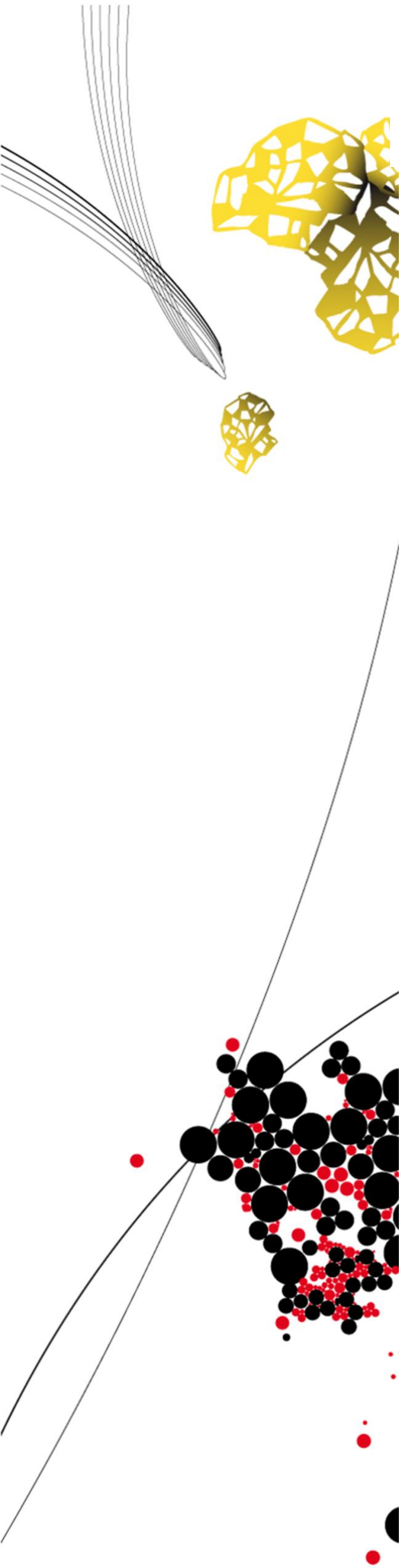

UNIVERSITY OF TWENTE.

STORK
A Fluor Company

Internship report :

Development of an hydrogen combustion plant

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from: 27th September 2018
to: 21st December 2018

Summary

1. Abstract

The project aims to analyze a pilot plant for the direct combustion of hydrogen that must be realized by the company Stork Thermeq B.V. together with DNVGL. The system in consideration has a nominal power of 500 kW (LHV) and is designed to operate in a power range between 10% and 100%. Under stationary and standard conditions, hydrogen is burned with pure oxygen, produced by electrolysis of water, the reaction creates steam, which is recirculated to cool the combustion chamber at temperatures that are sustainable for the equipment. Moreover, the recycled steam is used to cool the combustion to avoid the production of NOx pollutants that would be created if the reaction was cooled with air.

The plant can also operate with air or a mixture of nitrogen and oxygen introduced separately. The system also has the possibility to be powered by natural gas, in order to analyze the differences in combustion of the two different types of fuel and to implement the system for future commercialization. Since Stork Thermeq B.V. is a leading company in the production of boilers and burners, these two components will be analyzed in detail. The target is to give a flexible plant which can offer the chance to substitute the single parts of the system and measure the differences. The project therefore focuses on analyzing the plant from the point of view of the energy and mass balance in various conditions, standard and transition; analyzes various possibilities that can be developed. After defining a set-up for the plant, the individual equipment are sized and designed to be printed in 3D, P&ID's and PFD's are drawn for standard conditions. Finally, a proposal for the internal HAZOP analysis and a related philosophy of operation of the plant are drawn up.

2. Introduction

Due to the carbon-free nature of hydrogen gas, hydrogen has been considered as future energy in replacement of hydrocarbon resources. Hydrogen could be produced by varies means including water electrolysis, reforming, thermocatalytic cracking, thermolysis, biotechnology, photonic and etc. It can be stored and when energy is required it can be used for combustion in a boiler such that the resulting flue-gas only consists of steam. The direct combustion of hydrogen can be used to generate electricity by a semi-closed hydrogen combustion turbine, which emits no COx, SOx, or NOx [1-2]. If such a system could be realized it allows to have a production of energy with a clean combustion. The applications of H2 in various energy devices, such as fuel cell, internal combustion engine, have drawn many interests. However, the application of H2 on industrial burner for thermal supply has not widely applied. [1-3]

The combustion of hydrogen with oxygen results in temperatures that current boilers cannot withstand. This problem was chosen as focus for the current research in order to create a better understanding of the technical challenges and possibilities. In order to reduce the high temperature during combustion the recirculation of the steam produced is considered. First a simplified thermodynamic model was created to gain insight in the system parameters. This was used to determine the mass and energy balances for a system according to preliminary design considerations and to determine system optimization. Then a set-up of the system is provided and P&ID's and PFD's will be designed to serve as a basis for defining a system operating philosophy. The hydrogen combustion reaction can be cooled either with recycled steam or air, or with a mixture of nitrogen and oxygen introduced separately. In this way the combustion differences can be analyzed with different types of oxidizing gases and during the first tests of the plant it will be possible to precisely define the parameters for a functioning in optimal conditions. The report focuses on defining the design of the instrumentation of the system; the burner and the static mixer are drawn, the optimal sizes of the pipes of the plant lines are calculated and the characteristics of the recirculation fan, of the air fan and of the electric heater are defined. Furthermore, the operating philosophy of the system and of the control (DCS) and safety (BMS) systems are analyzed. Finally, a hazard and operability study (HAZOP) is drawn up and will serve as a reference for the future internal analysis that will be conducted by Stork Thermeq B.V. together with DNVGL, before the installation of the system. The designed plant is meant to generate hot water as final product, which can be used as a heat source. Future implementations of the system will have to consider the production of superheated steam in order to produce electricity in steam turbines.

3. Mass And Energy Balances

3.1 Stoichiometric Case

The analysis of the system begins considering the chemical reaction that takes place inside the boiler, in which hydrogen and oxygen are burnt together to obtain steam (standard condition). The combustion of hydrogen in oxygen has an adiabatic flame temperature higher than 3000°C [1-4]. The boiler that is used in the plant is thought to be used with natural gas as fuel, hence it doesn't support such high temperature. The recirculation of the steam produced by the combustion in the boiler is analyzed to reduce the adiabatic flame temperature up to 2000 ° C. To this end, a simple thermodynamic model is created in order to study the temperature variation as a function of the recirculated steam mass flow rate. As first estimation the adiabatic temperature results to be 4171°C, which is much higher than the value indicated from the literature, this is because the recombination effect is not considered in this calculation; in fact the partial combustion of hydrogen can also give $OH^{-} and\, H_3O^{+}$ as well as the steam produced can recombine with oxygen creating H_2O_2, subtracting energy from the reaction.[1-3] Recombination has a greater effect with high temperature[1-2], hence lowering the temperature to 2000°C will also reduce this undesired effect.

In order to give a first estimation of the required amount of recycled steam it is necessary to define the operating condition of the plant. Nitrogen, oxygen, natural gas and hydrogen are supposed to be stored in cylinders at a pressure of 7 bara and at atmospheric temperature. Oxygen is expanded to a pressure of about 50 mbarg with the use of a control valve and inserted in the burner at a pressure of about 40 mbarg. Correspondingly when the system is powered by oxygen and nitrogen, the nitrogen is expanded to the same oxygen pressure. Hydrogen and Natural gas are instead expanded to a pressure relatively equal to 2 bara and 1,6 bara before enter the burner, they both are considered to be at a ambient temperature of 25°C. This temperature has a wide impact on the amount of required recycled steam or combustion air, in fact the higher the inlet temperature, the more cooling medium is required which can either be air or recycled steam. This analysis could lead to the statement that if the hydrogen and oxygen are provided to an enough low temperature the recycled steam can be reduced to a minimum level; but in this case the low temperature of the inlet gases would condensate the small amount of recycled steam. Condensation is an unwanted phenomena which can leads to a reduction of the steam inside the combustion chamber, and a relative temperature rise, furthermore the water has to be removed from the burner and the wind box, complicating the design. [1-5] Previous analysis conducted [1-1] found out that the minimum inlet temperature of oxygen and hydrogen, to prevent condensation, are around 21°C; for this condition the minimum inlet temperature of the recycled steam for which condensation doesn't occur is roughly 132°C. It is considered that the recycled steam is mixed with oxygen at a temperature of 150°C, in order to prevent any possible condensation. Using these data the dependency of the temperature on the recycled steam is plotted. Calculation made can be seen in Appendix {1}.

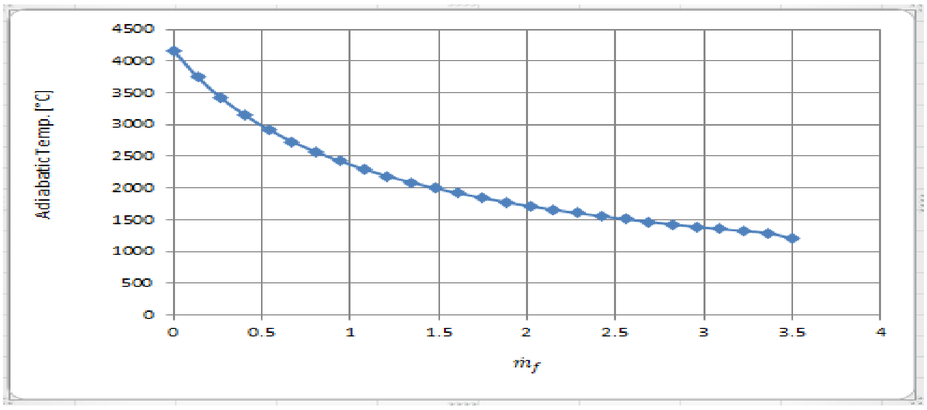


Figure 1: Dependency of the adiabatic flame temperature on the recirculation ratio

The recirculation ratio is defined as the ratio of the mass flow rate of the recycled stream and the sum of the mass flow rate of the hydrogen and oxygen supplied.

$$\dot{m}_f = \frac{\dot{m}_{recycled\ stream}}{\dot{m}_{H_2} + \dot{m}_{O_2}}$$

The results obtained shows that the ideal adiabatic temperature can be reached with a recycled mass flow rate ratio of about 1,5. Unfortunately the large variation of the heat capacity factor of the three gases involved doesn't allow to determine precisely the adiabatic temperature with simple calculation. In order to give a better estimation of the required recycled steam the software AnsysPlus is used to analyze the thermodynamic model. Results will be presented further on.

3.2 Excess of Oxygen Case

The combustion of hydrogen has been considered complete, but in reality the incomplete mixing of fuel with oxygen could lead to a percentage of hydrogen in the flue gases, thus creating problems for safety (unburned hydrogen) and reducing the potential producible energy. For this reason an excess of oxygen is required for the complete combustion of the fuel. Unfortunately it is impossible to estimate the excess of oxygen actually necessary for the complete combustion, in fact this depends on too many factors, among which many are not measurable, in particular the efficiency of gas mixing, both in the static mixer and in the boiler itself. The optimal excess of oxygen can be adjusted manually and will be fixed during commissioning.

Then through the use of AspenPlus the recycled steam, necessary to reach 2000 ° C in the boiler, was measured for different lambda "λ" values (excess of oxygen supplied). The graph has been plotted for the interested range of operation. λ is defined as:

$$\lambda = \frac{AFR}{AFR_{stoich}} = 1 + \varepsilon \qquad ;\ where\ AFR\ is\ the\ Air\ to\ Fuel\ Ratio \qquad [1]$$

In this case the reaction equation will become:

$$H_2 + \frac{1}{2}(1 + \varepsilon)O_2 \rightarrow H_2O + \varepsilon\frac{1}{2}O_2 \qquad [2]$$

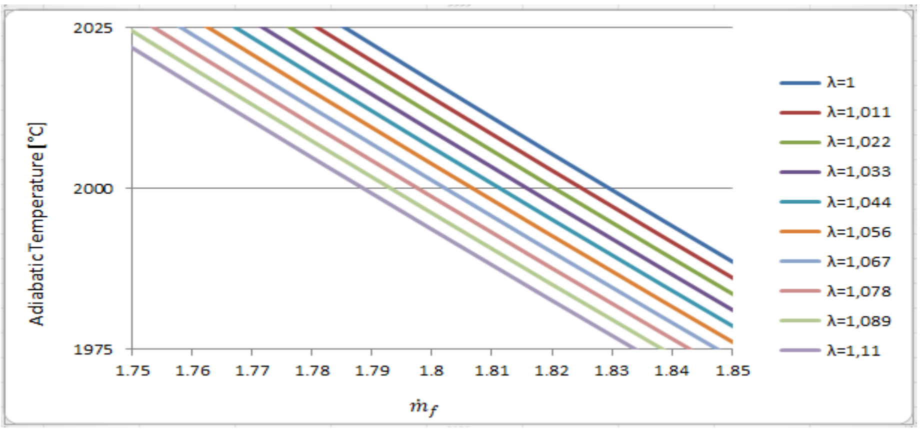


Figure 2: Dependency of the adiabatic flame temperature on the recirculation ratio for different excess of oxygen

The graph clearly shows that the increasing amount of oxygen leads to a reduction of the recycled steam. However in standard condition at 100% load the recycled stream ratio $\dot{m}_f$ varies from 1,79 to 1,83.

It is important to note that the situation analyzed is not the same if the load is changed. In the case in which the plant power is 50 kW (10% of the nominal power; which is the minimum load limit that can be applied to this system) the speed of the gases introduced into the boiler will be much lower, in fact, by reducing the gas flow and keeping the pressure unchanged by the 100% case, the speeds are reduced by about 10 times, as it will be explained later. For this reason it will be more difficult to obtain a complete mixing of the fuel with oxygen. Therefore the lower is the load, the more excess of air is necessary to ensure a complete combustion. It is assumed that for the 100% case the excess of oxygen in the flue gases has to be around 3% in volume, instead the 10% case is assumed to work with an excess of oxygen equal to 20,73%. These will be optimized during the tests.

As it will be described later, the boiler's heat exchanger, which is designed to cool the combustion product gas and at the same time produce hot water, is sized for 100% load, when switching to a 10 times lower load some problems arise in the transfer of heat, as the heat exchanger area will be too large. One of the possible solutions that will be adopted is to reduce the adiabatic flame temperature, increasing the recycled steam. For this reason the 10% case is analyzed in a lower range of temperature (between 515 °C and 1750°C).

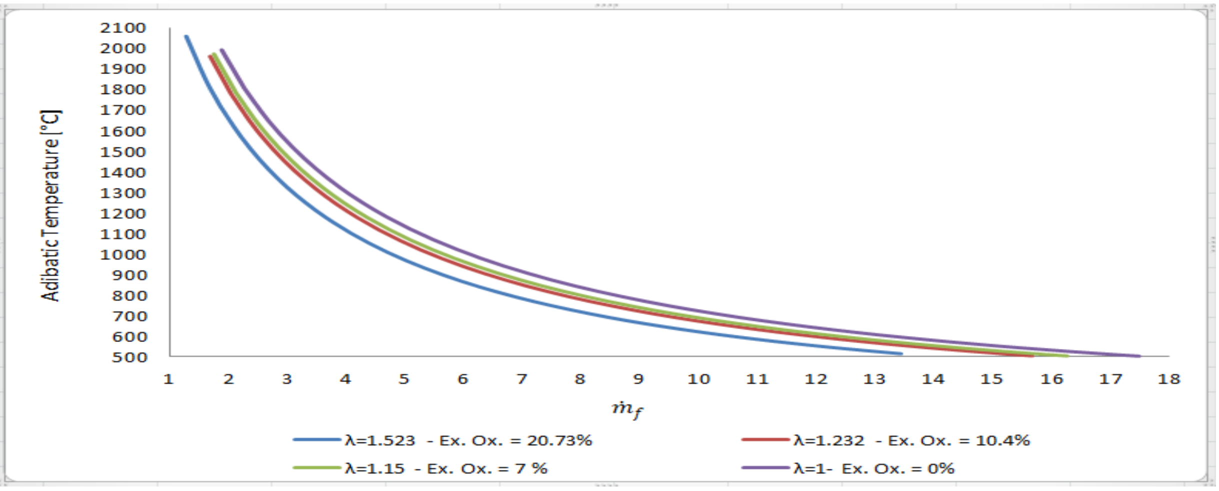


Figure 3: Dependency of the adiabatic flame temperature on the recirculation ratio for different excess of oxygen for a larger range

According to the hypothesis, the results shows that when the excess of oxygen increases the required recycled steam decreases, whereas lower adiabatic temperature can be reached increasing the recycled steam flow rate or increasing the excess of oxygen.

3.3 Combustion with Air

The pilot plant is designed to be able to study and optimize the equipment necessary for combustion of hydrogen with different set-ups. For this reason it is important to study the combustion of hydrogen with air. The plant has been designed in such a way that the air used for combustion can either be normally introduced through a fan, or oxygen and nitrogen can be supplied from two different pressure storage (roughly 7 bara) and expanded through a control valve to the burner working pressure. In both cases the oxygen-nitrogen volume ratio remains at 29-71.

In this case the reaction can be written as:

$$H_2 + \frac{1}{2}(1+\ \varepsilon)O_2 + \frac{79}{21}\frac{1}{2}(1+\ \varepsilon)N_2 \rightarrow H_2O + \varepsilon\frac{1}{2}O_2 + \frac{79}{21}\frac{1}{2}(1+\ \varepsilon)N_2 \qquad [3]$$

The analysis has been conducted for all the two limit cases (100% load and 10% load). The adiabatic temperature is only function of the excess of air in this case, because there is no presence of the recycled stream. In any case, the variation of load doesn’t influence the results.

The same method has also been applied to the case in which the fuel is natural gas, instead of hydrogen. For simplicity the natural gas is considered to be completely composed of methane, also because the detailed analysis of the combustion of natural gas is not the mean subject of the study. In this case the reaction can be written as:

$$CH_4 + 2(1+\ \varepsilon)O_2 + \frac{79}{21}2(1+\ \varepsilon)N_2 \rightarrow 2H_2O + CO_2 + \varepsilon 2O_2 + \frac{79}{21}2(1+\ \varepsilon)N_2 \qquad [4]$$

The graph below shows the results:

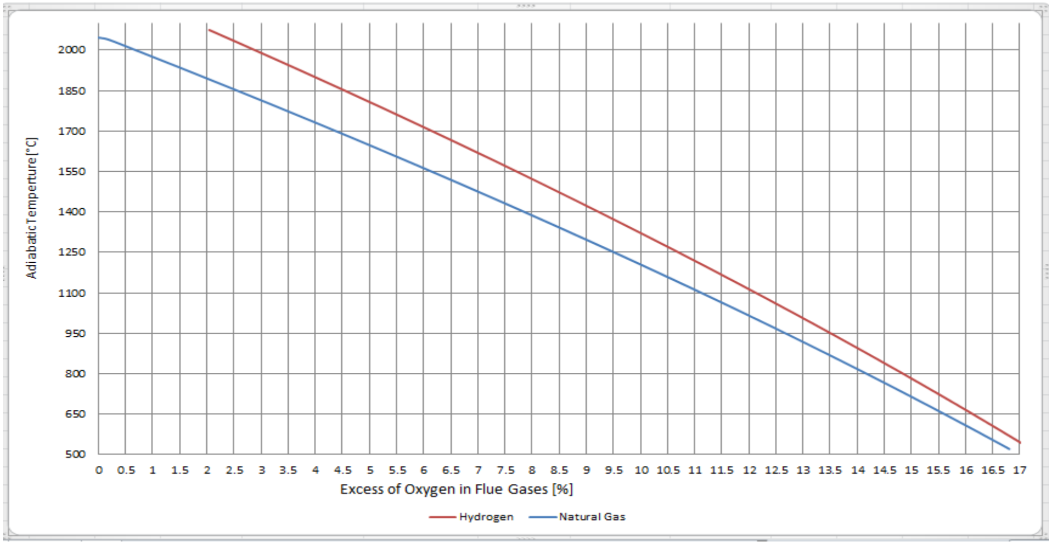


Figure 4: Dependency of the adiabatic temperature of Methane and Hydrogen on the excess of oxygen in the wet flue gases

The graph shows the variation of the adiabatic theoretical temperature of combustion in function of the percentage of the excess of oxygen in the wet flue gases. For the same excess of oxygen level the adiabatic flame temperature of hydrogen firing is higher than with natural gas firing.

3.4 Memo with Data sheet

In order to proceed with the analysis of the plant it is necessary to establish the conditions of standard operability, i.e. the situations in which the thermodynamic cycle is stable. In this way the bases are laid to define the dynamic conditions of transition from one standard condition to another.

Memos with the design data were created for each individual condition. For reasons of space these are presented in Appendix {2}, while here the results obtained will be explained and described.

The system is supposed to work with a thermal load of 500 kW, an adiabatic combustion temperature of about 2000 ° C and an excess of oxygen in the wet exhaust gases of 3% in volume. Exhaust gases coming out of the boiler are splitted into two streams, one is recirculated by a fan back to the wind-box and the remaining goes into the stack, it is important to note that the composition of the two streams does not vary from the original. Therefore a part of the oxygen introduced into the boiler inlet is also recirculated. To find a mass balance between the oxygen coming from the recycled stream and the oxygen coming from the storage it is necessary to know the amount of steam necessary to maintain the temperature at 2000 ° C. The calculations are shown in Appendix {2}; the results were compared and adapted to the simulation made with AspenPlus, in order to obtain more reliable values.

For standard conditions at 75% and 50% of the nominal load, the same flame temperature is assumed, while the excess of oxygen in the wet exhausted gases is brought to 4% and 6% in volume, respectively. Besides these two conditions is also analyzed with an excess of oxygen equal to 3% in volume, because in the case where 3% was sufficient for the complete combustion, it would be preferable to use the same stoichiometric fuel-oxygen ratio of the 100% case, to simplify the operability.

The case in which the load is reduced to 25% has been studied keeping the adiabatic temperature constant and varying the excess of oxygen in the exhausted gases to 6% and 10.4% in volume. Furthermore, the case in which the excess of oxygen is 10.4% has also been studied for an adiabatic temperature of 1745 ° C. The same method has been applied to the case in which the load is 10%: this condition was analyzed by maintaining the excess of oxygen equal to 20.73% and changing the adiabatic temperature of flame to 1750 ° C, 1500 ° C and 515 ° C. The decision to reduce the temperature in the boiler to lower loads is given by the fact that by reducing the load, the amount of steam necessary to cool the system is also reduced, in this way the quantity of gas flowing in the boiler will be reduced to such an extent that it could cool too much. Furthermore there is a need to keep the amount of combustion gases inside the furnace at the same level of the 100% load case also because when reducing the flow rate of the gases, all the stream's velocity will be reduced and this could cause a bad mixing of the gases in the static mixer, a reduction in the speed of rotation of the gases inside the wind-box. For the 10% case, in which the adiabatic temperature is 1750 ° C the amount of recycled stream is almost 10 times less than in the case 100%, instead if the desired adiabatic temperature is 515 ° C the amount of recycled stream needed is similar to that required in the 100% case, ensuring a better functioning of the entire system, as the equipment is designed to operate in optimal conditions at 100% load. The recycled stream in the latter case is 264 kg/hr with a recycled stream ratio $\dot{m}_f$ equal to 13,64 and it is actually higher than the recycled stream in the 100% case which is 254 kg/hr. Substantially it is advisable to decrease the adiabatic temperature of flame when decreasing the load (by increasing the recirculation stream flow rate) to assure a better functioning of the system. The optimal adiabatic temperature for the 10% case will be optimized during the tests.

Furthermore, the cases in which hydrogen is burned using normal air or a mixture of nitrogen and oxygen from storage have also been analyzed. Both cases were analyzed for two different conditions: considering the adiabatic flame temperature to be 2000 ° C which gives a lambda equal to about 1.2 and a lambda equal to 1.1 which increase the adiabatic temperature to 2113 ° C.

The same procedure is applied in case the fuel is changed to natural gas, the cases considered for this condition are: lambda = 1.16 and adiabatic temperature = 1845 ° C. and lambda = 1.1 and adiabatic temperatures = 1916 ° C. Both the cases have been analyzed for the situation in which the air is provided from the environment and from the two storages.

4. Boiler

The boiler used in the system is a Novumax H2R 475, of which the technical data sheet and a drawing are noticeable in Appendix {3}. The considered boiler is normally used for the combustion of methane or natural gas, for this reason its physiology has numerous limitations, among them the most important requires that the cooling water leaving the boiler does not exceed the temperature of 110 ° C. For this reason the gases leaving the boiler will have a temperature slightly higher than 110 ° C, while the recycled stream requires a temperature of about 150 ° C. This problem will be treated later and possible solutions will be investigated. The boiler itself can be divided into two parts: in the first part the heat is transferred by radiation to the water flowing in the walls of the furnace; in the second part the flue gases pass through a heat exchanger cooled by the same water that flows in the walls, this part is characterized by a convective heat transfer.

Analyzing the system it is natural to wonder why the steam produced by combustion is used to heat other water and is not used directly e.g. to produce electricity. the reasons are numerous: first of all the plant is designed to study in particular the burner and the boiler and there is no interest in complicating the thermodynamic cycle, secondly the steam produced by combustion should pass into a turbine, be condensed, heated again and reintroduced in the boiler; being able to accumulate impurities during this long path and requiring an instrumentation much more complicated than necessary, thirdly, the walls of the boiler still need to be cooled in some way, so a second cycle of water would still be necessary.

4.1 Radiation Heat Transfer

The following analysis will provide a simplified calculation method for quantify the amount of radiation heat transferred from the gases inside the furnace to the walls in boilers with a dominating thermal radiation component. Analysis are performed for the designed combustion chamber with water-cooled walls. In order to measure the absorbed radiation heat flux in the first part, it is best to measure the enthalpy increase of the process medium. This method is used in practice to monitor the total thermal output of a furnace or boiler. [2-1]. Hence it is necessary to define an energy balance for the entropic variation that occurs inside the furnace, which is given by the following formula. It is assumed that fuel combusts completely instantaneously at the burner exit and reaches adiabatic combustion temperature Ta, and that heat transfer only occurs in the radial direction of the furnace axis; ignore heat transfer in the axial direction (one-dimensional model).

$$Q[Kw] = \varrho\, \dot{m}_{fuel}(h_a - h_F) \tag{5}$$

Where h_a and h_F represent respectively the enthalpy of the gas at theoretical adiabatic combustion temperature and at the exit of the furnace; ϱ is the heat preservation coefficient which takes into account the heat loss due to the cooling effect of the furnace wall (assumed to be 0.80 from reference [2-5]).

Furthermore the radiation heat transferred to the walls can be written as:

$$Q = \sigma a_F A({T_g}^4 - {T_w}^4) \tag{6}$$

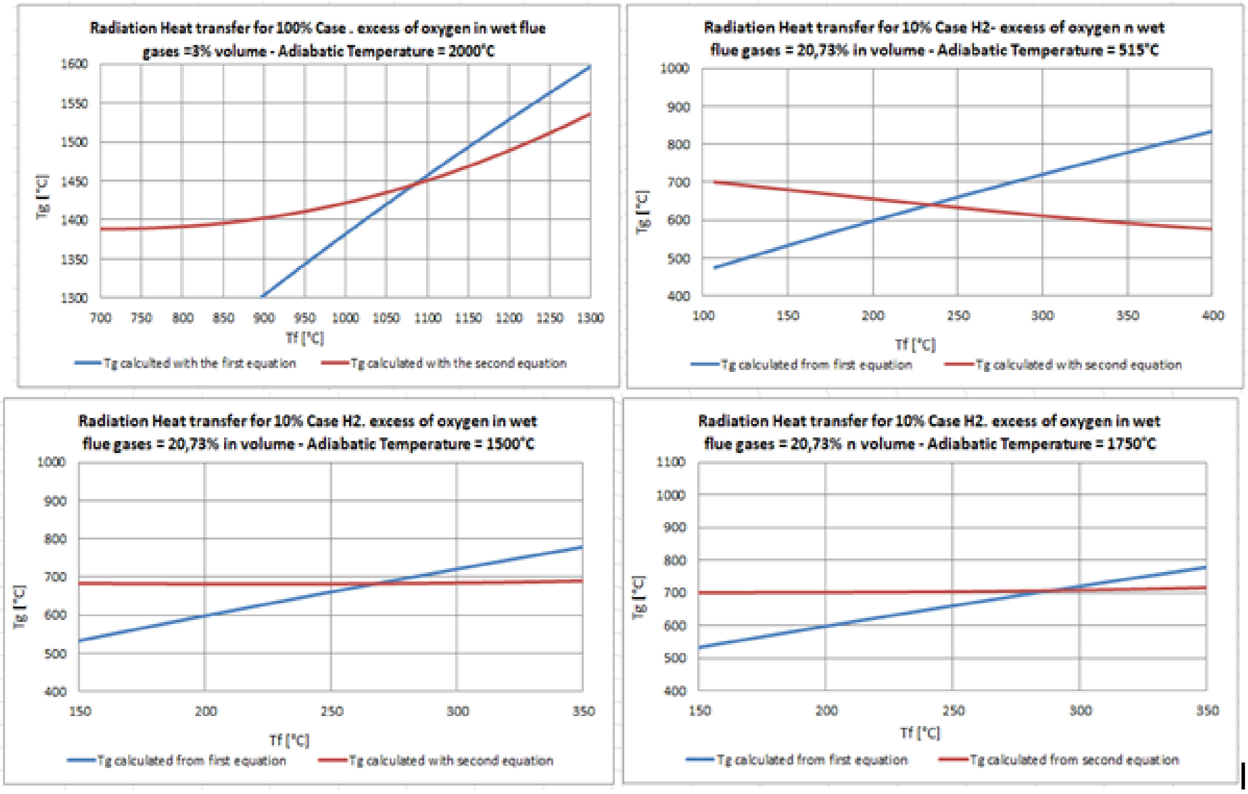
Where σ is the Stefan-Boltzman constant; A is the total area of the furnace (first part). T_w is the average temperature of the wall assumed to be (25°C above the cooling water outlet temperature) and T_g is the average temperature of the gas inside the furnace. The latter will vary with the three dimensions; to simplify the calculation T_g represents the average of all the temperatures of the gas. a_F is the emissivity factor which depends on the characteristic of the gas and the geometry of the boiler. Values are assumed from reference [2-7] and can be seen in Appendix {4}.

The combination of the equations of the heat released by the gas and absorbed by the walls lead to two different expressions for the average temperature T_g in function of the temperature at the boiler outlet T_F:

$${T_g}^4 = \frac{3{T_f}^4}{\left(\frac{{T_F}^3}{{T_a}^3}\right) + \left(\frac{{T_F}^2}{{T_a}^2}\right) + \left(\frac{T_F}{T_a}\right)} \tag{7}$$

$${T_g}^4 = \frac{\varrho\, \dot{m}_{fuel}(h_a - C_{p(T_F)} T_F)}{\sigma a_F A} + {T_w}^4 \tag{8}$$

Detailed calculation is visible in Appendix and the results are plotted in the following graphs:



Radiation Heat transfer for 100% Case . excess of oxygen in wet flue gases =3% volume - Adiabatic Temperature = 2000°C

Radiation Heat transfer for 10% Case H2- excess of oxygen n wet flue gases = 20,73% in volume - Adiabatic Temperature = 515°C

Radiation Heat transfer for 10% Case H2. excess of oxygen in wet flue gases = 20,73% in volume - Adiabatic Temperature = 1500°C

Radiation Heat transfer for 10% Case H2. excess of oxygen in wet flue gases = 20,73% n volume - Adiabatic Temperature = 1750°C

Figure 5: Radiation heat transfer - Determination of the outlet temperature of the flue gases for different cases of combustion of hydrogen with oxygen and recycled stream

From the analysis conducted it is possible to know approximately the temperature of the flue gases at the entrance of the convective heat exchanger in the second part of the boiler for all the standard cases analyzed and the relative heat exchanged with the walls. For the first case (100% load) the temperature of the gases after the boiler area characterized by the transfer of radiant heat is approximately 1085 ° C and the heat exchanged in this area is 267 kW (53.4% of the primary source) .It is important to note that since the temparature of flue gases is still high, there is still a considerable amount of heat which has to be assorved by the second heat exchanger.The average temperature of the gases inside the furnace is approximatly 1440°C For the 10% case instead, all three situations have been analyzed in detail and the results are as follows:

- case with T adibatic equal to 1750 ° C - Gas outlet temperatures equal to ~ 290 ° C and radiation heat transfered equal to ~ 43 kW (86% of the input power)
- case with T adibatic equal to 1500 ° C - Gas outlet temperatures equal to ~ 270 ° C and radiation heat transfered equal to ~ 39 kW (78% of the input power)
- case with T adibatic equal to 515 ° C - Gas outlet temperatures equal to ~ 220 ° C and radiation heat transfered equal to ~ 32 kW (64% of the input power)

It is possible to notice that decreasing the adiabatic temperature the efficiency of the system decrease, in fact, not only is the radiativ heat transfer less, but also the temperature exiting the first part of the boiler decreases; thus reducing the exchangeable heat in the second heat exchanger. This effect occurs because while the power supplied by the fuel is always the same the heat is transferred to a larger ammount of recirculated steam, depleting the usable energy.

The same method of analysis was used to calculate the radiation heat transfer in case the system works with air as oxidant or with the mixture of nitrogen and oxygen. The study was applied both in the case where the fuel is hydrogen or natural gas. The results are visible in the following graphs.

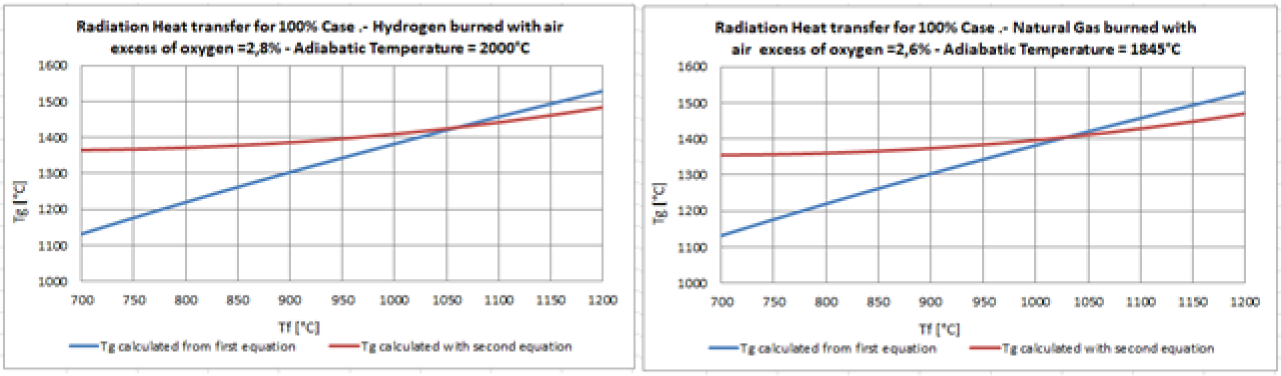


Figure 6: Radiation Heat transfer - determination of the oulet temperature of the flue gases for the combustion of Hydrogen and Methane with air

- The results of the first case "100% load Hydrogen burned with air with adiabatic flame temperature = 2000°C" are very similar to the one obtained using fresh oxygen and the recycled stream as oxydant (standard condition). In fact the average temperature of the gases inside the furnace is 1430°C, (1440°C for standard condition) while the oulet gas temperature is ~ 1070°C (1085°C for standard condition). The calcualted radiation heat transfer in this case is 260 kW, 52% of the inlet power (267 kW for standard condition). This result demonstrate that is possible to obtain a similar condition of radiation heat transfer of the standard case by cooling the combustion chamber with air instead of with the recycled steam.
- in the case where the natural gas is used as fuel the results are similar too. The average furnace temperature is about 1400 ° C, the gas outlet temperature is 1030 ° C and the energy exchanged by irradiation is about 253 kW. (50.6%). As well as the previous situation, the condition of radiation heat transfer are similar to the standard case.

This analysis demonstrated that is possible to obtain a similar behaviour of the fluegases in furnace for different condition and types of fuels. This fact guarantees that it is possible to use the boiler in all the analyzed situations avoiding substantial variations in the thermodynamic conditions of the cycle.

4.2 Heat exchanger

This chapter is purely theoretical and is meant to make understand the problems that may occur due to a wrong sizing of the heat exchanger. The boiler is divided into two parts, one characterized by radiated heat exchange and one by convective heat exchange. This second part of the boiler is considered removable or internally exchangeable with other types of heat exchanger. This analysis will in fact deal with three different types of system set-ups based on three different heat exchanger concepts. Calculation and data assumed can be seen in Appendix {5}, the results will be discussed here. All the cases are analyzed using the same method: in first place the heat exchanger is sized in order to produce the desired proprieties of the streams in case the system runs at 100% load; this is done by using the LMTD method. LMTD is the Logarithmic Mean Temperature Difference and it is calculated in two different ways according to the type of heat exchanger (cross or parallel flows). Subsequently the results obtained are used to see the behavior of the heat exchanger when the load is reduced to 10%. In particular the case study has been done in the case in which the adiabatic temperature is 1750°C because the amount of flue gases in the boiler is the least in this case, hence it can be defined as the limit condition, since a positive result in this case would result in a positive behavior in all the intermediate cases between 10% and 100% load.

The inlet temperature of the hot side is assumed to be equal to the oulet temperature of the radiation boiler part: for the 100% load the inlet temperature is 1085°C and for the 10% load is 280°C (from radiation heat transfer analysis).

-**The first case** analyzes the normal heat exchanger already present in the boiler with a counter-current operation. Working fluid is considered to be water that enters at a temperature of about 60 ° C and exits at a temperature of about 105 ° C. In this case it is necessary the introduction of a pump to have a forced circulation inside the heat exchanger and an electric heater to bring the flue gases from about 107 ° C to the desired temperature of 150 ° C to be mixed with the fresh oxygen. The cooling water is pressurized to 4,5 bar at the inlet of the heat exchanger and it assumed that it will have a pressure drop of 0,5 bar. The obtained results shows that the energy subtracted from the flue gases to cool it up from 1085°C to 107°C is 152 kW. Together with the heat transfer by radiation, the total energy given to the water will be 267+152=419 kW which is 83.8% of the input power. On the other side 0,11 kg/s of water are heated up from 60°C to 105°C, extracting 152 kW. The cooling water flows in 40 pipes with an intern diameter of 18 mm, a thickness of 2 mm and a length of 1,4 m with a velocity of 27 m/s (velocity is assumed). These pipes are surrounded by flue gases which flow in a larger conduct of about 7 cm diameter. Hence the heat exchanger can be easily sized for the standard condition at 100% load. In this case the work needed for the pump is about 0,1 Kw and the required energy for the electric heater is about 5 kW. The total efficiency of the system can be calculated : (419 kW-5,1kW)/500kW= 82.8%.

Applying the calculated dimensions to the 10% load case it is possible to see that the exchanging area is way too large for the small amount of flue gases. For this reason is not possible to achieve a desired setup, in fact if the mass flow rate of cooling water is very large the flue gases will cool down to about the inlet temperature of the cold side (60°C) and condense. If instead the mass flow rate of water is low this will be heated up to the inlet temperature of the hot side (274°C) almost reaching the inlet temperature of the hot side (280°C); this temperature exceed the safety limit of the boiler.

-**The second case** considers that the flue gases are cooled with air at room temperature in a heat exchanger dimensioned according to the 100% case in cross flow. In this case the flue gases would be cooled up to the required temperature of 150° C, requiring no heater, but requiring a compressor for the cooling air. In this case the flue gases flow in the inner pipes surrounded by the air streamflow. The heat exchanged is 149 kW, lower than the previous case because the flue gases are cooled down to 150°C. On the other hand air is heated up from 25°C to 370°C. In this case the flue gases flows in 100 pipes with intern diameter of 18 mm, a thickness of 2 mm and a length of 1,37 m; while the air flows in a larger conduct with an intern diameter of about 25 cm at a speed of 20 m/s. The air is considered to be pressurized at 70 mbarg, which requires a compressor work of about 1 kW. The total efficiency in this case is 82.6% if we consider the heat given to the cooling air as an useful product which is not in reality. In fact in this case only the heat transferred by radiation is considerable as useful and all the heat given to the air can be considered wasted; according to this consideration the efficiency is (267 kW-1kW-1,1kW)/500kW=53.2%.

When applying the calculated sizes of the heat exchanger to the 10% load case the same previous issue occurs. In fact the exchanging area is so big that doesn't allow, in this condition, to have a temperature difference between the hot and cold outlets: if the amount of air is large the flue gases will be cool down to 25°C provoking condensation and if it is low the air will reach 280°C. Since in this case the heat exchanger is considered to be external to the boiler it is not influenced by the safety limits of the boiler, hence this set up could actually be applicable to the system.

-**The third case** considers that the flue gases are cooled in a huge vessel of saturated water at ambient pressure. In this way the flue gases are cooled to the water saturation temperature of 100 ° C and must be reheated with an electric heater to 150° C. Nerveless, to keep the amount of water steady during the operation of the system, the water which evaporates has to be replaced with other saturated water that somehow has to be heated up from environmental temperature to 100°C, this is supposed to be done using an electric heater. The flue gases flow in 8 pipes with an intern diameter of 58 mm, a thickness of 2 mm and a length of 4,67 m. Of course the total exchanging area has to be larger than the two previous cases because in this case the cooling medium is not flowing, reducing the convective heat transfer.

In this situation the flue gases release 194 kW to the vessel, because they are cool down to 100°C. This amount of energy is entirely used to change the phase of water. In standard condition at 100% load the evaporated water rate is 0,09 Kg/s. This vapor has to be replaced with saturated water, which has been heated up previously. The power required to increase the temperature of 0,09 kg/s of water of 75°C is about 28 kW. The power required to heat up the recycled stream from 100°C to 150°C is about 7 kW. As the second case the useful energy can be considered only the radiation heat transfer and this fact gives an overall efficiency of (267kW-28kW-7kW)/500kW=46,2%.

As it is possible to foresee there are no problems for the case in which the load is 10%, in fact the water vessel is considered so large as to have no temperature gradient inside it and whatever the quantity of flue gas to cool, the temperature at the end of the process will always be equal to the saturation temperature of the water. In particular, if the quantity of flue gas is lower, it will reach the temperature of 100 ° C even more quickly than in the case of 100% load.

From the point of view of the simplicity of the system, the last case is certainly the best because it does not require particular control and safety instruments, does not imply the use of a real heat exchanger, but only a vessel and an electric heater and ensures a constant temperature of the flue gases at the output of the heat exchanger, in this way, whatever the load is, there will be no problems of condensation or ultra- heating of the cooling medium. However, it should be considered that this is the case with the least efficiency and would also require a large amount of space. The "air cooled" case instead requires a smaller space (about 25 cm x 150 cm), but problems occur when the load has decreased. Furthermore both of these cases have a lower efficiency compared to the first one.

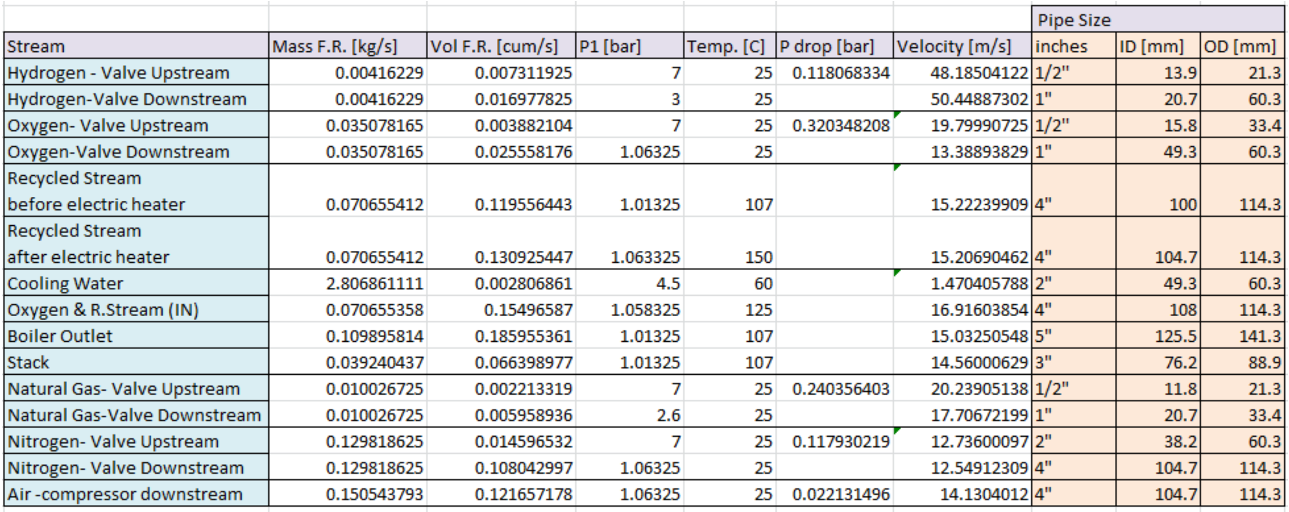
Another consideration, which favors the first case, is that the most economical option is certainly to use the heat exchanger already present in the boiler itself, in this way only an electric heater should be purchased. For this reason it is necessary to rethink the sizing of the first heat exchanger; it has been thought that using this exchanger in a parallel flows configuration it would force the cooling medium to have the desired temperature at the outlet. Calculation and data assumed can be seen in Appendix {5}.

In this case the total heat extracted remains the same, what changes is the sizes of the heat exchanger, in fact to extract the same amount of power a parallel flows heat exchanger requires around 1,5 times the area of a cross flow heat exchanger [2-3]. Keeping the size and the number of the inner pipes the same, the length of the heat exchanger becomes 2,8 m. By varying the size and the number of pipes, the length can be reduced; however, because this is a just a demonstrative analysis, the real surface doesn't really matter. With the data obtained is possible to estimate the outlet temperature of the cooling water which will be the same as the flue gas outlet temperature (107°C). This configuration insures that the outlet temperature of the cooling media doesn't exceed the safety temperature imposed by the boiler. By adjusting the mass flow rate of water (regulating the pump load) it will be possible to optimize the outlet temperature of the boiler. This last configuration will be used in the pilot plant and the required mass flow input will be adjusted for all the standard cases during the operation of the system.

5. Sizing of the pipes

For the sake of completeness of the project and for the drafting of the PFD it is necessary to know the dimensions of the pipes used in the plant. In order to achieve this target it is required to know the maximum volumetric flow rates that the system will have to support, therefore the size of the pipes is based on the standard 100% load combustion case. Furthermore it is essential to know the speed of the streams that is desired to obtain. To this end, some references were analyzed [4-4] and then compared with the company's experience. The basic idea to keep in mind is that the pressure losses, both localized and distributed, are proportional to the square of the speed, for this reason it's necessary to reduce the speed in such a way that it allows an optimal operation of the system while minimizing the pressure losses.

The speeds assumed are the following: hydrogen should flow approximately at a speed of 50 m/s, Natural gas at 30 m/s, while the oxidizing gases (oxygen, nitrogen and air) should have a speed of about 20 m/s, then recycled stream needs to flow at about 15-17.5 m/s and finally the cooling water should flow at 1.5 m/s. Assuming these speeds, the optimal diameter of the pipes can be calculated and compared with those available on the market. Although an infinite number of tube sizes are available on the market, the most easily available ones are the 1/2 inch multiplexes. So only these will be taken into consideration. However, for all the cases it was decided to choose the available size just bigger than necessary, in order to reduce the speed of the streams. The results are shown below:



							Pipe Size		
Stream	Mass F.R. [kg/s]	Vol F.R. [cum/s]	P1 [bar]	Temp. [C]	P drop [bar]	Velocity [m/s]	inches	ID [mm]	OD [mm]
Hydrogen - Valve Upstream	0.00416229	0.007311925	7	25	0.118068334	48.18504122	1/2"	13.9	21.3
Hydrogen-Valve Downstream	0.00416229	0.016977825	3	25		50.44887302	1"	20.7	60.3
Oxygen- Valve Upstream	0.035078165	0.003882104	7	25	0.320348208	19.79990725	1/2"	15.8	33.4
Oxygen-Valve Downstream	0.035078165	0.025558176	1.06325	25		13.38893829	1"	49.3	60.3
Recycled Stream before electric heater	0.070655412	0.119556443	1.01325	107		15.22239909	4"	100	114.3
Recycled Stream after electric heater	0.070655412	0.130925447	1.063325	150		15.20690462	4"	104.7	114.3
Cooling Water	2.806861111	0.002806861	4.5	60		1.470405788	2"	49.3	60.3
Oxygen & R.Stream (IN)	0.070655358	0.15496587	1.058325	125		16.91603854	4"	108	114.3
Boiler Outlet	0.109895814	0.185955361	1.01325	107		15.03250548	5"	125.5	141.3
Stack	0.039240437	0.066398977	1.01325	107		14.56000629	3"	76.2	88.9
Natural Gas- Valve Upstream	0.010026725	0.002213319	7	25	0.240356403	20.23905138	1/2"	11.8	21.3
Natural Gas-Valve Downstream	0.010026725	0.005958936	2.6	25		17.70672199	1"	20.7	33.4
Nitrogen- Valve Upstream	0.129818625	0.014596532	7	25	0.117930219	12.73600097	2"	38.2	60.3
Nitrogen- Valve Downstream	0.129818625	0.108042997	1.06325	25		12.54912309	4"	104.7	114.3
Air -compressor downstream	0.150543793	0.121657178	1.06325	25	0.022131496	14.1304012	4"	104.7	114.3

Figure 7: Sizing of the pipes section of the plant

As can be seen from the table the feed lines have been divided into "control valve upstream" and "downstream". this is because after the control valve the gases are expanded to the desired pressure (from 6 barg to 50 mbarg for oxygen and nitrogen and to 2.5 barg for hydrogen). It is therefore necessary to make sure that the pressure losses that occur before reaching the valve are not reducing the pressure below the desired level. In Appendix {6} it can be seen how the calculations of the localized losses were made. The result is that the pressure losses are between 100 mbar and 300 mbar, reducing the pressure from 6 barg to about 5,7-5,9 barg and therefore causing no problem. The distributed pressure losses have also been calculated for the air line, resulting in a pressure loss of about 20 mbar, which must be taken into account when sizing the air fan. It is important to know that the storages of nitrogen, hydrogen and oxygen become empty over time, thus reducing the pressure inside the cylinders. So there will be a time when the storage pressure will not be enough to counter pressure losses in the pipes and deliver the gas to the optimal burner pressure. This situation, as will be seen below, will be safeguarded by the security and control transmitters of the system. In any case it is desirable that the storage is re-filled before this limit point is reached.

6. Burner and wind box

Once the mass balance of the system, the thermodynamic conditions at each point and the dimensions of the pipes have been defined, it is possible to proceed with the planning of the necessary equipment. The first one that will be analyzed is the burner and its wind-box.

