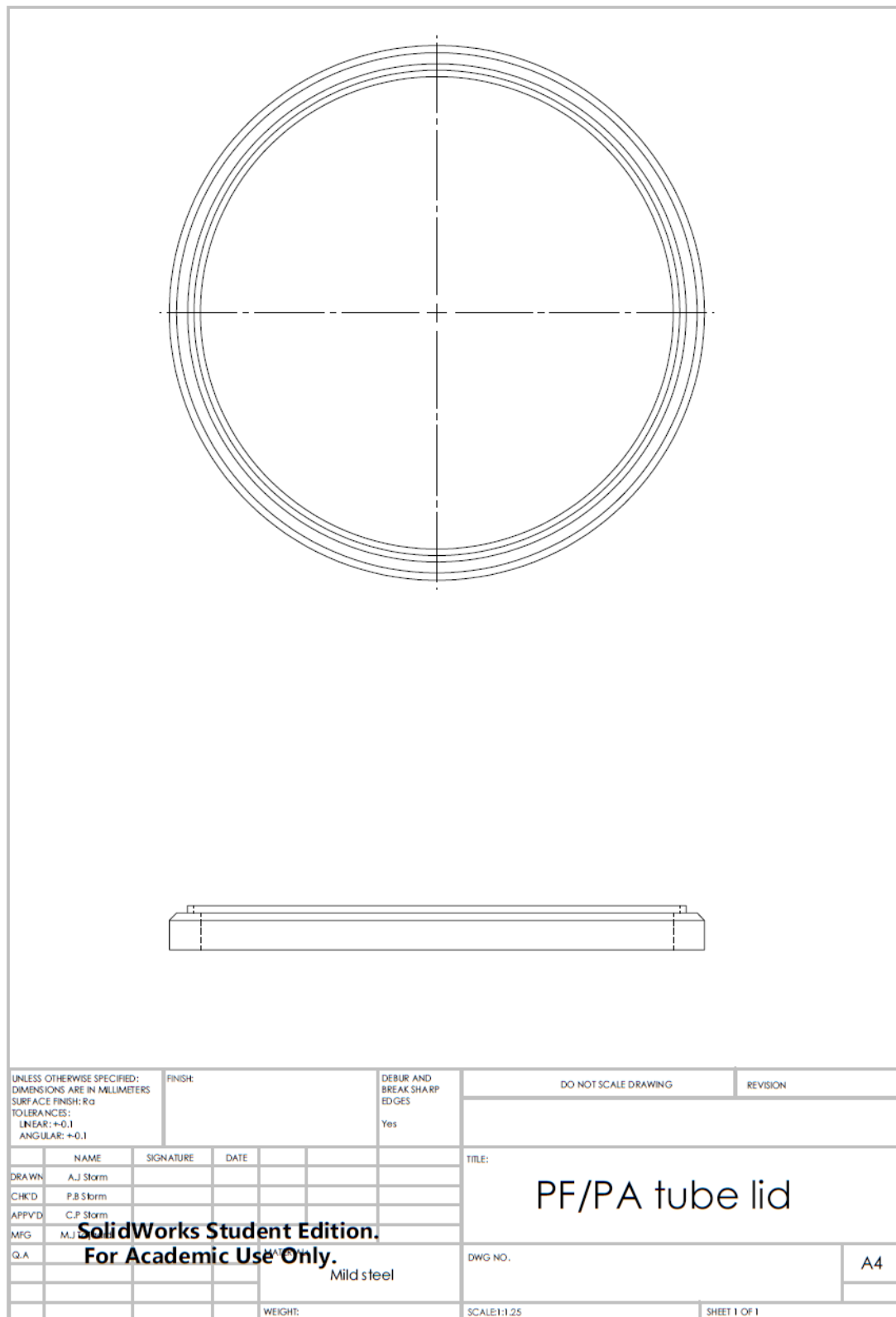


FIGURE D.14: MACHINE DRAWING OF PF/PA TUBE LID

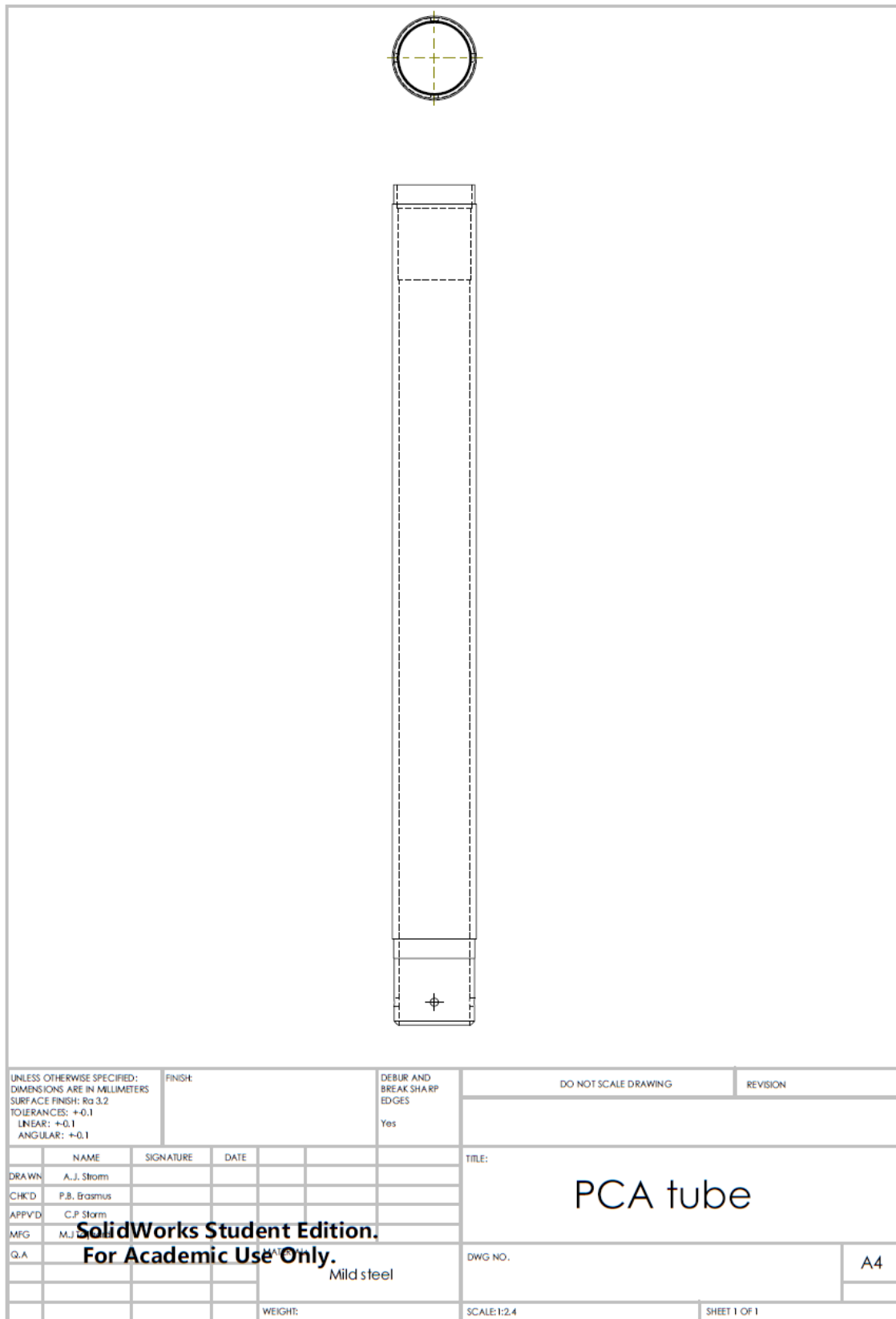


FIGURE D.15: MACHINE DRAWING OF PCA TUBE

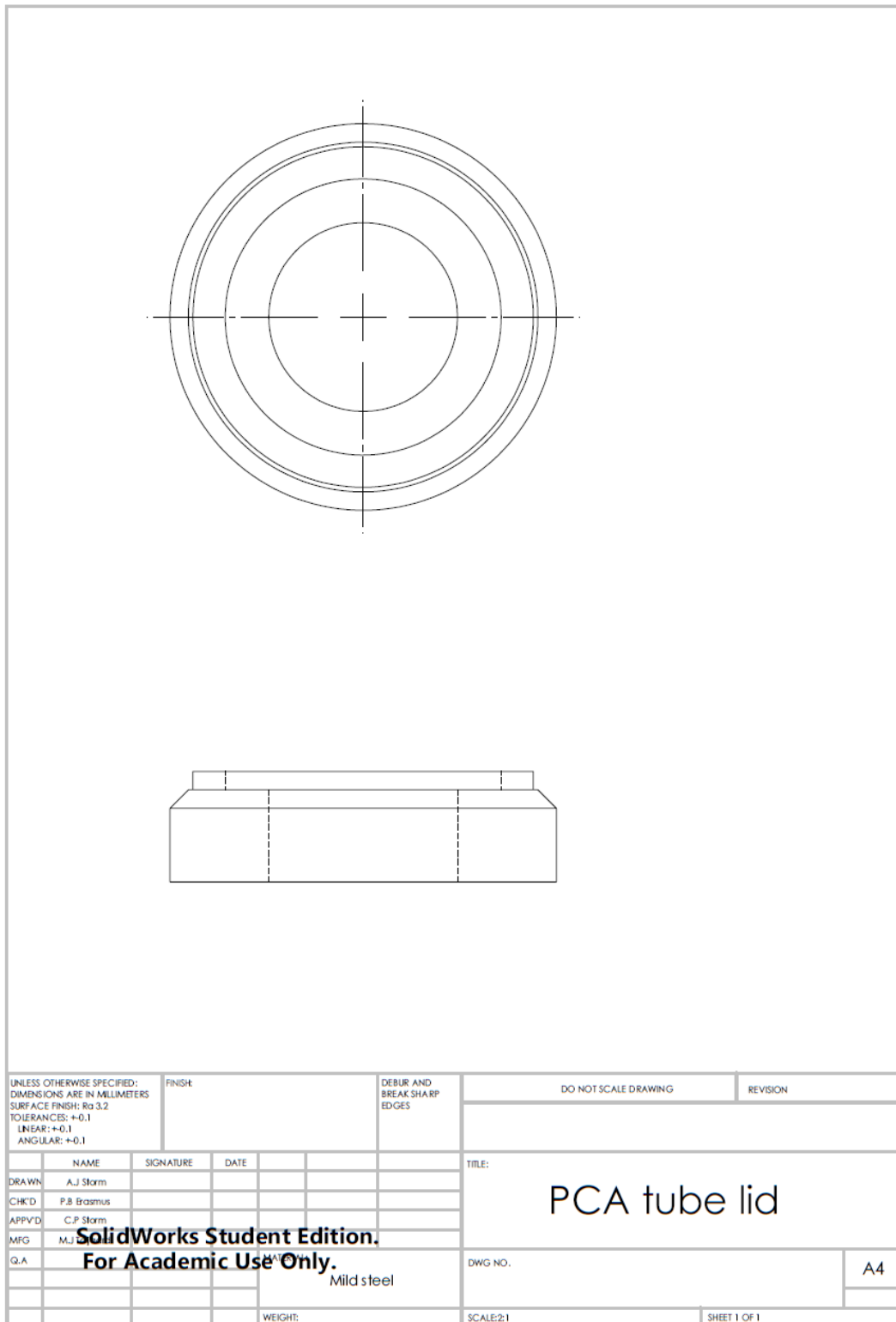
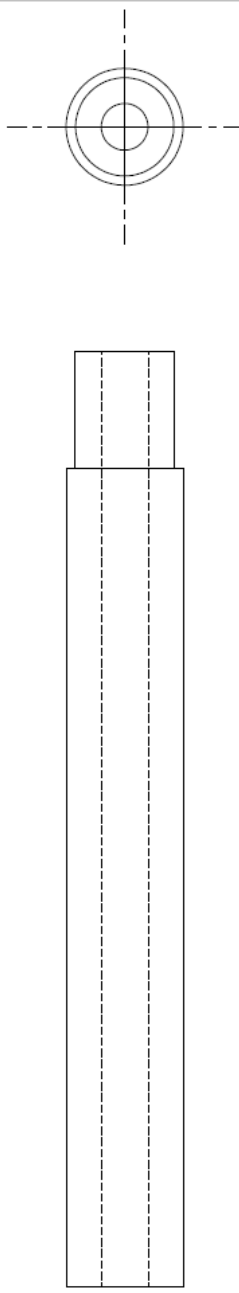


FIGURE D.16: MACHINE DRAWING OF PCA TUBE LID



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: Ra 3.2 TOLERANCES: 0.1 LINEAR: ANGULAR:		FINISH:		DEBUR AND BREAK SHARP EDGES Yes		DO NOT SCALE DRAWING		REVISION	
DRAWN: A.J. Storm		SIGNATURE		DATE		TITLE:		PCA inlet nozzle	
CHK'D: P.B. Erasmus									
APPV'D: C.P. Storm									
MFG: M.J. D. ...									
Q.A:						Mild steel		A4	
						WEIGHT:		SCALE: 4:1	
								SHEET 1 OF 1	

FIGURE D.17: MACHINE DRAWING OF PCA INLET NOZZLE