A burner is a device that produces a controlled flame by mixing a fuel gas such as natural gas, methane or hydrogen with an oxidizer such as the ambient air or supplied oxygen, and allowing for ignition and combustion. The wind box instead is a large pressurized cavity, from which the burner takes its combustion air or oxidizing gas. [3-1].

A compact burner will be used for this pilot plant; the basic dimensions were given directly by the company Stork Thermeq B.V., as one of the core buisiness of the company are precisely the burners. The objective of this assignment was to design the burner in AutoCad 3D and divide it into components so that the burner itself can be removed and the individual parts replaced; to make possible to analyze the behavior of the system with different burner set-ups.

In addition, the design of the burner must be appropriate to the requisites necessary for 3D printing; since at the end of the internship the entire burner could be printed.

To simplify the understanding of how the burner has been designed, an image showing the assembly of the burner and the burner in its entirety are shown below. The 2D drawings of the individual parts are visible in Appendix {7}.

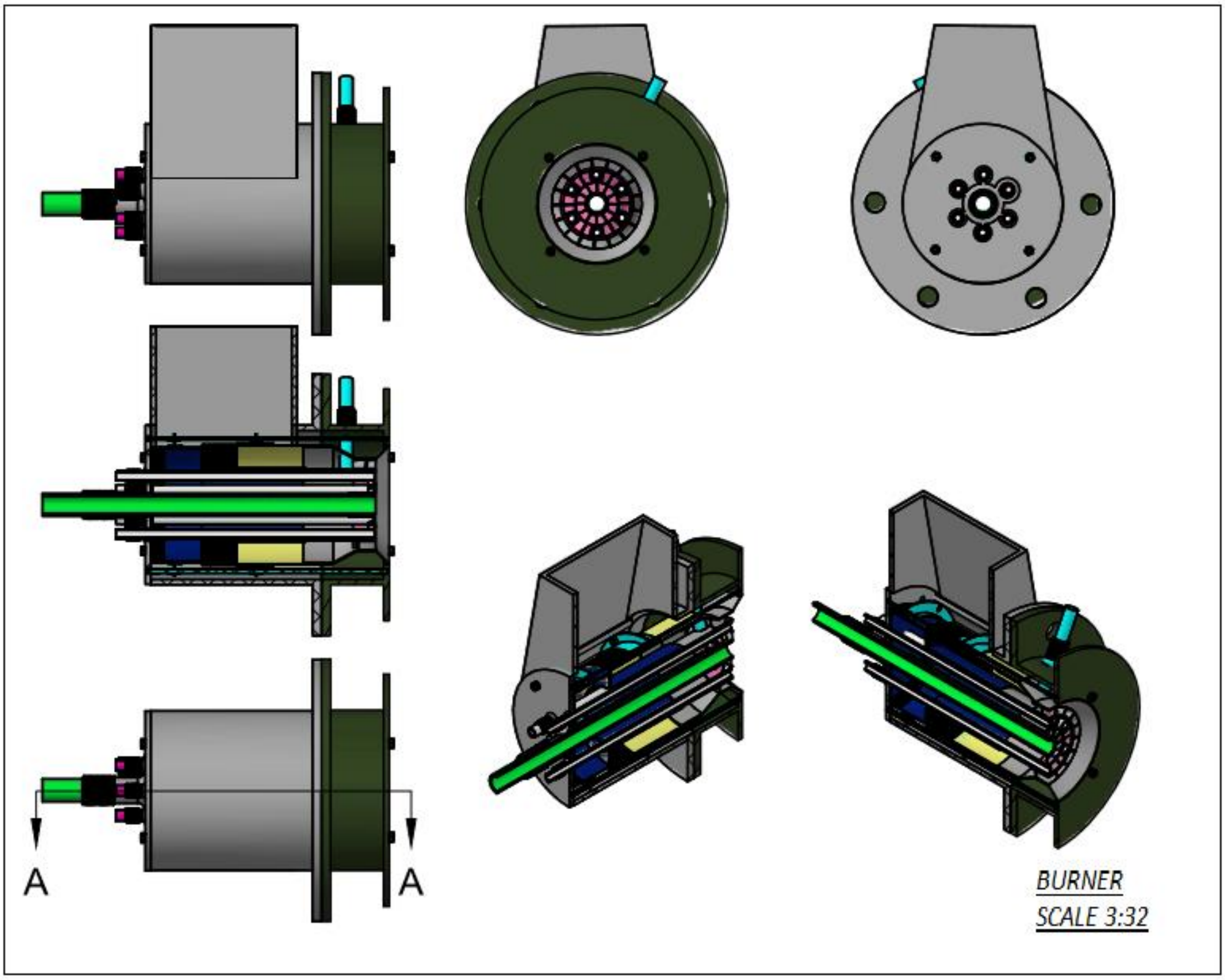


Figure 8: complete 2D and 3D drawing of the burner and wind-box

The burner has 6 spuds (OD Outside Diameter = Ø 12 mm, thickness = 2 mm) and a central gas gun (OD = Ø25 mm, thickness = 2,8 mm) that run the entire length of the burner. The fuel gas is introduced through the central gun and the 6 spuds, while the oxidizing gas is introduced by the entrance of the wind box. From the wind box the oxidizing gas can enter the burner through two perforated sections which are the primary air inlet and the secondary air inlet. Both of these two inlet can be adjusted via a sliding plate that can cover the holes or not.

The front of the burner is made so that the set of blades, used to get swirl inside the burner, can be removed and changed if necessary. These blades are divided into three parts. The first set of blades has an angle of 70 °, while the second one has an angle of 40 ° and the third has an angle of 30°. These dimensions have been given directly by the company and are studied in such a way that the triangles of speed of the fuel and of the oxidizing gas allow an efficient and complete mixing. The length of the blades has been calculated in such a way that there is 0.5 mm overlapping between a blade and the next one. In other words, the gas passing through the blades will certainly be deflected and will create a swirl; it will never enter the burner in a direction parallel to the burner geometry.

The first and second set of blades (smallest diameters) are meant for the passage of the primary combustion air, while the third set of blades (wider) is designated at the inlet of the secondary combustion air. All the three set of blades were thought to have 12 blades each, after a first sketch of the drawing, it has been notice that it was necessary to increase the number of blades of the wider set to 17, in order to reduce the length of those blades from 35 mm to 25 mm, in this way there is more space left for a possible entrance of the ignitor, as will be discussed later.

The set of blades in the middle is instead designed so that every two blades there is enough space for the insertion of a spud. In other words, the second set has 12 blades interspersed every two by a hole for the passage of the spud.

The entire set of blades is mounted on the burner by a joint on the central gun and another on the section that connects primary air to secondary air. Drawings are visible in Appendix {7}.

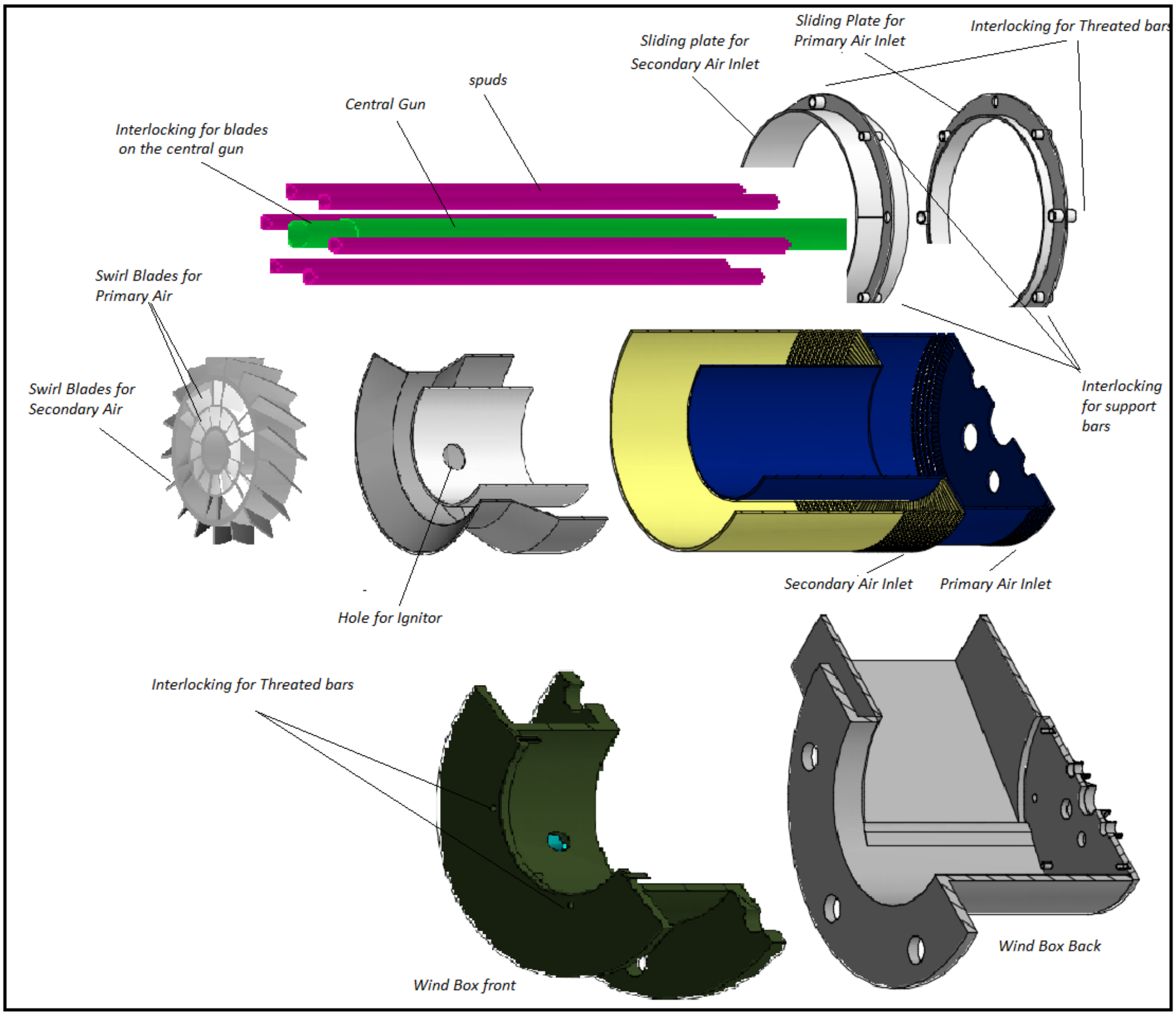


Figure 9: representation of the burner and wind-box assembly

The primary and secondary air inlets are designed in such a way that the pressure inside the wind box is such as to keep a vortex around the burner and to ensure that the oxidizing gas enters the burner in a homogeneous manner. The perforated part of the primary and secondary air inlet are 15 mm and 40 mm long, respectively. both of these two parts have holes with a diameter of 1 mm and the number of holes is calculated in such a way that the surface is about 50% perforated.

For each of the two air inlets there is a sliding plate that can be adjusted in such a way as to reduce or increase the air intake in the burner (in this way it is also possible to vary the split ratio between the primary and the secondary air). These two sliding plates have the same length as the relative perforated part of the burner and are supported by 4 bars with a diameter of 4.5 mm. To adjust the position of the sliding plates there are 4 other threaded bars, (two for each sliding plate); these threaded rods adjust the position of the sliding plates using a screw drive system. The air inlet regulation system is usually done through a register, in this case the burner geometry does not allow such a complicated instrumentation; the sliding plate system was designed during the internship using old projects of the company as reference.

The burner and the wind box are both divisible into two parts, so that both the front and the back can be replaced. the back part of the burner is slightly wider than the front part and this allows an interlocking of the two components; unfortunately this decision implies that there may be leakage of gas from primary air to secondary air and vice versa. However, considering that the fuel gas flows only in the central gun and in the spuds, there can be no leakage of fuel; therefore the system should not encounter serious problems from this configuration, as the only effect that can slightly vary is the ratio between primary and secondary air. It is also possible that part of the air in the wind box unwantedly enters the burner from this connection; this effect should be marginal as the backlash should be smaller than 0.5 mm. However, this fact can lead to a small pressure drop in the wind box which can be reduced by partially covering the air and secondary air inlet. Once the design of the burner and the wind box has been defined, it is necessary to understand where the ignitor can be inserted. The complete technical characteristics of the selected ignitor are visible in Appendix {7}: it has a power of 2 kW, which is sufficient for ignite boilers up to 500 kW. The external diameter of the ignitor is 15 mm. At first, it has been thought of inserting the ignitor between the central gun and the spuds, so that the ignitor outlet is positioned just behind the blades for secondary air. Unfortunately for reasons of geometry, there was not enough space to insert it in this position. Hence it was decided to insert the ignitor perpendicular to the burner, near the burner outlet (25 mm from the end of the burner). For this reason the length of the secondary air blades had to be reduced, increasing the number of blades from 12 to 17 in order to maintain the same angle of curvature, as already mentioned. The pilot flame created by the ignitor is 8 cm long, so a distance of 2.5 cm is sufficient for the ignition of the boiler. The end of the ignitor is positioned perpendicular to the burner outlet and the pilot flame will bend according to the direction of the air toward the end of the burner.

Another possible solution to this problem is to substitute one spud with the ignitor. For this reason one of the hole in the wind box and in the burner meant for the spud is expanded to a diameter of 15 mm instead of 12 mm. In this way there is a double choice for the ignitor position and the two different configurations can be analyzed. In any case it is preferable to use the standard configuration with 6 spuds and the perpendicular ignitor, so that the distribution of fuel gas in the boiler is more homogeneous, resulting in a more stable flame.

7. Static Mixer

A static mixer is a device inserted into pipeline with the objective of manipulating fluid streams. As a result of these alterations in the fluid flow, mixture components are brought into intimate contact. [4-1] Flow in an empty pipe produces some degree of radial mixing but in most cases, adequate mixing can only be achieved by an impractical length of pipe. Inserting a static mixer significantly accelerates inline mixing. From all static mixers, the Kenics mixer or K-mixer has been analyzed most. The General K in-line mixer consists of a number of elements of alternating right and left-hand 180◦ helices. The elements are positioned such that the leading edge of each element is perpendicular to the trailing edge of the preceding element. The length of the elements is typically one and a half tube diameters. [4-2][4-3] The twisted elements with left- and right-hand twists caused the material to move from the wall to the center and from the center to the wall. The flow in each channel circulates around its own hydraulic center causing radial mixing. After traveling through a number of these elements, the fluid is homogenized with respect to age, composition, and temperature.[6-3] The best static mixer is the one that delivers the mixing quality desired at the lowest pressure drop, for lowest installed cost and fits in the space available.

The static mixer that will be used for this pilot plan is a K-mixer designed appositely for the system; its objective is to homogeneously mix the recycled stream and fresh oxygen in case of standard condition, but also to mix the fresh oxygen with nitrogen in case the system works using these as oxidizing gases. In the situation in which air is used as an oxidizing gas, it too will pass through the static mixer, although there is no need for it. The reason is that otherwise the burner's wind box should have a double input, one for the stream coming from the static mixer and one for the air, thus complicating the system. To determine the size of the static mixer it is necessary to know all the proprieties of the gases that are supposed to be mixed, in particular the speed of the stream plays a main role in determine the pressure drop caused by the mixer.

The intern diameter of the static mixer will be equal to 110% of the largest inlet pipe which is connect to it: in this case the largest pipe is the one of the recycled stream (ID 104.7 mm); thus the intern diameter of static mixer is considered to 108 mm. The length of single element is 1,5 times the ID and it will result in 160 mm. The number of elements can be selected by looking at the Reynold number of the mixer inlet stream. In this case, in standard condition at 100% load the Reynold number is about 15000, while in standard condition at 10% load and adiabatic temperature=1750°C it results to be about 1700. For Reynolds number included between 1000 and 10000 the number of required elements is 6, instead for Reynold number higher than 10000 only 4 elements are required. We have opted for a configuration with 6 elements, so as to ensure complete mixing even for low loads; to the detriment of pressure losses when the load is high. Hence the total length of the mixer will be 0,96 m.

It was evaluated whether it was better to buy the static mixer directly or print it in 3D and it was decided that printing could be of greater interest to the company. For this reason the static mixer has been divided into three parts connected by flanges (because of the length restriction of a 3D printer). In this way it is possible to eliminate a pair of elements and check if 4 elements are still sufficient even for low loads. The following picture shows the drawing of the designed static mixer.

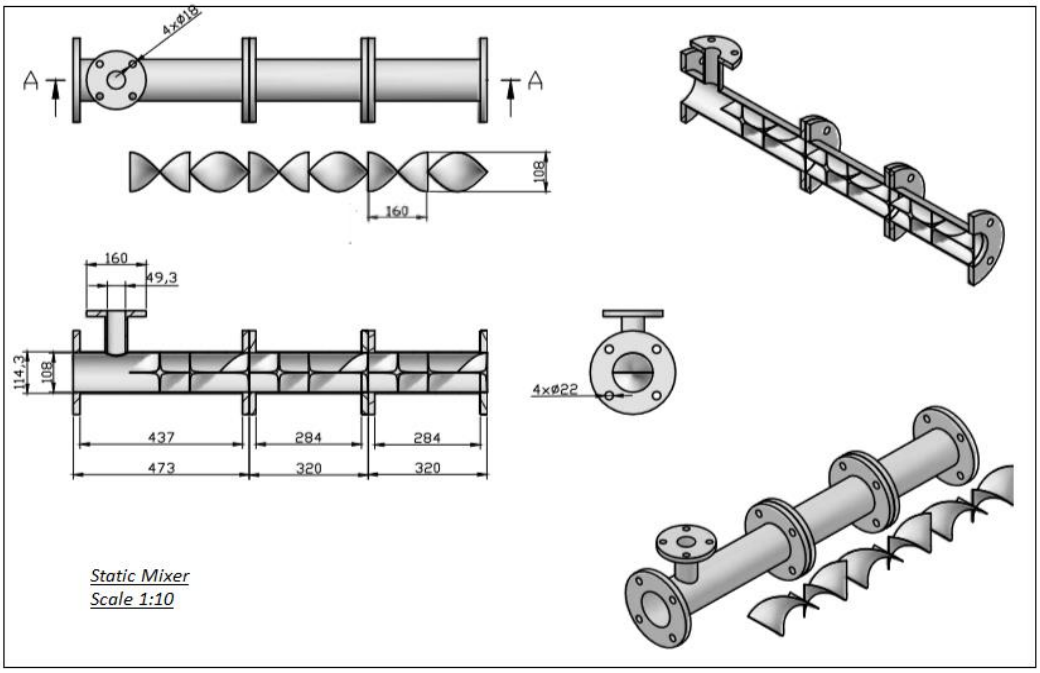


Figure 10: 2D drawing of the Static Mixer

Furthermore it is important to know the pressure drop that will occur in the static mixer. Since there are a host of parameters affecting pressure loss in the Kenics static mixer, it is convenient to represent pressure drop correlation in terms of fewer dimensionless groups. It is well known that the pressure drop correlation in an empty tube is simply given in terms of the friction factor (Cf) and Reynolds number (Re).[4-4] Different correlation have been used to analyze the pressure drop inside the static mixer, the only one which gives a reliable result is the following [4-5]. In Appendix {8} it is possible to see all the correlation which has been used and the calculation made for three different cases: 100% load – hydrogen burnt with oxygen and steam; 10% load- Hydrogen burnt with oxygen and steam at adiabatic temperature equal to 1750°C and 100% load- Natural gas burnt with mixture of nitrogen and oxygen provided by the storages (which is the same as if it is burnt with air).

$$\Delta P_{drop} = 4{,}59 * Re^{-0.22} * \rho * v^2 \left(\frac{L}{ID}\right) \qquad [9]$$

The pressure losses in the standard case at 100% load is about 5 mbar, while for the standard case, 10% load is about 0.2 mbar. The results obtained are optimal for the pre-established functioning of the system. In fact it was assumed that the static mixer had at most a pressure loss of about 10 mbar. In the case 100% with natural gas the pressure losses are about 11 mbar. Any value below this ensures that the oxidizing gas has enough pressure to reach the burner. The velocities of the fluids considered have been selected with the help and experience of the company and have been modified in such a way as to minimize pressure losses, not only in the static mixer, but also in their normal flow in the pipes. In fact, both localized and distributed pressure losses are proportional to the velocity square.

8. Electric Heater

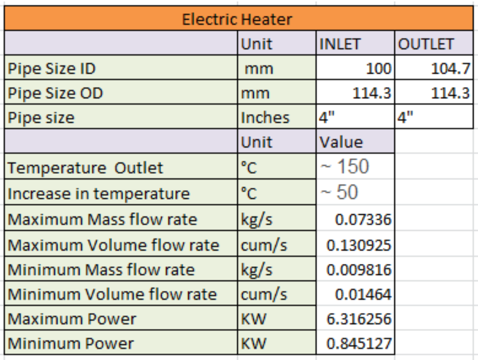
The flue gases coming out of the boiler (which consist out of steam and oxygen in standard condition) has a temperature of about 107°C, these will be mixed with fresh oxygen in the static mixer and recycled back to the boiler. The required temperature of the recycled stream at the mixer inlet is 150°C; hence a heater is required. For the sake of simplicity of the system it has been decided to use an electric heater. To determine and chose the optimal electric heater which can be used in this system, it is necessary to know the thermodynamic proprieties of the stream that is supposed to be heated. The memo design data which has already been presented is the base to define the electric heater data.

The load of the electric heater can be calculated with:

$$P[Kw] = \left(\dot{m}_{O_2 in\, rec.stream} * \overline{Cp}_{O_2} + \dot{m}_{H_2O\; in\, rec.stream} * \overline{Cp}_{H_2O}\right) * (150°C - 107°C) \qquad [10]$$

The aim of this chapter is to provide a memo design data for the electric heater, in order to simplify the selection. Therefore the minimum and the maximum load that the heater should have has to be defined; The formula used doesn't consider any efficiency of the heater or pressure drop along it. Hence the real load which will be required should actually be slightly higher that the value indicated. For instance a reliable value for the efficiency of an electric heater could be in a range of 85% to 93%.

In the 100% load case the recycled stream has to absorb a power of 6,1 kW. For the 75%, 50% and 25% load cases the required powers are 4,6 kW, 3,1 kW and 1,6 kW respectively. The 10% load case instead has been analyzed in the situations in which the adiabatic temperature in furnace is 1750°C and 515°C; the former represent the minimum load that the heat exchanger should have because in this case the amount of recycled stream will be the least. In fact the required power in this case is 0,8 kW. For the latter situation, the amount of recycled stream is the highest among all the other cases (even higher than the 100% load), hence this condition represents the maximum load of the heat exchanger: the power required is about 6,3 kW. In appendix {9} it is possible to see the table with the calculation for all the cases; while the main results are presented below:



Electric Heater			
	Unit	INLET	OUTLET
Pipe Size ID	mm	100	104.7
Pipe Size OD	mm	114.3	114.3
Pipe size	Inches	4"	4"
	Unit	Value	
Temperature Outlet	°C	~ 150	
Increase in temperature	°C	~ 50	
Maximum Mass flow rate	kg/s	0.07336	
Maximum Volume flow rate	cum/s	0.130925	
Minimum Mass flow rate	kg/s	0.009816	
Minimum Volume flow rate	cum/s	0.01464	
Maximum Power	KW	6.316256	
Minimum Power	KW	0.845127	

Figure 11: Design Data for the Electric Heater

It has been thought to use an “Air Duct” electric heater from the company Sinus Jevi, which could be either rectangular (model EEFK) or circular (model EEF) [5-1]. The final selection of the heater will be done by the company.

9. Recirculation Fan

The analysis of the gas recirculation fan was made analogous to that of the electric heater. The aim is to produce a memo design data to which the company can refer for the selection of the fan. Exactly as in the previous case the analysis is based on the thermodynamic properties of the affected streams. The fan in question is downstream the electric heater and its purpose is to bring the pressure of the recycled stream to a level that allows it to overcome pressure losses in the static mixer and be inserted into the burner wind-box. It is important to note that both the electric heater and the recirculation fan only work when the system uses fresh oxygen and recycled steam as an oxidizer. In all the other cases where the oxidant is different these two components are not used.

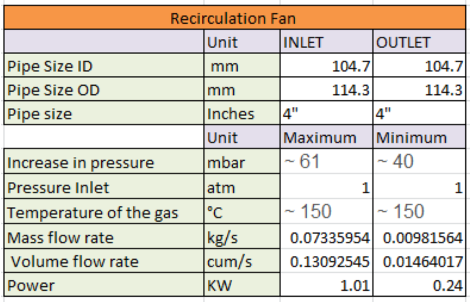
It was assumed that the fan doesn’t increase the temperature of the recycled stream, despite the fact that there is a slight increase of about maximum 1 degree.

The determination of the required load of the fan was done using the difference of enthalpy of the stream between the inlet and the outlet. (values are obtained by the use of AspenPlus)

$$P[Kw] = \dot{m}_{O_2 in\, rec.stream}(h_{O_2;50\, mbarg} - h_{O_2;0\, mbarg}) + \dot{m}_{H_2O\, in\, rec.stream}(h_{H_2O;50\, mbarg} - h_{H_2O;0\, mbarg}) \qquad [11]$$

In order to define the data necessary to select the recirculation fan it is essential to analyze the pressure drops that will occur during the path of the gas. It was considered that the gases needs to reach the inlet of the wind-box with a pressure of 40 mbarg. It is assumed that the in the burner and in the wind-box a pressure drop of 25 mbar and 15 mbar respectively will occur. The pressure at the inlet of the fan is close to the ambient pressure 1,01325 bar and the pressure increase is supposed to be 50 mbar. The pressure drop in the static mixer where considered to be maximum 10 mbar (in reality it’s even lower: 5-6 mbar); however it has already been calculated for some of the standard cases and the same method of analysis was used for missing situations. Furthermore the localized pressure drop along the pipe was considered. The same procedure used previously was exploited for this analysis. In this case it has been considered that the localized pressure drop are caused by two ball valves (friction factor - K=0,17), one flow meter (K = 7); one damper (K = 6,4) and two 90° curves (K = 4). [5-1]

The following table presents the results, while the calculation table is visible in Appendix {10}.



Recirculation Fan			
	Unit	INLET	OUTLET
Pipe Size ID	mm	104.7	104.7
Pipe Size OD	mm	114.3	114.3
Pipe size	Inches	4"	4"
	Unit	Maximum	Minimum
Increase in pressure	mbar	~ 61	~ 40
Pressure Inlet	atm	1	1
Temperature of the gas	°C	~ 150	~ 150
Mass flow rate	kg/s	0.07335954	0.00981564
Volume flow rate	cum/s	0.13092545	0.01464017
Power	KW	1.01	0.24

Figure 12: Design Data for the recirculation fan

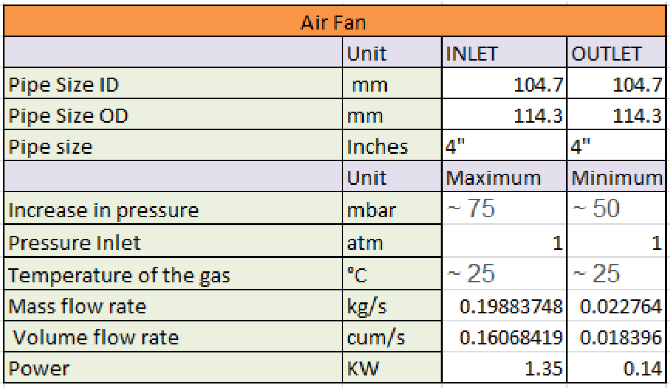
As well as the heat exchanger the maximum and minimum load are defined for the 10% load case; respectively for the situations in which the adiabatic temperature of combustion is 515°C and 1750°C.

The maximum load of the fan is about 1.01 kW; while the minimum load corresponds 0,24 kW. When the load is 100% the pressure drop in the static mixer and along the pipe are respectively equal to 5,2 mbar an 15,2 mbar, which means that the recirculation fan will have to increase the pressure of around 60,3 mbar. A similar result is obtained for the case with 10% load and adiabatic temperature equal to 515°C; in fact the total increase in pressure that the fan must give is 61 mbar (the static mixer pressure drop and localized pressure drops are equal to 5,7 mbar and 15,3 mbar). When the adiabatic temperature is set to 1750°C for the 10% case the pressure drop are the least (0,2mbar in the static mixer; 0,3 mbar along the pipe and 40,5 mbar in total). All the cases with a load within the maximum and the minimum will require a pressure increase between 60 mbar and 40 mbar.

The selection of the fan will be finalized by the company.

10. Air Fan

The study method for the selection of the air fan is very similar to that used for the recirculation fan. In this case, however, the system is analyzed in the situation in which it works with the use of air as oxidant. In this situation it is also considered that the pressure losses in the static mixer vary with the speed of the stream which in turn varies with the volumetric flow rate. In other words, the lower the load, the lower the pressure losses in the pipes and in the static mixer. For this analysis the localized pressure losses were also considered. These have already been calculated previously and the same iterative method has been applied to the other cases. In this way it was defined that to have an air pressure at the entrance of the wind box of 40 mbarg, it is necessary that the pressure increase of the fan is about 75 mbarg for cases where the load is greater; while for cases with 10% load the pressure increase can be limited to about 50 mbarg. Using the enthalpy variation formula given above for the gas conditions considered, the required power of the fan can be defined. All the situations reported in the main design data in which air is used as oxidant have been taken into consideration; the calculation table is available in Appendix {11}. Here below the results are presented:



Air Fan			
	Unit	INLET	OUTLET
Pipe Size ID	mm	104.7	104.7
Pipe Size OD	mm	114.3	114.3
Pipe size	Inches	4"	4"
	Unit	Maximum	Minimum
Increase in pressure	mbar	~ 75	~ 50
Pressure Inlet	atm	1	1
Temperature of the gas	°C	~ 25	~ 25
Mass flow rate	kg/s	0.19883748	0.022764
Volume flow rate	cum/s	0.16068419	0.018396
Power	KW	1.35	0.14

Figure 13

The power required by the air fan is slightly more than the one required by the recirculation fan: the maximum power required is about 1,35 kW and the minimum 0,15 kW (11%). This is because the required pressure increase is slightly higher as well as the volumetric flow rate, the reason is that in this case the oxygen is not coming from the storage, but from the air and it has to be pressurized to the desired level, while in the other case the control valve expands the oxygen and it doesn't have to pass through the fan. Furthermore it has been considered that the air is flowing at around 20 m/s while the recycled stream flows at around 15 m/s. Since the speed has a huge effect on the pressure drops, the air fan will absorb an higher power than the recirculation fan. The tables in appendix {11} shows that for the case with 100% load – combustion of hydrogen with air with the adiabatic temperature set to 2000°C the pressure drops in the SM and along the pipe are respectively 9,8 mbar and 21,3 mbar, about one third more than the pressure drops when using the recycled stream. For the combustion of methane with the adiabatic temperature set at 1845°C the pressure drops are similar to the hydrogen case: 11 mbar in the static mixer and 22,7 mbar along the pipes.

It has been considered that when the hydrogen is burning with air the adiabatic temperature should be around 1620°C with an excess of air in the wet flue gases equal to 7% in volume. This case cannot be compared functionally with the case in which the recycled stream is used because in that situation the adiabatic temperature is set in a different range of temperature. However it is expected that to reach the same adiabatic temperature the pressure drops in the static mixer and in the pipe are about one third more than the relative case with the recycled stream. However for the 10% load case (H2) analyzed the pressure drops in the Static Mixer and in the pipes are 2,5mbar and 7,8 mbar. For the 10% load case (NG) they are 2,9 mbar and 8,1 mbar.

Both for the recirculation fan and for the air fan also the pressure losses in the burner and in the wind-box actually change with the variation of flow rate quantity. However, these are considered constant in the applied calculations. It is assumed that in the case in which the system works with 10% load it is better to use a much lower adiabatic temperature compared to the standard conditions of 2000° C. If an adiabatic flame temperature of 515° C is considered, the amount of flow rate is similar to that of 100%. In any case it is advisable to have higher temperatures inside the boiler. For this reason during the tests the adiabatic flame temperature will be optimized for the 10% case in such a way as to be between the two limit conditions (1750 ° - 515 ° C).

11. Operating Phylosophy and Control & Safeguarding

This chapter is dedicated to the design of the plant's safety and control system. To this end, during the internship, the P&ID (Piping & Instrumentation Diagram) and the PFD (Process Flow Diagram) were designed.

The former shows all the valves, connections and transmitters that will be employed by the system. The latter instead shows the size of the pipes and the optimal values of pressure, temperature and mass flow rate that each line should have for every standard condition.

The realization of the P&ID was made with the guidance of the experience of the company Stork Thermeq B.V.. It describes the control system, also called DCS (Distributed Control System), and the security system, namely BMS (Burner Management System). The transmitters on each line can belong either to the DCS or to the BMS, and transmit the detected measurements to the relative CPU (Central Processing Unit). Each of the two systems is controlled by a different computer which will act differently on the equipment of the installation. The BMS is responsible for the safe start-up, operation and shutdown of the boiler. It monitors and controls igniters and main burners; utilizes flame scanners to detect and discriminate between the igniter and main flames; employs safety shut-off valves, pressure, temperature, flow and valve position limit switches and uses blowers to cool the scanners and/or provide combustion air for the igniters. Its proper operation is crucial to the safety of the boiler.

The DCS consists of a large number of local controllers in various sections of plant control area and are connected via a high speed communication network. The data acquisition and control functions are carried through a number of DCS transmitters distributed functionally over the plant and are situated near area where control or data gathering functions being performed. These controllers are able to communicate among themselves and also with other controllers like supervisory terminals, control valves, fans, dampers, etc.

11.1 Process Flow diagram

One of the objectives of the internship was to produce PFDs for a better understanding of the operation of the plant. The diagrams were created for the 100% standard conditions in several cases:

- 100% load – hydrogen- combustion with oxygen and recycled stream (excess oxygen in wet fumes = 3% in volume and adiabatic flame temperature = 2000 ° C)

-100% load – hydrogen- combustion with oxygen and nitrogen (excess oxygen in wet fumes = 2,89% in volume and adiabatic flame temperature = 2000 ° C)

-100% load – hydrogen- combustion with air (excess oxygen in wet fumes = 2,89% in volume and adiabatic flame temperature = 2000 ° C)

-100% load – Natural Gas combustion with air (excess oxygen in wet fumes = 2,63% in volume and adiabatic flame temperature = 2000 ° C)

-100% load – Natural Gas combustion with nitrogen and oxygen (excess oxygen in wet fumes = 2,63% in volume and adiabatic flame temperature = 2000 ° C)

The diagrams are visible in Appendix {12}.

11.2 Description of the P&ID

The P&ID has been divided into three parts. The first shows the lines for the supply of hydrogen and natural gas. The second presents the supply lines of the oxidizing gases (oxygen, nitrogen and air) to the static mixer and the last one shows the combustion chamber, the recirculation line and the cooling system of the boiler. The P&ID are visible in appendix {12}.

The lines that come from the storages (natural gas, hydrogen, oxygen and nitrogen) are more or less all similar and have more or less all the same equipment: the storages are connected to the line with a ball valve and a flange that separates the line from the storage, both can be operated manually. A filter ensures that there are no solid residues inside the gas or impurities. After that, all the lines present 7 transmitters. The first one is a pressure indicator used to show the pressure level of the storage; then three different transmitters for flow, pressure and temperature gives a signal to the BMS, the measured values are elaborated by the CPU and in this way it is possible to identify possible storage leaks, pipe obstructions or control valve malfunctions. Other three transmitters of temperature, flow and pressure report the measured values to the DCS, which processes them and acts on the control valve of the corresponding line, increasing or decreasing the incoming flow. It is considered that under the operation of the system the position of the control valve will vary continuously with respect to a stable position.

Subsequently the lines have two safety valves called "shut-off valves", which are solenoids valves managed by the BMS and can only be either closed or open; between the two valves there is another safety valve called "vent valve" which is open, when the two shut-off valves are closed (allowing the release of gases remaining in the line into the atmosphere) and it is closed when the shut-off valves are open (allowing the gas to normally flow in the line). After these, there is the control valve which is managed by the DCS and expands the gases to the required pressure, the correct functioning of this valve is measured by two pressure transmitters downstream the control valve (one for the BMS and one for the DCS). The lines end with a manual ball valve, a flexible pipe and a flange that allow the connection with the rest of the system.

The instrumentation described above is common to all the lines coming from the storages, but there are some differences among them:

The natural gas (NG) line has a second line (ignition line) that starts upstream the shut-off valves. It is connected by a manual ball valve to the main NG line and also has two shut-off valves, one vent valve, (controlled by the BMS) and a control valve, in this case self-regulated to the pre-set pressure, and ends with a manual valve, flexible pipe and flange which will then be connected to the ignitor.

The principal NG line is connected at the end with the hydrogen line; the properties of the flow are detected by a pressure and a temperature transmitter (downstream the connection) linked to the DCS, both the transmitters are meant mostly for indication and their detected measures do not actually act on any instrumentation. This line is then connected to the burner through a flexible pipe and a manual ball valve. In this way it is possible to supply different fuels to the burner without the need to change the connection. The oxygen line has a deviation, upstream the shut-off valves, which is connected to the burner flame scanner. This part of the line has a manual valve and a safety valve (shut-off valve) and is then connected to the flame scanner with a flexible pipe and a flange. This ensures that the flame scanner is cooled during the system operation. It has been chosen to use oxygen as cooling media, while air is normally used, because if the system works using fresh oxygen and recycled stream for hydrogen combustion, there could be some trace of nitrogen in the boiler if air was used.

The air supply line is slightly different from the rest of the supply lines: the air is taken from the environment, so there is no need for a flange connection and manual valve, in fact the first instrumentation on the line is the filter, after which there are the temperature and flow transmitters (for the DCS and for the BMS) upstream the air fan and pressure transmitters right downstream the air fan. A damper allows to close the line or reduce the incoming flow. A safety valve is positioned before the flexible pipe, manual valve and flange. In this case the transmitters connected to the DCS operate on the load of the air fan and on the damper and regulate the flow according to the request. The air supply line and the nitrogen line are connected together at their ends with the line of the recycled stream. This common pipe is then connected with a manual valve, a flexible pipe and a flange to the static mixer. The reason for this configuration has already been explained previously.

The static mixer outlet is then connected with a flange to the wind-box supply line. On this line there are three transmitters linked to the DCS (for pressure, temperature and composition) and it is entirely thermally insulated, to allow the recycled stream not to lose the temperature given by the electric heater and therefore not to allow a temperature drop or condensation of steam inside the pipe. Thanks to the three transmitters it is possible to know the pressure losses in the static mixer, the temperature reached by the mixture, in case the system works with the recycled stream, and the composition. If, for instance, there is a trace of incombustible hydrogen in the flue gases, it is possible to know it from this measurement. Furthermore it is possible to know the volume ratio of nitrogen and oxygen (in case they are coming from the storages) and adjust the flows in order to have the proper ratio.

The CPU of the DCS analyzes the data measured by the transmitters (described above), calculates the volumetric oxygen-fuel ratio and adjusts the control valves (or the air or recirculation fan) so as to obtain the desired ratio for the selected standard condition.

The burner has a flame scanner meant to detect the presence of the pilot flame which is connected to the BMS; another flame scanner, linked to the safety system, is positioned on top of the burner right upstream the cooled-water heat exchanger. This scanner is cooled by the oxygen line described above. In this position there are also two pressure transmitters and two temperature transmitters (for both, one is linked to the BMS and the other to the DCS). The safety transmitters make sure that the pressure and the temperature are in the range suitable for the optimal functioning of the system, in fact if the temperatures and pressures are too high the boiler could be damaged, while if these were too low there could be condensation inside the combustion chamber. The control transmitters indicate the temperature and the pressure values at the end of the part of the boiler characterized by the radiation heat transfer and they will be essential to define the quantities of oxidizing gasses and cooling water that are optimal for the standard operation of the plant.
At the boiler outlet the exhausted gases are divided, using the pressure difference caused by the recirculation fan; the gases directed towards the stack pass through two transmitters linked to the DCS. These two transmitters indicate the temperature and pressure of the stream directed towards the stack, but also of the recycled stream, since the properties of the two streams will be the same. On this line there is also a transmitter which will detect the level of oxygen inside the flue gases. These information, elaborated by the CPU, acts on the control valves of the fuel and oxidizing gases (or on the recirculation or air fan), so that there are no traces of unburnt fuel in the flue gas and so that the temperature is in the desired range. To this end, these measurements can also influence the load of the cooling water pump and adjust it in such a way as to obtain an outlet gas temperature of about 107 ° C. The stack line has a damper which can be closed or opened to force the exhausted gases to enter the line of the recycled stream. This damper is used in particular for the purge of the system and to regulate the split ratio between stack and recirculation line, as it will be explained later.

The recycled stream line is thermally insulated in order to avoid temperature losses. The line has a temperature indicator before the electric heater and a temperature transmitter after, the latter is linked to the DCS and regulates the load of the electric heater in such a way that the outlet temperature remains 150° C. After this there is the recirculation fan, followed by pressure, temperature and flow transmitters for both the DCS and the BMS. These transmitters work for the same purpose as the transmitters on the supply lines already described above. The transmitters of the BMS make sure that the recirculation fan works (positive pressure), that there is no leakage from the pipes and that there are no obstructions in the pipes. While the DCS transmitters regulate the flow rate of the recycled stream by acting on the fan load and on the damper positioned on the line. This damper can force the gases to pass through the stack line when it’s closed and can help to regulate the recirculation stream flow rate. The line is then connected with a ball valve, a flexible pipe and flange to the air line. Which is then connected to the nitrogen line and then to the static mixer.
Finally the cooling water line has a flow transmitter before the pump, this is linked to the DCS, and regulates the pump load. Downstream the pump there is a temperature control transmitter that indicates the inlet temperature of the cooling water into the boiler; this should be fixed at 60 ° C, but slight variations may occur, in

this case the water flow should be adjusted so as to maintain the outlet temperature within the boiler maximum limit temperature of 110 ° C and the outlet flue gases temperature around 107°C.

At the boiler outlet the pressure and temperature of the cooling water are measured for both the control system and the safety system. In this case the DCS acts on the load of the pump, increasing it and decreasing it so as to maintain the outlet temperature close to 105 ° C, furthermore it measures the pressure losses inside the boiler. The BMS checks that the temperature is in an optimum range for system operation, ensures that the temperature does not exceed the safety limit and that there are no excessive pressure losses that could be caused by leakage in the line or malfunction of the pump.

11.3 Start-up and shutdown

The system start-up and shutdown procedures are regulated by the security system (BMS). The plant can be turned on using air or the mixture of nitrogen and oxygen as oxidizing gas. When the system is on it can be used with the starting oxidizing gas and increase the load to the desired level, or it can switch to the fresh oxygen and recycled stream condition. The system can be turned on using either hydrogen or natural gas; the possibility of changing the fuel type during the operation of the system has not been considered.

The procedures for switching on, off and switching from one condition to another are described precisely in the Appendix {13}. Here the philosophy with which these procedures have been made will be described.

Before starting-up the system the BMS must check for the starting condition: all the shut-off valves on all the lines have to be closed, and all the vent valves has to be open. Once selected the type of fuel that will be used, the control valve on the selected line has to be in the starting position (it is considered that the system starts up with a load of about 25% the maximum power). The BMS must check that there is no presence of flame in the burner and in the boiler; that the damper of the air line and on the stack line are open, and the damper on the recirculation line is closed. Then it can start purging the system by activating the air fan, (it has to make sure that air fan is working by checking the positive pressure in the air line). The fan has to run for at least 30 seconds in order to make sure that the volume of air that will flow inside the boiler is at least 5 times the volume of boiler itself. Then the recycled line has to be purged to make sure that there is no trace of fuel inside the line, to this purpose the damper of the stack will be closed, the damper on the recirculation line will be open, the air fan will ran for at least other 30 seconds, together with the recirculation fan, then the two dampers comes back to the initial position and the recirculation fan is switched off . The system could also be purged with pure nitrogen, which would actually be better because it is an inert gas, but the costs of storing it doesn’t make this solution economically feasible. Once the purged had been done the system can start up by keeping the air fan on or by using the mixture of oxygen and nitrogen coming from the storages. In the latter case the air fan is switched off, the nitrogen and oxygen shut-off valves are opened and their vent valves are closed (after checking that their control valve are in the starting position); the volume ratio oxygen-nitrogen is checked by the composition transmitter downstream the static mixer and regulated in order to have an oxygen percentage in volume between 20% and 20,9%. Then the ignitor starts sparking and after 5 seconds, the NG ignition line is opened. At this point the pilot flame should be on, and this checked by the flame scanner in the burner. If the pilot flame is on the shut-off valves, on the desired fuel line, can be opened (and the relative vent valve closed). The BMS check the flame presence in the boiler for 5 seconds, if the response is positive, the pilot flame is switched off by closing the shut off valves on the ignition line.

When the system is running at a power higher than 25% the load must be brought to the 25% condition, then the system can be shut-down using the normal procedure. Nevertheless it's possible that, for safety reasons, the plant must be switched off when running at any load. In this case the same procedure of normal shutting down will be applied and however considered as an emergency shutdown.

In case the transmitters of the composition detect a volume percentage of fuel in the flue gases or a lower excess of oxygen than the one expected, there will be unburned fuel in the exhausted gases leading to fire or explosion risk, hence the fuel flow rate must be reduced to the start condition (25%) for 15 seconds, to make sure that all the fuel is fully burnt.

The first thing to do to turn off the system is to stop feeding the system with the fuel. So the shut-off valves of the used fuel line must be closed and the relative vent valve must be opened. However it still necessary to keep running the system to make sure that there is no unburned fuel in the exhausted gases. Hence if the system is running using the recycled stream, the recirculation fan must keep running and slowly its load must be reduced, as well as the fresh oxygen flow must be reduced. If it's running using the mixture of nitrogen and oxygen the control valves on those two lines has to slowly be closed. In any case the air fan must be switched on in order to purge the system and prevent any fuel accumulation in any part of the system. The load of the cooling water pump must be reduced, until it is off. When all the control valves are closed and the pump and the recirculation fan are off, the air fan must still run for 2 more minutes to make sure that there is not undesired fuel in any part of the system. To this purpose the recirculation damper must be closed and the stack damper must be fully open.

11.4 Switching from one standard condition to another

It was already mentioned that the system can start-up only using air or the nitrogen-oxygen mixture, therefore it is necessary to know the procedure that the system must do when switching to the standard situation that use the recycled stream to cool down the combustion. It has been considered that the change of condition can only be done for loads lower than 25%, in particular it has been analyzed for 10% load, but the same procedure can be applied for any load within the range.

The case in which the system switch from using the oxygen-nitrogen mixture to the oxygen-recycled steam mixture as oxidizing gas has been analyzed in detail, however the same philosophy of operation has to be applied for any switching procedure.

The basic procedure to be applied is based on the fact that before decreasing the flow rate of the oxidizing gas previously used, the flow of the next oxidizing gas must be increased, to maintain the adiabatic temperature constant. In the example of the case study analyzed, the recirculation fan must be turned on before the nitrogen flow rate is reduced. After that the procedure works by step: the flow rate of recycled gas is increased and the nitrogen flow rate is reduced, a stable situation is reached, in which the adiabatic combustion temperature must remain constant and then it must be continued to proceed by steps until the nitrogen flow rate is zeroed and the system runs only with the recycled stream. Vice versa: to switch from the condition with the recycled stream to the one with the nitrogen-oxygen mixture, it is first needed to increase the nitrogen flow rate and then reduce the flow rate of the recycled stream, wait for a stable condition and repeat the process step by step until the system reaches the final condition.

To analyze the case study in which the system passes from the combustion of hydrogen (10% load) with the mixture of oxygen-nitrogen and an excess of oxygen in the wet fumes equal to 4% in volume to the hydrogen combustion condition (10% load) with the recycled stream and an excess of oxygen in the wet fumes equal to 7% in volume. This final condition is in fact unrealizable, since it has been established that the excess of air required in the fumes for the 10% load case should be about 20% in volume. The analysis made is to be considered a pure example to understand the behavior of the system during the change from one condition to another, but the results will not be really usable for the construction of the plant. The analysis has been conducted using the software AnsysPlus and using an iteration process made step by step, changing the input data in order to keep the adiabatic flame temperature stable at 1750°C. It has been considered that the flow rate of fresh oxygen doesn't change with the change of the condition; only the nitrogen flow rate and the recirculation flow rate changes over the time. The iteration has been done for "arbitrary steps" which represents the amount of time that is necessary to reach the stable condition.

The results are represented in the following graphs:

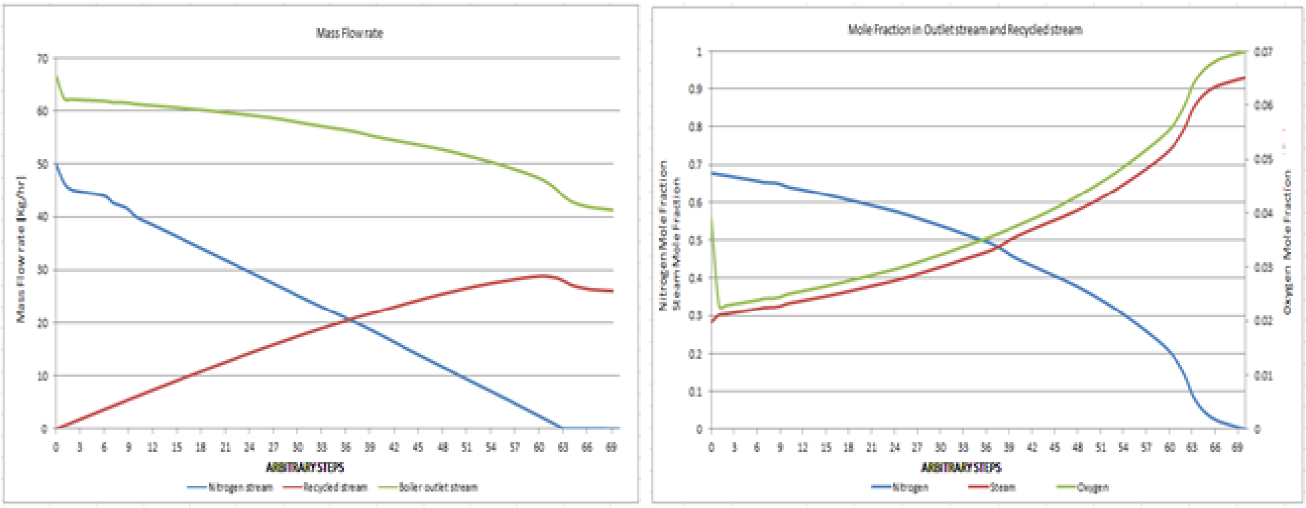


Figure 14: Switching condition from "10% load combustion oh Hydrogen with Nitrogen and oxygen" to "10% load combustion oh Hydrogen with Oxygen and recycled stream"

The graphs shows that as soon as the reciruclation fan starts working, the nitrogen flow rate must be reduced fastly in order to mainten the same temperature in the boiler. Since there is a large ammount of steam and nitrogen in the boiler, the excess of oxygen in the wet fumes drops from 4% to 3%; while the oxygen-fuel ratio remains stable. Then the changment is stable: the nitrogen keeps decreasing, while the recycled stream keeps increasing, causing an increase of the the excess of oxygen in the wet gases. When the nitrogen flow rate is equal to zero, there will still be some nitrogen in the recycled stream, which will all go slowly to the stack. Since the heat capacity of the steam is higher than the one of nitrogen, a lower ammount of recycled stream is needed when there is no more nitrogen in the recycled stream. Hence this analysis shows that to keep the temeprature in boiler stable in is necessary to decrease the recirculation flow rate, once the nitrogen flow rate is equal to zero, otherwise there will be a temperature drop in the boiler.

The last consideration that it is important to mention regards the change of load within the same operation conditions, e.g. when the load increases and the system keeps running using the recycled stream. Whenever the load increase it is always important to first increase the oxiding gas flow rate (which is being used) and then increasing the fuel flow rate. This is because if it was the other way around there will be some unburnerd fuel in the system which it's necessary to avoid. Whenever the load decrease the procedure is the other way around: first the fuel flow rate must be decreased and then the oxydizing gas flow rate.

12. HAZOP

The Hazard and Operability (HAZOP) study technique is a widely recognized and well-established method of safety review. The primary objective of the HAZOP study is to identify the causes, consequences and existing safeguards for credible hazards. A major concern of using hydrogen as fuel is the safety issues which could also be an obstacle to expand its commercial implementation. Hydrogen safety issue is always of significant importance to secure the property. In order to develop a dedicated safety analysis method for hydrogen energy system in power industry, the risk analysis for the plant was made. The HAZOP study will be performed before the installation, to identify the most problematic parts of the system in view of hydrogen safety and possible failure modes and consequences. Important safeguards and mitigation measures are proposed based on the risk and safety studies.[6-1][6-2]

The methodology involves a structured, systematic and comprehensive examination of process flow sheets, flow diagrams, plant/facility layouts or procedures in order to identify potential hazards and operability problems. The study is undertaken by a multi-disciplinary team familiar with the process undergoing examination and a chairman. The role of the chairman is to guide and encourage the study team through the examination process to identify all possible hazard scenarios. The team also requires a secretary to formally record the discussions and findings of the study. HAZOPs, thus, provide a method for individuals in a team to visualize ways in which a plant can malfunction or mal-operate. [6-2] It was acknowledged that the safe operation of the system is highly dependent on the knowledge and experience of the current hydrogen experts. [6-3]

Therefore the HAZOP analysis should be done during a meeting of experts on the topic; this will be conducted after the end of the internship. For this reason this chapter will be limited to describe a first approach that could be used to carry out the study, and it will provide an initial development of the structure that the HAZOP will have.

To develop the analysis it was thought that any system malfunction will be detected in some way by the safety and control transmitters as anomalies. That is to say, for each standard condition, each transmitter should detect a measurement within an acceptable range for an optimal system operation. If the measured value deviates from the desired range, it means that there are malfunctions in the system. In the table shown in Appendix 14, it was considered that each transmitter can detect a value which can differ in excess or in default from the reference range. For each possible deviation it was indicated which could be the possible causes, which could be the possible effects that this variation may cause and the action that the BMS or the DCS will have to do to resolve or report the problem.

Despite the table has the possibility to fully report all the possible problems that may occur to the system and to give an action procedure for each case, the structure of the table does not make the analysis easy to read; a better and simplified analysis will have to be written during the HAZOP meeting.

13. Conclusion and recommendations

The internship aimed to define a set-up for the pilot plant for direct hydrogen combustion with a nominal power of 500 kW (LHV). The technical characteristics of the instrumentation used have been defined. The system has been presented from the point of view of the mass and energy balances, instrumentation involved and from the point of view of the operation. The report drawn up will serve as the basis for the final construction of the plant. The P&ID's and PFD's drawn will be used as a guide for the installation of the system and the HAZOP study could be used as a reference for internal analysis that will be carried out subsequently. The analysis of the plant has shown that there are numerous technical difficulties in combustion of hydrogen compared to that of natural gas: a higher flame temperature, a lower density that forces to have higher speeds in the pipes. Moreover the differences that occur when the system works with a power load lower than the standard one were shown. Since the combustion of hydrogen is considered to be a clean energy source the potential of this technology is enormous and the research done will help the its commercial implementation.

In order to have a more complete view of the technology, it is necessary to consider the efficiencies of the hydrogen fuel production and its storage, as these can have very negative influences on the final energy efficiency. To have a 360 ° view of the system it would be necessary to consider what is the ratio between the final energy output and the primary energy source with which the fuel is produced (example: the electricity used for electrolysis). Furthermore, for the commercial implementation of the technology, the system should be able to produce superheated steam which could be used for the production of electricity; to do this, the configuration of the plant should be changed to (in particular the design of the boiler) in such a way that the cooling medium reaches much higher temperatures.

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15. Appendix

APPENDIX 1 : Mass and Energy Balance

The reaction considered is the following:

$$H_2 + \frac{1}{2} O_2 \rightarrow H_2O \qquad \{1\}$$

Hess' law is used to determine the Low Heating Value (LHV) of hydrogen, in this case it can be written as:

$$\Delta H_{H_2} = \sum_i n_i \, \Delta H^0_{f;i_{products}} - \sum_j n_j \, \Delta H^0_{f;j_{reactants}} = -241{,}82 \frac{KJ}{mol} \qquad \{2\}$$

Where ΔH^0_f is the enthalpy of formation of every specie involved in the reaction, "i" represents the products and "j" the reactants; "n" is the stoichiometric coefficient of the relative specie. $\Delta H_{H_2}$is the energy produced by the reaction. The LHV can be written as the energy produced per mole of hydrogen divided the molecular weight:

$$LHV = \frac{\Delta H_{H_2}}{PM_{H_2}} \approx 120 \frac{MJ}{Kg} \qquad \{3\}$$

The required mass flow rate of hydrogen to give a nominal power P_n of 500 MW (LHV) is:

$$\dot{m}_{H_2} = \frac{P_n}{LHV} \left[\frac{kg}{s}\right] \qquad \{4\}$$

By dividing the mass flow rate by the molecular weight of hydrogen the molar flow rate can easily be found. Then the molar flow rate for the stoichiometric oxygen can be calculated by multiplying the latter for the stoichiometric coefficient (0,5). The molar flow rate of the produced water by the reaction is equal to the molar flow rate of hydrogen. The relative mass flow rates can be founded by multiplying for the relative molecular weight. The energy contained in the inlet streams is calculated by multiplying for the relative heat capacity and inlet temperature in Kelvin.

The enthalpy of the steam produced is calculated using the following energy balance:

$$\sum (\dot{m}_{in} * h) + P_n = \sum \dot{m}_{out} * h \qquad \{5\}$$

For this case equation {5} can be written as:

$$h_{steam-out(adiabatic\,T)} = \frac{\dot{m}_{H_2(25°C)} * (h_{H_2} + LHV) + \dot{m}_{O_2(25°C)} * h_{O_2}}{\dot{m}_{steam-out}} \qquad \{6\}$$

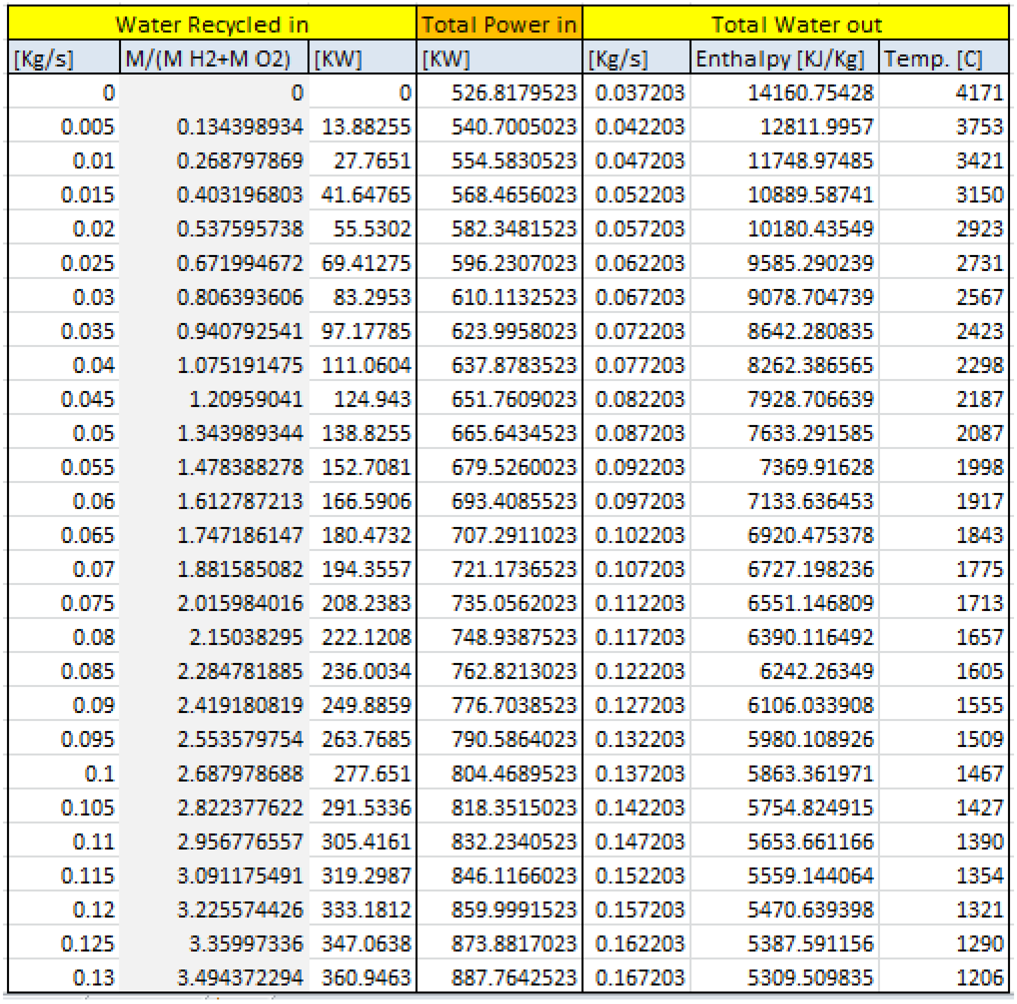
Hence it is found the in absence of any cooling media the adiabatic temperature of flame of hydrogen burnt in pure oxygen is 4171°C.

By adding the recirculation stream to the balance expression 6 can be written as:

$$h_{steam-out(adiabatic\,T)} = \frac{\dot{m}_{H_2(25°C)} * (h_{H_2} + LHV) + \dot{m}_{O_2(25°C)} * h_{O_2} + \dot{m}_{steam-in\,(150°C)}}{\dot{m}_{steam-out}} \qquad \{7\}$$

Then, equation {5} is readapted for the recirculation case. The following table summarizes the results. The column " M/ (M H2 +M O2)" represents the recirculated mass flow rate ratio, expressed as:

$$\dot{m}_f = \frac{\dot{m}_{recycled\ stream}}{\dot{m}_{H_2} + \dot{m}_{O_2}} \qquad \{8\}$$



Water Recycled in			Total Power in	Total Water out		
[Kg/s]	M/(M H2+M O2)	[KW]	[KW]	[Kg/s]	Enthalpy [KJ/Kg]	Temp. [C]
0	0	0	526.8179523	0.037203	14160.75428	4171
0.005	0.134398934	13.88255	540.7005023	0.042203	12811.9957	3753
0.01	0.268797869	27.7651	554.5830523	0.047203	11748.97485	3421
0.015	0.403196803	41.64765	568.4656023	0.052203	10889.58741	3150
0.02	0.537595738	55.5302	582.3481523	0.057203	10180.43549	2923
0.025	0.671994672	69.41275	596.2307023	0.062203	9585.290239	2731
0.03	0.806393606	83.2953	610.1132523	0.067203	9078.704739	2567
0.035	0.940792541	97.17785	623.9958023	0.072203	8642.280835	2423
0.04	1.075191475	111.0604	637.8783523	0.077203	8262.386565	2298
0.045	1.20959041	124.943	651.7609023	0.082203	7928.706639	2187
0.05	1.343989344	138.8255	665.6434523	0.087203	7633.291585	2087
0.055	1.478388278	152.7081	679.5260023	0.092203	7369.91628	1998
0.06	1.612787213	166.5906	693.4085523	0.097203	7133.636453	1917
0.065	1.747186147	180.4732	707.2911023	0.102203	6920.475378	1843
0.07	1.881585082	194.3557	721.1736523	0.107203	6727.198236	1775
0.075	2.015984016	208.2383	735.0562023	0.112203	6551.146809	1713
0.08	2.15038295	222.1208	748.9387523	0.117203	6390.116492	1657
0.085	2.284781885	236.0034	762.8213023	0.122203	6242.26349	1605
0.09	2.419180819	249.8859	776.7038523	0.127203	6106.033908	1555
0.095	2.553579754	263.7685	790.5864023	0.132203	5980.108926	1509
0.1	2.687978688	277.651	804.4689523	0.137203	5863.361971	1467
0.105	2.822377622	291.5336	818.3515023	0.142203	5754.824915	1427
0.11	2.956776557	305.4161	832.2340523	0.147203	5653.661166	1390
0.115	3.091175491	319.2987	846.1166023	0.152203	5559.144064	1354
0.12	3.225574426	333.1812	859.9991523	0.157203	5470.639398	1321
0.125	3.35997336	347.0638	873.8817023	0.162203	5387.591156	1290
0.13	3.494372294	360.9463	887.7642523	0.167203	5309.509835	1206

Fig. 1: Calculation for the dependency of the adiabatic flame temperature on the recirculation ratio

APPENDIX 2 : Memo with design data

To determine the proprieties of all the streams involved in the process it is needed to relate all the flow rates to the hydrogen flow rate. The amount of oxygen which enters the boiler, after the static mixer will be:

$$\dot{m}_{O_2-static\ mixer\ downstream} = \frac{\dot{m}_{H_2}}{PM_{H_2}} * \frac{1}{2} * (1+\varepsilon) * PM_{O_2} \qquad \{9\}$$

The amount of water produced in the combustion is:

$$\dot{m}_{steam-produced} = \frac{\dot{m}_{H_2}}{PM_{H_2}} * PM_{H2O} \qquad \{10\}$$

The amount of recycled stream and the amount of the fresh oxygen required has to be calculated together, taking into consideration the mass and energy balances for the static mixer and the boiler itself. The energy and mass balances for the static mixer are:

$$\dot{m}_{O_2-fresh} * Cp(T) * T_{O_2(25°C)} + \dot{m}_{H20-rec.str.} * Cp(T) * T_{H_2O(150°C)} + \dot{m}_{O2-rec.str.} * Cp(T) * T_{O2(150°C)} = \dot{m}_{O_2-static\ mixer\ downstream} Cp(T) * T_{Final} + \dot{m}_{H20-rec.str.} * Cp(T) * T_{Final} \quad \{11\}$$

$$\dot{m}_{O_2-fresh} + \dot{m}_{O2-rec.str.} = \dot{m}_{O_2-static\ mixer\ downstream} \quad \{12\}$$

$$\dot{m}_{O2-rec.str.} = \frac{\dot{m}_{rec.str.}}{PM_{rec.str.}} \varepsilon * PM_{O2} = \frac{\dot{m}_{rec.str.} * \varepsilon * PM_{O2}}{(1-\varepsilon) * PM_{H2O} + \varepsilon * PM_{O2}} \quad \{13\}$$

Using the three equations written above it is possible to calculate the T_{Final}, which is the temperature of the stream after the static mixer, which will be composed by the recycled stream (water and oxygen) and the fresh oxygen coming from the storage. The final temperature will be a function of the mass flow rate of recycled stream $\dot{m}_{rec.str.}$. which can be calculated using the energy and mass balance of the boiler, or more precisely only of the combustion. Because the energy balance of the boiler itself should also consider the cooling water which is supposed to extract the heat from the exhausted gases and bring them from a temperature of 2000°c to roughly about 107°C. As explained in the report the heat transfer with the cooling medium in the boiler has been done separately, assuming the heat exchanger as external of the boiler. The energy balance for the combustion is:

$$\dot{m}_{O_2-static\ mixer\ downstream} Cp(T) * T_{Final} + \dot{m}_{H20-rec.str.} * Cp(T) * T_{Final} + \dot{m}_{H_2(25°C)} * (h_{H_2} + LHV) = = (\dot{m}_{steam-produced} + \dot{m}_{H20-rec.str.}) Cp(T) * T_{adiabatic} + \dot{m}_{O_2-static\ mixer\ downstream} Cp(T) * T_{adiabatic} \quad \{14\}$$

From the equations cited above it is possible to notice that all the parameters of the streams involved in the system can be calculated by assuming two data: the adiabatic temperature of flame and the excess of oxygen in the exhausted gases. Of course the fresh oxygen required will be the difference of the total oxygen required in the boiler (which is expressed with $\dot{m}_{O_2-static\ mixer\ downstream}$) minus the amount of oxygen already present in the recycled stream $\dot{m}_{O2-rec.str.}$ which is depending on the amount of recycled stream which depends on the desired adiabatic temperature.

For all the other cases analyzed and presented here below the calculations has been made in the same way, applying mass and energy balances where applicable and considering the different reactions involved. i.e. combustion of methane instead of hydrogen; no use of static mixer for the cases which involves nitrogen, because nitrogen becomes the cooling medium substituting the recycled steam.

The following picture helps to visualize the streams and the process involved in the system. (note that the boiler and the heat exchanger downstream are in reality one single element). Note that the heat exchanger after the boiler is in reality part of the boiler. To simplify the calculation it is assumed that the combustion reaction takes place in the boiler and its outlet is at the combustion adiabatic temperature: the Stream "OUT" is cooled by cooling water in the heat exchanger, but this happen in reality in the boiler itself, as it is analyzed in the relative chapter.

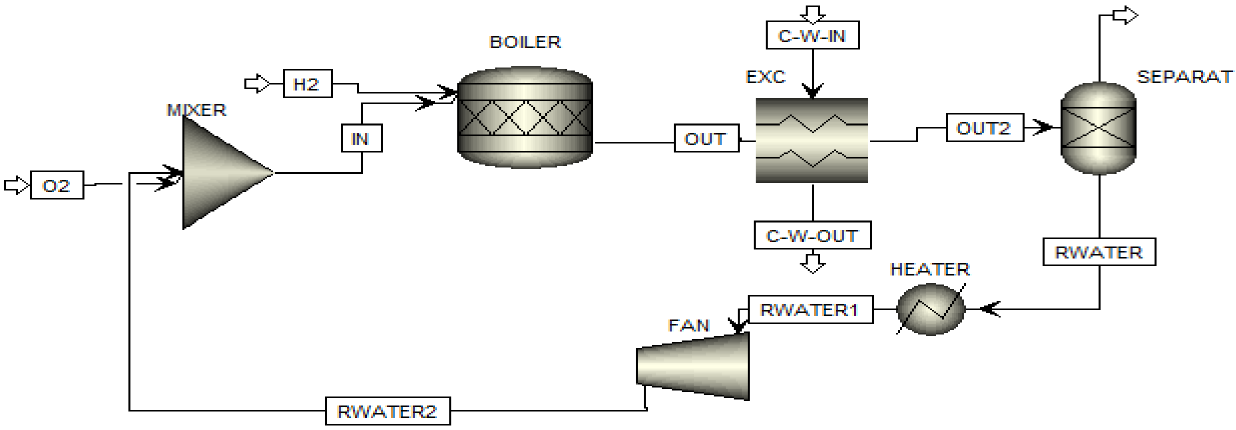
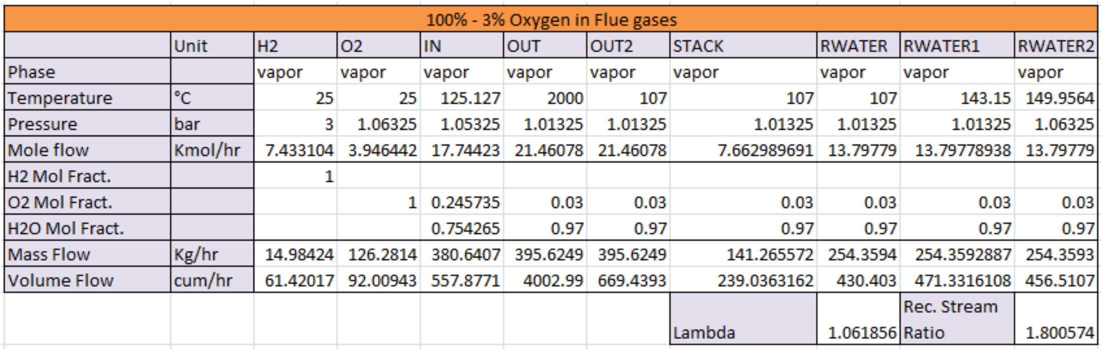
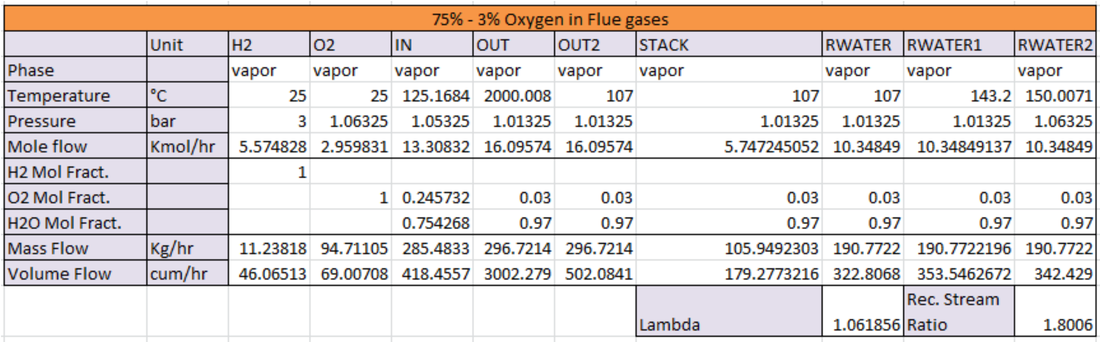


Fig. 2: scheme of the plant for operation with the use of the recycled stream



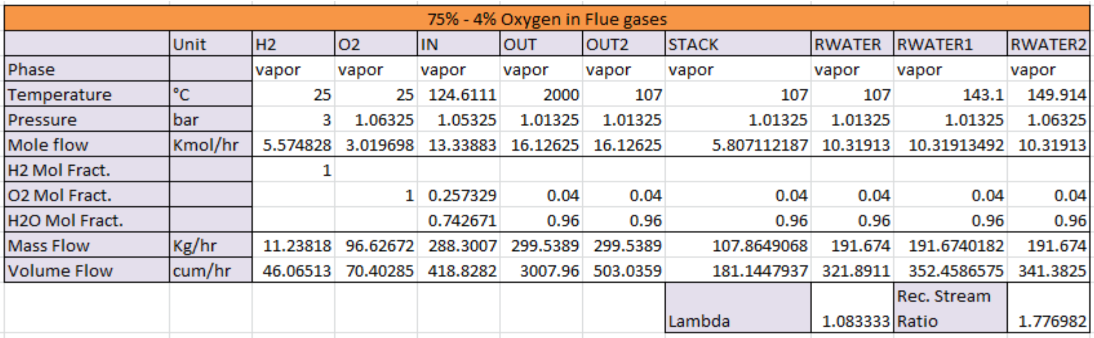
100% - 3% Oxygen in Flue gases										
	Unit	H2	O2	IN	OUT	OUT2	STACK	RWATER	RWATER1	RWATER2
Phase		vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	125.127	2000	107	107	107	143.15	149.9564
Pressure	bar	3	1.06325	1.05325	1.01325	1.01325	1.01325	1.01325	1.01325	1.06325
Mole flow	Kmol/hr	7.433104	3.946442	17.74423	21.46078	21.46078	7.662989691	13.79779	13.79778938	13.79779
H2 Mol Fract.		1								
O2 Mol Fract.			1	0.245735	0.03	0.03	0.03	0.03	0.03	0.03
H2O Mol Fract.				0.754265	0.97	0.97	0.97	0.97	0.97	0.97
Mass Flow	Kg/hr	14.98424	126.2814	380.6407	395.6249	395.6249	141.265572	254.3594	254.3592887	254.3593
Volume Flow	cum/hr	61.42017	92.00943	557.8771	4002.99	669.4393	239.0363162	430.403	471.3316108	456.5107
							Lambda	1.061856	Rec. Stream Ratio	1.800574

Fig. 3: Memo design data – Combustion of hydrogen with oxygen and recycled stream – 100% load – Adiabatic Temperature =2000°C – Excess of oxygen in the wet flue gases = 3% in volume



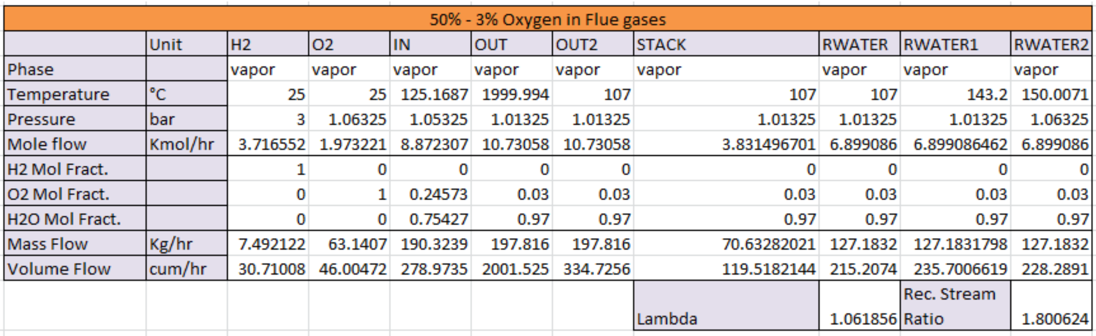
75% - 3% Oxygen in Flue gases										
	Unit	H2	O2	IN	OUT	OUT2	STACK	RWATER	RWATER1	RWATER2
Phase		vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	125.1684	2000.008	107	107	107	143.2	150.0071
Pressure	bar	3	1.06325	1.05325	1.01325	1.01325	1.01325	1.01325	1.01325	1.06325
Mole flow	Kmol/hr	5.574828	2.959831	13.30832	16.09574	16.09574	5.747245052	10.34849	10.34849137	10.34849
H2 Mol Fract.		1								
O2 Mol Fract.			1	0.245732	0.03	0.03	0.03	0.03	0.03	0.03
H2O Mol Fract.				0.754268	0.97	0.97	0.97	0.97	0.97	0.97
Mass Flow	Kg/hr	11.23818	94.71105	285.4833	296.7214	296.7214	105.9492303	190.7722	190.7722196	190.7722
Volume Flow	cum/hr	46.06513	69.00708	418.4557	3002.279	502.0841	179.2773216	322.8068	353.5462672	342.429
							Lambda	1.061856	Rec. Stream Ratio	1.8006

Fig. 4: Memo design data – Combustion of hydrogen with oxygen and recycled stream – 75% load – Adiabatic Temperature =2000°C – Excess of oxygen in the wet flue gases = 3% in volume



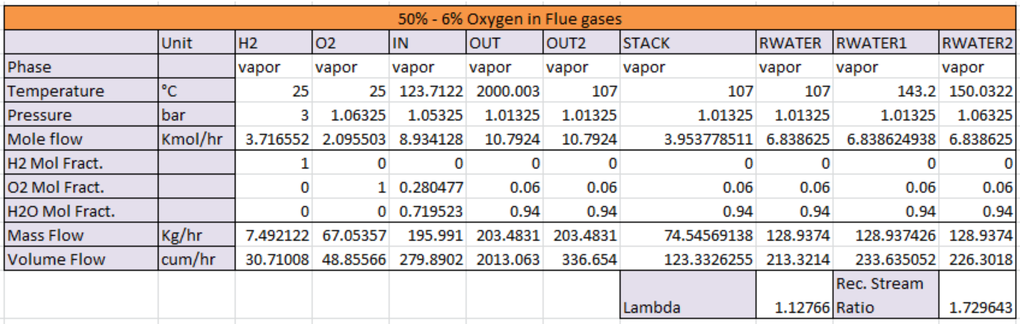
75% - 4% Oxygen in Flue gases										
	Unit	H2	O2	IN	OUT	OUT2	STACK	RWATER	RWATER1	RWATER2
Phase		vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	124.6111	2000	107	107	107	143.1	149.914
Pressure	bar	3	1.06325	1.05325	1.01325	1.01325	1.01325	1.01325	1.01325	1.06325
Mole flow	Kmol/hr	5.574828	3.019698	13.33883	16.12625	16.12625	5.807112187	10.31913	10.31913492	10.31913
H2 Mol Fract.		1								
O2 Mol Fract.			1	0.257329	0.04	0.04	0.04	0.04	0.04	0.04
H2O Mol Fract.				0.742671	0.96	0.96	0.96	0.96	0.96	0.96
Mass Flow	Kg/hr	11.23818	96.62672	288.3007	299.5389	299.5389	107.8649068	191.674	191.6740182	191.674
Volume Flow	cum/hr	46.06513	70.40285	418.8282	3007.96	503.0359	181.1447937	321.8911	352.4586575	341.3825
							Lambda	1.083333	Rec. Stream Ratio	1.776982

Fig. 5: Memo design data – Combustion of hydrogen with oxygen and recycled stream – 75% load – Adiabatic Temperature =2000°C – Excess of oxygen in the wet flue gases = 4% in volume



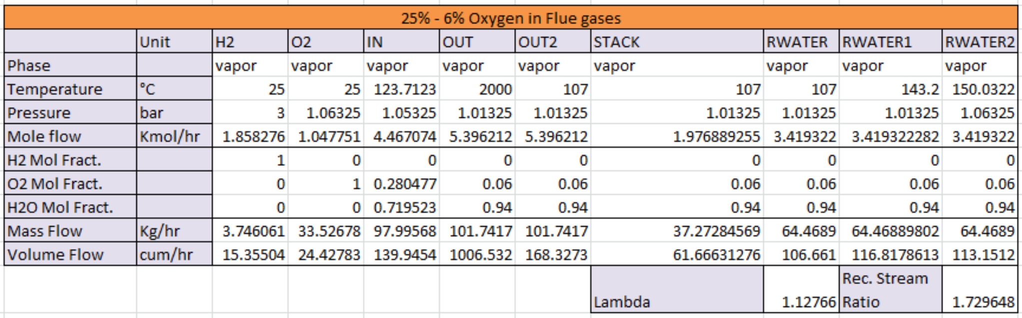
50% - 3% Oxygen in Flue gases										
	Unit	H2	O2	IN	OUT	OUT2	STACK	RWATER	RWATER1	RWATER2
Phase		vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	125.1687	1999.994	107	107	107	143.2	150.0071
Pressure	bar	3	1.06325	1.05325	1.01325	1.01325	1.01325	1.01325	1.01325	1.06325
Mole flow	Kmol/hr	3.716552	1.973221	8.872307	10.73058	10.73058	3.831496701	6.899086	6.899086462	6.899086
H2 Mol Fract.		1	0	0	0	0	0	0	0	0
O2 Mol Fract.		0	1	0.24573	0.03	0.03	0.03	0.03	0.03	0.03
H2O Mol Fract.		0	0	0.75427	0.97	0.97	0.97	0.97	0.97	0.97
Mass Flow	Kg/hr	7.492122	63.1407	190.3239	197.816	197.816	70.63282021	127.1832	127.1831798	127.1832
Volume Flow	cum/hr	30.71008	46.00472	278.9735	2001.525	334.7256	119.5182144	215.2074	235.7006619	228.2891
							Lambda	1.061856	Rec. Stream Ratio	1.800624

Fig. 6: Memo design data – Combustion of hydrogen with oxygen and recycled stream – 50% load – Adiabatic Temperature =2000°C – Excess of oxygen in the wet flue gases = 3% in volume



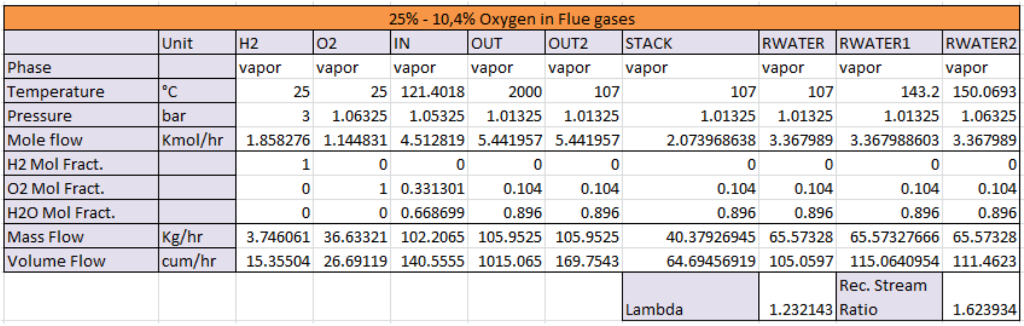
50% - 6% Oxygen in Flue gases										
	Unit	H2	O2	IN	OUT	OUT2	STACK	RWATER	RWATER1	RWATER2
Phase		vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	123.7122	2000.003	107	107	107	143.2	150.0322
Pressure	bar	3	1.06325	1.05325	1.01325	1.01325	1.01325	1.01325	1.01325	1.06325
Mole flow	Kmol/hr	3.716552	2.095503	8.934128	10.7924	10.7924	3.953778511	6.838625	6.838624938	6.838625
H2 Mol Fract.		1	0	0	0	0	0	0	0	0
O2 Mol Fract.		0	1	0.280477	0.06	0.06	0.06	0.06	0.06	0.06
H2O Mol Fract.		0	0	0.719523	0.94	0.94	0.94	0.94	0.94	0.94
Mass Flow	Kg/hr	7.492122	67.05357	195.991	203.4831	203.4831	74.54569138	128.9374	128.937426	128.9374
Volume Flow	cum/hr	30.71008	48.85566	279.8902	2013.063	336.654	123.3326255	213.3214	233.635052	226.3018
							Lambda	1.12766	Rec. Stream Ratio	1.729643

Fig. 7: Memo design data – Combustion of hydrogen with oxygen and recycled stream – 50% load – Adiabatic Temperature =2000°C – Excess of oxygen in the wet flue gases = 6% in volume



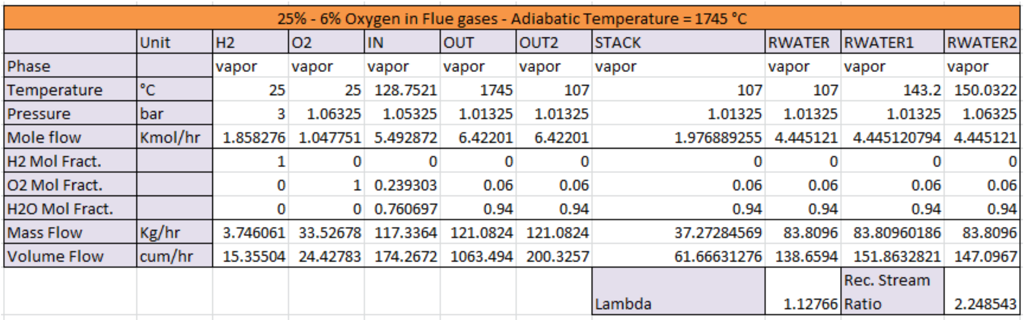
25% - 6% Oxygen in Flue gases										
	Unit	H2	O2	IN	OUT	OUT2	STACK	RWATER	RWATER1	RWATER2
Phase		vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	123.7123	2000	107	107	107	143.2	150.0322
Pressure	bar	3	1.06325	1.05325	1.01325	1.01325	1.01325	1.01325	1.01325	1.06325
Mole flow	Kmol/hr	1.858276	1.047751	4.467074	5.396212	5.396212	1.976889255	3.419322	3.419322282	3.419322
H2 Mol Fract.		1	0	0	0	0	0	0	0	0
O2 Mol Fract.		0	1	0.280477	0.06	0.06	0.06	0.06	0.06	0.06
H2O Mol Fract.		0	0	0.719523	0.94	0.94	0.94	0.94	0.94	0.94
Mass Flow	Kg/hr	3.746061	33.52678	97.99568	101.7417	101.7417	37.27284569	64.4689	64.46889802	64.4689
Volume Flow	cum/hr	15.35504	24.42783	139.9454	1006.532	168.3273	61.66631276	106.661	116.8178613	113.1512
							Lambda	1.12766	Rec. Stream Ratio	1.729648

Fig. 8: Memo design data – Combustion of hydrogen with oxygen and recycled stream – 25% load – Adiabatic Temperature =2000°C – Excess of oxygen in the wet flue gases = 6% in volume



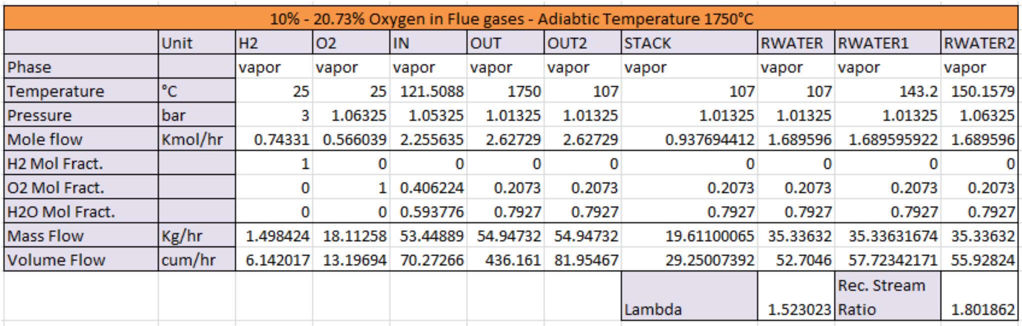
25% - 10,4% Oxygen in Flue gases										
	Unit	H2	O2	IN	OUT	OUT2	STACK	RWATER	RWATER1	RWATER2
Phase		vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	121.4018	2000	107	107	107	143.2	150.0693
Pressure	bar	3	1.06325	1.05325	1.01325	1.01325	1.01325	1.01325	1.01325	1.06325
Mole flow	Kmol/hr	1.858276	1.144831	4.512819	5.441957	5.441957	2.073968638	3.367989	3.367988603	3.367989
H2 Mol Fract.		1	0	0	0	0	0	0	0	0
O2 Mol Fract.		0	1	0.331301	0.104	0.104	0.104	0.104	0.104	0.104
H2O Mol Fract.		0	0	0.668699	0.896	0.896	0.896	0.896	0.896	0.896
Mass Flow	Kg/hr	3.746061	36.63321	102.2065	105.9525	105.9525	40.37926945	65.57328	65.57327666	65.57328
Volume Flow	cum/hr	15.35504	26.69119	140.5555	1015.065	169.7543	64.69456919	105.0597	115.0640954	111.4623
							Lambda	1.232143	Rec. Stream Ratio	1.623934

Fig. 9: Memo design data – Combustion of hydrogen with oxygen and recycled stream – 25% load – Adiabatic Temperature =2000°C – Excess of oxygen in the wet flue gases = 10,4% in volume



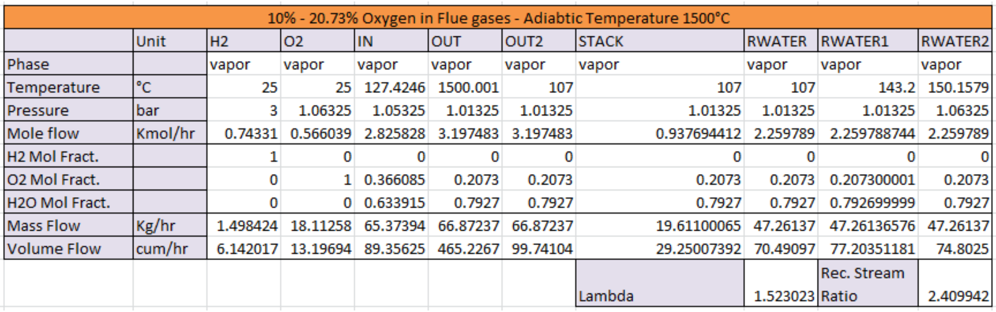
25% - 6% Oxygen in Flue gases - Adiabatic Temperature = 1745 °C										
	Unit	H2	O2	IN	OUT	OUT2	STACK	RWATER	RWATER1	RWATER2
Phase		vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	128.7521	1745	107	107	107	143.2	150.0322
Pressure	bar	3	1.06325	1.05325	1.01325	1.01325	1.01325	1.01325	1.01325	1.06325
Mole flow	Kmol/hr	1.858276	1.047751	5.492872	6.42201	6.42201	1.976889255	4.445121	4.445120794	4.445121
H2 Mol Fract.		1	0	0	0	0	0	0	0	0
O2 Mol Fract.		0	1	0.239303	0.06	0.06	0.06	0.06	0.06	0.06
H2O Mol Fract.		0	0	0.760697	0.94	0.94	0.94	0.94	0.94	0.94
Mass Flow	Kg/hr	3.746061	33.52678	117.3364	121.0824	121.0824	37.27284569	83.8096	83.80960186	83.8096
Volume Flow	cum/hr	15.35504	24.42783	174.2672	1063.494	200.3257	61.66631276	138.6594	151.8632821	147.0967
							Lambda	1.12766	Rec. Stream Ratio	2.248543

Fig. 10: Memo design data – Combustion of hydrogen with oxygen and recycled stream – 25% load – Adiabatic Temperature =1745°C – Excess of oxygen in the wet flue gases = 6% in volume



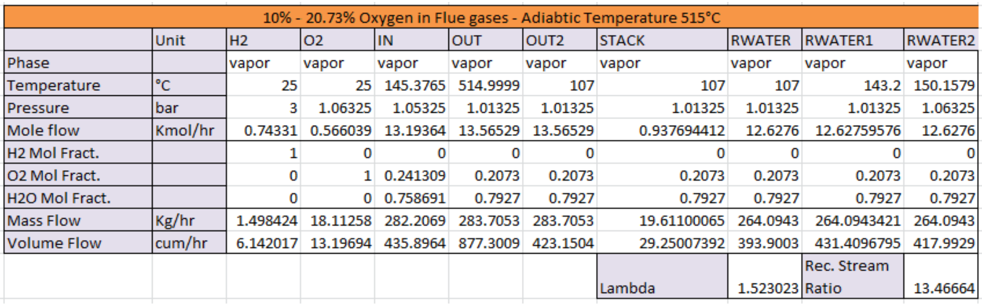
10% - 20.73% Oxygen in Flue gases - Adiabtic Temperature 1750°C										
	Unit	H2	O2	IN	OUT	OUT2	STACK	RWATER	RWATER1	RWATER2
Phase		vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	121.5088	1750	107	107	107	143.2	150.1579
Pressure	bar	3	1.06325	1.05325	1.01325	1.01325	1.01325	1.01325	1.01325	1.06325
Mole flow	Kmol/hr	0.74331	0.566039	2.255635	2.62729	2.62729	0.937694412	1.689596	1.689595922	1.689596
H2 Mol Fract.		1	0	0	0	0	0	0	0	0
O2 Mol Fract.		0	1	0.406224	0.2073	0.2073	0.2073	0.2073	0.2073	0.2073
H2O Mol Fract.		0	0	0.593776	0.7927	0.7927	0.7927	0.7927	0.7927	0.7927
Mass Flow	Kg/hr	1.498424	18.11258	53.44889	54.94732	54.94732	19.61100065	35.33632	35.33631674	35.33632
Volume Flow	cum/hr	6.142017	13.19694	70.27266	436.161	81.95467	29.25007392	52.7046	57.72342171	55.92824
							Lambda	1.523023	Rec. Stream Ratio	1.801862

Fig. 11: Memo design data – Combustion of hydrogen with oxygen and recycled stream – 10% load – Adiabatic Temperature =1750°C – Excess of oxygen in the wet flue gases = 20,73% in volume



10% - 20.73% Oxygen in Flue gases - Adiabtic Temperature 1500°C										
	Unit	H2	O2	IN	OUT	OUT2	STACK	RWATER	RWATER1	RWATER2
Phase		vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	127.4246	1500.001	107	107	107	143.2	150.1579
Pressure	bar	3	1.06325	1.05325	1.01325	1.01325	1.01325	1.01325	1.01325	1.06325
Mole flow	Kmol/hr	0.74331	0.566039	2.825828	3.197483	3.197483	0.937694412	2.259789	2.259788744	2.259789
H2 Mol Fract.		1	0	0	0	0	0	0	0	0
O2 Mol Fract.		0	1	0.366085	0.2073	0.2073	0.2073	0.2073	0.207300001	0.2073
H2O Mol Fract.		0	0	0.633915	0.7927	0.7927	0.7927	0.7927	0.792699999	0.7927
Mass Flow	Kg/hr	1.498424	18.11258	65.37394	66.87237	66.87237	19.61100065	47.26137	47.26136576	47.26137
Volume Flow	cum/hr	6.142017	13.19694	89.35625	465.2267	99.74104	29.25007392	70.49097	77.20351181	74.8025
							Lambda	1.523023	Rec. Stream Ratio	2.409942

Fig. 122: Memo design data – Combustion of hydrogen with oxygen and recycled stream – 10% load – Adiabatic Temperature =1500°C – Excess of oxygen in the wet flue gases = 20,73% in volume



10% - 20.73% Oxygen in Flue gases - Adiabtic Temperature 515°C										
	Unit	H2	O2	IN	OUT	OUT2	STACK	RWATER	RWATER1	RWATER2
Phase		vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	145.3765	514.9999	107	107	107	143.2	150.1579
Pressure	bar	3	1.06325	1.05325	1.01325	1.01325	1.01325	1.01325	1.01325	1.06325
Mole flow	Kmol/hr	0.74331	0.566039	13.19364	13.56529	13.56529	0.937694412	12.6276	12.62759576	12.6276
H2 Mol Fract.		1	0	0	0	0	0	0	0	0
O2 Mol Fract.		0	1	0.241309	0.2073	0.2073	0.2073	0.2073	0.2073	0.2073
H2O Mol Fract.		0	0	0.758691	0.7927	0.7927	0.7927	0.7927	0.7927	0.7927
Mass Flow	Kg/hr	1.498424	18.11258	282.2069	283.7053	283.7053	19.61100065	264.0943	264.0943421	264.0943
Volume Flow	cum/hr	6.142017	13.19694	435.8964	877.3009	423.1504	29.25007392	393.9003	431.4096795	417.9929
							Lambda	1.523023	Rec. Stream Ratio	13.46664

Fig. 13: Memo design data – Combustion of hydrogen with oxygen and recycled stream – 10% load – Adiabatic Temperature =515°C – Excess of oxygen in the wet flue gases = 20,73% in volume

The following picture represents the configuration of the system when it is working with air. The stream fuel can be either natural gas or hydrogen.

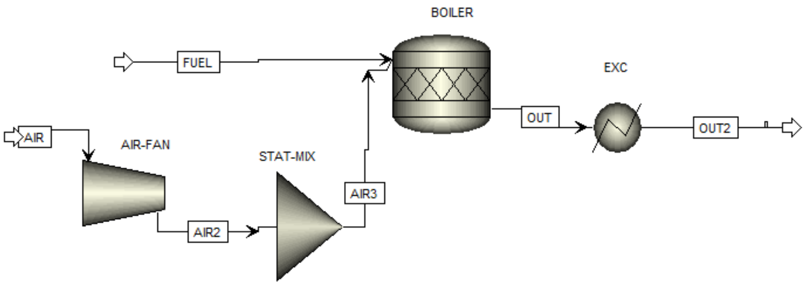
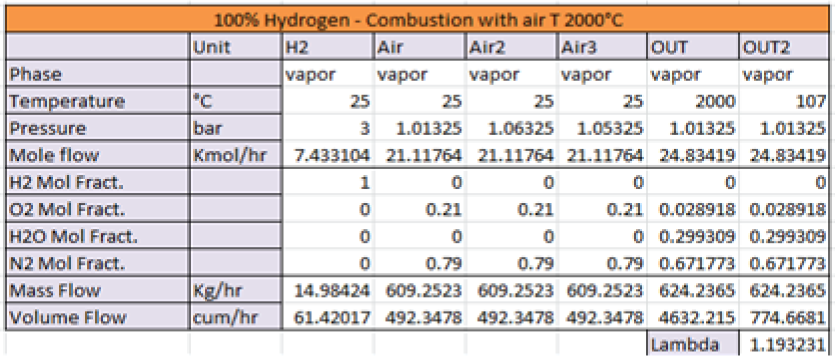
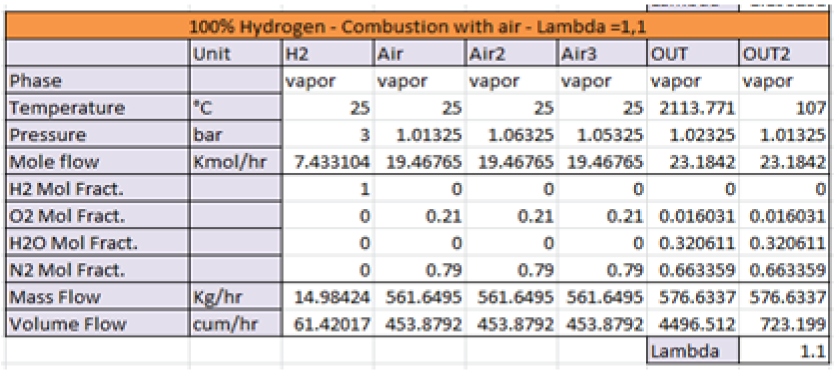


Fig. 14: scheme of the plant for operation with the use of air as oxidizing gas



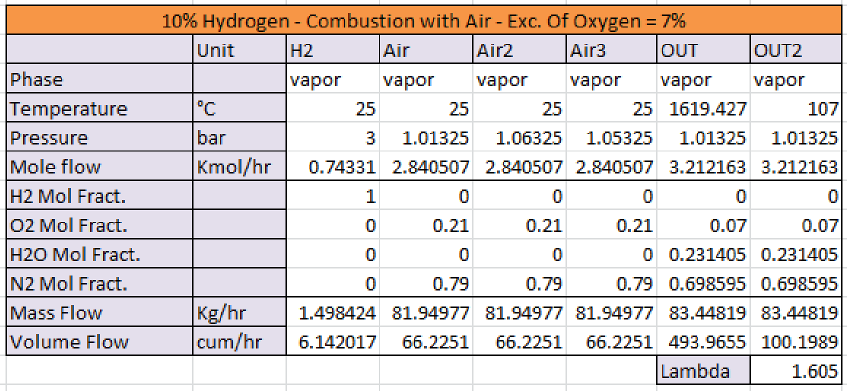
100% Hydrogen - Combustion with air T 2000°C							
	Unit	H2	Air	Air2	Air3	OUT	OUT2
Phase		vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	25	25	2000	107
Pressure	bar	3	1.01325	1.06325	1.05325	1.01325	1.01325
Mole flow	Kmol/hr	7.433104	21.11764	21.11764	21.11764	24.83419	24.83419
H2 Mol Fract.		1	0	0	0	0	0
O2 Mol Fract.		0	0.21	0.21	0.21	0.028918	0.028918
H2O Mol Fract.		0	0	0	0	0.299309	0.299309
N2 Mol Fract.		0	0.79	0.79	0.79	0.671773	0.671773
Mass Flow	Kg/hr	14.98424	609.2523	609.2523	609.2523	624.2365	624.2365
Volume Flow	cum/hr	61.42017	492.3478	492.3478	492.3478	4632.215	774.6681
						Lambda	1.193231

Fig. 15: Memo design data – Combustion of hydrogen with air – 100% load – Adiabatic Temperature =2000°C – Excess of oxygen in the wet flue gases = 2,89% in volume



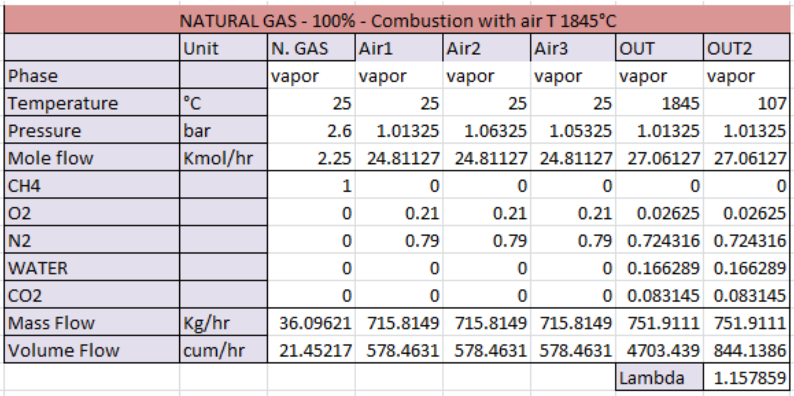
100% Hydrogen - Combustion with air - Lambda =1,1							
	Unit	H2	Air	Air2	Air3	OUT	OUT2
Phase		vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	25	25	2113.771	107
Pressure	bar	3	1.01325	1.06325	1.05325	1.02325	1.01325
Mole flow	Kmol/hr	7.433104	19.46765	19.46765	19.46765	23.1842	23.1842
H2 Mol Fract.		1	0	0	0	0	0
O2 Mol Fract.		0	0.21	0.21	0.21	0.016031	0.016031
H2O Mol Fract.		0	0	0	0	0.320611	0.320611
N2 Mol Fract.		0	0.79	0.79	0.79	0.663359	0.663359
Mass Flow	Kg/hr	14.98424	561.6495	561.6495	561.6495	576.6337	576.6337
Volume Flow	cum/hr	61.42017	453.8792	453.8792	453.8792	4496.512	723.199
						Lambda	1.1

Fig. 16: Memo design data – Combustion of hydrogen with air – 100% load – Adiabatic Temperature =2110°C – Excess of oxygen in the wet flue gases = 1,6% in volume



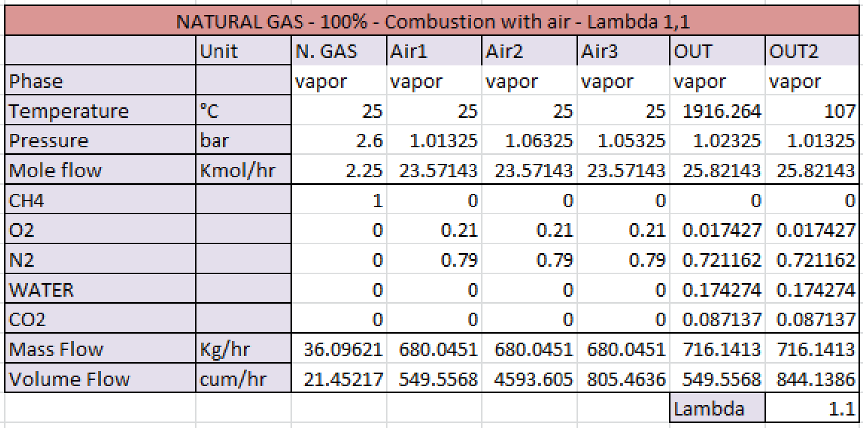
10% Hydrogen - Combustion with Air - Exc. Of Oxygen = 7%							
	Unit	H2	Air	Air2	Air3	OUT	OUT2
Phase		vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	25	25	1619.427	107
Pressure	bar	3	1.01325	1.06325	1.05325	1.01325	1.01325
Mole flow	Kmol/hr	0.74331	2.840507	2.840507	2.840507	3.212163	3.212163
H2 Mol Fract.		1	0	0	0	0	0
O2 Mol Fract.		0	0.21	0.21	0.21	0.07	0.07
H2O Mol Fract.		0	0	0	0	0.231405	0.231405
N2 Mol Fract.		0	0.79	0.79	0.79	0.698595	0.698595
Mass Flow	Kg/hr	1.498424	81.94977	81.94977	81.94977	83.44819	83.44819
Volume Flow	cum/hr	6.142017	66.2251	66.2251	66.2251	493.9655	100.1989
						Lambda	1.605

Fig. 17: Memo design data – Combustion of hydrogen with air – 10% load – Adiabatic Temperature =1620°C – Excess of oxygen in the wet flue gases = 7% in volume



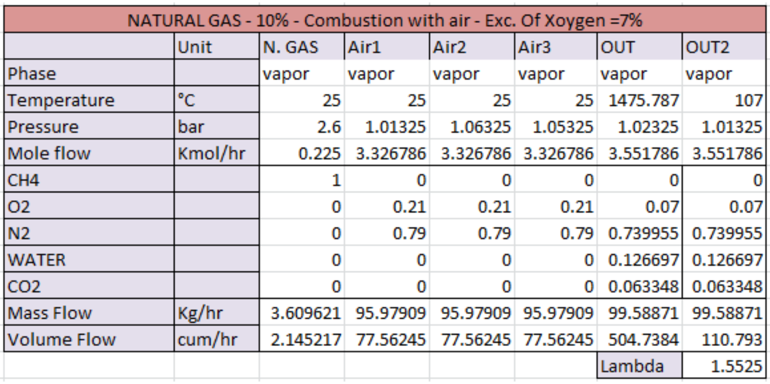
NATURAL GAS - 100% - Combustion with air T 1845°C							
	Unit	N. GAS	Air1	Air2	Air3	OUT	OUT2
Phase		vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	25	25	1845	107
Pressure	bar	2.6	1.01325	1.06325	1.05325	1.01325	1.01325
Mole flow	Kmol/hr	2.25	24.81127	24.81127	24.81127	27.06127	27.06127
CH4		1	0	0	0	0	0
O2		0	0.21	0.21	0.21	0.02625	0.02625
N2		0	0.79	0.79	0.79	0.724316	0.724316
WATER		0	0	0	0	0.166289	0.166289
CO2		0	0	0	0	0.083145	0.083145
Mass Flow	Kg/hr	36.09621	715.8149	715.8149	715.8149	751.9111	751.9111
Volume Flow	cum/hr	21.45217	578.4631	578.4631	578.4631	4703.439	844.1386
						Lambda	1.157859

Fig. 18: Memo design data – Combustion of Natural Gas with air – 100% load – Adiabatic Temperature =1845°C – Excess of oxygen in the wet flue gases = 2,63% in volume



NATURAL GAS - 100% - Combustion with air - Lambda 1,1							
	Unit	N. GAS	Air1	Air2	Air3	OUT	OUT2
Phase		vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	25	25	1916.264	107
Pressure	bar	2.6	1.01325	1.06325	1.05325	1.02325	1.01325
Mole flow	Kmol/hr	2.25	23.57143	23.57143	23.57143	25.82143	25.82143
CH4		1	0	0	0	0	0
O2		0	0.21	0.21	0.21	0.017427	0.017427
N2		0	0.79	0.79	0.79	0.721162	0.721162
WATER		0	0	0	0	0.174274	0.174274
CO2		0	0	0	0	0.087137	0.087137
Mass Flow	Kg/hr	36.09621	680.0451	680.0451	680.0451	716.1413	716.1413
Volume Flow	cum/hr	21.45217	549.5568	4593.605	805.4636	549.5568	844.1386
						Lambda	1.1

Fig. 19: Memo design data – Combustion of Natural Gas with air – 100% load – Adiabatic Temperature =1915°C – Excess of oxygen in the wet flue gases = 1,74% in volume



NATURAL GAS - 10% - Combustion with air - Exc. Of Xoygen =7%							
	Unit	N. GAS	Air1	Air2	Air3	OUT	OUT2
Phase		vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	25	25	1475.787	107
Pressure	bar	2.6	1.01325	1.06325	1.05325	1.02325	1.01325
Mole flow	Kmol/hr	0.225	3.326786	3.326786	3.326786	3.551786	3.551786
CH4		1	0	0	0	0	0
O2		0	0.21	0.21	0.21	0.07	0.07
N2		0	0.79	0.79	0.79	0.739955	0.739955
WATER		0	0	0	0	0.126697	0.126697
CO2		0	0	0	0	0.063348	0.063348
Mass Flow	Kg/hr	3.609621	95.97909	95.97909	95.97909	99.58871	99.58871
Volume Flow	cum/hr	2.145217	77.56245	77.56245	77.56245	504.7384	110.793
						Lambda	1.5525

Fig. 20: Memo design data – Combustion of Natural Gas with air – 10% load – Adiabatic Temperature =1475°C – Excess of oxygen in the wet flue gases = 7% in volume

The following picture represents the configuration of the system when it is working with nitrogen and oxygen coming from the respective cylinder. The stream fuel can be either natural gas or hydrogen.

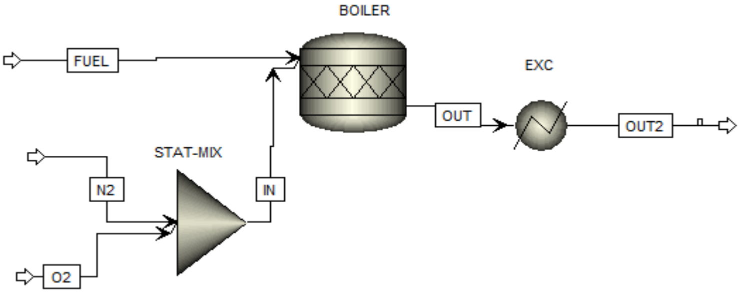
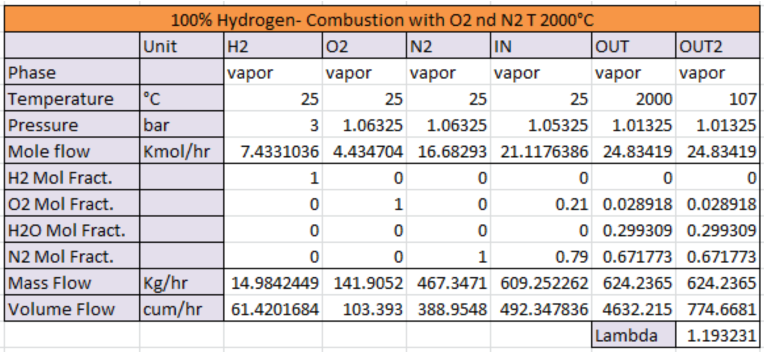
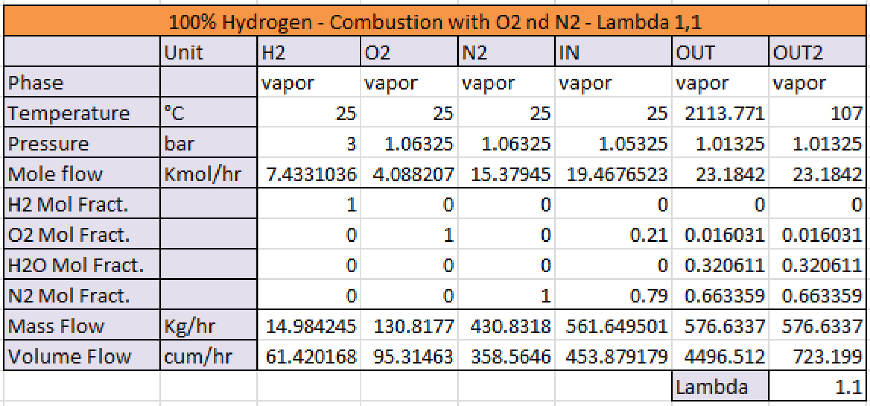


Fig. 21: scheme of the plant for operation with the use of the mixture of nitrogen and oxygen as oxidizing gas



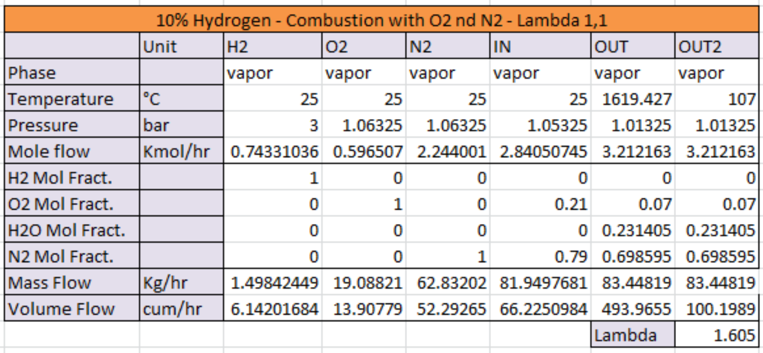
100% Hydrogen- Combustion with O2 nd N2 T 2000°C							
	Unit	H2	O2	N2	IN	OUT	OUT2
Phase		vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	25	25	2000	107
Pressure	bar	3	1.06325	1.06325	1.05325	1.01325	1.01325
Mole flow	Kmol/hr	7.4331036	4.434704	16.68293	21.1176386	24.83419	24.83419
H2 Mol Fract.		1	0	0	0	0	0
O2 Mol Fract.		0	1	0	0.21	0.028918	0.028918
H2O Mol Fract.		0	0	0	0	0.299309	0.299309
N2 Mol Fract.		0	0	1	0.79	0.671773	0.671773
Mass Flow	Kg/hr	14.9842449	141.9052	467.3471	609.252262	624.2365	624.2365
Volume Flow	cum/hr	61.4201684	103.393	388.9548	492.347836	4632.215	774.6681
						Lambda	1.193231

Fig. 22: Memo design data – Combustion of hydrogen with mixture Nitrogen-Oxygen – 100% load – Adiabatic Temperature =2000°C – Excess of oxygen in the wet flue gases = 2,89% in volume



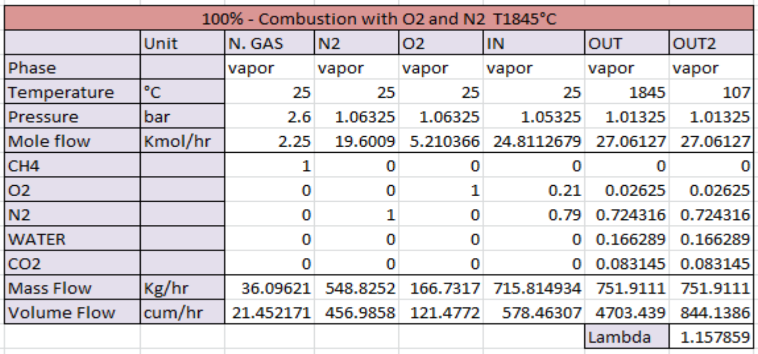
100% Hydrogen - Combustion with O2 nd N2 - Lambda 1,1							
	Unit	H2	O2	N2	IN	OUT	OUT2
Phase		vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	25	25	2113.771	107
Pressure	bar	3	1.06325	1.06325	1.05325	1.01325	1.01325
Mole flow	Kmol/hr	7.4331036	4.088207	15.37945	19.4676523	23.1842	23.1842
H2 Mol Fract.		1	0	0	0	0	0
O2 Mol Fract.		0	1	0	0.21	0.016031	0.016031
H2O Mol Fract.		0	0	0	0	0.320611	0.320611
N2 Mol Fract.		0	0	1	0.79	0.663359	0.663359
Mass Flow	Kg/hr	14.984245	130.8177	430.8318	561.649501	576.6337	576.6337
Volume Flow	cum/hr	61.420168	95.31463	358.5646	453.879179	4496.512	723.199
						Lambda	1.1

Fig. 23: Memo design data – Combustion of hydrogen with mixture Nitrogen-Oxygen – 100% load – Adiabatic Temperature =2110°C – Excess of oxygen in the wet flue gases = 1,6% in volume



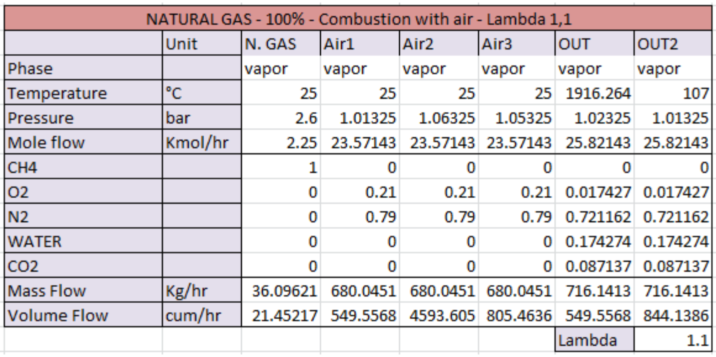
10% Hydrogen - Combustion with O2 nd N2 - Lambda 1,1							
	Unit	H2	O2	N2	IN	OUT	OUT2
Phase		vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	25	25	1619.427	107
Pressure	bar	3	1.06325	1.06325	1.05325	1.01325	1.01325
Mole flow	Kmol/hr	0.74331036	0.596507	2.244001	2.84050745	3.212163	3.212163
H2 Mol Fract.		1	0	0	0	0	0
O2 Mol Fract.		0	1	0	0.21	0.07	0.07
H2O Mol Fract.		0	0	0	0	0.231405	0.231405
N2 Mol Fract.		0	0	1	0.79	0.698595	0.698595
Mass Flow	Kg/hr	1.49842449	19.08821	62.83202	81.9497681	83.44819	83.44819
Volume Flow	cum/hr	6.14201684	13.90779	52.29265	66.2250984	493.9655	100.1989
						Lambda	1.605

Fig. 24: Memo design data – Combustion of hydrogen with mixture Nitrogen-Oxygen – 10% load – Adiabatic Temperature =1620°C – Excess of oxygen in the wet flue gases = 7% in volume



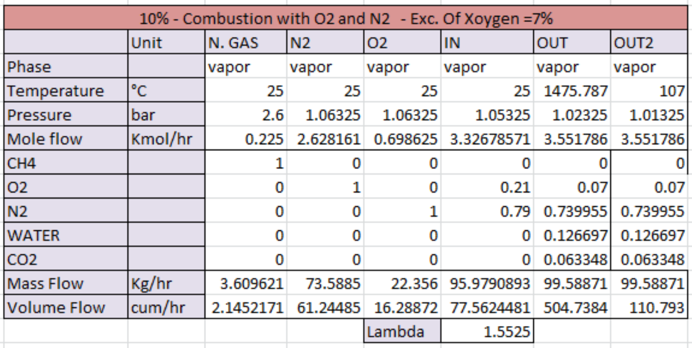
100% - Combustion with O2 and N2 T1845°C							
	Unit	N. GAS	N2	O2	IN	OUT	OUT2
Phase		vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	25	25	1845	107
Pressure	bar	2.6	1.06325	1.06325	1.05325	1.01325	1.01325
Mole flow	Kmol/hr	2.25	19.6009	5.210366	24.8112679	27.06127	27.06127
CH4		1	0	0	0	0	0
O2		0	0	1	0.21	0.02625	0.02625
N2		0	1	0	0.79	0.724316	0.724316
WATER		0	0	0	0	0.166289	0.166289
CO2		0	0	0	0	0.083145	0.083145
Mass Flow	Kg/hr	36.09621	548.8252	166.7317	715.814934	751.9111	751.9111
Volume Flow	cum/hr	21.452171	456.9858	121.4772	578.46307	4703.439	844.1386
						Lambda	1.157859

Fig. 25: Memo design data – Combustion of Natural Gas with mixture Nitrogen-Oxygen – 100% load – Adiabatic Temperature =1845°C – Excess of oxygen in the wet flue gases = 2,63% in volume



NATURAL GAS - 100% - Combustion with air - Lambda 1,1							
	Unit	N. GAS	Air1	Air2	Air3	OUT	OUT2
Phase		vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	25	25	1916.264	107
Pressure	bar	2.6	1.01325	1.06325	1.05325	1.02325	1.01325
Mole flow	Kmol/hr	2.25	23.57143	23.57143	23.57143	25.82143	25.82143
CH4		1	0	0	0	0	0
O2		0	0.21	0.21	0.21	0.017427	0.017427
N2		0	0.79	0.79	0.79	0.721162	0.721162
WATER		0	0	0	0	0.174274	0.174274
CO2		0	0	0	0	0.087137	0.087137
Mass Flow	Kg/hr	36.09621	680.0451	680.0451	680.0451	716.1413	716.1413
Volume Flow	cum/hr	21.45217	549.5568	4593.605	805.4636	549.5568	844.1386
						Lambda	1.1

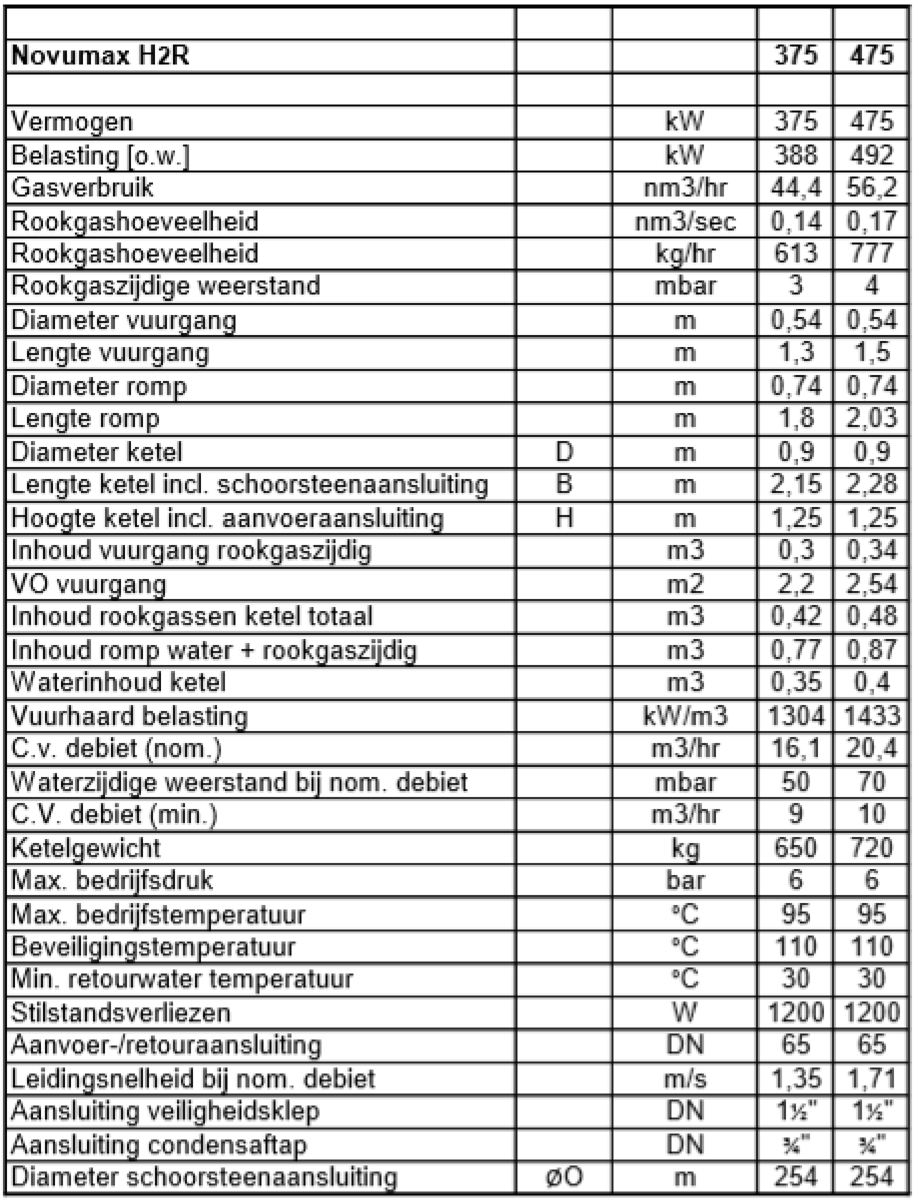
Fig. 26: Memo design data – Combustion of Natural Gas with air – 100% load – Adiabatic Temperature =1915°C – Excess of oxygen in the wet flue gases = 1,74% in volume



10% - Combustion with O2 and N2 - Exc. Of Xoygen =7%							
	Unit	N. GAS	N2	O2	IN	OUT	OUT2
Phase		vapor	vapor	vapor	vapor	vapor	vapor
Temperature	°C	25	25	25	25	1475.787	107
Pressure	bar	2.6	1.06325	1.06325	1.05325	1.02325	1.01325
Mole flow	Kmol/hr	0.225	2.628161	0.698625	3.32678571	3.551786	3.551786
CH4		1	0	0	0	0	0
O2		0	1	0	0.21	0.07	0.07
N2		0	0	1	0.79	0.739955	0.739955
WATER		0	0	0	0	0.126697	0.126697
CO2		0	0	0	0	0.063348	0.063348
Mass Flow	Kg/hr	3.609621	73.5885	22.356	95.9790893	99.58871	99.58871
Volume Flow	cum/hr	2.1452171	61.24485	16.28872	77.5624481	504.7384	110.793
				Lambda	1.5525		

Fig. 27: Memo design data – Combustion of Natural Gas with air – 10% load – Adiabatic Temperature =1475°C – Excess of oxygen in the wet flue gases = 7% in volume

APPENDIX 3 : Boiler



Novumax H2R			**375**	**475**
Vermogen		kW	375	475
Belasting [o.w.]		kW	388	492
Gasverbruik		nm3/hr	44,4	56,2
Rookgashoeveelheid		nm3/sec	0,14	0,17
Rookgashoeveelheid		kg/hr	613	777
Rookgaszijdige weerstand		mbar	3	4
Diameter vuurgang		m	0,54	0,54
Lengte vuurgang		m	1,3	1,5
Diameter romp		m	0,74	0,74
Lengte romp		m	1,8	2,03
Diameter ketel	D	m	0,9	0,9
Lengte ketel incl. schoorsteenaansluiting	B	m	2,15	2,28
Hoogte ketel incl. aanvoeraansluiting	H	m	1,25	1,25
Inhoud vuurgang rookgaszijdig		m3	0,3	0,34
VO vuurgang		m2	2,2	2,54
Inhoud rookgassen ketel totaal		m3	0,42	0,48
Inhoud romp water + rookgaszijdig		m3	0,77	0,87
Waterinhoud ketel		m3	0,35	0,4
Vuurhaard belasting		kW/m3	1304	1433
C.v. debiet (nom.)		m3/hr	16,1	20,4
Waterzijdige weerstand bij nom. debiet		mbar	50	70
C.V. debiet (min.)		m3/hr	9	10
Ketelgewicht		kg	650	720
Max. bedrijfsdruk		bar	6	6
Max. bedrijfstemperatuur		°C	95	95
Beveiligingstemperatuur		°C	110	110
Min. retourwater temperatuur		°C	30	30
Stilstandsverliezen		W	1200	1200
Aanvoer-/retouraansluiting		DN	65	65
Leidingsnelheid bij nom. debiet		m/s	1,35	1,71
Aansluiting veiligheidsklep		DN	1½"	1½"
Aansluiting condensaftap		DN	¾"	¾"
Diameter schoorsteenaansluiting	øO	m	254	254

Fig. 28: Novumax H2R Data Sheet

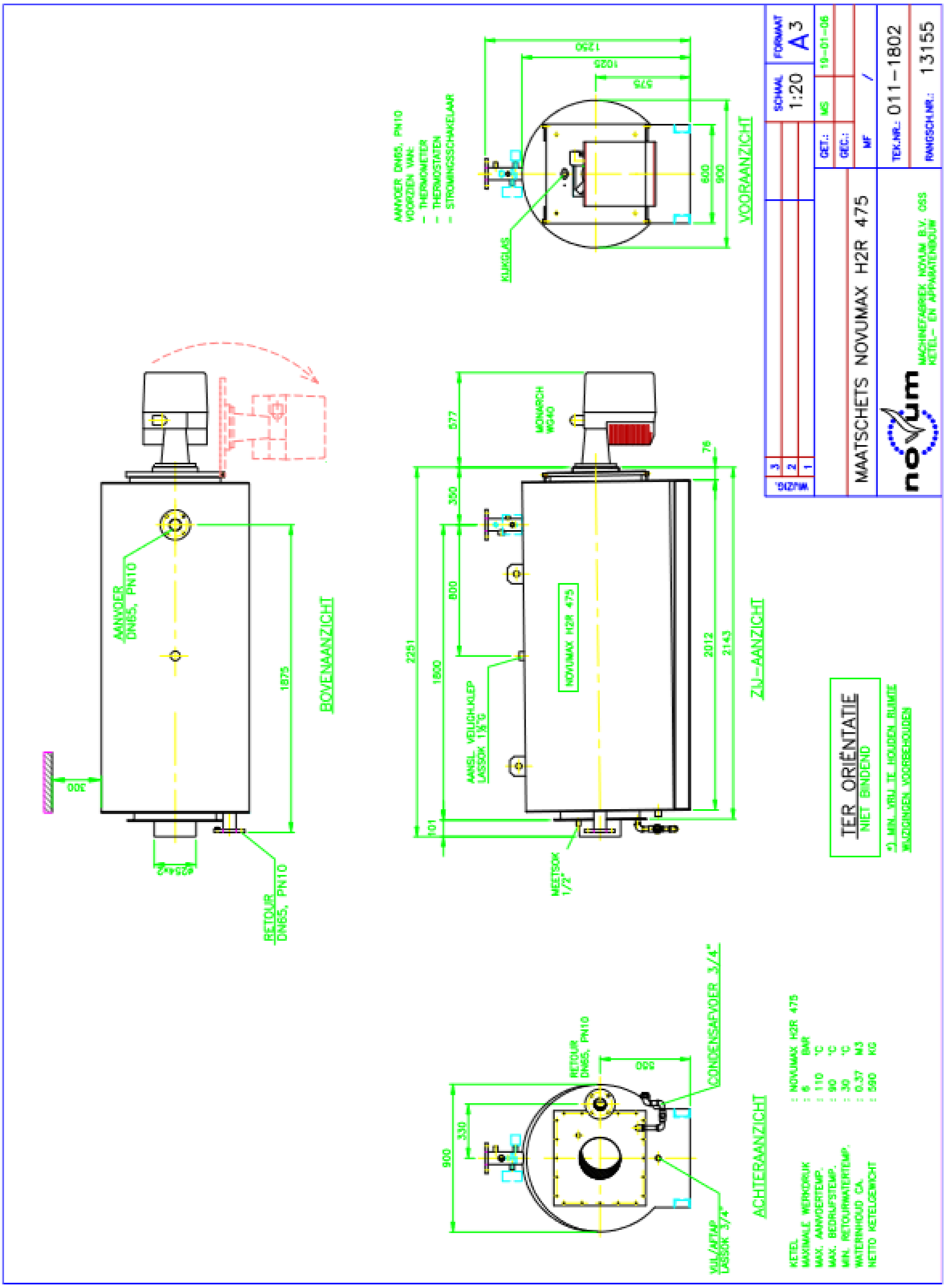


Fig. 29: Novumax H2R – 2D drawing

APPENDIX 4: Radiation Heat Transfer calculation

Here, the furnace heat transfer calculation method is introduced using a zero dimensional model. This method is primarily based on the energy conservation equation and the radiative heat transfer equation.[4-4]. The thermal balance equation is as follows

$$Q = \varrho\, \dot{m}_{fuel} V\bar{C}(T_a - T_F)$$

{15}

Where ϱ is the heat preservation coefficient which takes into account the heat loss due to the cooling effect of the furnace wall (assumed to be 0.80 from reference [4-5]). T_a is the theoretical adiabatic temperature of combustion. T_F is the temperature of the exit of the furnace. $V\bar{C}$ is the mean overall heat capacity of the combustion products between $T_a and\, T_F$ expressed as follow (ha and hf are the enthalpies of the gas at theoretical combustion temperature and at the exit of the furnace):

$$V\bar{C} = \frac{h_a - h_F}{T_a - T_F}$$

{16}

On the other and hand the heat irradiated by the gas to the furnaces wall can also be written using the Hottel method with the following equation:

$$Q = \sigma a_F A({T_g}^4 - {T_w}^4)$$

{17}

Where σ is the Stefan-Boltzman constante qual to 5.670367(13)×10−8 W·m−2·K−4; A is the total area of the furnace. T_w is the temperature of the wall assumed to be (15-20°C less than the temperature of the gas inside the furnace) and T_g is the average temperature of the gas inside the furnace. The temperature of the gas inside the furnace will vary with the three dimensions; to simplify the calculation T_g represents the average of all the temperatures of the gas. a_F is the emissivity factor which can be calculated with the following formula which include the emissivity factor of the wall $\epsilon_w = 0.7$ and the emissivity factor of the gas inside the furnace ϵ_g =0.43. Values assumed are taken from reference [2-7]

$$a_F = \frac{1}{\frac{1}{\epsilon_w} + \left(\frac{1}{\epsilon_g} - 1\right)}$$

{18}

According to the Gurvich method, equation {17} can also be written as:

$$Q = \sigma \overline{a_F} A\psi\ ({T_g}^4) = \frac{\psi A \sigma T_g^4}{1+\psi\left(\frac{1}{\epsilon_g} - 1\right)}$$

{19}

By the combination of equation of equation {15} and {19} it is possible to obtain:

$$\frac{T_a - T_F}{{T_g}^4} = \frac{\sigma\psi\overline{a_F}A}{\varrho\, \dot{m}_{fuel} V\bar{C}}$$

{20}

Assume that fuel combusts completely instantaneously at the burner exit and reaches adiabatic combustion temperature Ta, and that heat transfer only occurs in the radial direction of the furnace axis; ignore heat transfer in the axial direction (one-dimensional model).Equations {15} and {19} can also be written as:

$$-\varrho \dot{m}_{fuel} V\bar{C}\ \ dT_g = \sigma\psi\overline{a_F}{T_g}^4\, dA$$

{21}

$$\int_{T_a}^{T_F} -\frac{dT_g}{{T_g}^4} = \int_0^A \frac{\sigma\psi\overline{a_F}}{\varrho\, \dot{m}_{fuel} V\bar{C}}\, dA$$

{22}

$$\frac{1}{3}\left(\frac{1}{{T_f}^3} - \frac{1}{{T_a}^3}\right) = \frac{\sigma\psi\overline{a_F}}{\varrho\, \dot{m}_{fuel} V\bar{C}}\, A$$

{23}

Equations {20} and {23} can be combined in:

$$\frac{T_a - T_F}{{T_g}^4} = \frac{1}{3}\left(\frac{1}{{T_F}^3} - \frac{1}{{T_a}^3}\right) \qquad \{24\}$$

By solving the equation it is possible to obtain:

$$ {T_g}^4 = \frac{3{T_f}^4}{\left(\frac{{T_F}^3}{{T_a}^3}\right) + \left(\frac{{T_F}^2}{{T_a}^2}\right) + \left(\frac{T_F}{T_a}\right)} \qquad \{25\}$$

From equation {15} and equation {17} the average gas temperature an also be calculated with:

$$ {T_g}^4 = \frac{\varrho\, \dot{m}_{fuel}(h_a - h_F)}{\sigma a_F A} + {T_w}^4 \qquad \{26\}$$

So it is now possible to calculate the average temperature of the gas in the furnace in two different ways, by varying the temperature at the outlet of the furnace T_F. With an iteration process it is possible to find the radiation heat transfer for which the two average temperature are the same. The heat capacity of the mixture is calculated with a weighted average by using the following formula.

$$ C_{p\ (T);mixture} = \sum_{x=1}^{n} \frac{(PM_x\, Y_x\ C_{p\,x})}{(PM_x\, Y_x\ \ )} \qquad \{27\}$$

The mixture is made "n" elements, the single element "x" has a molecular weight "PM_x" a volume fraction "Y_x" and a heat capacity factor "$C_{p\,x}$" at the designed temperature.

APPENDIX 5: Heat Exchanger

Case 1

To size the heat exchanger it is first necessary to set the thermodynamic properties of the gas in consideration before and after the heat exchanger. The values assumed are shown in the following tables, while the calculation method is as follows:

$$ C_{p\,,flue\ gas;\,average} = \frac{(PM_{H20}\, Y_{H20}\ C_{p\,,H20;\,average} + PM_{O2}\, Y_{O2}\ C_{p\,,O2})}{(PM_{H20}\, Y_{H20}\ \ + PM_{O2}\, Y_{O2}\ \ )} \qquad \{28\}$$

Where PM_x is the molecular weight of (x) and Y_x is the molar fraction of (x).The same weighted average is used to calculate the average viscosity μ_x, the average density ρ_x and the average conductivity α_x.

Thus it is required to assume a diameter, a thickness and an amount of the small pipes in which the cooling water is flowing; the assumed values are changed in order to obtain an optimal result and the final values are visible in the tables. The method used to analyze the heat exchanger is LMTD method (Logarithmic Mean temperature Difference). The water flowing in the small pipes is assumed to have a velocity of 0,2 m/s, while the flue gases which flows surrounding the water pipes is assumed to have a velocity of about 25 m/s. Using these information the Reynolds number and the Prandtl number can be calculated:

$$ Re = \frac{\rho\ D\ V}{\mu} \qquad \{29\}$$

Where V is the velocity of the stream considered and D is the intern diameter of the pipes for the cold side and the equivalent diameter for the hot side (the equivalent diameter defined as 4 times the perimeter divided the length).

$$Pr = \frac{Cp\,\mu}{\alpha} \qquad \{30\}$$

Using the two dimensionless numbers it is possible to obtain the Nusselt number with the following equation:

$$Nu = 0{,}023 * Re^{0,8} * Pr^{0,3} \qquad \{31\}$$

From the Nusselt number the coefficient of heat transfer can be calculated:

$$hi = Nu * \frac{\alpha}{D} \qquad \{32\}$$

With the same method it is possible to calculate hi and he, which are the heat transfer coefficient internal and external. And using this information it is possible to calculate the global heat transfer coefficient K:

$$K = \frac{1}{\frac{1}{hi} + \frac{D_{int}}{2\lambda_{steel}} \ln\left(\frac{D_{ext}}{D_{int}}\right) + \frac{D_{int}}{D_{ext}}\frac{1}{he}} \qquad \{33\}$$

Then the LMTD is calculated: The LMTD is logarithmic average of the temperature difference between the hot and cold feeds at each end of the double pipe exchanger.

$$LMTD = \frac{(T_{Hot-in} - T_{Cold-out}) - (T_{Hot-out} - T_{Cold-in})}{LN\,\frac{(T_{Hot-in} - T_{Cold-out})}{(T_{Hot-out} - T_{Cold-in})}} \qquad \{34\}$$

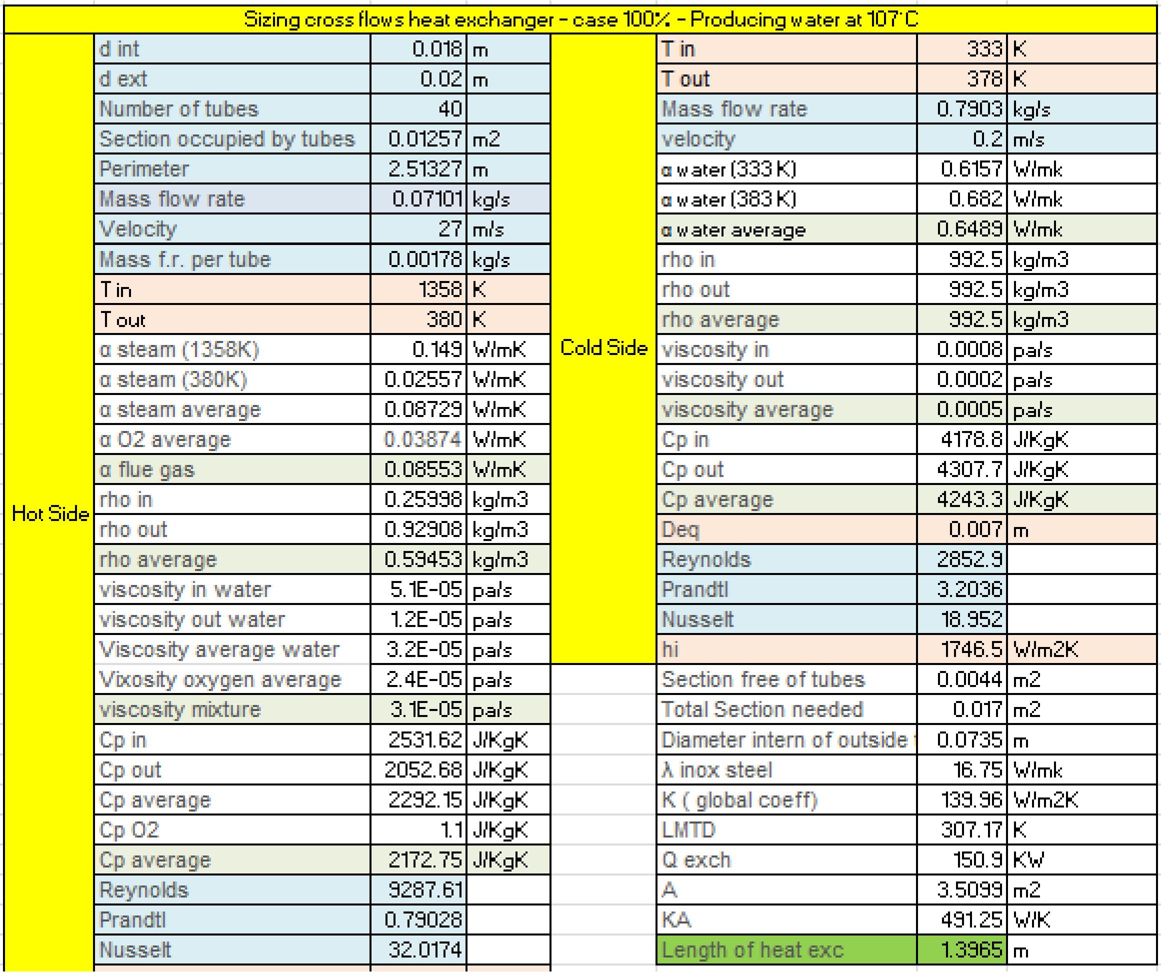
The heat transferred from the hot side to the cold side is equal to the enthalpy difference of the two fluids:

$$Q = \dot{m}_{flue\ gas} * Cp_{average,flue\ gas} * (T_{Hot-in} - T_{Hot-out}) = \dot{m}_{H20} Cp_{H2O} (T_{c-out} - T_{c-in}) \qquad \{35\}$$

From this equation it is possible to calculate the required mass flow rate of cooling water. From the LMTD method it is also possible to write:

$$Q = LMTD * K * A \qquad \{36\}$$

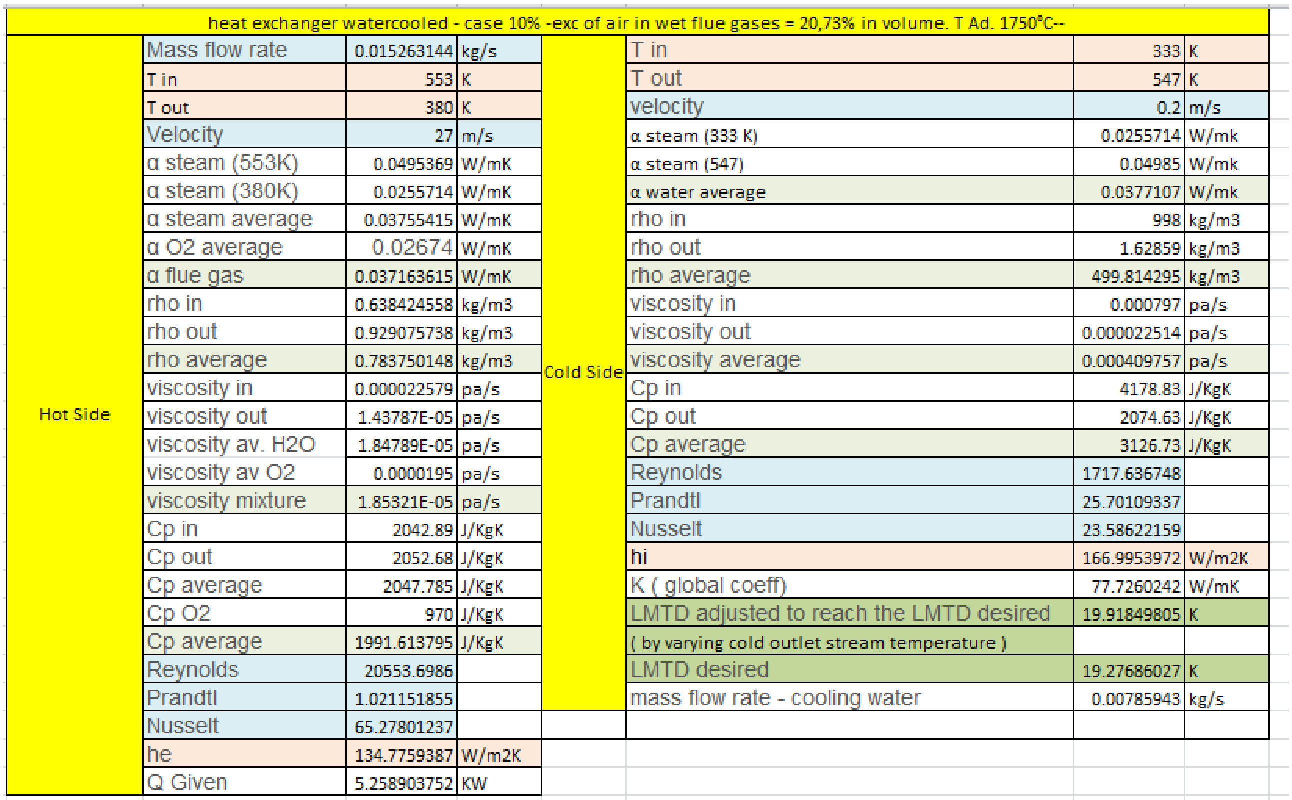
Hence it is easily possible to calculate the required exchanging area A and the relative length of the heat exchanger. Results are presented in the following table:



Sizing cross flows heat exchanger - case 100% - Producing water at 107°C							
Hot Side	d int	0.018	m	Cold Side	T in	333	K
	d ext	0.02	m		T out	378	K
	Number of tubes	40			Mass flow rate	0.7903	kg/s
	Section occupied by tubes	0.01257	m2		velocity	0.2	m/s
	Perimeter	2.51327	m		α water (333 K)	0.6157	W/mk
	Mass flow rate	0.07101	kg/s		α water (383 K)	0.682	W/mk
	Velocity	27	m/s		α water average	0.6489	W/mk
	Mass f.r. per tube	0.00178	kg/s		rho in	992.5	kg/m3
	T in	1358	K		rho out	992.5	kg/m3
	T out	380	K		rho average	992.5	kg/m3
	α steam (1358K)	0.149	W/mK		viscosity in	0.0008	pa/s
	α steam (380K)	0.02557	W/mK		viscosity out	0.0002	pa/s
	α steam average	0.08729	W/mK		viscosity average	0.0005	pa/s
	α O2 average	0.03874	W/mK		Cp in	4178.8	J/KgK
	α flue gas	0.08553	W/mK		Cp out	4307.7	J/KgK
	rho in	0.25998	kg/m3		Cp average	4243.3	J/KgK
	rho out	0.92908	kg/m3		Deq	0.007	m
	rho average	0.59453	kg/m3		Reynolds	2852.9	
	viscosity in water	5.1E-05	pa/s		Prandtl	3.2036	
	viscosity out water	1.2E-05	pa/s		Nusselt	18.952	
	Viscosity average water	3.2E-05	pa/s		hi	1746.5	W/m2K
	Vixosity oxygen average	2.4E-05	pa/s		Section free of tubes	0.0044	m2
	viscosity mixture	3.1E-05	pa/s		Total Section needed	0.017	m2
	Cp in	2531.62	J/KgK		Diameter intern of outside	0.0735	m
	Cp out	2052.68	J/KgK		λ inox steel	16.75	W/mk
	Cp average	2292.15	J/KgK		K (global coeff)	139.96	W/m2K
	Cp O2	1.1	J/KgK		LMTD	307.17	K
	Cp average	2172.75	J/KgK		Q exch	150.9	KW
	Reynolds	9287.61			A	3.5099	m2
	Prandtl	0.79028			KA	491.25	W/K
	Nusselt	32.0174			Length of heat exc	1.3965	m

Fig. 30: Calculation for the size of the heat exchanger–Water cooled – counter flow – based on the case "Combustion of hydrogen with oxygen and recycled stream – 100% load – Adiabatic Temperature =2000°C – Excess of oxygen in the wet flue gases = 3% in volume"

Once the size of the heat exchanger is calculated it is important to analyze the behavior of the heat exchanger when the load of the system is reduced to the minimum (10%); in this case the LMTD is calculated in the same way it was calculated for the 100% case and the result is compared to the value obtained from equation {35}. Than using an iteration process the outlet temperature of the cold side has been changed in order that the two calculated LMTD matches.



heat exchanger watercooled - case 10% -exc of air in wet flue gases = 20,73% in volume. T Ad. 1750°C--							
Hot Side	Mass flow rate	0.015263144	kg/s	Cold Side	T in	333	K
	T in	553	K		T out	547	K
	T out	380	K		velocity	0.2	m/s
	Velocity	27	m/s		α steam (333 K)	0.0255714	W/mk
	α steam (553K)	0.0495369	W/mK		α steam (547)	0.04985	W/mk
	α steam (380K)	0.0255714	W/mK		α water average	0.0377107	W/mk
	α steam average	0.03755415	W/mK		rho in	998	kg/m3
	α O2 average	0.02674	W/mK		rho out	1.62859	kg/m3
	α flue gas	0.037163615	W/mK		rho average	499.814295	kg/m3
	rho in	0.638424558	kg/m3		viscosity in	0.000797	pa/s
	rho out	0.929075738	kg/m3		viscosity out	0.000022514	pa/s
	rho average	0.783750148	kg/m3		viscosity average	0.000409757	pa/s
	viscosity in	0.000022579	pa/s		Cp in	4178.83	J/KgK
	viscosity out	1.43787E-05	pa/s		Cp out	2074.63	J/KgK
	viscosity av. H2O	1.84789E-05	pa/s		Cp average	3126.73	J/KgK
	viscosity av O2	0.0000195	pa/s		Reynolds	1717.636748	
	viscosity mixture	1.85321E-05	pa/s		Prandtl	25.70109337	
	Cp in	2042.89	J/KgK		Nusselt	23.58622159	
	Cp out	2052.68	J/KgK		hi	166.9953972	W/m2K
	Cp average	2047.785	J/KgK		K (global coeff)	77.7260242	W/mK
	Cp O2	970	J/KgK		LMTD adjusted to reach the LMTD desired	19.91849805	K
	Cp average	1991.613795	J/KgK		(by varying cold outlet stream temperature)		
	Reynolds	20553.6986			LMTD desired	19.27686027	K
	Prandtl	1.021151855			mass flow rate - cooling water	0.00785943	kg/s
	Nusselt	65.27801237					
	he	134.7759387	W/m2K				
	Q Given	5.258903752	KW				

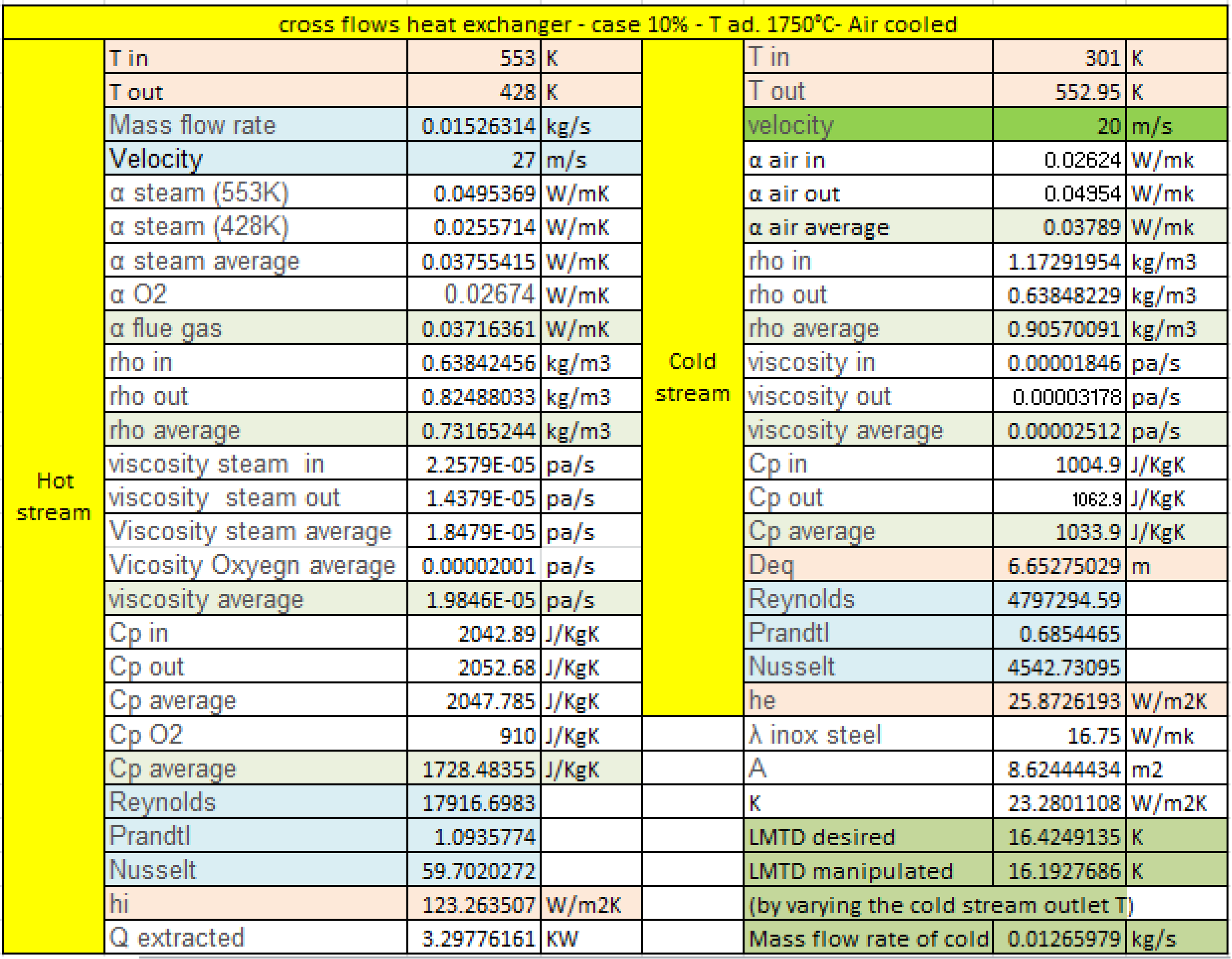
Fig. 31: calculation for the verification of the feasibility of the dimensions found - application of the heat exchangers - water cooled - counter flow to the case "Combustion of hydrogen with oxygen and recycled stream - 10% load - Adiabatic Temperature =1750 ° C - Excess of oxygen in the wet flue gases = 20,73% by volume"

The same method has been applied for the other two cases; the following table shows the results obtained when sizing the cross flows heat exchanger cooled with air at ambient temperature; the outlet temperature of the air is assumed to be 370°C.

Sizing cross flows heat exchanger - case 100% - Producing Hot Air at 370°C										
HOT SIDE	T in	1358	K	COLD SIDE	T in	298	K	Section free of tubes	0.02083465	m2
	T out	423	K		T out	643	K	Total Section needed	0.05225058	m2
	Velocity	27	m/s		Mass flow rate	0.36123	kg/s	Diameter intern of outside tube	0.25792926	m
	Mass flow rate	0.071012335	kg/s		velocity	20	m/s	Number of tubes	100	
	α steam (1358 K)	0.149	W/mK		α air in	0.02624	W/mk	Mass f.r. per tube	0.00071012	
	α steam (423K)	0.0289	W/mK		α air out	0.04954	W/mk	d int - small tubes	0.018	m
	α steam average	0.08895	W/mK		α air average	0.03789	W/mk	d ext - small tubes	0.02	m
	α O2	0.0342	W/mK		rho in	1.184727	kg/m3	Section occupied by tubes	0.03141593	m2
	α flue gas	0.084374963	W/mK		rho out	0.549065	kg/m3	Perimeter	6.28318531	m
	rho in	0.277336041	kg/m3		rho average	0.866896	kg/m3	λ inox steel	16.75	W/mk
	rho out	0.824880328	kg/m3		viscosity in	1.85E-05	pa/s	K (global coeff)	50.993144	W/mK
	rho average	0.551108185	kg/m3		viscosity out	3.178E-05	pa/s	LMTD	338.308804	K
	viscosity in steam	0.00005063	pa/s		viscosity average	2.51E-05	pa/s	Q exch	148.783994	KW
	viscosity out steam	1.41887E-05	pa/s		Cp in	1004.9	J/KgK	A	8.62444434	m2
	Viscosity average steam	3.24094E-05	pa/s		Cp out	1062.9	J/KgK			
	Viscosity average oxygen	0.00002163	pa/s		Cp average	1033.9	J/KgK	Length of heat exc	1.37262295	m
	viscosity average	3.18476E-05	pa/s		Deq	0.033264	m			
	Cp in	2563.25	J/KgK		Reynolds	22958.77				
	Cp out	2052.68	J/KgK		Prandtl	0.685447				
	Cp average	2307.965	J/KgK		Nusselt	63.28181				
	Cp O2	1020	J/KgK		hi	72.0829	W/m2K			
	Cp average	2240.839788	J/KgK							
	Reynolds	8410.018843								
	Prandtl	0.845811089								
	Nusselt	30.18195891								
	he	141.4778698	W/m2K							

Fig. 32: Calculation for the size of the heat exchanger–Air cooled – counter flow – based on the case "Combustion of hydrogen with oxygen and recycled stream – 100% load – Adiabatic Temperature =2000°C – Excess of oxygen in the wet flue gases = 3% in volume"

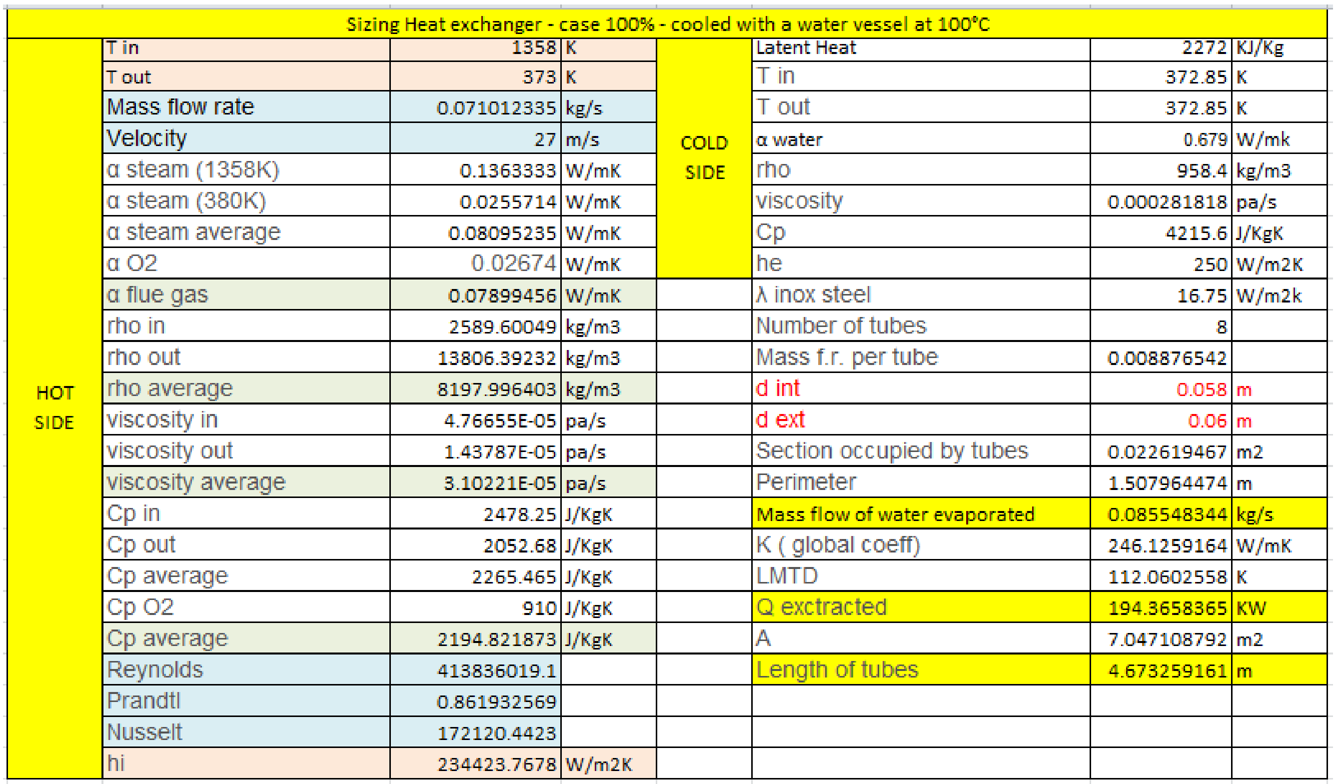
The behavior of the heat exchanger when the load is reduced to 10% is described in the following table:



cross flows heat exchanger - case 10% - T ad. 1750°C- Air cooled							
Hot stream	T in	553	K	Cold stream	T in	301	K
	T out	428	K		T out	552.95	K
	Mass flow rate	0.01526314	kg/s		velocity	20	m/s
	Velocity	27	m/s		α air in	0.02624	W/mk
	α steam (553K)	0.0495369	W/mK		α air out	0.04954	W/mk
	α steam (428K)	0.0255714	W/mK		α air average	0.03789	W/mk
	α steam average	0.03755415	W/mK		rho in	1.17291954	kg/m3
	α O2	0.02674	W/mK		rho out	0.63848229	kg/m3
	α flue gas	0.03716361	W/mK		rho average	0.90570091	kg/m3
	rho in	0.63842456	kg/m3		viscosity in	0.00001846	pa/s
	rho out	0.82488033	kg/m3		viscosity out	0.00003178	pa/s
	rho average	0.73165244	kg/m3		viscosity average	0.00002512	pa/s
	viscosity steam in	2.2579E-05	pa/s		Cp in	1004.9	J/KgK
	viscosity steam out	1.4379E-05	pa/s		Cp out	1062.9	J/KgK
	Viscosity steam average	1.8479E-05	pa/s		Cp average	1033.9	J/KgK
	Vicosity Oxyegn average	0.00002001	pa/s		Deq	6.65275029	m
	viscosity average	1.9846E-05	pa/s		Reynolds	4797294.59	
	Cp in	2042.89	J/KgK		Prandtl	0.6854465	
	Cp out	2052.68	J/KgK		Nusselt	4542.73095	
	Cp average	2047.785	J/KgK		he	25.8726193	W/m2K
	Cp O2	910	J/KgK		λ inox steel	16.75	W/mk
	Cp average	1728.48355	J/KgK		A	8.62444434	m2
	Reynolds	17916.6983			K	23.2801108	W/m2K
	Prandtl	1.0935774			LMTD desired	16.4249135	K
	Nusselt	59.7020272			LMTD manipulated	16.1927686	K
	hi	123.263507	W/m2K		(by varying the cold stream outlet T)		
	Q extracted	3.29776161	KW		Mass flow rate of cold	0.01265979	kg/s

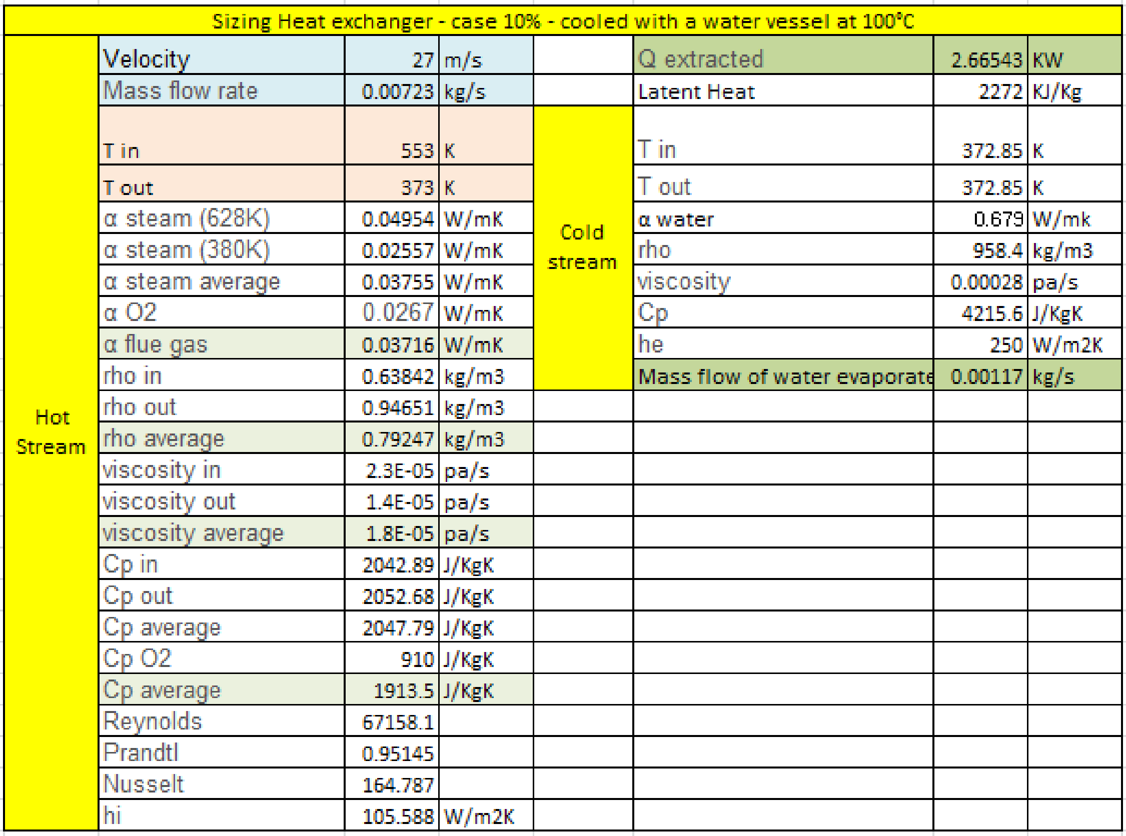
Fig. 33: calculation for the verification of the feasibility of the dimensions found - application of the heat exchangers – Air cooled - counter flow to the case "Combustion of hydrogen with oxygen and recycled stream - 10% load - Adiabatic Temperature =1750 ° C - Excess of oxygen in the wet flue gases = 20,73% by volume"

The following two tables shows the results for the third case; calculation has been made using the previous method; in this case there is the difference that the water is not flowing and it keeps the same temperature, hence the convective transfer will be reduced and the conduction will be increased. The global transfer coefficient is calculated according to this fact; reference[4-5].



Sizing Heat exchanger - case 100% - cooled with a water vessel at 100°C							
HOT SIDE	T in	1358	K	COLD SIDE	Latent Heat	2272	KJ/Kg
	T out	373	K		T in	372.85	K
	Mass flow rate	0.071012335	kg/s		T out	372.85	K
	Velocity	27	m/s		α water	0.679	W/mk
	α steam (1358K)	0.1363333	W/mK		rho	958.4	kg/m3
	α steam (380K)	0.0255714	W/mK		viscosity	0.000281818	pa/s
	α steam average	0.08095235	W/mK		Cp	4215.6	J/KgK
	α O2	0.02674	W/mK		he	250	W/m2K
	α flue gas	0.07899456	W/mK		λ inox steel	16.75	W/m2k
	rho in	2589.60049	kg/m3		Number of tubes	8	
	rho out	13806.39232	kg/m3		Mass f.r. per tube	0.008876542	
	rho average	8197.996403	kg/m3		d int	0.058	m
	viscosity in	4.76655E-05	pa/s		d ext	0.06	m
	viscosity out	1.43787E-05	pa/s		Section occupied by tubes	0.022619467	m2
	viscosity average	3.10221E-05	pa/s		Perimeter	1.507964474	m
	Cp in	2478.25	J/KgK		Mass flow of water evaporated	0.085548344	kg/s
	Cp out	2052.68	J/KgK		K (global coeff)	246.1259164	W/mK
	Cp average	2265.465	J/KgK		LMTD	112.0602558	K
	Cp O2	910	J/KgK		Q exctracted	194.3658365	KW
	Cp average	2194.821873	J/KgK		A	7.047108792	m2
	Reynolds	413836019.1			Length of tubes	4.673259161	m
	Prandtl	0.861932569					
	Nusselt	172120.4423					
	hi	234423.7678	W/m2K				

Fig. 34: Calculation for the size of the heat exchanger–cooled with a water vessel at 100°C – based on the case “Combustion of hydrogen with oxygen and recycled stream – 100% load – Adiabatic Temperature =2000°C – Excess of oxygen in the wet flue gases = 3% in volume”



Sizing Heat exchanger - case 10% - cooled with a water vessel at 100°C							
Hot Stream	Velocity	27	m/s		Q extracted	2.66543	KW
	Mass flow rate	0.00723	kg/s		Latent Heat	2272	KJ/Kg
	T in	553	K	Cold stream	T in	372.85	K
	T out	373	K		T out	372.85	K
	α steam (628K)	0.04954	W/mK		α water	0.679	W/mk
	α steam (380K)	0.02557	W/mK		rho	958.4	kg/m3
	α steam average	0.03755	W/mK		viscosity	0.00028	pa/s
	α O2	0.0267	W/mK		Cp	4215.6	J/KgK
	α flue gas	0.03716	W/mK		he	250	W/m2K
	rho in	0.63842	kg/m3		Mass flow of water evaporate	0.00117	kg/s
	rho out	0.94651	kg/m3				
	rho average	0.79247	kg/m3				
	viscosity in	2.3E-05	pa/s				
	viscosity out	1.4E-05	pa/s				
	viscosity average	1.8E-05	pa/s				
	Cp in	2042.89	J/KgK				
	Cp out	2052.68	J/KgK				
	Cp average	2047.79	J/KgK				
	Cp O2	910	J/KgK				
	Cp average	1913.5	J/KgK				
	Reynolds	67158.1					
	Prandtl	0.95145					
	Nusselt	164.787					
	hi	105.588	W/m2K				

Fig. 35: alculation for the verification of the feasibility of the dimensions found - application of the heat exchangers – cooled with a water vessel at 100°C - counter flow to the case "Combustion of hydrogen with oxygen and recycled stream - 10% load - Adiabatic Temperature =1750 ° C - Excess of oxygen in the wet flue gases = 20,73% by volume"

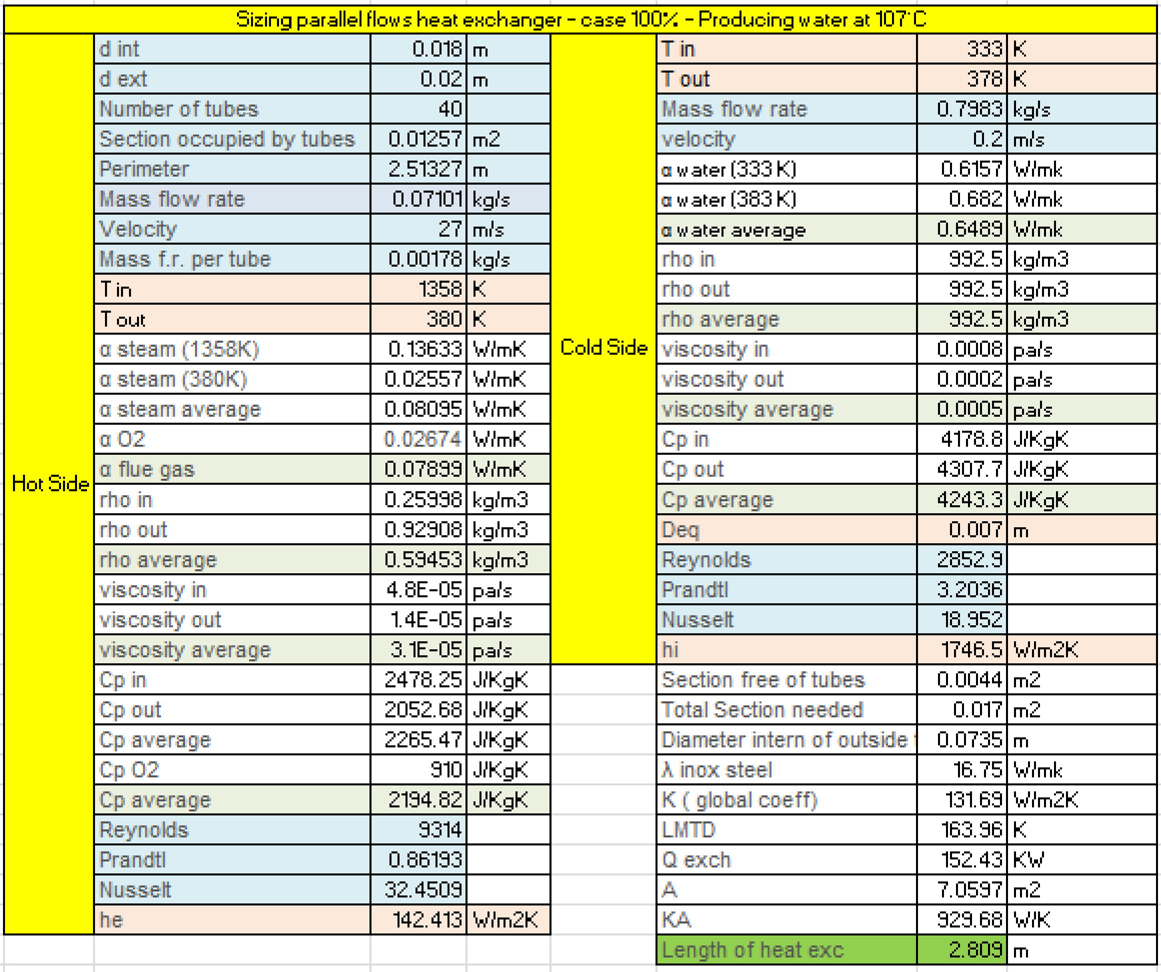
The last final case analyzes the first heat exchanger in the case in which it works with parallel flows, the method which is used to find the result is the same that has been used until now, the only difference is that the Nusselt number will be calculated with the following formula:

$$Nu = 0{,}023 * Re^{0{,}8} * Pr^{0{,}4} \tag{37}$$

And the LMTD becomes:

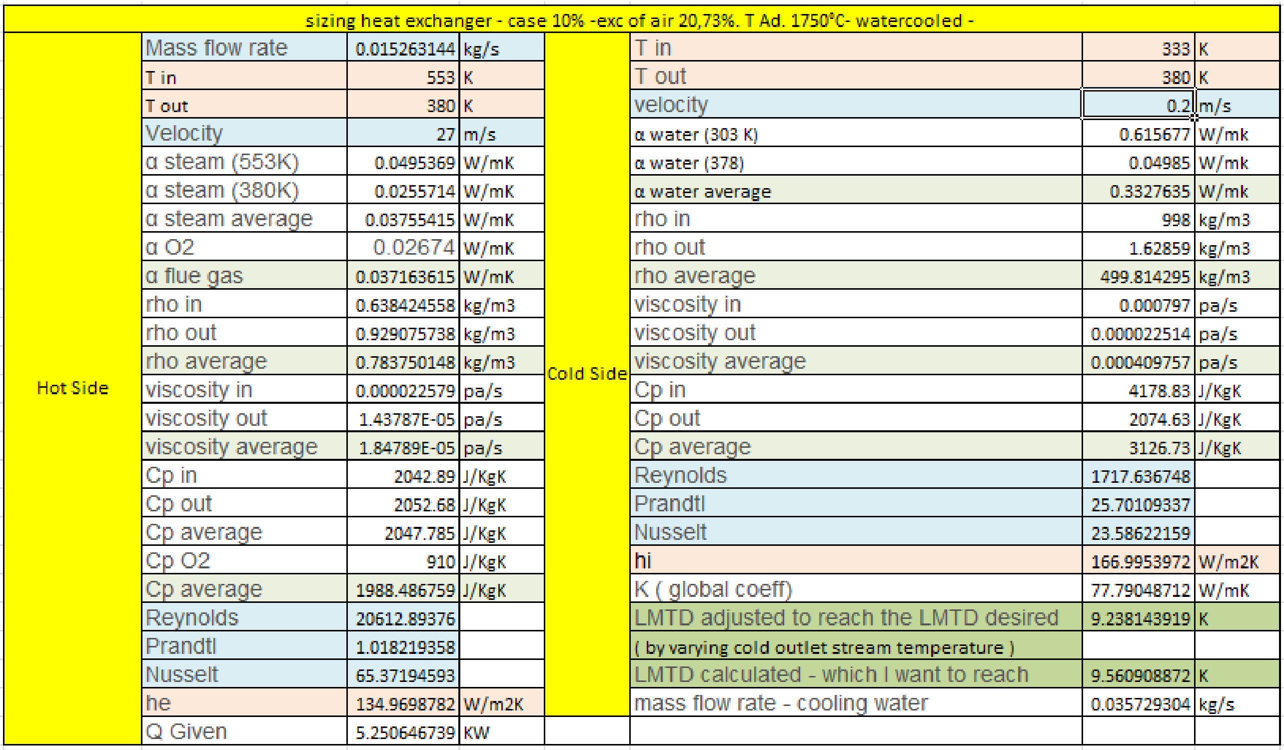
$$LMTD = \frac{(T_{Hot-in} - T_{Cold-in}) - (T_{Hot-out} - T_{Cold-out})}{LN\,\frac{(T_{Hot-in} - T_{Cold-in})}{(T_{Hot-out} - T_{Cold-out})}} \tag{38}$$

Results are shown is the two tables below:



Sizing parallel flows heat exchanger - case 100% - Producing water at 107°C							
Hot Side	d int	0.018	m	Cold Side	T in	333	K
	d ext	0.02	m		T out	378	K
	Number of tubes	40			Mass flow rate	0.7983	kg/s
	Section occupied by tubes	0.01257	m2		velocity	0.2	m/s
	Perimeter	2.51327	m		α water (333 K)	0.6157	W/mk
	Mass flow rate	0.07101	kg/s		α water (383 K)	0.682	W/mk
	Velocity	27	m/s		α water average	0.6489	W/mk
	Mass f.r. per tube	0.00178	kg/s		rho in	992.5	kg/m3
	T in	1358	K		rho out	992.5	kg/m3
	T out	380	K		rho average	992.5	kg/m3
	α steam (1358K)	0.13633	W/mK		viscosity in	0.0008	pa/s
	α steam (380K)	0.02557	W/mK		viscosity out	0.0002	pa/s
	α steam average	0.08095	W/mK		viscosity average	0.0005	pa/s
	α O2	0.02674	W/mK		Cp in	4178.8	J/KgK
	α flue gas	0.07899	W/mK		Cp out	4307.7	J/KgK
	rho in	0.25998	kg/m3		Cp average	4243.3	J/KgK
	rho out	0.92908	kg/m3		Deq	0.007	m
	rho average	0.59453	kg/m3		Reynolds	2852.9	
	viscosity in	4.8E-05	pa/s		Prandtl	3.2036	
	viscosity out	1.4E-05	pa/s		Nusselt	18.952	
	viscosity average	3.1E-05	pa/s		hi	1746.5	W/m2K
	Cp in	2478.25	J/KgK		Section free of tubes	0.0044	m2
	Cp out	2052.68	J/KgK		Total Section needed	0.017	m2
	Cp average	2265.47	J/KgK		Diameter intern of outside	0.0735	m
	Cp O2	910	J/KgK		λ inox steel	16.75	W/mk
	Cp average	2194.82	J/KgK		K (global coeff)	131.69	W/m2K
	Reynolds	9314			LMTD	163.96	K
	Prandtl	0.86193			Q exch	152.43	KW
	Nusselt	32.4509			A	7.0597	m2
	he	142.413	W/m2K		KA	929.68	W/K
					Length of heat exc	2.809	m

Fig. 36: Calculation for the size of the heat exchanger–water cooled – parallel flow – based on the case “Combustion of hydrogen with oxygen and recycled stream – 100% load – Adiabatic Temperature =2000°C – Excess of oxygen in the wet flue gases = 3% in volume”



sizing heat exchanger - case 10% -exc of air 20,73%. T Ad. 1750°C- watercooled -							
Hot Side	Mass flow rate	0.015263144	kg/s	Cold Side	T in	333	K
	T in	553	K		T out	380	K
	T out	380	K		velocity	0.2	m/s
	Velocity	27	m/s		α water (303 K)	0.615677	W/mk
	α steam (553K)	0.0495369	W/mK		α water (378)	0.04985	W/mk
	α steam (380K)	0.0255714	W/mK		α water average	0.3327635	W/mk
	α steam average	0.03755415	W/mK		rho in	998	kg/m3
	α O2	0.02674	W/mK		rho out	1.62859	kg/m3
	α flue gas	0.037163615	W/mK		rho average	499.814295	kg/m3
	rho in	0.638424558	kg/m3		viscosity in	0.000797	pa/s
	rho out	0.929075738	kg/m3		viscosity out	0.000022514	pa/s
	rho average	0.783750148	kg/m3		viscosity average	0.000409757	pa/s
	viscosity in	0.000022579	pa/s		Cp in	4178.83	J/KgK
	viscosity out	1.43787E-05	pa/s		Cp out	2074.63	J/KgK
	viscosity average	1.84789E-05	pa/s		Cp average	3126.73	J/KgK
	Cp in	2042.89	J/KgK		Reynolds	1717.636748	
	Cp out	2052.68	J/KgK		Prandtl	25.70109337	
	Cp average	2047.785	J/KgK		Nusselt	23.58622159	
	Cp O2	910	J/KgK		hi	166.9953972	W/m2K
	Cp average	1988.486759	J/KgK		K (global coeff)	77.79048712	W/mK
	Reynolds	20612.89376			LMTD adjusted to reach the LMTD desired	9.238143919	K
	Prandtl	1.018219358			(by varying cold outlet stream temperature)		
	Nusselt	65.37194593			LMTD calculated - which I want to reach	9.560908872	K
	he	134.9698782	W/m2K		mass flow rate - cooling water	0.035729304	kg/s
	Q Given	5.250646739	KW				

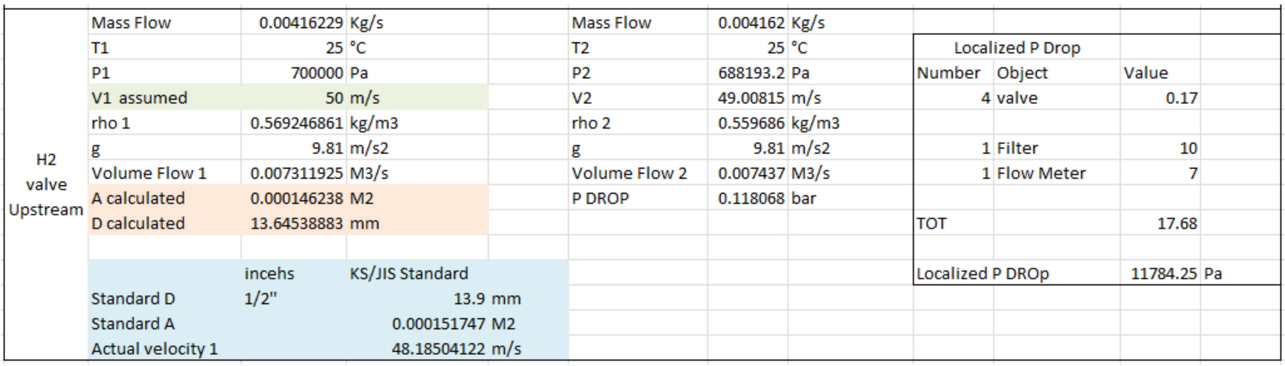
Fig. 37: calculation for the verification of the feasibility of the dimensions found - application of the heat exchangers - water cooled - parallel flow to the case "Combustion of hydrogen with oxygen and recycled stream - 10% load - Adiabatic Temperature =1750 ° C - Excess of oxygen in the wet flue gases = 20,73% by volume"

APPENDIX 6 : Pipe Sizing

Here are some of the calculation tables used to define the pressure losses localized along the lines. The formula of pressure loss that has been used is the following[6-1]:

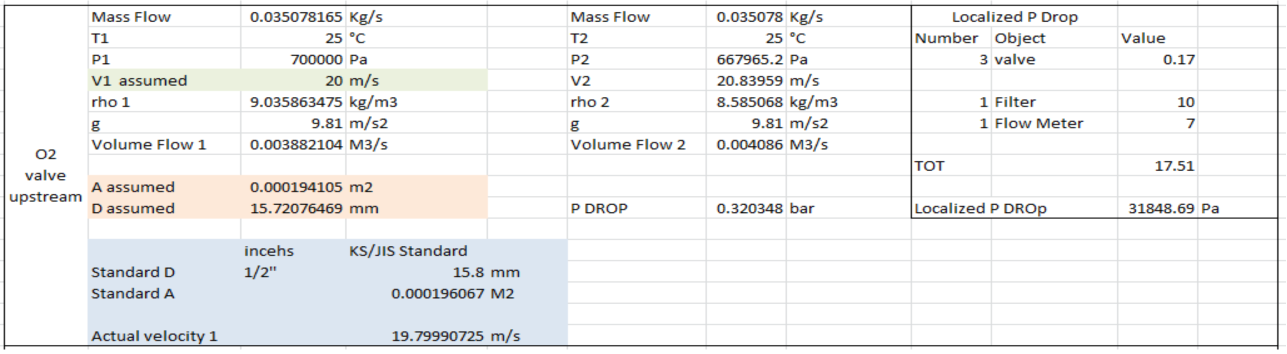
$$\Delta P = \frac{V^2}{2} * \rho * \sum K \tag{39}$$

Where K are the pressure drop factors characterizing the equipment on the line: such as valves, filters, transmitters or pipe bending. V is the average velocity in the pipe and ρ the average density. [6-1]



H2 valve Upstream	Mass Flow	0.00416229	Kg/s		Mass Flow	0.004162	Kg/s				
	T1	25	°C		T2	25	°C	Localized P Drop			
	P1	700000	Pa		P2	688193.2	Pa	Number	Object	Value	
	V1 assumed	50	m/s		V2	49.00815	m/s	4	valve	0.17	
	rho 1	0.569246861	kg/m3		rho 2	0.559686	kg/m3				
	g	9.81	m/s2		g	9.81	m/s2	1	Filter	10	
	Volume Flow 1	0.007311925	M3/s		Volume Flow 2	0.007437	M3/s	1	Flow Meter	7	
	A calculated	0.000146238	M2		P DROP	0.118068	bar				
	D calculated	13.64538883	mm					TOT		17.68	
		incehs	KS/JIS Standard					Localized P DROp		11784.25	Pa
	Standard D	1/2"	13.9	mm							
	Standard A		0.000151747	M2							
	Actual velocity 1		48.18504122	m/s							

Fig. 38: Example of the calculation of the pressure drop on the hydrogen line before the control valve



O2 valve upstream	Mass Flow	0.035078165	Kg/s		Mass Flow	0.035078	Kg/s	Localized P Drop			
	T1	25	°C		T2	25	°C	Number	Object	Value	
	P1	700000	Pa		P2	667965.2	Pa	3	valve	0.17	
	V1 assumed	20	m/s		V2	20.83959	m/s				
	rho 1	9.035863475	kg/m3		rho 2	8.585068	kg/m3	1	Filter	10	
	g	9.81	m/s2		g	9.81	m/s2	1	Flow Meter	7	
	Volume Flow 1	0.003882104	M3/s		Volume Flow 2	0.004086	M3/s				
								TOT		17.51	
	A assumed	0.000194105	m2								
	D assumed	15.72076469	mm		P DROP	0.320348	bar	Localized P DROp		31848.69	Pa
		incehs	KS/JIS Standard								
	Standard D	1/2"	15.8	mm							
	Standard A		0.000196067	M2							
	Actual velocity 1		19.79990725	m/s							

Fig. 39: example of the calculation of the pressure drop on the oxygen line before the control valve

It is reported only the calculation made from the hydrogen and the oxygen lines, but the same method has been applied also for the other lines and the results are reported in the main report.

APPENDIX 7 : Burner

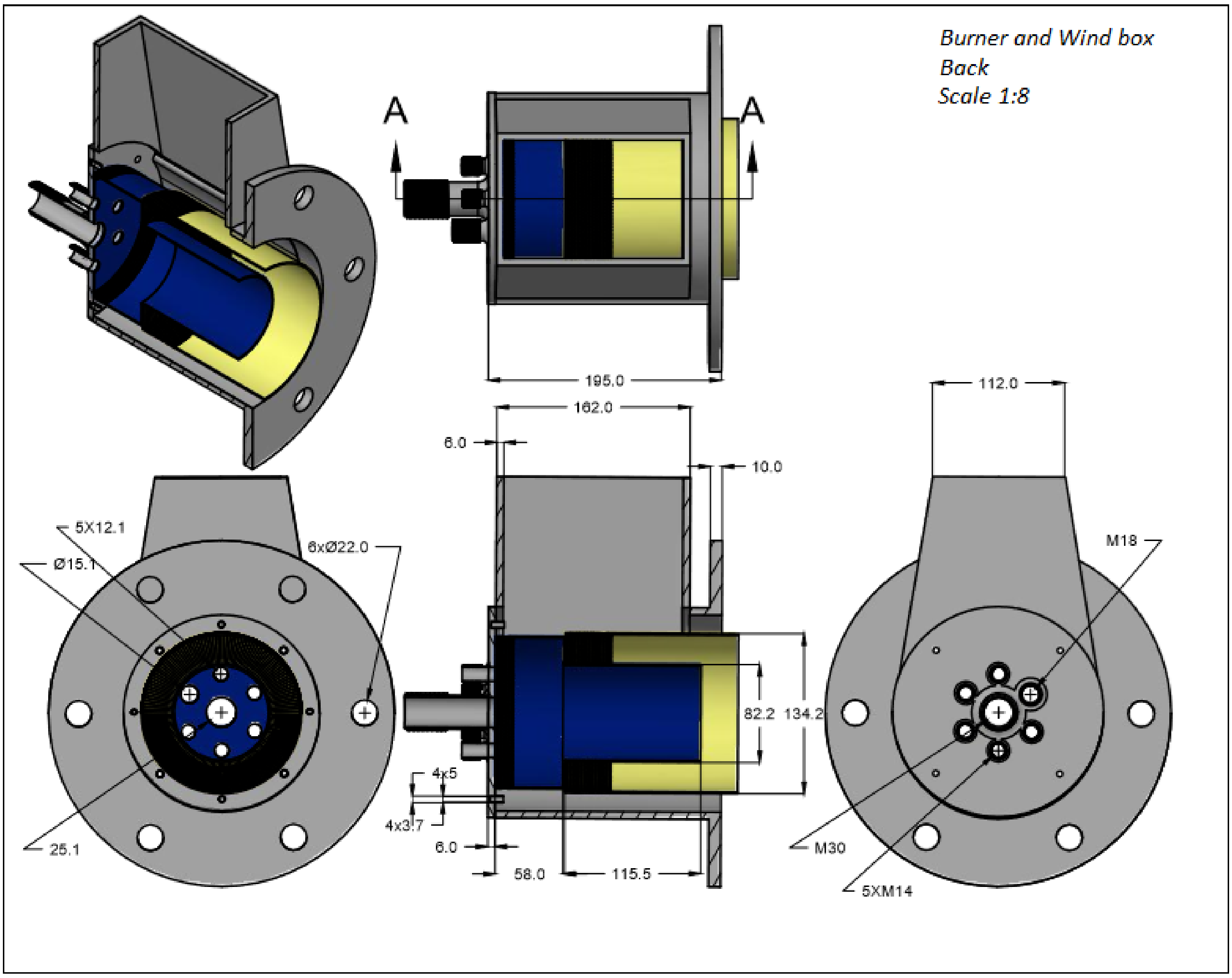


Fig. 40: Burner and wind-box back - 2D drawing

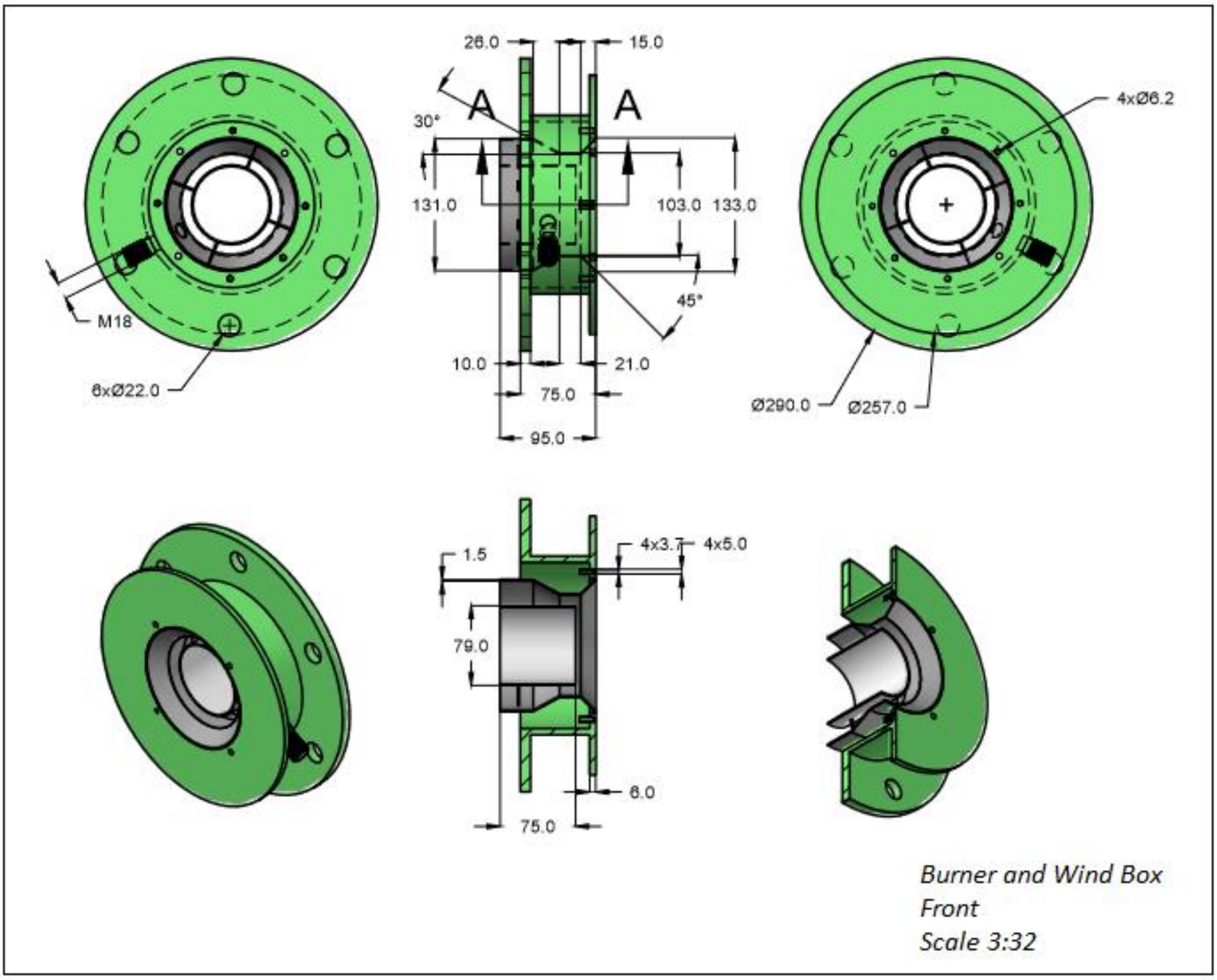


Fig. 41: Burner and wind-box front - 2D drawing

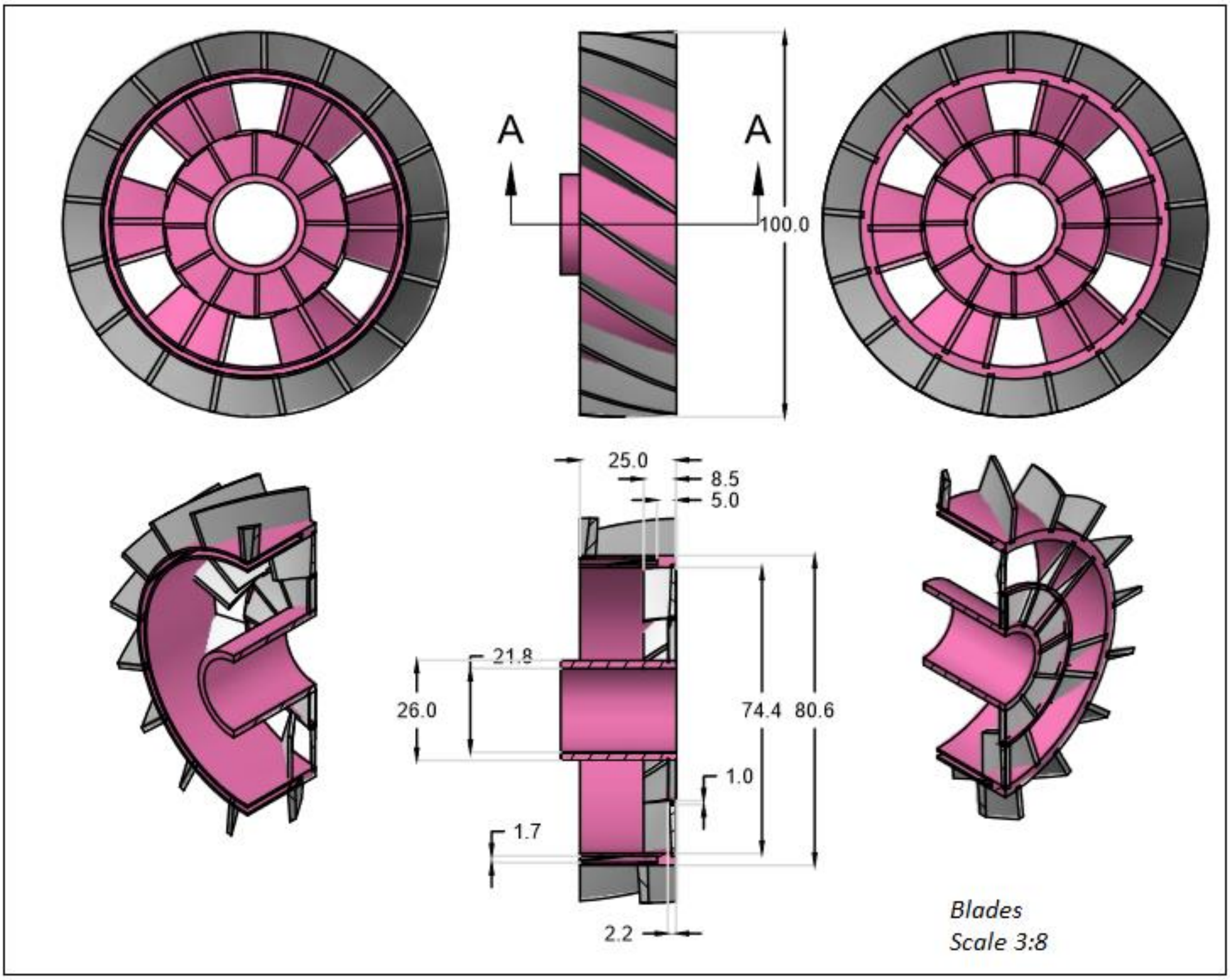


Fig. 42: Blades of the burner - 2D drawing

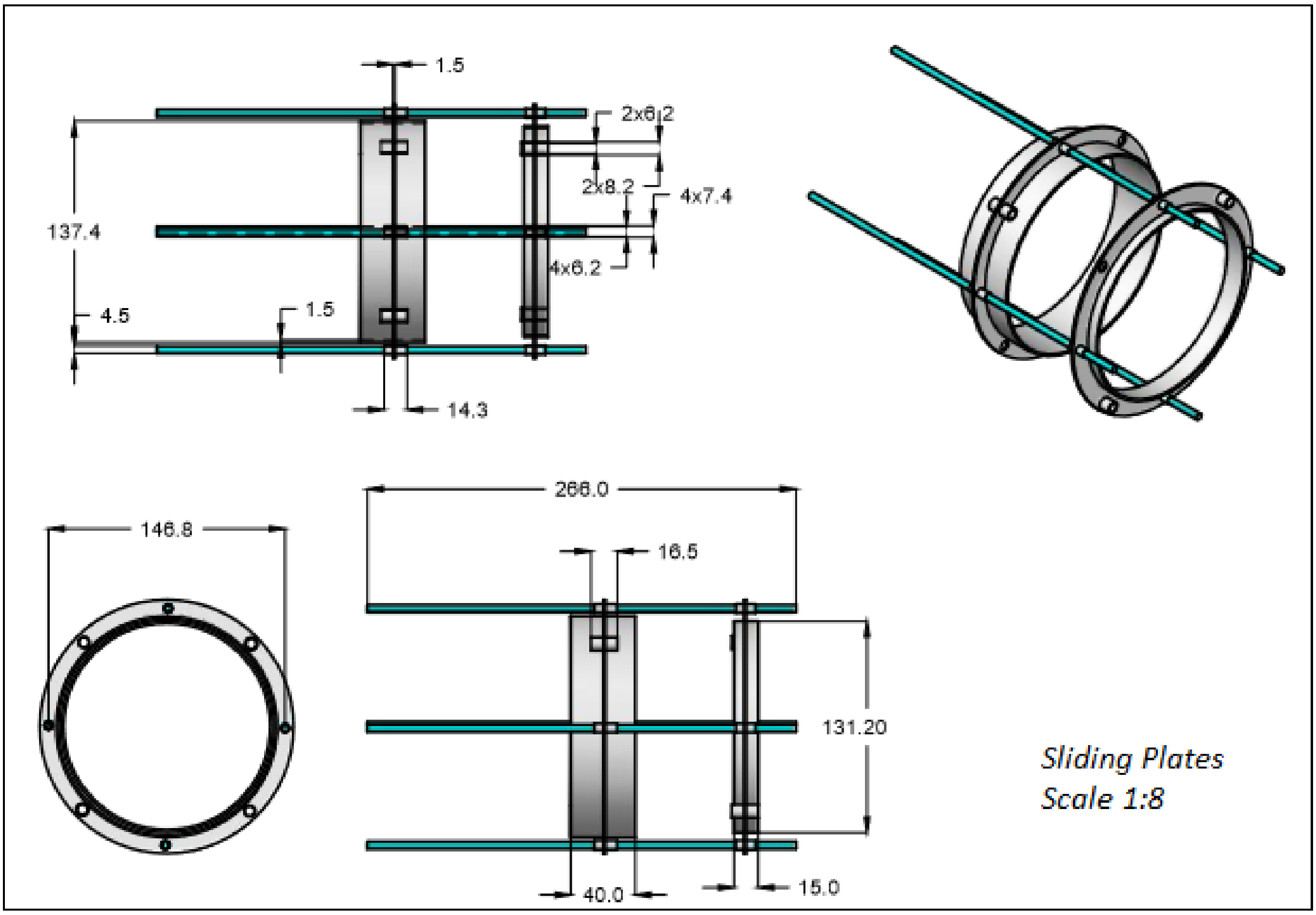
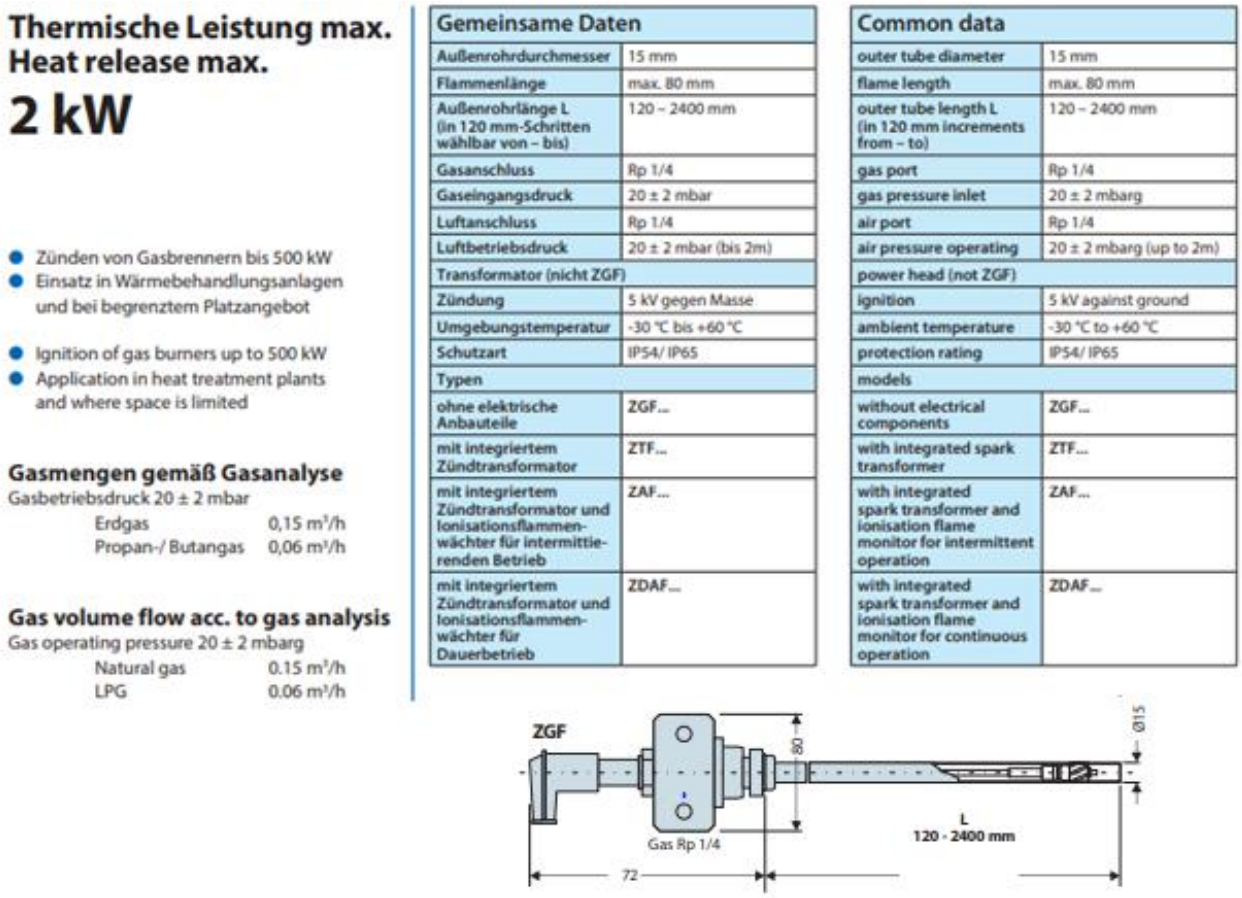


Fig. 43: Sliding plate system for the primary and secondary air inlets - 2D drawing



Thermische Leistung max.
Heat release max.
2 kW

- Zünden von Gasbrennern bis 500 kW
- Einsatz in Wärmebehandlungsanlagen und bei begrenztem Platzangebot

- Ignition of gas burners up to 500 kW
- Application in heat treatment plants and where space is limited

Gasmengen gemäß Gasanalyse
Gasbetriebsdruck 20 ± 2 mbar

Erdgas	0,15 m³/h
Propan-/ Butangas	0,06 m³/h

Gas volume flow acc. to gas analysis
Gas operating pressure 20 ± 2 mbarg

Natural gas	0.15 m³/h
LPG	0.06 m³/h

Gemeinsame Daten	
Außenrohrdurchmesser	15 mm
Flammenlänge	max. 80 mm
Außenrohrlänge L (in 120 mm-Schritten wählbar von – bis)	120 – 2400 mm
Gasanschluss	Rp 1/4
Gaseingangsdruck	20 ± 2 mbar
Luftanschluss	Rp 1/4
Luftbetriebsdruck	20 ± 2 mbar (bis 2m)
Transformator (nicht ZGF)	
Zündung	5 kV gegen Masse
Umgebungstemperatur	-30 °C bis +60 °C
Schutzart	IP54/ IP65
Typen	
ohne elektrische Anbauteile	ZGF...
mit integriertem Zündtransformator	ZTF...
mit integriertem Zündtransformator und Ionisationsflammenwächter für intermittierenden Betrieb	ZAF...
mit integriertem Zündtransformator und Ionisationsflammenwächter für Dauerbetrieb	ZDAF...

Common data	
outer tube diameter	15 mm
flame length	max. 80 mm
outer tube length L (in 120 mm increments from – to)	120 – 2400 mm
gas port	Rp 1/4
gas pressure inlet	20 ± 2 mbarg
air port	Rp 1/4
air pressure operating	20 ± 2 mbarg (up to 2m)
power head (not ZGF)	
ignition	5 kV against ground
ambient temperature	-30 °C to +60 °C
protection rating	IP54/ IP65
models	
without electrical components	ZGF...
with integrated spark transformer	ZTF...
with integrated spark transformer and ionisation flame monitor for intermittent operation	ZAF...
with integrated spark transformer and ionisation flame monitor for continuous operation	ZDAF...

Fig. 44: Selected Ignitor – Data sheet

APPENDIX 8 : Static Mixer

Different correlations has been used to analyze the pressure drop inside the static mixer. Reference [4-4] suggests to use the following formula:

$$\Delta P_{drop} = \frac{0{,}0135 * L * Q^2 * \rho * f * F}{D_{eq}} \qquad \{40\}$$

Where L is the length if static mixer [m]; Q is the volume flow rate [cum/s]; f is given for table according to the number of elements which compose the mixer and F depends on the Reynolds number:

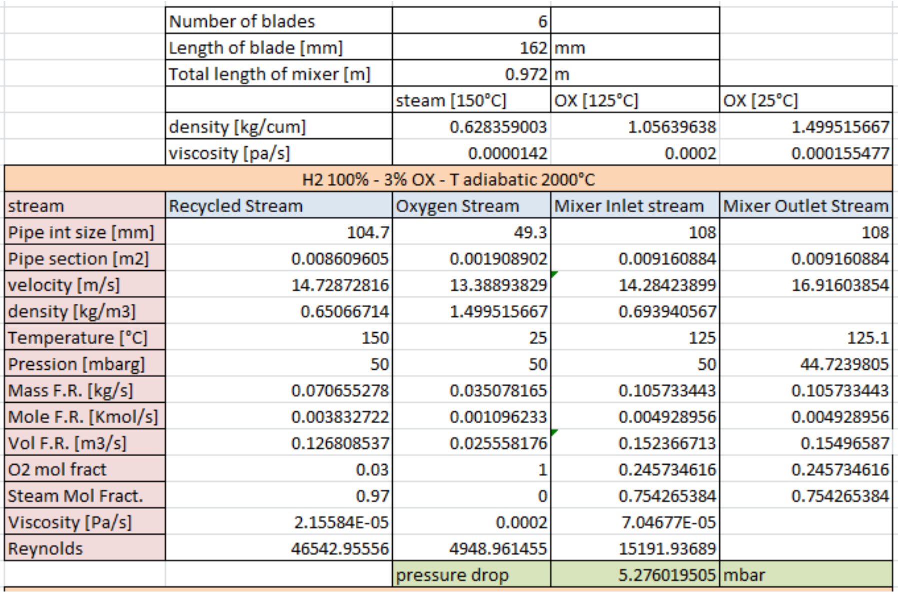
F=6,5 0< Re <100
F=1,53*Re^(0,45) 100 < Re < 1000
F= 8,5 *LN (Re) -16 Re>1000
Reference [7-5] instead suggests to use a different correlation according to the number of Reynolds:
ΔP_{drop}=0,0008*Re^(2) 0< Re <1000
ΔP_{drop}=0,0028*Re^(1,78) 1000 < Re < 10000
ΔP_{drop}= 0,0004 *(Re)^(1,998) Re>10000

Both of these correlation does not give reliable results. Reference [7-3] instead propose the two following formulas:

$$\Delta P_{drop} = f * \frac{v^2}{2} * \rho * \left(\frac{L}{ID}\right) \quad where\ f = 0{,}0032 + \frac{0{,}221}{Re^{0{,}257}} \tag{41}$$

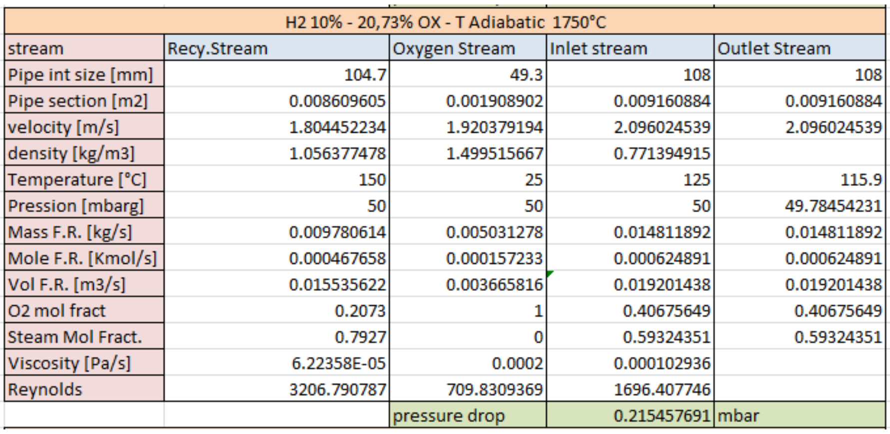
$$\Delta P_{drop} = 4{,}59 * Re^{-0.22} * \rho * v^2 \left(\frac{L}{ID}\right) \tag{42}$$

Where v is the velocity and ID the intern diameter. Among all the these formulas only the last one {38} gives reliable results. The following tables shows the calculation made:



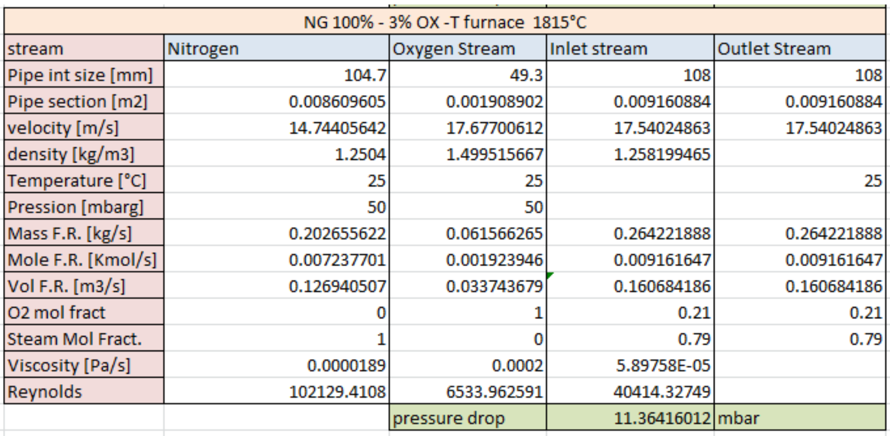
	Number of blades	6		
	Length of blade [mm]	162	mm	
	Total length of mixer [m]	0.972	m	
		steam [150°C]	OX [125°C]	OX [25°C]
	density [kg/cum]	0.628359003	1.05639638	1.499515667
	viscosity [pa/s]	0.0000142	0.0002	0.000155477
H2 100% - 3% OX - T adiabatic 2000°C				
stream	Recycled Stream	Oxygen Stream	Mixer Inlet stream	Mixer Outlet Stream
Pipe int size [mm]	104.7	49.3	108	108
Pipe section [m2]	0.008609605	0.001908902	0.009160884	0.009160884
velocity [m/s]	14.72872816	13.38893829	14.28423899	16.91603854
density [kg/m3]	0.65066714	1.499515667	0.693940567	
Temperature [°C]	150	25	125	125.1
Pression [mbarg]	50	50	50	44.7239805
Mass F.R. [kg/s]	0.070655278	0.035078165	0.105733443	0.105733443
Mole F.R. [Kmol/s]	0.003832722	0.001096233	0.004928956	0.004928956
Vol F.R. [m3/s]	0.126808537	0.025558176	0.152366713	0.15496587
O2 mol fract	0.03	1	0.245734616	0.245734616
Steam Mol Fract.	0.97	0	0.754265384	0.754265384
Viscosity [Pa/s]	2.15584E-05	0.0002	7.04677E-05	
Reynolds	46542.95556	4948.961455	15191.93689	
		pressure drop	5.276019505	mbar

Fig. 45: calculation of the pressure drop in the Static mixer for the case "Combustion of hydrogen with oxygen and recycled stream – 100% load – Adiabatic Temperature =2000°C – Excess of oxygen in the wet flue gases = 3% in volume"



H2 10% - 20,73% OX - T Adiabatic 1750°C				
stream	Recy.Stream	Oxygen Stream	Inlet stream	Outlet Stream
Pipe int size [mm]	104.7	49.3	108	108
Pipe section [m2]	0.008609605	0.001908902	0.009160884	0.009160884
velocity [m/s]	1.804452234	1.920379194	2.096024539	2.096024539
density [kg/m3]	1.056377478	1.499515667	0.771394915	
Temperature [°C]	150	25	125	115.9
Pression [mbarg]	50	50	50	49.78454231
Mass F.R. [kg/s]	0.009780614	0.005031278	0.014811892	0.014811892
Mole F.R. [Kmol/s]	0.000467658	0.000157233	0.000624891	0.000624891
Vol F.R. [m3/s]	0.015535622	0.003665816	0.019201438	0.019201438
O2 mol fract	0.2073	1	0.40675649	0.40675649
Steam Mol Fract.	0.7927	0	0.59324351	0.59324351
Viscosity [Pa/s]	6.22358E-05	0.0002	0.000102936	
Reynolds	3206.790787	709.8309369	1696.407746	
		pressure drop	0.215457691	mbar

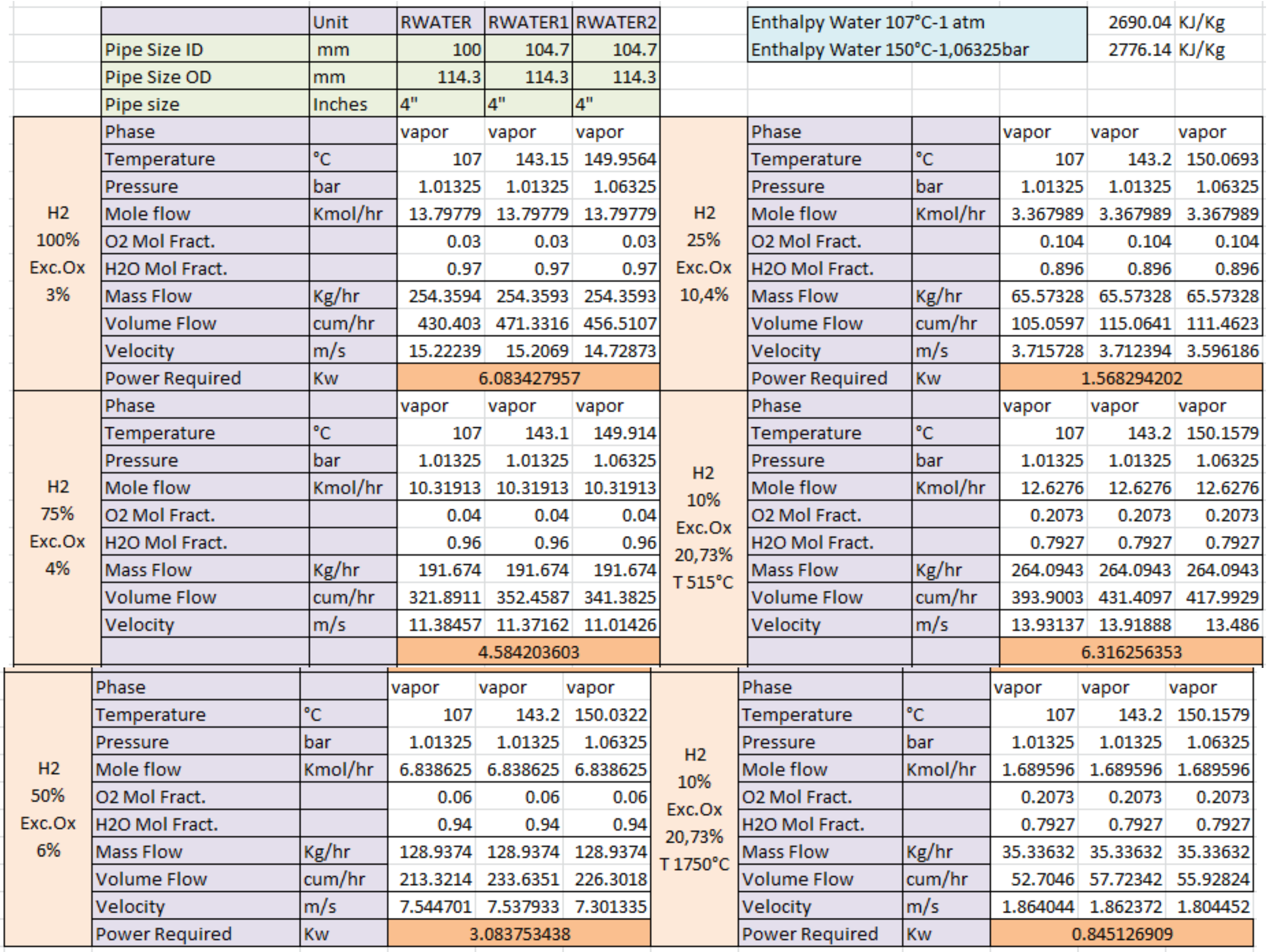
Fig. 46: calculation of the pressure drop in the Static mixer for the case "Combustion of hydrogen with oxygen and recycled stream – 10% load – Adiabatic Temperature =1750°C – Excess of oxygen in the wet flue gases = 20,73% in volume"



NG 100% - 3% OX -T furnace 1815°C				
stream	Nitrogen	Oxygen Stream	Inlet stream	Outlet Stream
Pipe int size [mm]	104.7	49.3	108	108
Pipe section [m2]	0.008609605	0.001908902	0.009160884	0.009160884
velocity [m/s]	14.74405642	17.67700612	17.54024863	17.54024863
density [kg/m3]	1.2504	1.499515667	1.258199465	
Temperature [°C]	25	25		25
Pression [mbarg]	50	50		
Mass F.R. [kg/s]	0.202655622	0.061566265	0.264221888	0.264221888
Mole F.R. [Kmol/s]	0.007237701	0.001923946	0.009161647	0.009161647
Vol F.R. [m3/s]	0.126940507	0.033743679	0.160684186	0.160684186
O2 mol fract	0	1	0.21	0.21
Steam Mol Fract.	1	0	0.79	0.79
Viscosity [Pa/s]	0.0000189	0.0002	5.89758E-05	
Reynolds	102129.4108	6533.962591	40414.32749	
		pressure drop	11.36416012	mbar

Fig. 47: calculation of the pressure drop in the Static mixer for the case “Combustion of Natural Gas with air – 100% load – Adiabatic Temperature =1815°C – Excess of oxygen in the wet flue gases = 2,6% in volume”

APPENDIX 9: Electric heater



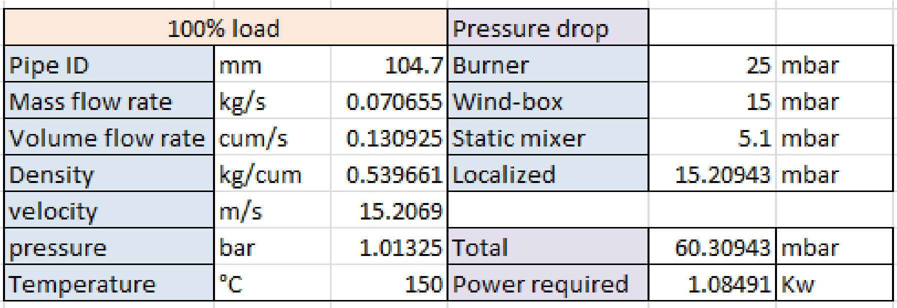
	Unit	RWATER	RWATER1	RWATER2
Pipe Size ID	mm	100	104.7	104.7
Pipe Size OD	mm	114.3	114.3	114.3
Pipe size	Inches	4"	4"	4"

Enthalpy Water 107°C-1 atm	2690.04	KJ/Kg
Enthalpy Water 150°C-1,06325bar	2776.14	KJ/Kg

H2 100% Exc.Ox 3%	Phase		vapor	vapor	vapor	H2 25% Exc.Ox 10,4%	Phase		vapor	vapor	vapor
	Temperature	°C	107	143.15	149.9564		Temperature	°C	107	143.2	150.0693
	Pressure	bar	1.01325	1.01325	1.06325		Pressure	bar	1.01325	1.01325	1.06325
	Mole flow	Kmol/hr	13.79779	13.79779	13.79779		Mole flow	Kmol/hr	3.367989	3.367989	3.367989
	O2 Mol Fract.		0.03	0.03	0.03		O2 Mol Fract.		0.104	0.104	0.104
	H2O Mol Fract.		0.97	0.97	0.97		H2O Mol Fract.		0.896	0.896	0.896
	Mass Flow	Kg/hr	254.3594	254.3593	254.3593		Mass Flow	Kg/hr	65.57328	65.57328	65.57328
	Volume Flow	cum/hr	430.403	471.3316	456.5107		Volume Flow	cum/hr	105.0597	115.0641	111.4623
	Velocity	m/s	15.22239	15.2069	14.72873		Velocity	m/s	3.715728	3.712394	3.596186
	Power Required	Kw	6.083427957				Power Required	Kw	1.568294202		
H2 75% Exc.Ox 4%	Phase		vapor	vapor	vapor	H2 10% Exc.Ox 20,73% T 515°C	Phase		vapor	vapor	vapor
	Temperature	°C	107	143.1	149.914		Temperature	°C	107	143.2	150.1579
	Pressure	bar	1.01325	1.01325	1.06325		Pressure	bar	1.01325	1.01325	1.06325
	Mole flow	Kmol/hr	10.31913	10.31913	10.31913		Mole flow	Kmol/hr	12.6276	12.6276	12.6276
	O2 Mol Fract.		0.04	0.04	0.04		O2 Mol Fract.		0.2073	0.2073	0.2073
	H2O Mol Fract.		0.96	0.96	0.96		H2O Mol Fract.		0.7927	0.7927	0.7927
	Mass Flow	Kg/hr	191.674	191.674	191.674		Mass Flow	Kg/hr	264.0943	264.0943	264.0943
	Volume Flow	cum/hr	321.8911	352.4587	341.3825		Volume Flow	cum/hr	393.9003	431.4097	417.9929
	Velocity	m/s	11.38457	11.37162	11.01426		Velocity	m/s	13.93137	13.91888	13.486
			4.584203603						6.316256353		
H2 50% Exc.Ox 6%	Phase		vapor	vapor	vapor	H2 10% Exc.Ox 20,73% T 1750°C	Phase		vapor	vapor	vapor
	Temperature	°C	107	143.2	150.0322		Temperature	°C	107	143.2	150.1579
	Pressure	bar	1.01325	1.01325	1.06325		Pressure	bar	1.01325	1.01325	1.06325
	Mole flow	Kmol/hr	6.838625	6.838625	6.838625		Mole flow	Kmol/hr	1.689596	1.689596	1.689596
	O2 Mol Fract.		0.06	0.06	0.06		O2 Mol Fract.		0.2073	0.2073	0.2073
	H2O Mol Fract.		0.94	0.94	0.94		H2O Mol Fract.		0.7927	0.7927	0.7927
	Mass Flow	Kg/hr	128.9374	128.9374	128.9374		Mass Flow	Kg/hr	35.33632	35.33632	35.33632
	Volume Flow	cum/hr	213.3214	233.6351	226.3018		Volume Flow	cum/hr	52.7046	57.72342	55.92824
	Velocity	m/s	7.544701	7.537933	7.301335		Velocity	m/s	1.864044	1.862372	1.804452
	Power Required	Kw	3.083753438				Power Required	Kw	0.845126909		

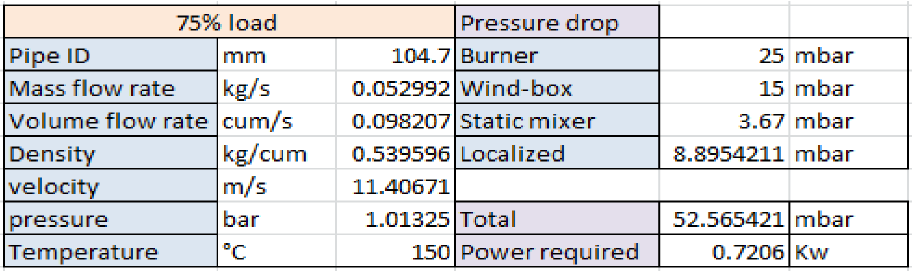
Fig. 48: Calculation for the design data of the electric heater for different standard condition

APPENDIX 10 : Recircultion fan



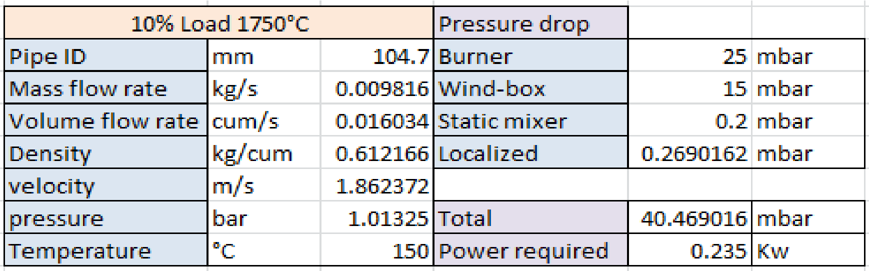
100% load			Pressure drop		
Pipe ID	mm	104.7	Burner	25	mbar
Mass flow rate	kg/s	0.070655	Wind-box	15	mbar
Volume flow rate	cum/s	0.130925	Static mixer	5.1	mbar
Density	kg/cum	0.539661	Localized	15.20943	mbar
velocity	m/s	15.2069			
pressure	bar	1.01325	Total	60.30943	mbar
Temperature	°C	150	Power required	1.08491	Kw

Fig. 49: Calculation of the required pressure increase and required power of the recirculation fan for the "Combustion of hydrogen with oxygen and recycled stream – 100% load – Adiabatic Temperature =2000°C – Excess of oxygen in the wet flue gases = 3% in volume"



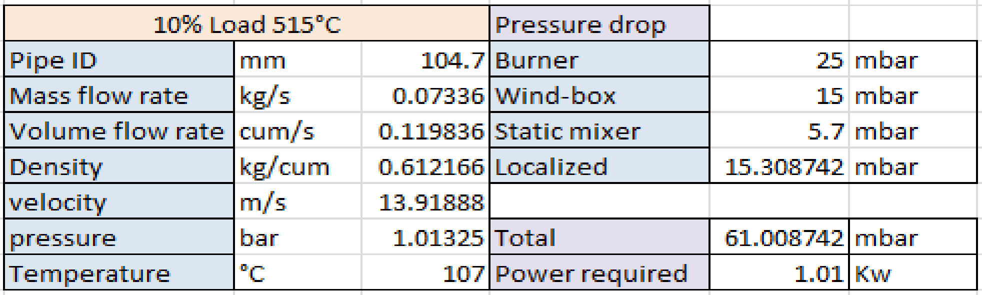
75% load			Pressure drop		
Pipe ID	mm	104.7	Burner	25	mbar
Mass flow rate	kg/s	0.052992	Wind-box	15	mbar
Volume flow rate	cum/s	0.098207	Static mixer	3.67	mbar
Density	kg/cum	0.539596	Localized	8.8954211	mbar
velocity	m/s	11.40671			
pressure	bar	1.01325	Total	52.565421	mbar
Temperature	°C	150	Power required	0.7206	Kw

Fig. 50: Calculation of the required pressure increase and required power of the recirculation fan for the "Combustion of hydrogen with oxygen and recycled stream – 75% load – Adiabatic Temperature =2000°C – Excess of oxygen in the wet flue gases = 4% in volume"



10% Load 1750°C			Pressure drop		
Pipe ID	mm	104.7	Burner	25	mbar
Mass flow rate	kg/s	0.009816	Wind-box	15	mbar
Volume flow rate	cum/s	0.016034	Static mixer	0.2	mbar
Density	kg/cum	0.612166	Localized	0.2690162	mbar
velocity	m/s	1.862372			
pressure	bar	1.01325	Total	40.469016	mbar
Temperature	°C	150	Power required	0.235	Kw

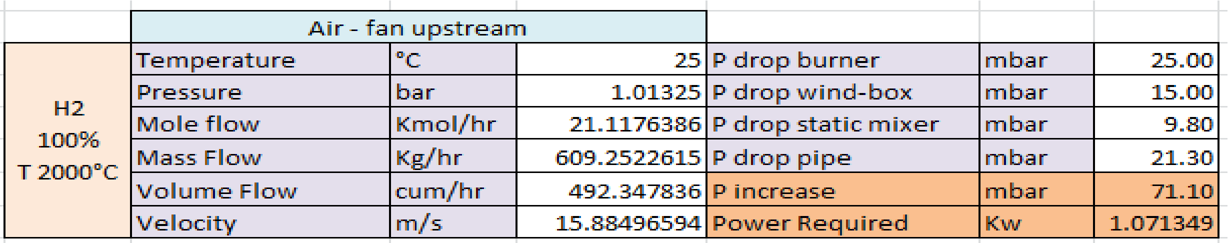
Fig. 51: Calculation of the required pressure increase and required power of the recirculation fan for the "Combustion of hydrogen with oxygen and recycled stream – 10% load – Adiabatic Temperature =1750°C – Excess of oxygen in the wet flue gases = 20,73% in volume"



10% Load 515°C			Pressure drop		
Pipe ID	mm	104.7	Burner	25	mbar
Mass flow rate	kg/s	0.07336	Wind-box	15	mbar
Volume flow rate	cum/s	0.119836	Static mixer	5.7	mbar
Density	kg/cum	0.612166	Localized	15.308742	mbar
velocity	m/s	13.91888			
pressure	bar	1.01325	Total	61.008742	mbar
Temperature	°C	107	Power required	1.01	Kw

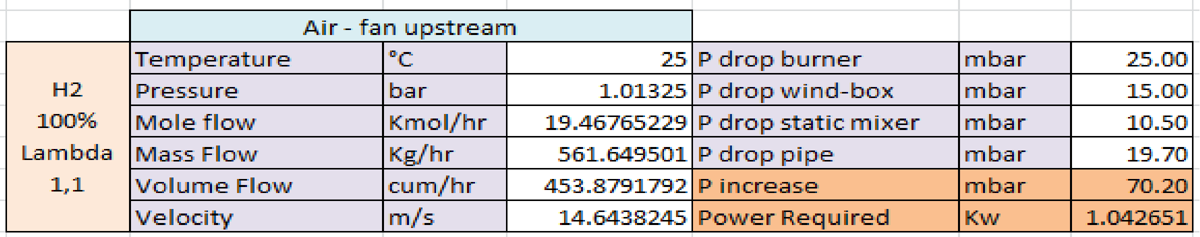
Fig. 52: Calculation of the required pressure increase and required power of the recirculation fan for the "Combustion of hydrogen with oxygen and recycled stream – 10% load – Adiabatic Temperature =515°C – Excess of oxygen in the wet flue gases = 20,73% in volume"

APPENDIX 11: AIR FAN



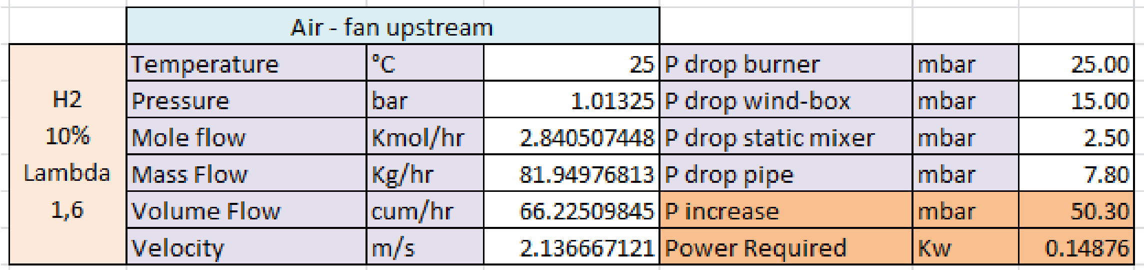
	Air - fan upstream					
H2 100% T 2000°C	Temperature	°C	25	P drop burner	mbar	25.00
	Pressure	bar	1.01325	P drop wind-box	mbar	15.00
	Mole flow	Kmol/hr	21.1176386	P drop static mixer	mbar	9.80
	Mass Flow	Kg/hr	609.2522615	P drop pipe	mbar	21.30
	Volume Flow	cum/hr	492.347836	P increase	mbar	71.10
	Velocity	m/s	15.88496594	Power Required	Kw	1.071349

Fig. 53: Calculation of the required pressure increase and required power of the air fan for the case "Combustion of hydrogen with air – 100% load – Adiabatic Temperature =2000°C – Excess of oxygen in the wet flue gases = 2,89% in volume"



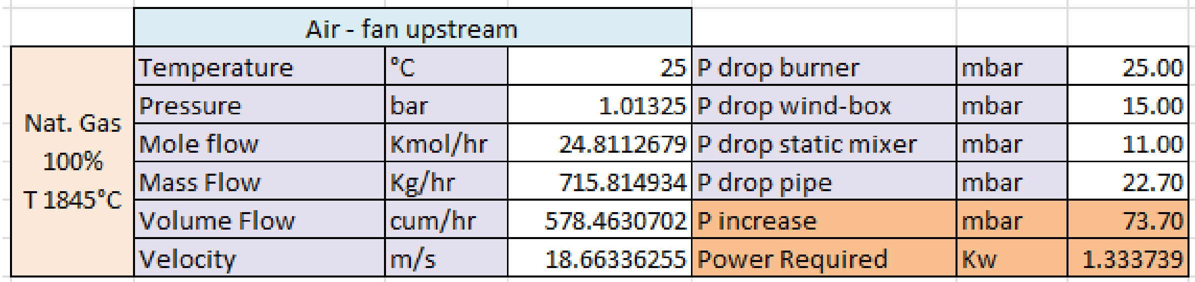
	Air - fan upstream					
H2 100% Lambda 1,1	Temperature	°C	25	P drop burner	mbar	25.00
	Pressure	bar	1.01325	P drop wind-box	mbar	15.00
	Mole flow	Kmol/hr	19.46765229	P drop static mixer	mbar	10.50
	Mass Flow	Kg/hr	561.649501	P drop pipe	mbar	19.70
	Volume Flow	cum/hr	453.8791792	P increase	mbar	70.20
	Velocity	m/s	14.6438245	Power Required	Kw	1.042651

Fig. 54: Calculation of the required pressure increase and required power of the air fan for the case "Combustion of hydrogen with air – 100% load – Adiabatic Temperature =2110°C – Excess of oxygen in the wet flue gases = 1,7% in volume"



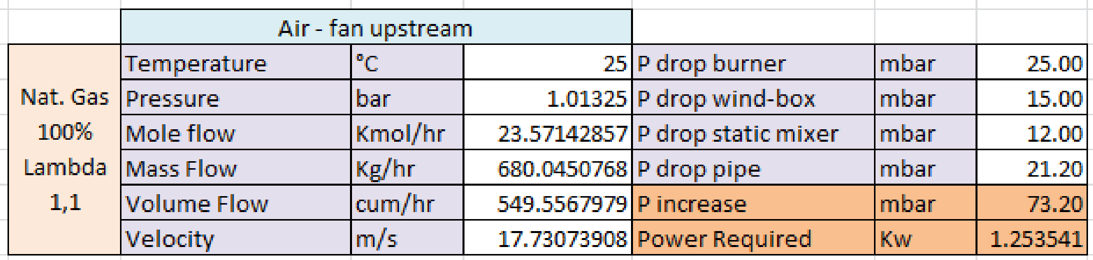
	Air - fan upstream					
H2 10% Lambda 1,6	Temperature	°C	25	P drop burner	mbar	25.00
	Pressure	bar	1.01325	P drop wind-box	mbar	15.00
	Mole flow	Kmol/hr	2.840507448	P drop static mixer	mbar	2.50
	Mass Flow	Kg/hr	81.94976813	P drop pipe	mbar	7.80
	Volume Flow	cum/hr	66.22509845	P increase	mbar	50.30
	Velocity	m/s	2.136667121	Power Required	Kw	0.14876

Fig. 55: Calculation of the required pressure increase and required power of the air fan for the case "Combustion of hydrogen with mixture air – 10% load – Adiabatic Temperature =1620°C – Excess of oxygen in the wet flue gases = 7% in volume"



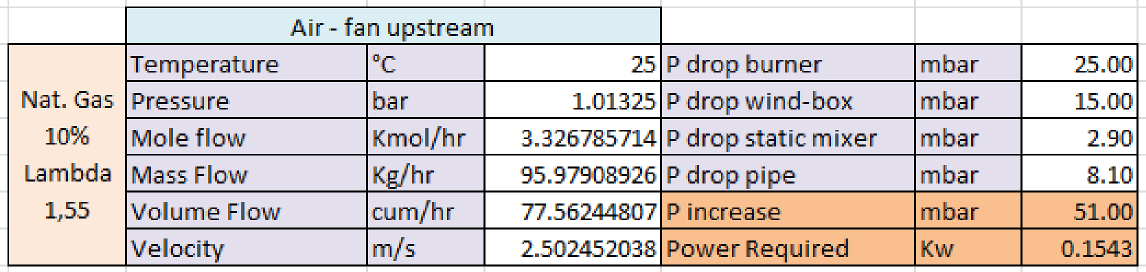
	Air - fan upstream					
Nat. Gas 100% T 1845°C	Temperature	°C	25	P drop burner	mbar	25.00
	Pressure	bar	1.01325	P drop wind-box	mbar	15.00
	Mole flow	Kmol/hr	24.8112679	P drop static mixer	mbar	11.00
	Mass Flow	Kg/hr	715.814934	P drop pipe	mbar	22.70
	Volume Flow	cum/hr	578.4630702	P increase	mbar	73.70
	Velocity	m/s	18.66336255	Power Required	Kw	1.333739

Fig. 56: Calculation of the required pressure increase and required power of the air fan for the case " Combustion of Natural Gas with air – 100% load – Adiabatic Temperature =1845°C – Excess of oxygen in the wet flue gases = 2,63% in volume"



	Air - fan upstream					
Nat. Gas 100% Lambda 1,1	Temperature	°C	25	P drop burner	mbar	25.00
	Pressure	bar	1.01325	P drop wind-box	mbar	15.00
	Mole flow	Kmol/hr	23.57142857	P drop static mixer	mbar	12.00
	Mass Flow	Kg/hr	680.0450768	P drop pipe	mbar	21.20
	Volume Flow	cum/hr	549.5567979	P increase	mbar	73.20
	Velocity	m/s	17.73073908	Power Required	Kw	1.253541

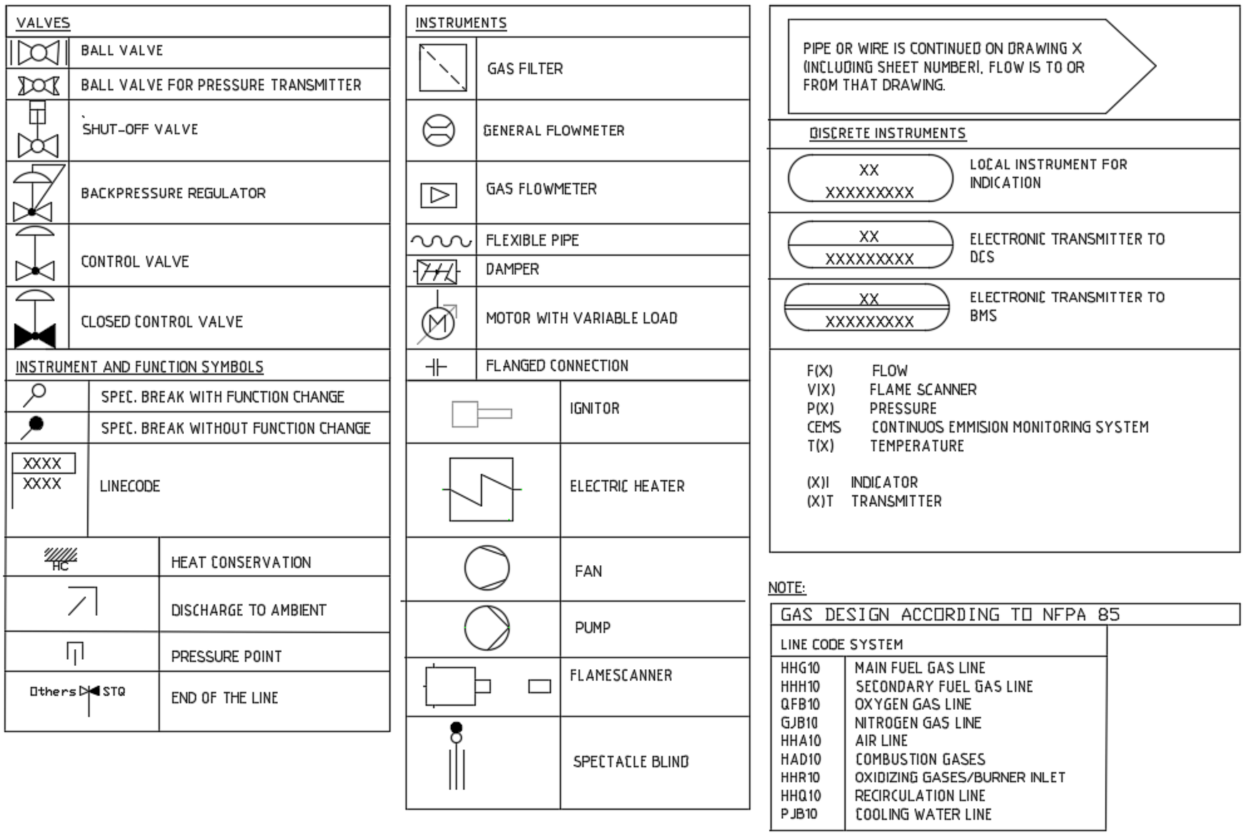
Fig. 57: Calculation of the required pressure increase and required power of the air fan for the case "Combustion of Natural Gas with air – 100% load – Adiabatic Temperature =1915°C – Excess of oxygen in the wet flue gases = 1,74% in volume"



	Air - fan upstream					
Nat. Gas	Temperature	°C	25	P drop burner	mbar	25.00
10%	Pressure	bar	1.01325	P drop wind-box	mbar	15.00
Lambda	Mole flow	Kmol/hr	3.326785714	P drop static mixer	mbar	2.90
1,55	Mass Flow	Kg/hr	95.97908926	P drop pipe	mbar	8.10
	Volume Flow	cum/hr	77.56244807	P increase	mbar	51.00
	Velocity	m/s	2.502452038	Power Required	Kw	0.1543

Fig. 58: Calculation of the required pressure increase and required power of the air fan for the case “Combustion of Natural Gas with air – 10% load – Adiabatic Temperature =1475°C – Excess of oxygen in the wet flue gases = 7% in volume”

APPENDIX 12 : PFD and P&ID



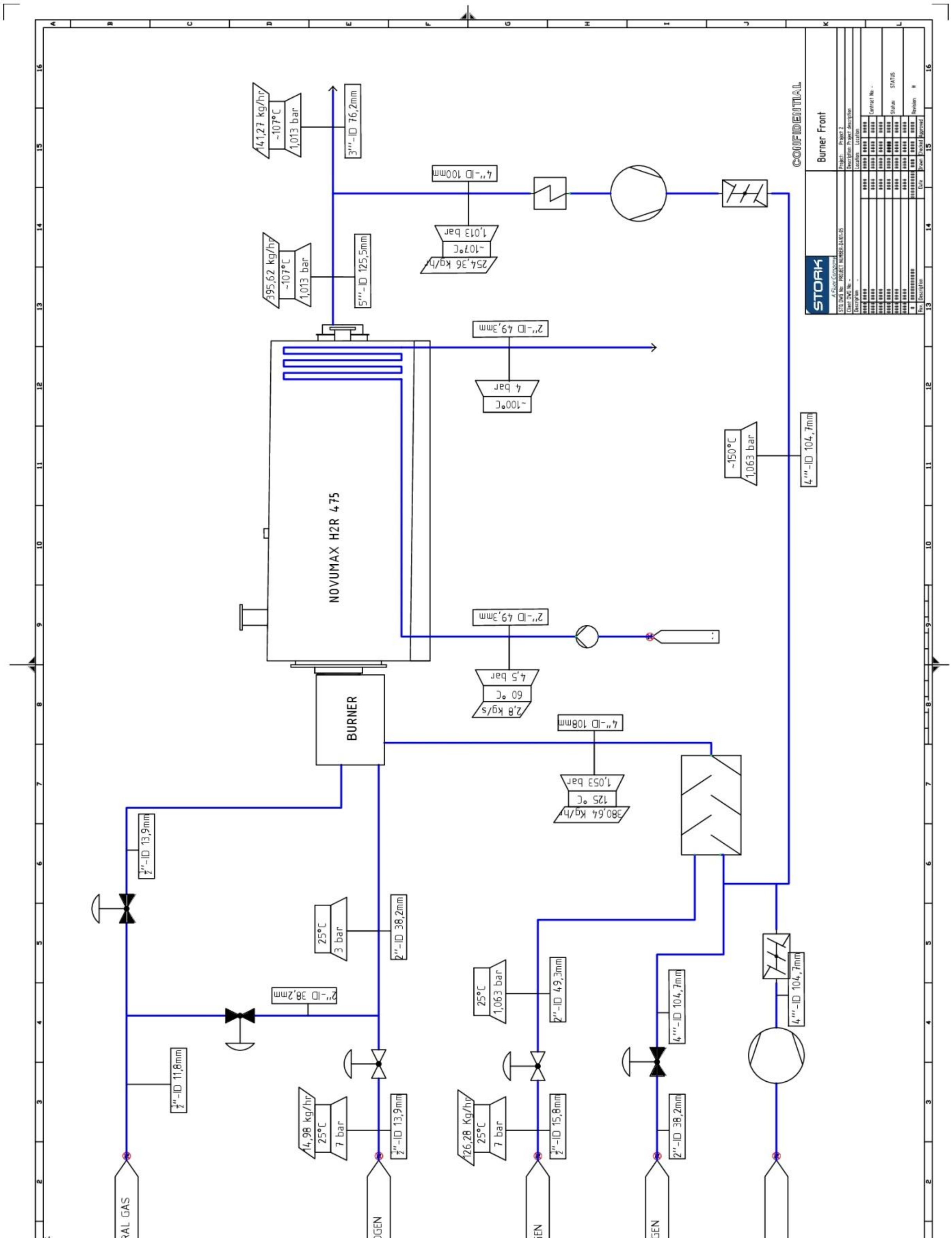
F(X) FLOW
V(X) FLAME SCANNER
P(X) PRESSURE
CEMS CONTINUOS EMMISION MONITORING SYSTEM
T(X) TEMPERATURE

(X)I INDICATOR
(X)T TRANSMITTER

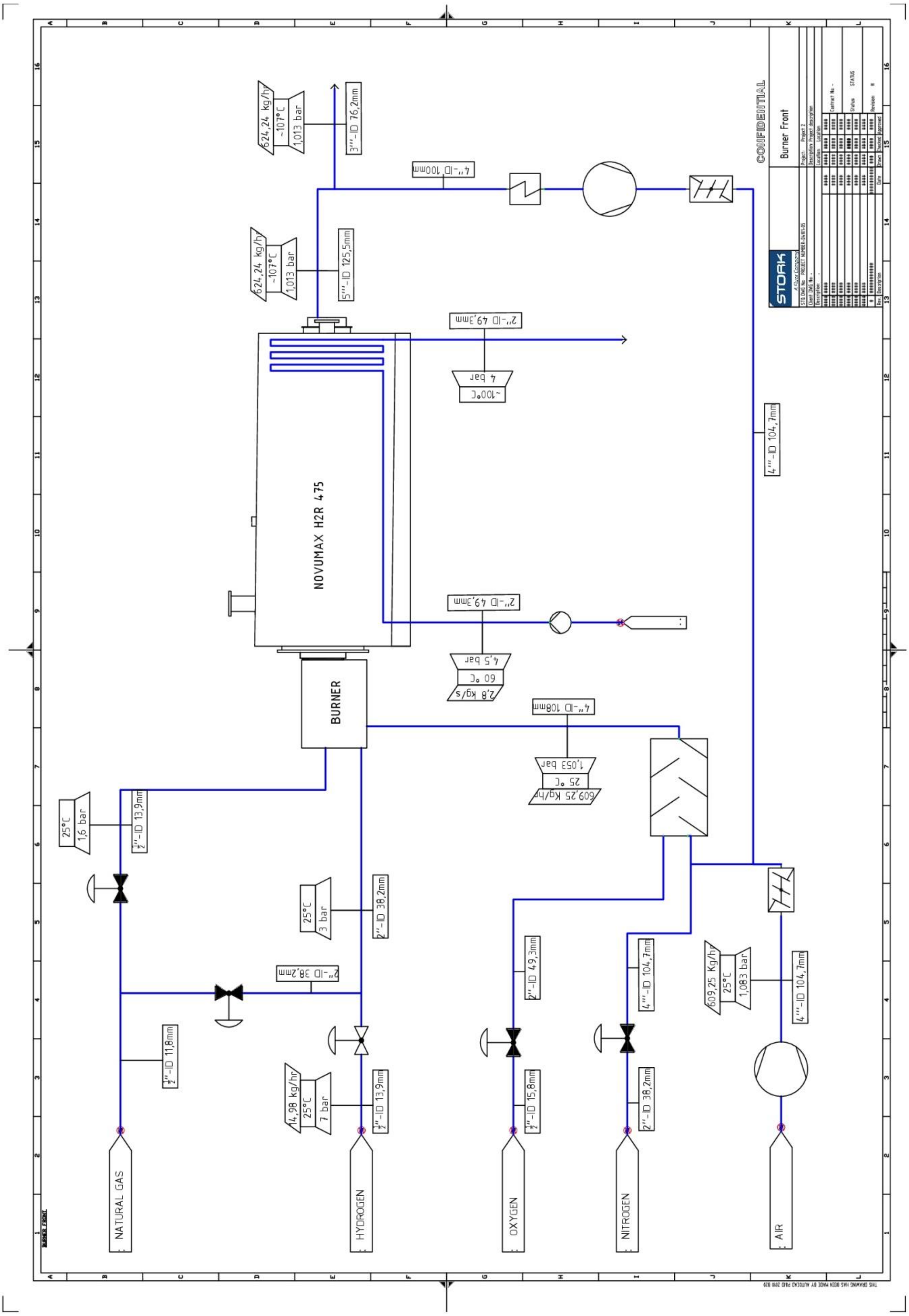
LINE CODE SYSTEM	
HHG10	MAIN FUEL GAS LINE
HHH10	SECONDARY FUEL GAS LINE
QFB10	OXYGEN GAS LINE
GJB10	NITROGEN GAS LINE
HHA10	AIR LINE
HAD10	COMBUSTION GASES
HHR10	OXIDIZING GASES/BURNER INLET
HHQ10	RECIRCULATION LINE
PJB10	COOLING WATER LINE

Fig. 59: list of the symbols used in the PFD and P&ID

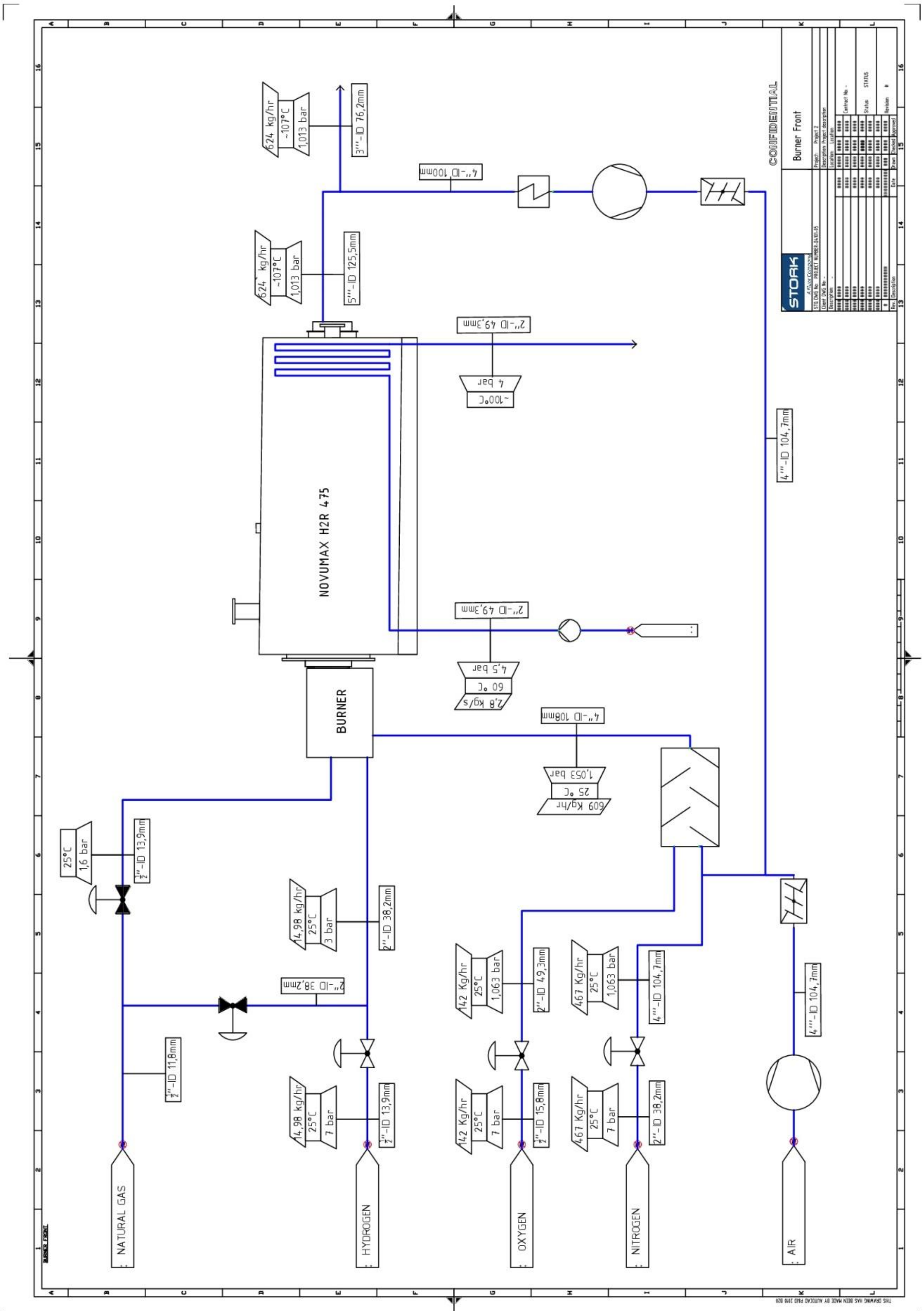
- 100% load – hydrogen- combustion with oxygen and recycled stream (excess oxygen in wet fumes = 3% in volume and adiabatic flame temperature = 2000 ° C)



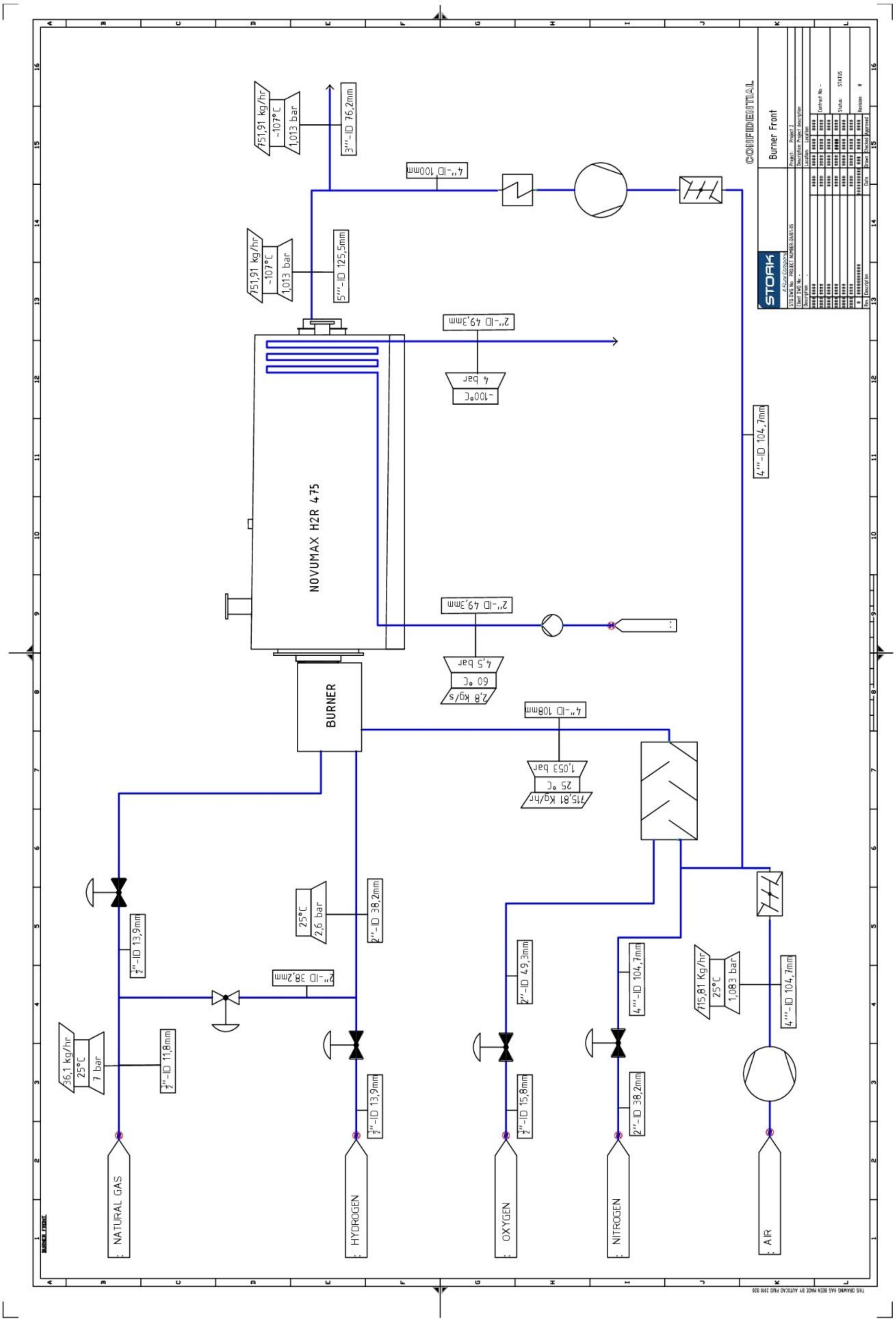
-100% load – hydrogen- combustion with air (excess oxygen in wet fumes = 2,89% in volume and adiabatic flame temperature = 2000 ° C)



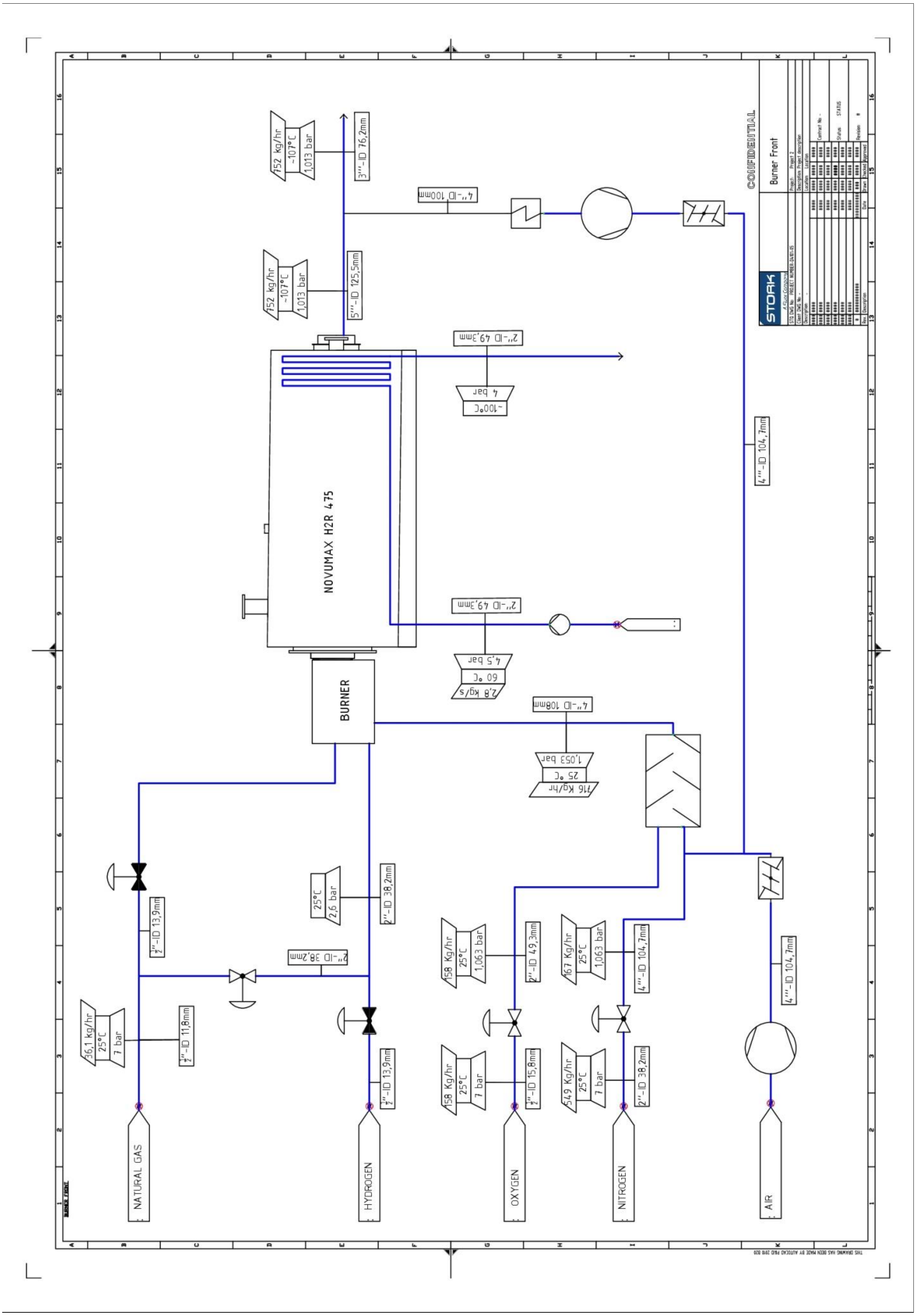
-100% load – hydrogen- combustion with oxygen and nitrogen (excess oxygen in wet fumes = 2,89% in volume and adiabatic flame temperature = 2000 ° C)



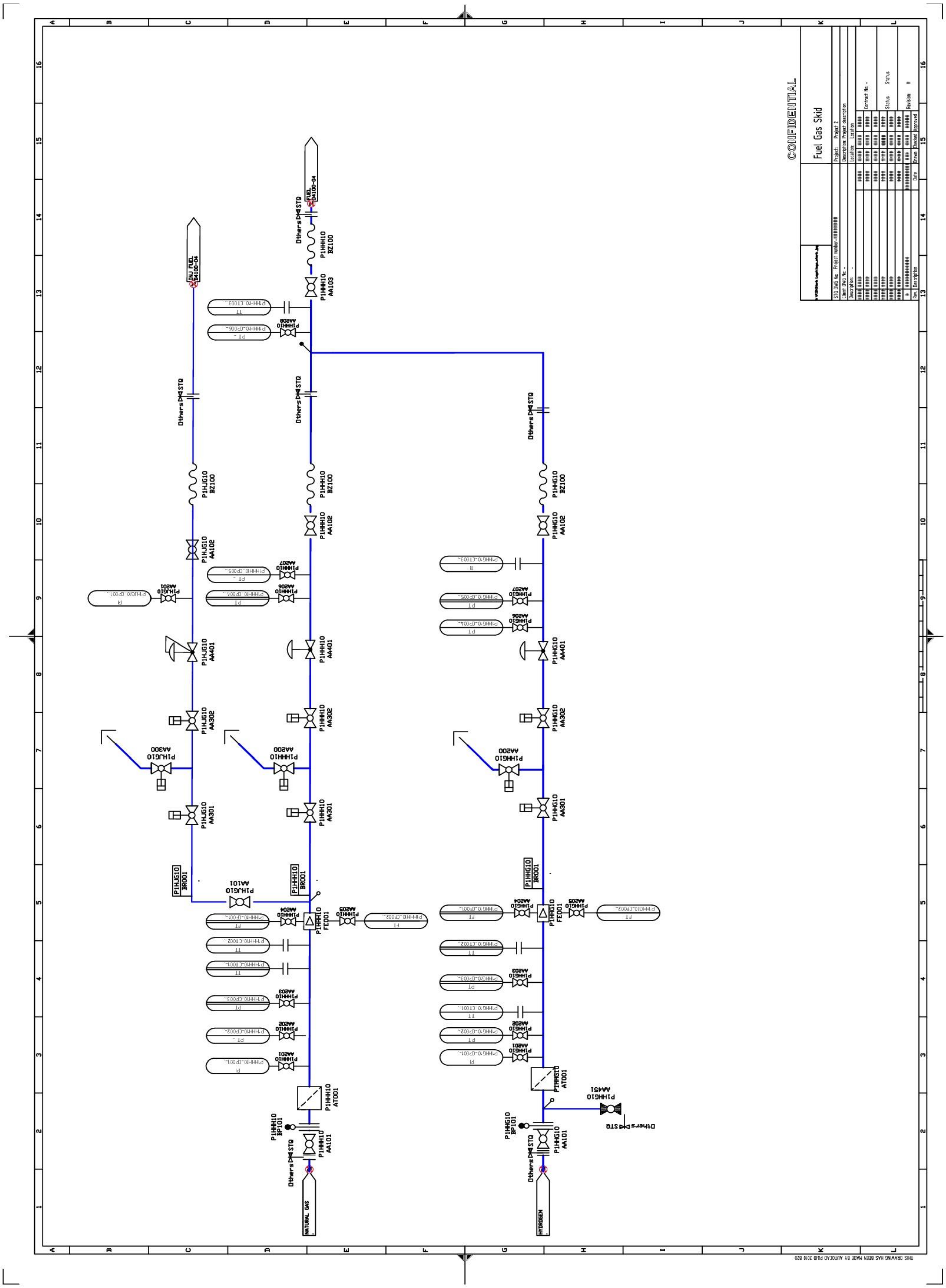
-100% load – Natural Gas combustion with air (excess oxygen in wet fumes = 2,63% in volume and adiabatic flame temperature = 2000 ° C)



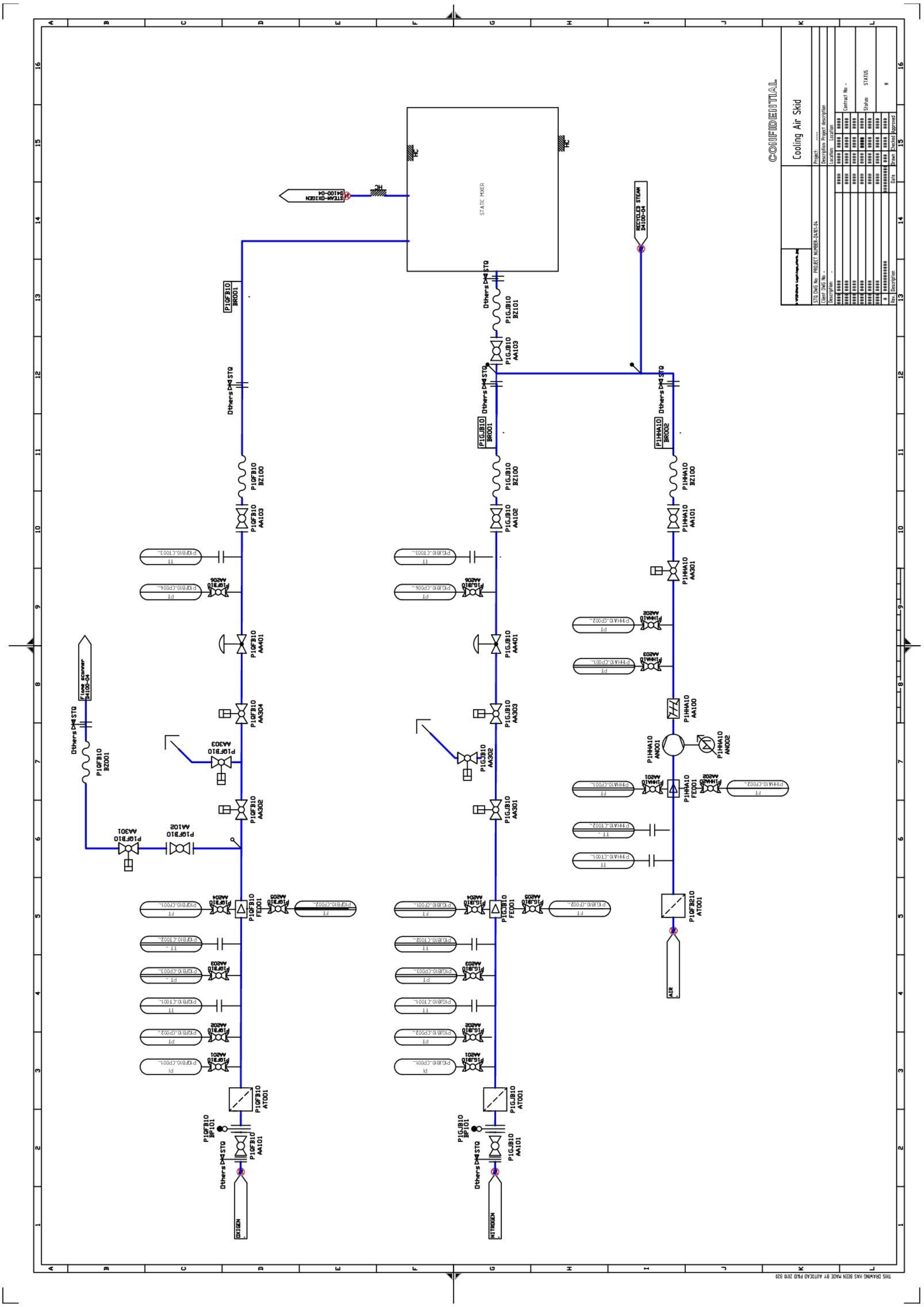
-100% load – Natural Gas combustion with nitrogen and oxygen (excess oxygen in wet fumes = 2,63% in volume and adiabatic flame temperature = 2000 ° C)



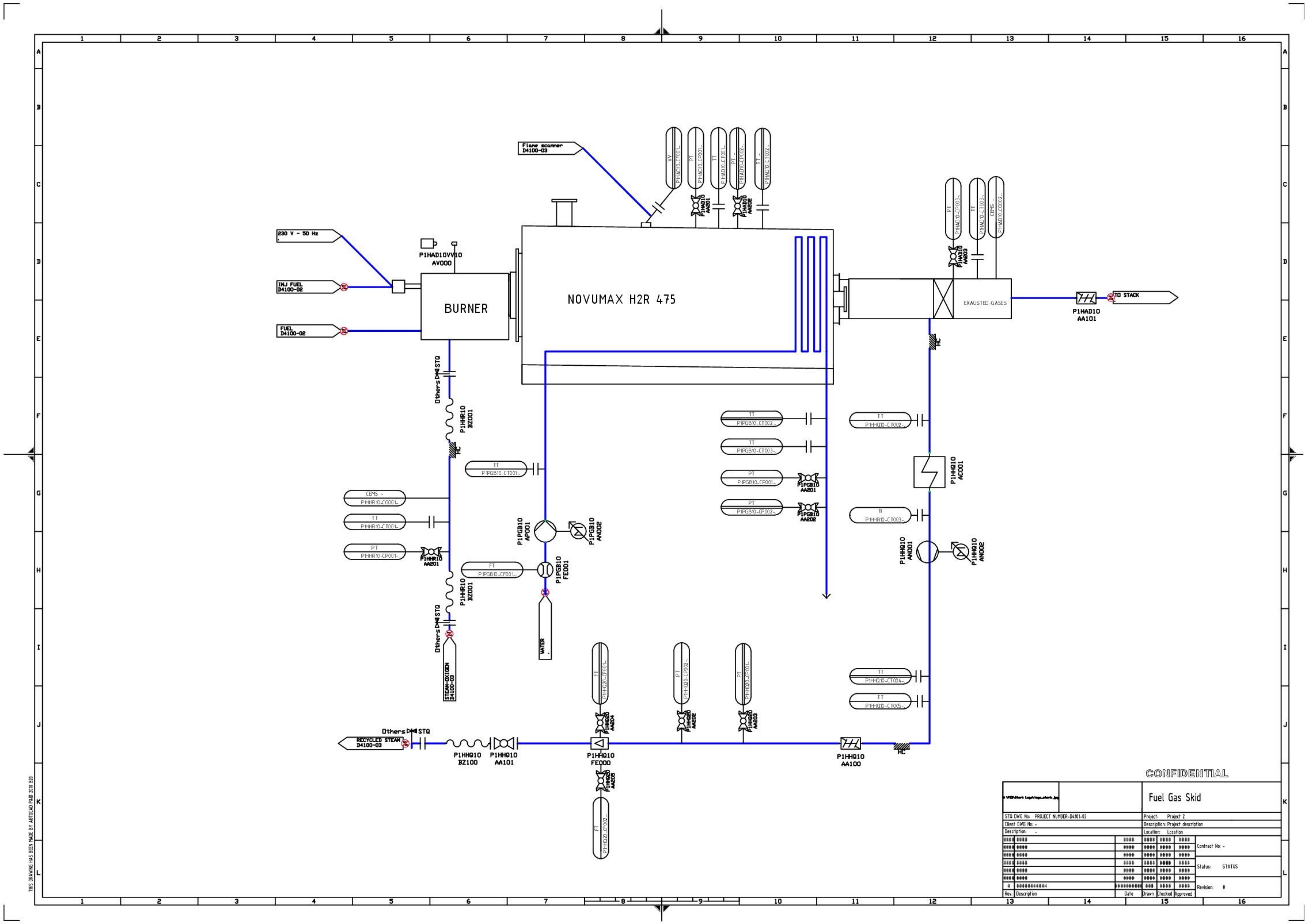
P&ID: Natural Gas - Hydrogen supply



P&ID: Air- Oxygen- Nitrogen supply



P&ID: Combustion Chamber



APPENDIX 13 : Procedure for start-up, shut down and switch the condition

System start up to condition " 25% load- combustion of hydrogen with air"

- Start conditions:

.hydrogen shut-off valves closed (P1HHG10AA301 P1HHG10AA302)
.Vent valve on hydrogen line open (P1HHG10AA200)
. Natural gas shut-off valves closed (P1HHH10AA301 P1HHH10AA302)
.Vent valve on natural gas line open (P1HHH10AA200)
. shut-off valves on ignition line closed (P1HJG10AA301 P1HJG10AA302)
.Vent valve on ignition line open (P1HJG10AA300)
. shut-off valves on nitrogen line closed (P1GJB10AA301 P1 GJB10AA302)
.Vent valve on nitrogen line open (P1GJB10AA303)
shut-off valves on oxygen line closed (P1QFB10AA303 P1QFB10AA304)
.Vent valve on oxygen line open (P1QFB10AA303)
. No flame in furnace (dark check)
.FGR damper closed.
.FD damper open
. Control valve on hydrogen valve at start position

- Switch on FD fan at >=25% maximum power for minimum 30 seconds (Volume of air > = 5 times volume of the boiler)
- Check for positive pressure in air line
- Open FGR damper
- Switch on recirculation at >= 25 % load and close stack dumper
- Check FGR working (positive pressure)
- Purge FGR duct with FGR fan >=25% with the air fan at >=25% maximum power for 30 seconds to purge the recycled stream line
- Switch off recirculation fan
- Close FGR damper and open stack dumper
- Start the spark for at least 5 seconds
- Open shut-off valves on Natural Gas ignition Line (P1HJG10-AA301 and P1HJG10-AA302) at the same time. Control valve on ignition line is auto regulated.
- Ignition of the pilot flame
- Control flame presence (5 sec)
- Stop sparking
- Open shut-off valves on Hydrogen line (P1HHG10-AA301 P1HHG10-AA302)
- Control presence of main H2 flame 5sec
- Close shut-off valve on Natural Gas ignition line (P1HJG10-AA301 and P1HJG10-AA302). Open vent valve on Natural gas ignition line P1HJG10-AA300

System start up to condition " 25% load- combustion of hydrogen with oxygen and nitrogen"

- Control safety condition:

.hydrogen shut-off valves closed (P1HHG10AA301 P1HHG10AA302)
.Vent valve on hydrogen line open (P1HHG10AA200)
. Natural gas shut-off valves closed (P1HHH10AA301 P1HHH10AA302)
.Vent valve on natural gas line open (P1HHH10AA200)

. shut-off valves on ignition line closed (P1HJG10AA301 P1HJG10AA302)
.Vent valve on ignition line open (P1HJG10AA300)
. shut-off valves on nitrogen line closed (P1GJB10AA301 P1 GJB10AA302)
.Vent valve on nitrogen line open (P1GJB10AA303)
shut-off valves on oxygen line closed (P1QFB10AA303 P1QFB10AA304)
.Vent valve on oxygen line open (P1QFB10AA303)
. No flame in furnace (dark check)
.FGR damper closed.
.FD damper open
. Control valve on hydrogen valve at start position
.control valve on oxygen line at start position
. Control valve on nitrogen line at start position
. Control valve on hydrogen valve at start position

- Switch on Air fan at maximum power for minimum 30 seconds (Volume of air > = 5 times volume of the boiler)
- Check for positive pressure in air line
- Open FGR damper
- Switch on recirculation at >= 25 % load and close stack dumper
- Check FGR working (positive pressure)
- Purge FGR duct with FGR fan >=25% with

the air fan at >=25% maximum power for 30 seconds to purge the recycled stream line

- Switch off recirculation fan
- Switch off the air fan
- Open shut-off valve on Nitrogen Line (in order P1GJB10-AA303 P1GJB10-AA301)
- Open shut-off valve on Oxygen Line (P1QFB10-AA304 P1QFB10-AA302)
- Create air mixture (control/safeguard mix O2) 20<O2<20.9 vol%.
- Close FGR damper and open stack dumper
- Start the spark for at least 5 seconds

Open shut-off valves on Natural Gas ignition Line (P1HJG10-AA301 and P1HJG10-AA302) at the same time. Control valve on ignition line is auto regulated.

- Ignition of the pilot flame
- Control flame presence (5 sec)
- Stop sparking
- Open shut-off valves on Hydrogen line (P1HHG10-AA301 P1HHG10-AA302)
- Control presence of main H2 flame 5sec
- Close shut-off valve on Natural Gas ignition line (P1HJG10-AA301 and P1HJG10-AA302).
- Open vent valve on Natural gas ignition line P1HJG10-AA300

Switching from condition “ 10% load- combustion of hydrogen with air” to condition “10%load – Combustion of hydrogen with Oxygen and recirculation stream”

- O2 valve at closed position.
- Switch on recirculation fan at minimum load
- Open shut off valves on oxygen line(P1QFB10-AA304 P1QFB10-AA302)
- Increase O2 flow (slowly close control valve P1QFB10-AA401)
- Decrease air fan load with
- Wait for stable condition

- Increase Oxygen flow rate (slowly open control valve P1QFB10-AA401) while

Increasing the load of the recirculation fan and decreasing air fan load

- Wait for stable condition
- Keep repeating the same procedure until it reaches a stable condition with air fan at minimum
- Switch off the air fan
- Close SSO's at air connection
- Release burner for control.
- Increase oxygen flow rate and recirculation fan load to reach stable condition

Switching from condition " 10% load- combustion of hydrogen with nitrogen and oxygen" to condition "10%load – Combustion of hydrogen with Oxygen and recirculation stream"

- switch on recirculation fan
- Decrease nitrogen flow (slowly close control valve P1GJB10-AA401)
- Increase recirculation fan load
- Check for stable condition
- Repeat the same procedure until the recirculation fan load is at the designed value for 10% load
- Close control valve on nitrogen line (P1GJB10-AA401) and Open vent valve on nitrogen line (P1GJB10-AA302) and close shut-off valves on nitrogen line (P1GJB10-AA301 P1GJB10-AA303)

Switching from condition "10%load – Combustion of hydrogen with Oxygen" to "100%load – Combustion of hydrogen with Oxygen"

- increase recirculation fan load,
- increase oxygen flow rate
- increase hydrogen flow rate
- check for stable condition
- repeat the same procedure until it reaches 100% load stable condition

Switching from condition "100%load – Combustion of hydrogen with Oxygen" to "10%load – Combustion of hydrogen with Oxygen"

- decrease hydrogen flow rate
- decrease oxygen flow rate
- decrease recirculation fan load
- check for stable condition
- repeat the same procedure until it reaches 10% load stable condition

Switching from condition "10%load – Combustion of hydrogen with Oxygen" to condition "10%load – Combustion of hydrogen with air"

- Switch on air fan
- decrease Oxygen flow rate (slowly close control valve P1QFB10-AA401)
- decrease the load of the recirculation fan
- Check for stable condition
- Increase air fan load till it reaches 100% load
- decrease Oxygen flow rate (slowly close control valve P1QFB10-AA401)
- decrease the load of the recirculation fan until it reaches the minimum load
- Check for stable condition

- Repeat the same procedure until it reaches a stable 100%load – Combustion of hydrogen with air stable condition
- Switch off recirculation fan
- close shut-off valve on Oxygen Line (P1QFB10-AA302 P1QFB10-AA304) and open vent valve on oxygen line P1QFB10-AA303

Emergency shut off of the system from condition "x% load – Combustion of hydrogen with Oxygen

- Close shut-off valves on hydrogen line (P1HHG10-AA301 P1HHG10-AA302) and Open purge valve on Hydrogen line (P1HHG10-AA300)
- Reduce FGR flow to min
- Start FD fan
- Slowly Close control valve on oxygen line (P1QFB10-AA401)
- Open purge valve on Oxygen line (P1QFB10-AA303) and Close shut-off valves on oxygen line (P1QFB10-AA302 P1QFB10-AA304)
- Switch off recirculation fan
- Close FGR damper
- Stop FD fan after 2 min.

Emergency Shut off of the system from condition "x%load – Combustion of hydrogen with air"

- Close shut-off valves on hydrogen line (P1HHG10-AA301 P1HHG10-AA302) and Open purge valve on Hydrogen line (P1HHG10-AA300)
- Decrease air fan load
- Stop fan after 2 min

Emergency Shut off of the system from condition "x%load – Combustion of hydrogen with Nitrogen and Oxygen"

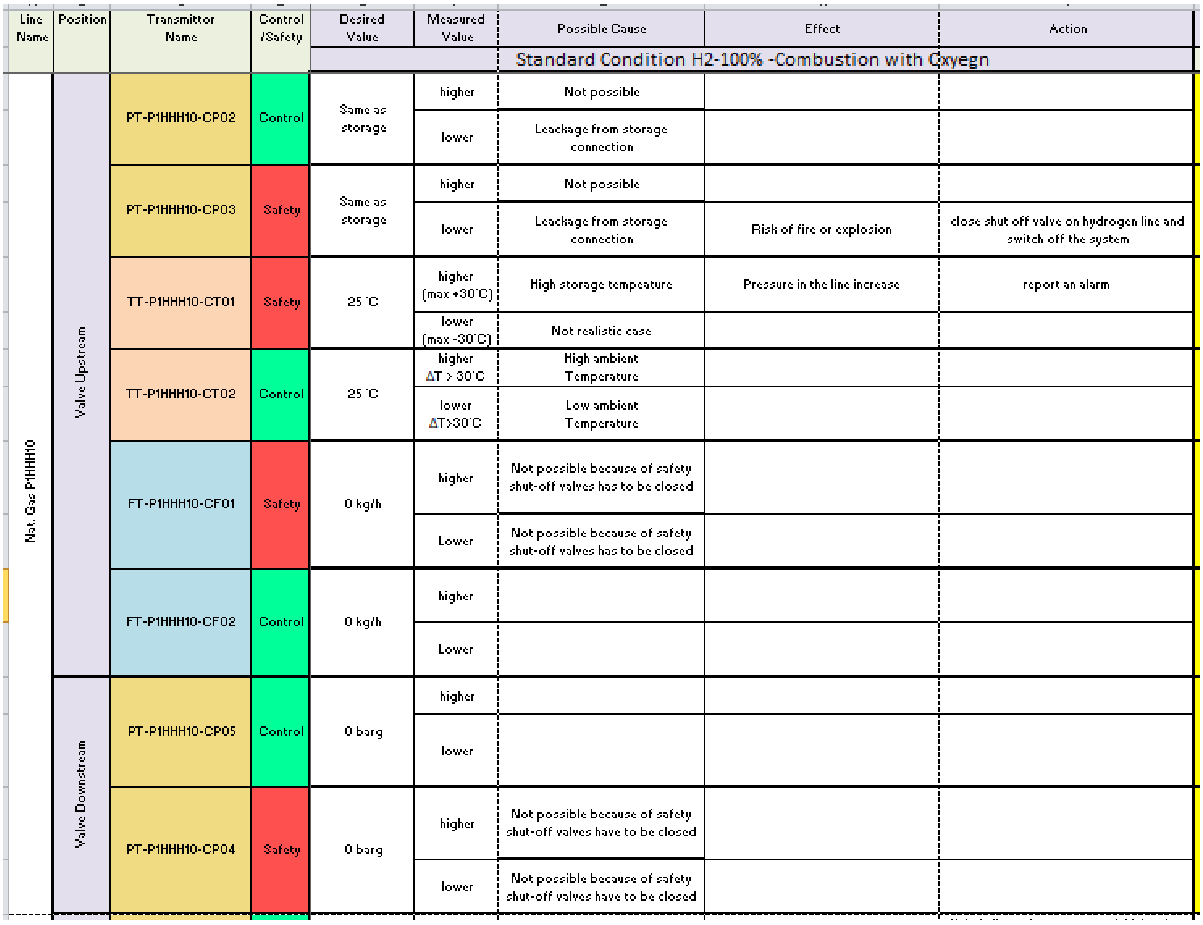
- Close shut-off valves on hydrogen line (P1HHG10-AA301 P1HHG10-AA302) and Open purge valve on Hydrogen line (P1HHG10-AA300)
- Close control valve on hydrogen line (P1HHG10-AA401)
- Start FD fan
- Decrease oxygen flow rate to minimum (close control valve P1QFB10-AA401)
- Decrease Nitrogen flow rate to minimum (close control valve P1GJB10-AA401)
- Close shut-off valves on oxygen line (P1QFB10-AA302 P1QFB10-AA304) and Open purge valve on Oxygen line (P1QFB10-AA303)
- Open purge valve on nitrogen line (P1GJB10-AA302) and Close shut-off valves on nitrogen line (P1GJB10-AA301 P1GJB10-AA303)
- Stop FD fan after 2 min.

APPENDIX 14: HAZOP

The following tables shows the results of the HAZOP analysis which has been carried out during the internship.

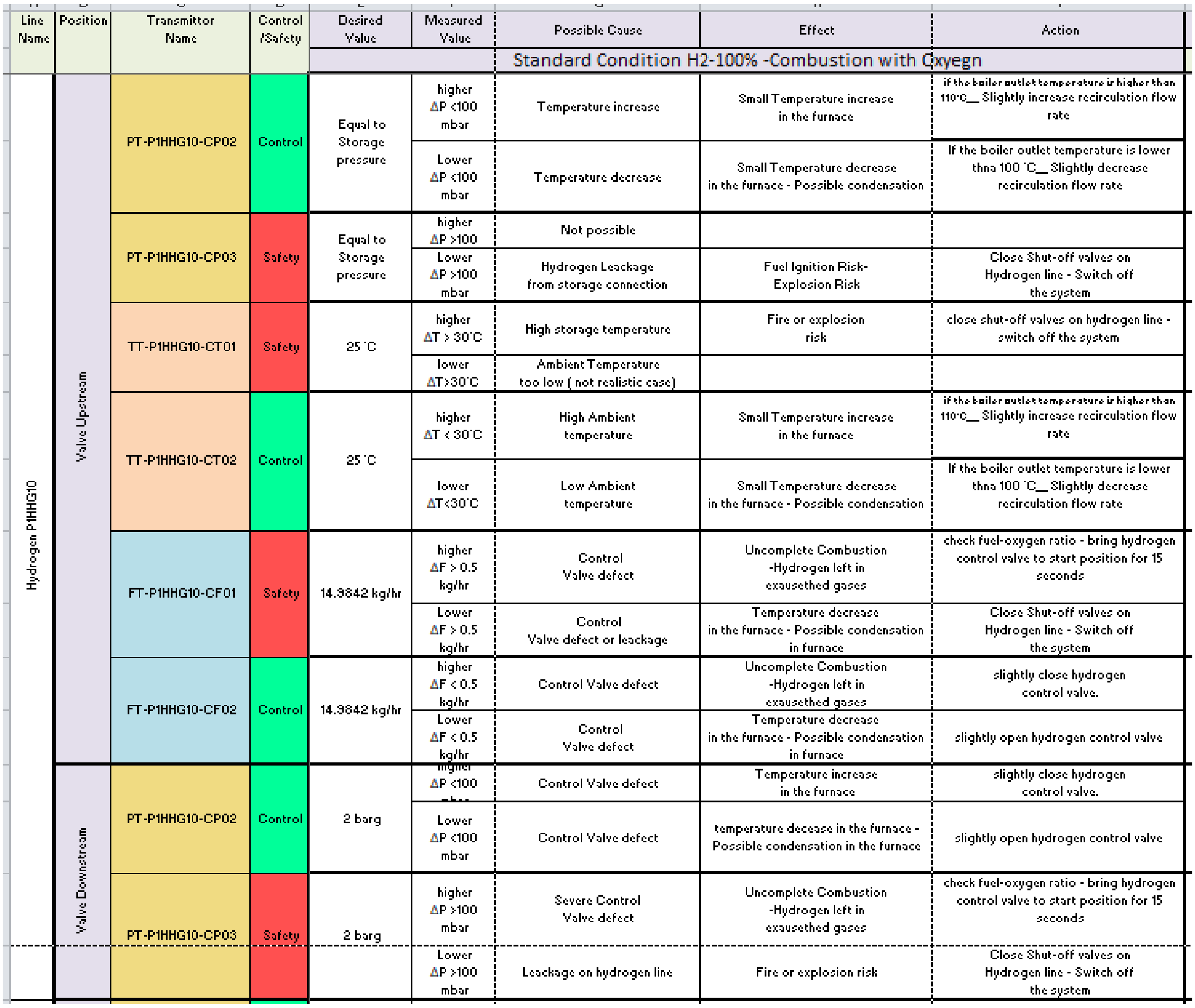
For every standard condition all the transmittors of all the lines has been analyzed, but here it is reported only the transmitters which are on the lines which are involved in the operation in the relative standard condition.

The first table shows the results for the standard condition " 100% load combustion of hydrogen with oxygen and recycled stream".Hence for this case there are reported only the transmitters on the lines of hydrogen, oxygen, static mixer downstream, boiler, stack, cooling water and recirculation line. There won't be reported the lines of nitrogen, natural gas and air because those are not involved in the operation of the system. However different problems can stillbe detected in those lines even during the operation in a standard condiotn in which they are not involved. For example the Natural Gas line is supposed to be at the same pressure of the storage (before the shut-off valves) which means that if there is a leackage in the line, the transmitter would detect a lower value than expected and then it will report an alarm to warn about the risk of a leakage of fuel which could lead to fire or risk explosion. This first table shows exactly this case. However this case has to be taken as reference for all the other lines which won't be used during the several standard operation.



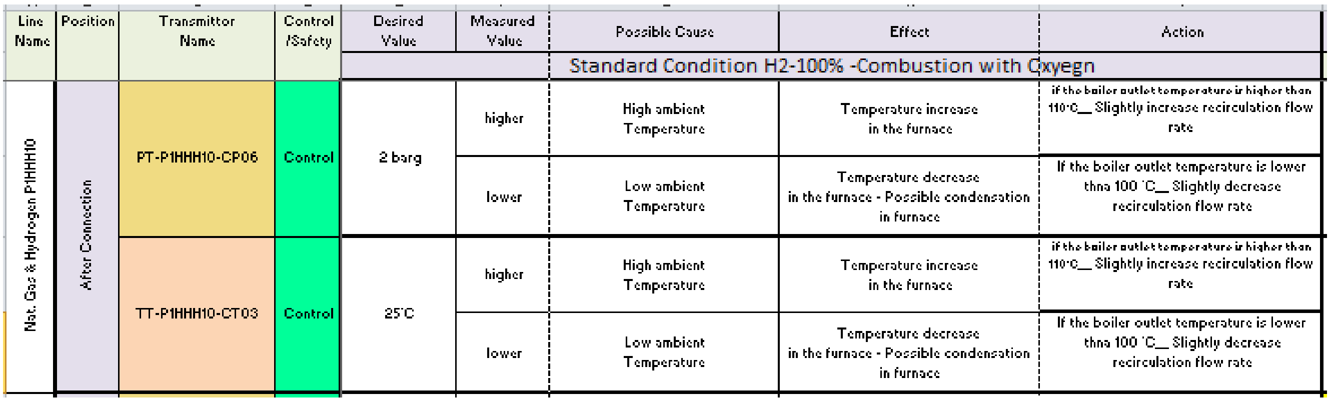
Line Name	Position	Transmittor Name	Control /Safety	Desired Value	Measured Value	Possible Cause	Effect	Action
				Standard Condition H2-100% -Combustion with Cxyegn				
Nat. Gas P1HHH10	Valve Upstream	PT-P1HHH10-CP02	Control	Same as storage	higher	Not possible		
					lower	Leackage from storage connection		
		PT-P1HHH10-CP03	Safety	Same as storage	higher	Not possible		
					lower	Leackage from storage connection	Risk of fire or explosion	close shut off valve on hydrogen line and switch off the system
		TT-P1HHH10-CT01	Safety	25 °C	higher (max +30°C)	High storage tempeature	Pressure in the line increase	report an alarm
					lower (max -30°C)	Not realistic case		
		TT-P1HHH10-CT02	Control	25 °C	higher ΔT > 30°C	High ambient Temperature		
					lower ΔT>30°C	Low ambient Temperature		
		FT-P1HHH10-CF01	Safety	0 kg/h	higher	Not possible because of safety shut-off valves has to be closed		
					Lower	Not possible because of safety shut-off valves has to be closed		
		FT-P1HHH10-CF02	Control	0 kg/h	higher			
					Lower			
	Valve Downstream	PT-P1HHH10-CP05	Control	0 barg	higher			
					lower			
		PT-P1HHH10-CP04	Safety	0 barg	higher	Not possible because of safety shut-off valves have to be closed		
					lower	Not possible because of safety shut-off valves have to be closed		

Fig. 60: Reference case _ HAZOP_ Natural gas line during standard operation at 100% load combustion of hydrogen with oxygen and recycled stream



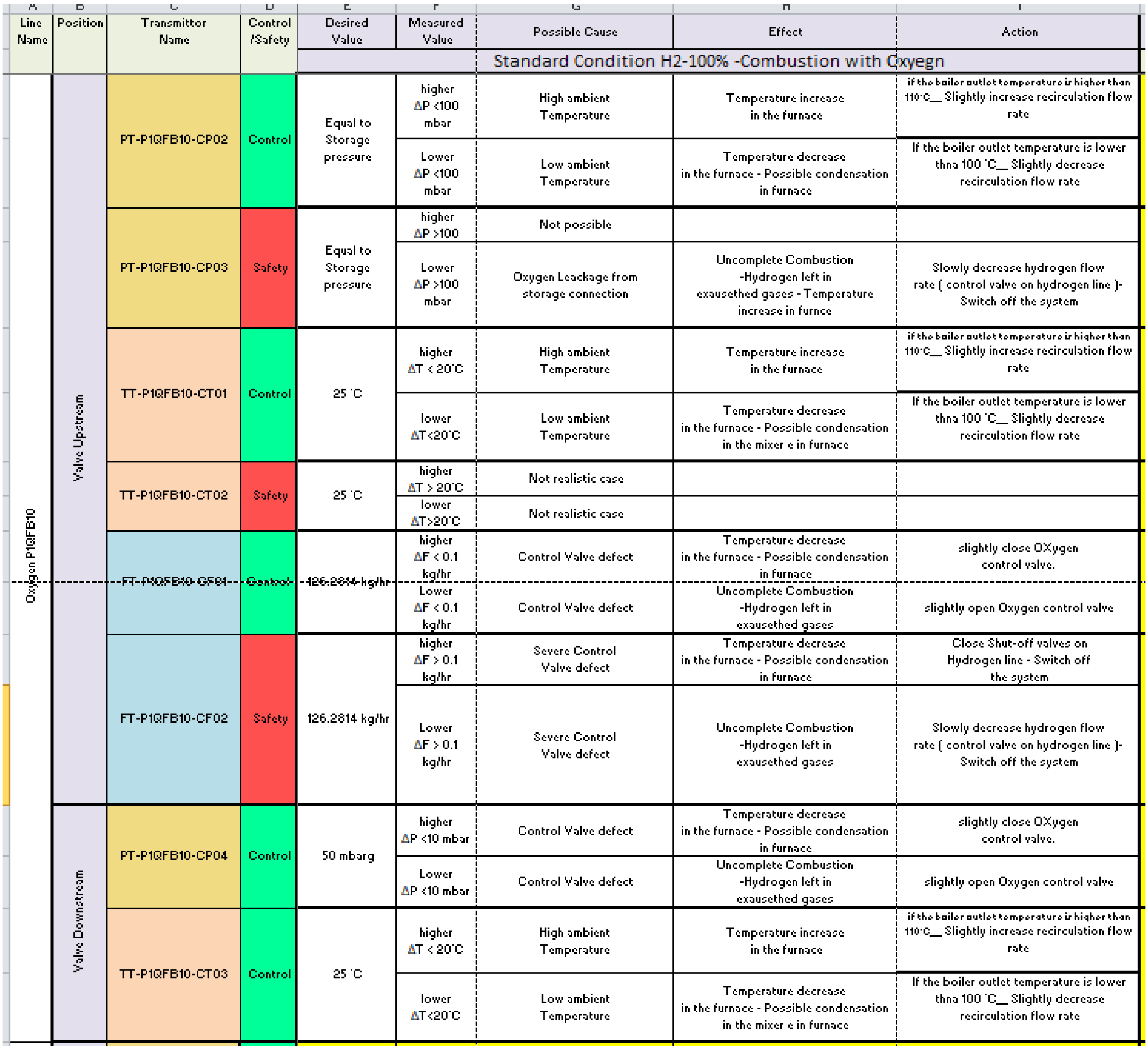
Line Name	Position	Transmittor Name	Control /Safety	Desired Value	Measured Value	Possible Cause	Effect	Action
						Standard Condition H2-100% -Combustion with Oxyegn		
Hydrogen P1HHG10	Valve Upstream	PT-P1HHG10-CP02	Control	Equal to Storage pressure	higher ΔP <100 mbar	Temperature increase	Small Temperature increase in the furnace	if the boiler outlet temperature is higher than 110°C__ Slightly increase recirculation flow rate
					Lower ΔP <100 mbar	Temperature decrease	Small Temperature decrease in the furnace - Possible condensation	If the boiler outlet temperature is lower thna 100 °C__ Slightly decrease recirculation flow rate
		PT-P1HHG10-CP03	Safety	Equal to Storage pressure	higher ΔP >100	Not possible		
					Lower ΔP >100 mbar	Hydrogen Leackage from storage connection	Fuel Ignition Risk- Explosion Risk	Close Shut-off valves on Hydrogen line - Switch off the system
		TT-P1HHG10-CT01	Safety	25 °C	higher ΔT > 30°C	High storage temperature	Fire or explosion risk	close shut-off valves on hydrogen line - switch off the system
					lower ΔT>30°C	Ambient Temperature too low (not realistic case)		
		TT-P1HHG10-CT02	Control	25 °C	higher ΔT < 30°C	High Ambient temperature	Small Temperature increase in the furnace	if the boiler outlet temperature is higher than 110°C__ Slightly increase recirculation flow rate
					lower ΔT<30°C	Low Ambient temperature	Small Temperature decrease in the furnace - Possible condensation	If the boiler outlet temperature is lower thna 100 °C__ Slightly decrease recirculation flow rate
		FT-P1HHG10-CF01	Safety	14.9842 kg/hr	higher ΔF > 0.5 kg/hr	Control Valve defect	Uncomplete Combustion -Hydrogen left in exausethed gases	check fuel-oxygen ratio - bring hydrogen control valve to start position for 15 seconds
					Lower ΔF > 0.5 kg/hr	Control Valve defect or leackage	Temperature decrease in the furnace - Possible condensation in furnace	Close Shut-off valves on Hydrogen line - Switch off the system
		FT-P1HHG10-CF02	Control	14.9842 kg/hr	higher ΔF < 0.5 kg/hr	Control Valve defect	Uncomplete Combustion -Hydrogen left in exausethed gases	slightly close hydrogen control valve.
					Lower ΔF < 0.5 kg/hr	Control Valve defect	Temperature decrease in the furnace - Possible condensation in furnace	slightly open hydrogen control valve
	Valve Downstream	PT-P1HHG10-CP02	Control	2 barg	higher ΔP <100 mbar	Control Valve defect	Temperature increase in the furnace	slightly close hydrogen control valve.
					Lower ΔP <100 mbar	Control Valve defect	temperature decease in the furnace - Possible condensation in the furnace	slightly open hydrogen control valve
		PT-P1HHG10-CP03	Safety	2 barg	higher ΔP >100 mbar	Severe Control Valve defect	Uncomplete Combustion -Hydrogen left in exausethed gases	check fuel-oxygen ratio - bring hydrogen control valve to start position for 15 seconds
					Lower ΔP >100 mbar	Leackage on hydrogen line	Fire or explosion risk	Close Shut-off valves on Hydrogen line - Switch off the system

Fig. 61: HAZOP - Hydrogen line - case: " 100% load combustion of hydrogen with oxygen and recycled stream"



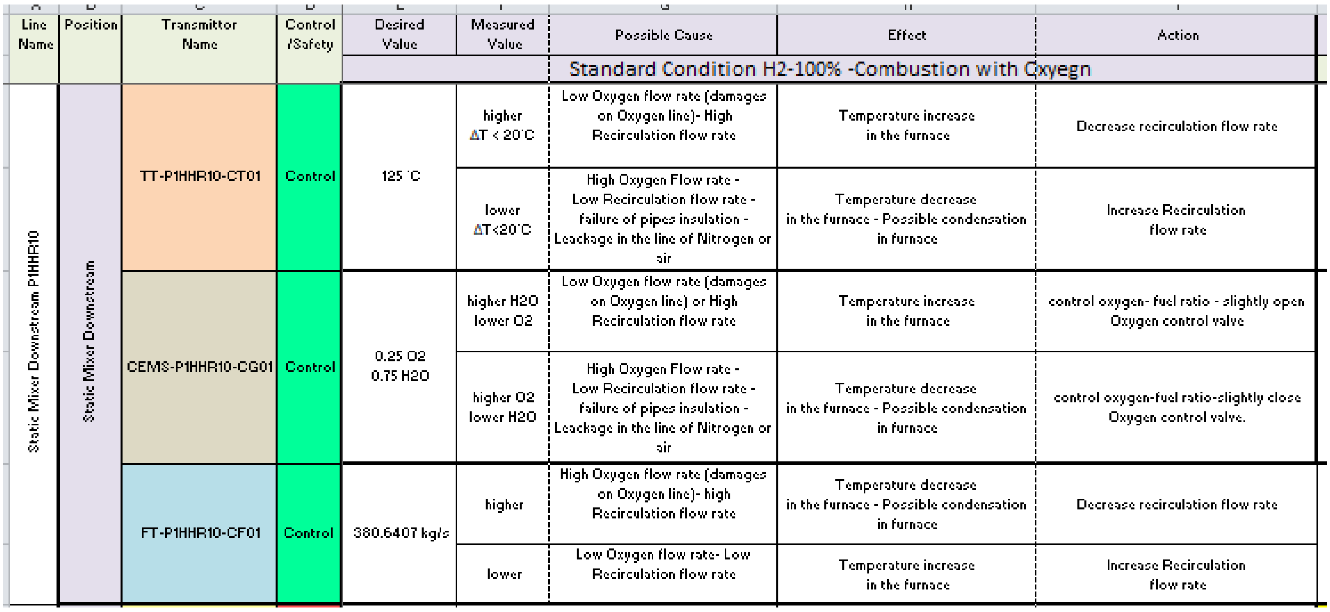
Line Name	Position	Transmitter Name	Control /Safety	Desired Value	Measured Value	Possible Cause	Effect	Action
						Standard Condition H2-100% -Combustion with Oxyegn		
Nat. Gas & Hydrogen P1HHH10	After Connection	PT-P1HHH10-CP06	Control	2 barg	higher	High ambient Temperature	Temperature increase in the furnace	if the boiler outlet temperature is higher than 110°C__ Slightly increase recirculation flow rate
					lower	Low ambient Temperature	Temperature decrease in the furnace - Possible condensation in furnace	If the boiler outlet temperature is lower thna 100 °C__ Slightly decrease recirculation flow rate
		TT-P1HHH10-CT03	Control	25°C	higher	High ambient Temperature	Temperature increase in the furnace	if the boiler outlet temperature is higher than 110°C__ Slightly increase recirculation flow rate
					lower	Low ambient Temperature	Temperature decrease in the furnace - Possible condensation in furnace	If the boiler outlet temperature is lower thna 100 °C__ Slightly decrease recirculation flow rate

Fig. 62: HAZOP - Hydrogen & Natural Gas line - case: " 100% load combustion of hydrogen with oxygen and recycled stream"



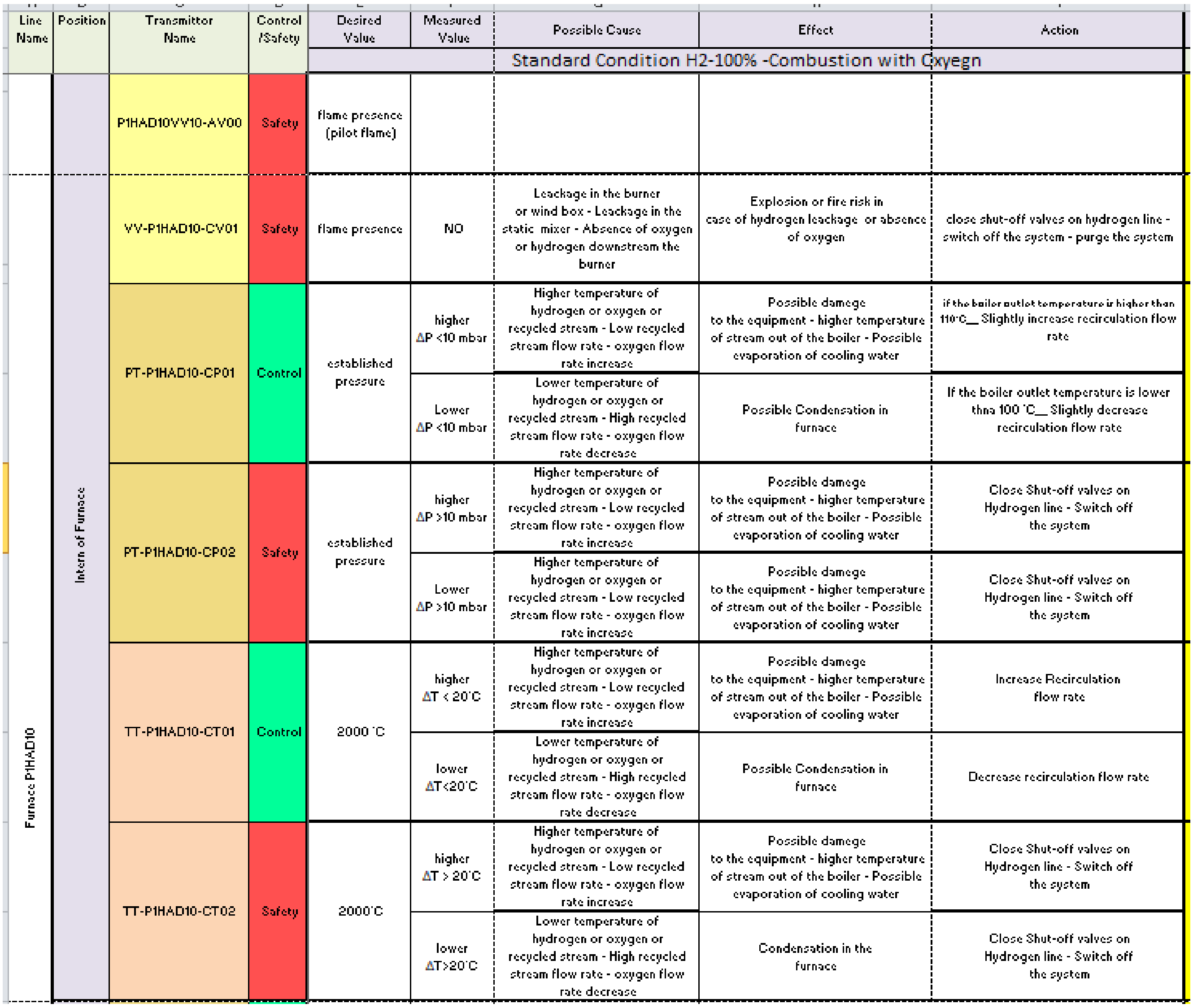
A	B	C	D	E	F	G	H	I
Line Name	Position	Transmittor Name	Control /Safety	Desired Value	Measured Value	Possible Cause	Effect	Action
						Standard Condition H2-100% -Combustion with Oxyegn		
Oxygen P1QFB10	Valve Upstream	PT-P1QFB10-CP02	Control	Equal to Storage pressure	higher $\Delta P < 100$ mbar	High ambient Temperature	Temperature increase in the furnace	if the boiler outlet temperature is higher than 110°C__ Slightly increase recirculation flow rate
					Lower $\Delta P < 100$ mbar	Low ambient Temperature	Temperature decrease in the furnace - Possible condensation in furnace	If the boiler outlet temperature is lower thna 100 °C__ Slightly decrease recirculation flow rate
		PT-P1QFB10-CP03	Safety	Equal to Storage pressure	higher $\Delta P > 100$	Not possible		
					Lower $\Delta P > 100$ mbar	Oxygen Leackage from storage connection	Uncomplete Combustion -Hydrogen left in exausethed gases - Temperature increase in furnce	Slowly decrease hydrogen flow rate (control valve on hydrogen line)- Switch off the system
		TT-P1QFB10-CT01	Control	25 °C	higher $\Delta T < 20°C$	High ambient Temperature	Temperature increase in the furnace	if the boiler outlet temperature is higher than 110°C__ Slightly increase recirculation flow rate
					lower $\Delta T < 20°C$	Low ambient Temperature	Temperature decrease in the furnace - Possible condensation in the mixer e in furnace	If the boiler outlet temperature is lower thna 100 °C__ Slightly decrease recirculation flow rate
		TT-P1QFB10-CT02	Safety	25 °C	higher $\Delta T > 20°C$	Not realistic case		
					lower $\Delta T > 20°C$	Not realistic case		
		~~FT-P1QFB10-CF01~~	~~Control~~	~~126.2814 kg/hr~~	higher $\Delta F < 0.1$ kg/hr	Control Valve defect	Temperature decrease in the furnace - Possible condensation in furnace	slightly close OXygen control valve.
					Lower $\Delta F < 0.1$ kg/hr	Control Valve defect	Uncomplete Combustion -Hydrogen left in exausethed gases	slightly open Oxygen control valve
		FT-P1QFB10-CF02	Safety	126.2814 kg/hr	higher $\Delta F > 0.1$ kg/hr	Severe Control Valve defect	Temperature decrease in the furnace - Possible condensation in furnace	Close Shut-off valves on Hydrogen line - Switch off the system
					Lower $\Delta F > 0.1$ kg/hr	Severe Control Valve defect	Uncomplete Combustion -Hydrogen left in exausethed gases	Slowly decrease hydrogen flow rate (control valve on hydrogen line)- Switch off the system
	Valve Downstream	PT-P1QFB10-CP04	Control	50 mbarg	higher $\Delta P < 10$ mbar	Control Valve defect	Temperature decrease in the furnace - Possible condensation in furnace	slightly close OXygen control valve.
					Lower $\Delta P < 10$ mbar	Control Valve defect	Uncomplete Combustion -Hydrogen left in exausethed gases	slightly open Oxygen control valve
		TT-P1QFB10-CT03	Control	25 °C	higher $\Delta T < 20°C$	High ambient Temperature	Temperature increase in the furnace	if the boiler outlet temperature is higher than 110°C__ Slightly increase recirculation flow rate
					lower $\Delta T < 20°C$	Low ambient Temperature	Temperature decrease in the furnace - Possible condensation in the mixer e in furnace	If the boiler outlet temperature is lower thna 100 °C__ Slightly decrease recirculation flow rate

Fig. 63: HAZOP – Oxygen line - case: " 100% load combustion of hydrogen with oxygen and recycled stream"



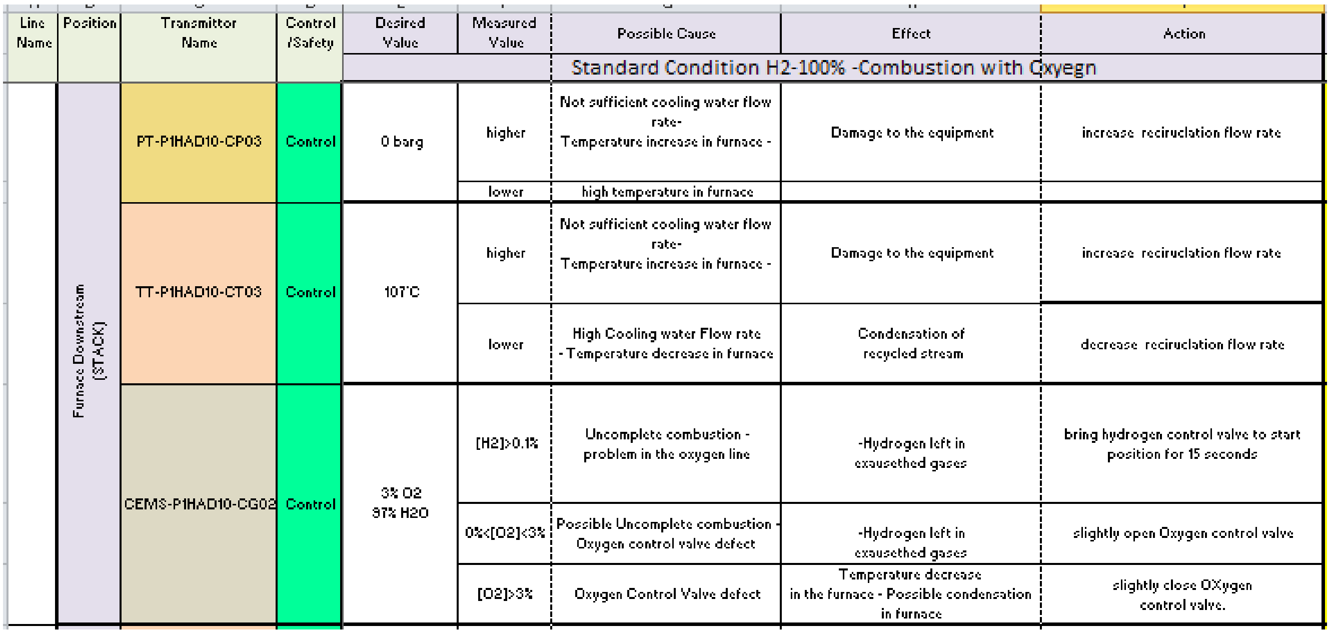
Line Name	Position	Transmittor Name	Control /Safety	Desired Value	Measured Value	Possible Cause	Effect	Action
						Standard Condition H2-100% -Combustion with Oxyegn		
Static Mixer Downstream P1HHR10	Static Mixer Downstream	TT-P1HHR10-CT01	Control	125 °C	higher $\Delta T < 20^{\circ}C$	Low Oxygen flow rate (damages on Oxygen line)- High Recirculation flow rate	Temperature increase in the furnace	Decrease recirculation flow rate
					lower $\Delta T < 20^{\circ}C$	High Oxygen Flow rate - Low Recirculation flow rate - failure of pipes insulation - Leackage in the line of Nitrogen or air	Temperature decrease in the furnace - Possible condensation in furnace	Increase Recirculation flow rate
		CEMS-P1HHR10-CG01	Control	0.25 O2 0.75 H2O	higher H2O lower O2	Low Oxygen flow rate (damages on Oxygen line) or High Recirculation flow rate	Temperature increase in the furnace	control oxygen- fuel ratio - slightly open Oxygen control valve
					higher O2 lower H2O	High Oxygen Flow rate - Low Recirculation flow rate - failure of pipes insulation - Leackage in the line of Nitrogen or air	Temperature decrease in the furnace - Possible condensation in furnace	control oxygen-fuel ratio-slightly close Oxygen control valve.
		FT-P1HHR10-CF01	Control	380.6407 kg/s	higher	High Oxygen flow rate (damages on Oxygen line)- high Recirculation flow rate	Temperature decrease in the furnace - Possible condensation in furnace	Decrease recirculation flow rate
					lower	Low Oxygen flow rate- Low Recirculation flow rate	Temperature increase in the furnace	Increase Recirculation flow rate

Fig. 64: HAZOP – Static mixer downstream - case: " 100% load combustion of hydrogen with oxygen and recycled stream"



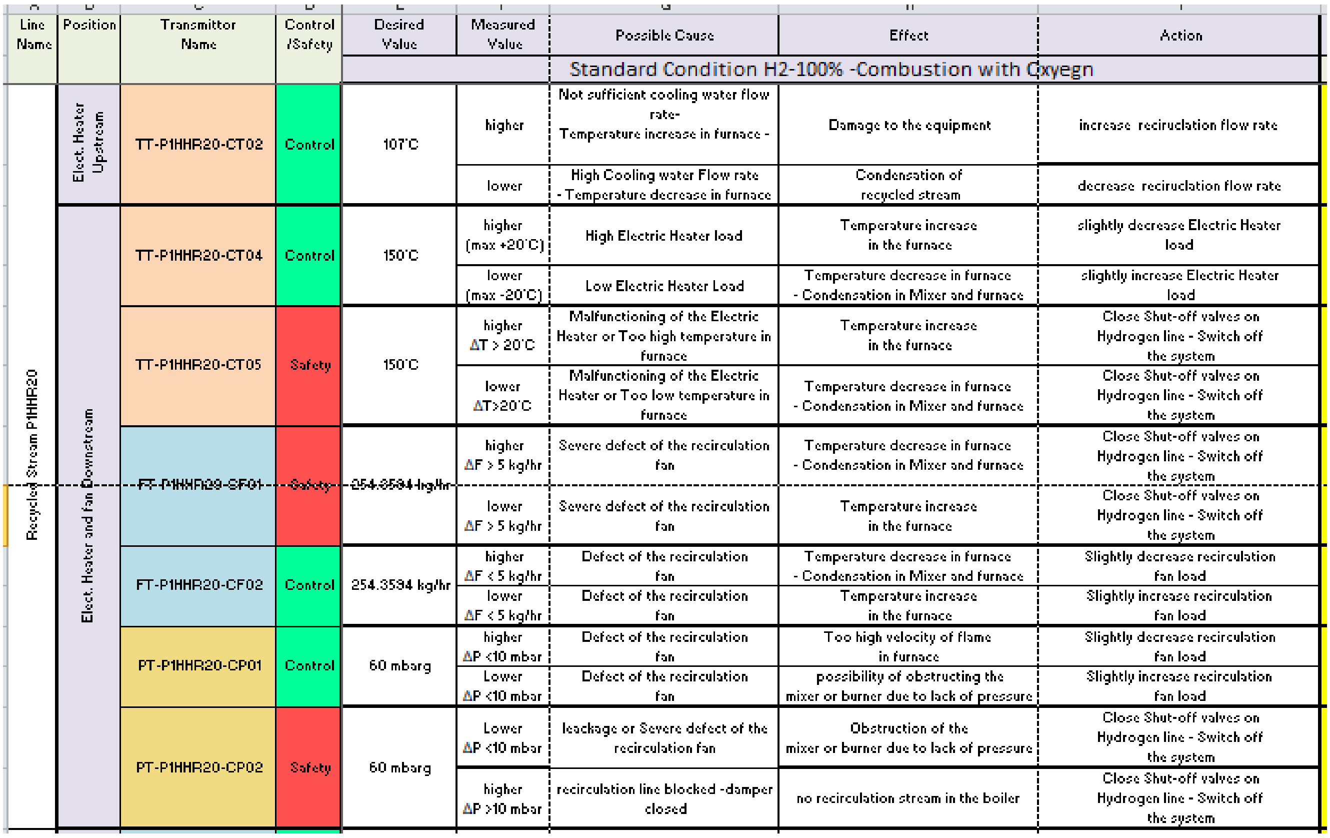
Line Name	Position	Transmittor Name	Control /Safety	Desired Value	Measured Value	Possible Cause	Effect	Action
						Standard Condition H2-100% -Combustion with Oxyegn		
		P1HAD10VV10-AV00	Safety	flame presence (pilot flame)				
Furnace P1HAD10	Intern of Furnace	VV-P1HAD10-CV01	Safety	flame presence	NO	Leackage in the burner or wind box - Leackage in the static mixer - Absence of oxygen or hydrogen downstream the burner	Explosion or fire risk in case of hydrogen leackage or absence of oxygen	close shut-off valves on hydrogen line - switch off the system - purge the system
		PT-P1HAD10-CP01	Control	established pressure	higher $\Delta P < 10$ mbar	Higher temperature of hydrogen or oxygen or recycled stream - Low recycled stream flow rate - oxygen flow rate increase	Possible damege to the equipment - higher temperature of stream out of the boiler - Possible evaporation of cooling water	if the boiler outlet temperature is higher than 110°C__ Slightly increase recirculation flow rate
					Lower $\Delta P < 10$ mbar	Lower temperature of hydrogen or oxygen or recycled stream - High recycled stream flow rate - oxygen flow rate decrease	Possible Condensation in furnace	If the boiler outlet temperature is lower thna 100 °C__ Slightly decrease recirculation flow rate
		PT-P1HAD10-CP02	Safety	established pressure	higher $\Delta P > 10$ mbar	Higher temperature of hydrogen or oxygen or recycled stream - Low recycled stream flow rate - oxygen flow rate increase	Possible damege to the equipment - higher temperature of stream out of the boiler - Possible evaporation of cooling water	Close Shut-off valves on Hydrogen line - Switch off the system
					Lower $\Delta P > 10$ mbar	Higher temperature of hydrogen or oxygen or recycled stream - Low recycled stream flow rate - oxygen flow rate increase	Possible damege to the equipment - higher temperature of stream out of the boiler - Possible evaporation of cooling water	Close Shut-off valves on Hydrogen line - Switch off the system
		TT-P1HAD10-CT01	Control	2000 °C	higher $\Delta T < 20^{\circ}C$	Higher temperature of hydrogen or oxygen or recycled stream - Low recycled stream flow rate - oxygen flow rate increase	Possible damege to the equipment - higher temperature of stream out of the boiler - Possible evaporation of cooling water	Increase Recirculation flow rate
					lower $\Delta T < 20^{\circ}C$	Lower temperature of hydrogen or oxygen or recycled stream - High recycled stream flow rate - oxygen flow rate decrease	Possible Condensation in furnace	Decrease recirculation flow rate
		TT-P1HAD10-CT02	Safety	2000°C	higher $\Delta T > 20^{\circ}C$	Higher temperature of hydrogen or oxygen or recycled stream - Low recycled stream flow rate - oxygen flow rate increase	Possible damege to the equipment - higher temperature of stream out of the boiler - Possible evaporation of cooling water	Close Shut-off valves on Hydrogen line - Switch off the system
					lower $\Delta T > 20^{\circ}C$	Lower temperature of hydrogen or oxygen or recycled stream - High recycled stream flow rate - oxygen flow rate decrease	Condensation in the furnace	Close Shut-off valves on Hydrogen line - Switch off the system

Fig. 65 : HAZOP – intern of the furnace - case: " 100% load combustion of hydrogen with oxygen and recycled stream"



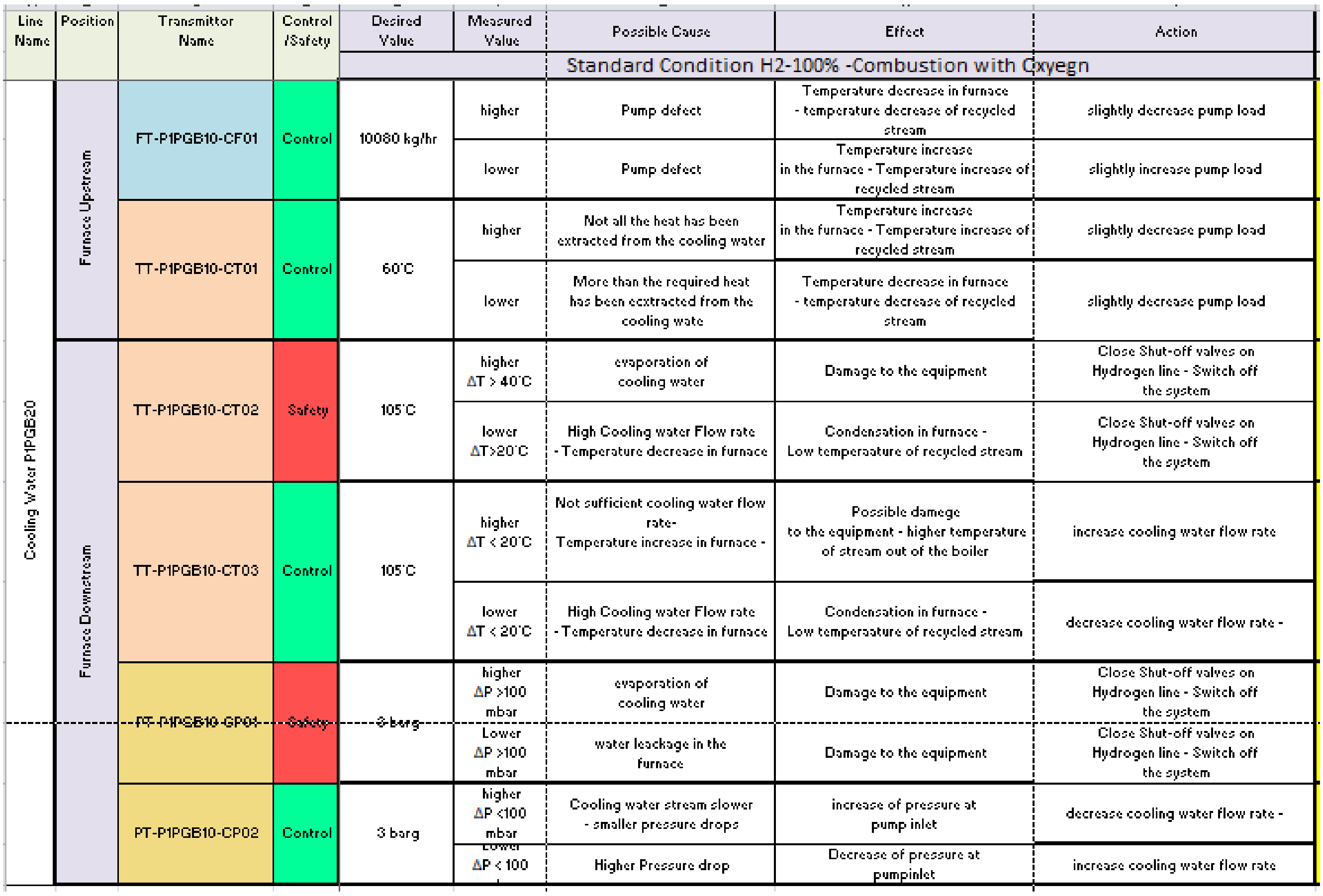
Line Name	Position	Transmittor Name	Control /Safety	Desired Value	Measured Value	Possible Cause	Effect	Action
						Standard Condition H2-100% -Combustion with Oxyegn		
	Furnace Downstream (STACK)	PT-P1HAD10-CP03	Control	0 barg	higher	Not sufficient cooling water flow rate- Temperature increase in furnace -	Damage to the equipment	increase reciruclation flow rate
					lower	high temperature in furnace		
		TT-P1HAD10-CT03	Control	107°C	higher	Not sufficient cooling water flow rate- Temperature increase in furnace -	Damage to the equipment	increase reciruclation flow rate
					lower	High Cooling water Flow rate - Temperature decrease in furnace	Condensation of recycled stream	decrease reciruclation flow rate
		CEMS-P1HAD10-CG02	Control	3% O2 97% H2O	[H2]>0.1%	Uncomplete combustion - problem in the oxygen line	-Hydrogen left in exausethed gases	bring hydrogen control valve to start position for 15 seconds
					0%<[O2]<3%	Possible Uncomplete combustion - Oxygen control valve defect	-Hydrogen left in exausethed gases	slightly open Oxygen control valve
					[O2]>3%	Oxygen Control Valve defect	Temperature decrease in the furnace - Possible condensation in furnace	slightly close OXygen control valve.

Fig. 66: HAZOP – Boiler downstream - case: " 100% load combustion of hydrogen with oxygen and recycled stream"



Line Name	Position	Transmittor Name	Control /Safety	Desired Value	Measured Value	Possible Cause	Effect	Action
						Standard Condition H2-100% -Combustion with Oxyegn		
Recycled Stream P1HHR20	Elect. Heater Upstream	TT-P1HHR20-CT02	Control	107°C	higher	Not sufficient cooling water flow rate- Temperature increase in furnace -	Damage to the equipment	increase reciruclation flow rate
					lower	High Cooling water Flow rate - Temperature decrease in furnace	Condensation of recycled stream	decrease reciruclation flow rate
	Elect. Heater and fan Downstream	TT-P1HHR20-CT04	Control	150°C	higher (max +20°C)	High Electric Heater load	Temperature increase in the furnace	slightly decrease Electric Heater load
					lower (max -20°C)	Low Electric Heater Load	Temperature decrease in furnace - Condensation in Mixer and furnace	slightly increase Electric Heater load
		TT-P1HHR20-CT05	Safety	150°C	higher ΔT > 20°C	Malfunctioning of the Electric Heater or Too high temperature in furnace	Temperature increase in the furnace	Close Shut-off valves on Hydrogen line - Switch off the system
					lower ΔT>20°C	Malfunctioning of the Electric Heater or Too low temperature in furnace	Temperature decrease in furnace - Condensation in Mixer and furnace	Close Shut-off valves on Hydrogen line - Switch off the system
		FT-P1HHR20-CF01	Safety	254.3594 kg/hr	higher ΔF > 5 kg/hr	Severe defect of the recirculation fan	Temperature decrease in furnace - Condensation in Mixer and furnace	Close Shut-off valves on Hydrogen line - Switch off the system
					lower ΔF > 5 kg/hr	Severe defect of the recirculation fan	Temperature increase in the furnace	Close Shut-off valves on Hydrogen line - Switch off the system
		FT-P1HHR20-CF02	Control	254.3594 kg/hr	higher ΔF < 5 kg/hr	Defect of the recirculation fan	Temperature decrease in furnace - Condensation in Mixer and furnace	Slightly decrease recirculation fan load
					lower ΔF < 5 kg/hr	Defect of the recirculation fan	Temperature increase in the furnace	Slightly increase recirculation fan load
		PT-P1HHR20-CP01	Control	60 mbarg	higher ΔP <10 mbar	Defect of the recirculation fan	Too high velocity of flame in furnace	Slightly decrease recirculation fan load
					Lower ΔP <10 mbar	Defect of the recirculation fan	possibility of obstructing the mixer or burner due to lack of pressure	Slightly increase recirculation fan load
		PT-P1HHR20-CP02	Safety	60 mbarg	Lower ΔP <10 mbar	leackage or Severe defect of the recirculation fan	Obstruction of the mixer or burner due to lack of pressure	Close Shut-off valves on Hydrogen line - Switch off the system
					higher ΔP >10 mbar	recirculation line blocked -damper closed	no recirculation stream in the boiler	Close Shut-off valves on Hydrogen line - Switch off the system

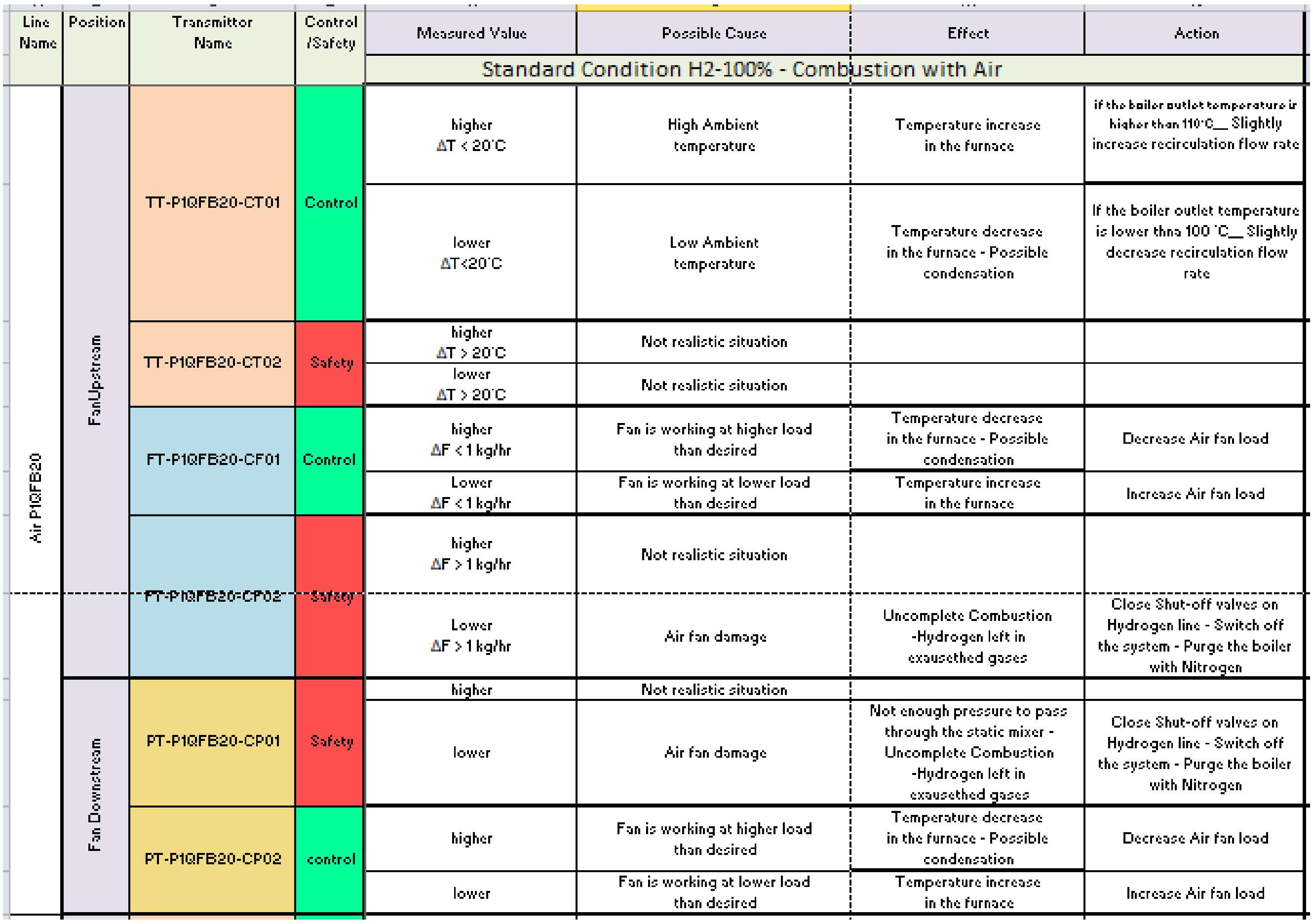
Fig. 67: HAZOP- Recirculation line - case: " 100% load combustion of hydrogen with oxygen and recycled stream"



Line Name	Position	Transmittor Name	Control /Safety	Desired Value	Measured Value	Possible Cause	Effect	Action
						Standard Condition H2-100% -Combustion with Oxyegn		
Cooling Water P1PGB20	Furnace Upstream	FT-P1PGB10-CF01	Control	10080 kg/hr	higher	Pump defect	Temperature decrease in furnace - temperature decrease of recycled stream	slightly decrease pump load
					lower	Pump defect	Temperature increase in the furnace - Temperature increase of recycled stream	slightly increase pump load
		TT-P1PGB10-CT01	Control	60°C	higher	Not all the heat has been extracted from the cooling water	Temperature increase in the furnace - Temperature increase of recycled stream	slightly decrease pump load
					lower	More than the required heat has been ecxtracted from the cooling wate	Temperature decrease in furnace - temperature decrease of recycled stream	slightly decrease pump load
	Furnace Downstream	TT-P1PGB10-CT02	Safety	105°C	higher ΔT > 40°C	evaporation of cooling water	Damage to the equipment	Close Shut-off valves on Hydrogen line - Switch off the system
					lower ΔT>20°C	High Cooling water Flow rate - Temperature decrease in furnace	Condensation in furnace - Low temperaature of recycled stream	Close Shut-off valves on Hydrogen line - Switch off the system
		TT-P1PGB10-CT03	Control	105°C	higher ΔT < 20°C	Not sufficient cooling water flow rate- Temperature increase in furnace -	Possible damege to the equipment - higher temperature of stream out of the boiler	increase cooling water flow rate
					lower ΔT < 20°C	High Cooling water Flow rate - Temperature decrease in furnace	Condensation in furnace - Low temperaature of recycled stream	decrease cooling water flow rate -
		PT-P1PGB10-GP01	Safety	3 barg	higher ΔP >100 mbar	evaporation of cooling water	Damage to the equipment	Close Shut-off valves on Hydrogen line - Switch off the system
					Lower ΔP >100 mbar	water leackage in the furnace	Damage to the equipment	Close Shut-off valves on Hydrogen line - Switch off the system
		PT-P1PGB10-CP02	Control	3 barg	higher ΔP <100 mbar	Cooling water stream slower - smaller pressure drops	increase of pressure at pump inlet	decrease cooling water flow rate -
					Lower ΔP < 100	Higher Pressure drop	Decrease of pressure at pumpinlet	increase cooling water flow rate

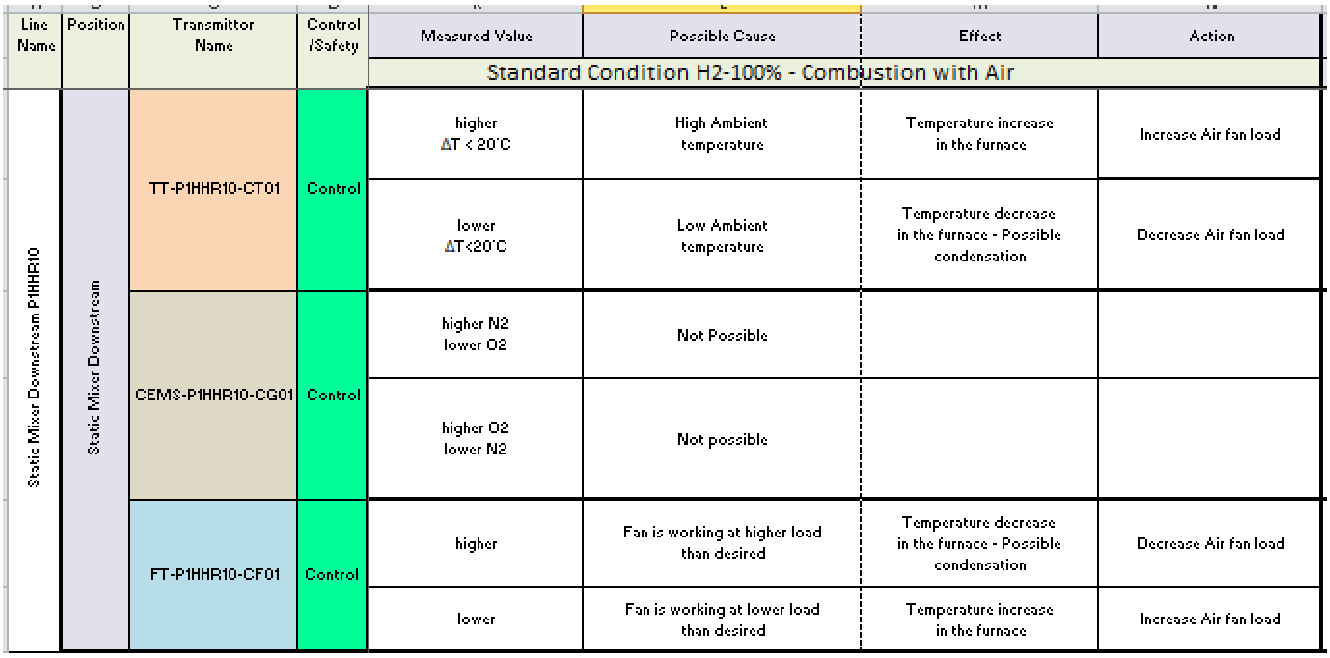
Fig. 68: HAZOP- Cooling water line - case: " 100% load combustion of hydrogen with oxygen and recycled stream"

In order to give a clear picture of what the BMS and DCS should do in case of malfunctioning of the system, it is not necessary to repeat all the lines for all the cases. In fact in general only the desired value which the transmitter should detect will vary from one standard condition to another. For example for the standard condition "– Combustion of hydrogen with air – 100% load – Adiabatic Temperature =2000°C" the BMS and DCS should behave in the same way for what concern the hydrogen line, the boiler and the stack. The recirculation line in this case is not involved, as well as the nitrogen, oxygen and natural gas line. The lines which are involved in this case and were not mentioned before are the air line and the line downstream the static mixer, which will actually vary its condition.



Line Name	Position	Transmitter Name	Control /Safety	Measured Value	Possible Cause	Effect	Action
				Standard Condition H2-100% - Combustion with Air			
Air P1QFB20	FanUpstream	TT-P1QFB20-CT01	Control	higher ΔT < 20°C	High Ambient temperature	Temperature increase in the furnace	if the boiler outlet temperature is higher than 110°C__ Slightly increase recirculation flow rate
				lower ΔT<20°C	Low Ambient temperature	Temperature decrease in the furnace - Possible condensation	If the boiler outlet temperature is lower thna 100 °C__ Slightly decrease recirculation flow rate
		TT-P1QFB20-CT02	Safety	higher ΔT > 20°C	Not realistic situation		
				lower ΔT > 20°C	Not realistic situation		
		FT-P1QFB20-CF01	Control	higher ΔF < 1 kg/hr	Fan is working at higher load than desired	Temperature decrease in the furnace - Possible condensation	Decrease Air fan load
				Lower ΔF < 1 kg/hr	Fan is working at lower load than desired	Temperature increase in the furnace	Increase Air fan load
		FT-P1QFB20-CF02	Safety	higher ΔF > 1 kg/hr	Not realistic situation		
				Lower ΔF > 1 kg/hr	Air fan damage	Uncomplete Combustion -Hydrogen left in exausthed gases	Close Shut-off valves on Hydrogen line - Switch off the system - Purge the boiler with Nitrogen
	Fan Downstream	PT-P1QFB20-CP01	Safety	higher	Not realistic situation		
				lower	Air fan damage	Not enough pressure to pass through the static mixer - Uncomplete Combustion -Hydrogen left in exausthed gases	Close Shut-off valves on Hydrogen line - Switch off the system - Purge the boiler with Nitrogen
		PT-P1QFB20-CP02	control	higher	Fan is working at higher load than desired	Temperature decrease in the furnace - Possible condensation	Decrease Air fan load
				lower	Fan is working at lower load than desired	Temperature increase in the furnace	Increase Air fan load

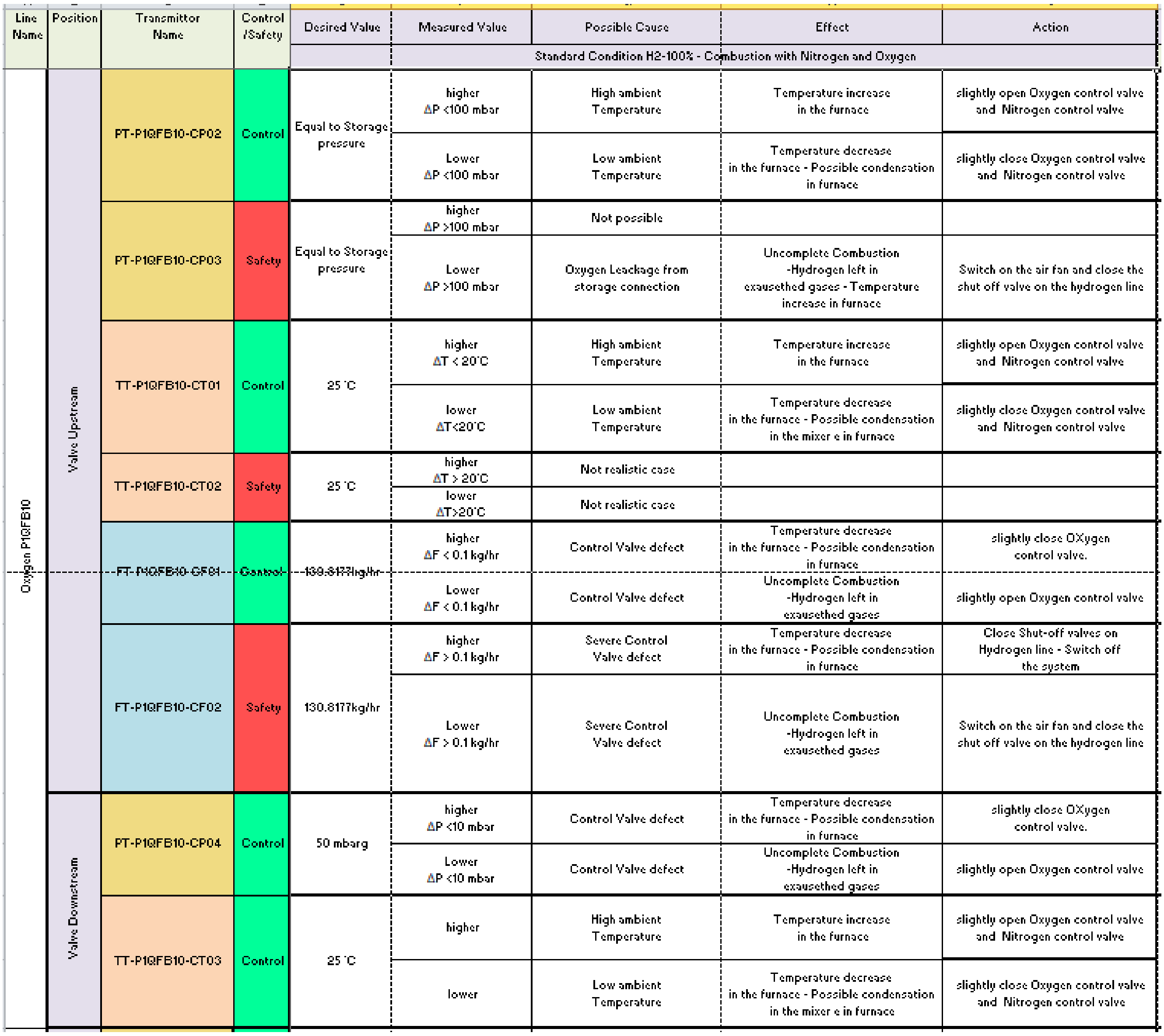
Fig. 69: HAZOP - AIr line - case: " 100% load combustion of hydrogen with air"



Line Name	Position	Transmitter Name	Control /Safety	Measured Value	Possible Cause	Effect	Action
				Standard Condition H2-100% - Combustion with Air			
Static Mixer Downstream P1HHR10	Static Mixer Downstream	TT-P1HHR10-CT01	Control	higher ΔT < 20°C	High Ambient temperature	Temperature increase in the furnace	Increase Air fan load
				lower ΔT<20°C	Low Ambient temperature	Temperature decrease in the furnace - Possible condensation	Decrease Air fan load
		CEMS-P1HHR10-CG01	Control	higher N2 lower O2	Not Possible		
				higher O2 lower N2	Not possible		
		FT-P1HHR10-CF01	Control	higher	Fan is working at higher load than desired	Temperature decrease in the furnace - Possible condensation	Decrease Air fan load
				lower	Fan is working at lower load than desired	Temperature increase in the furnace	Increase Air fan load

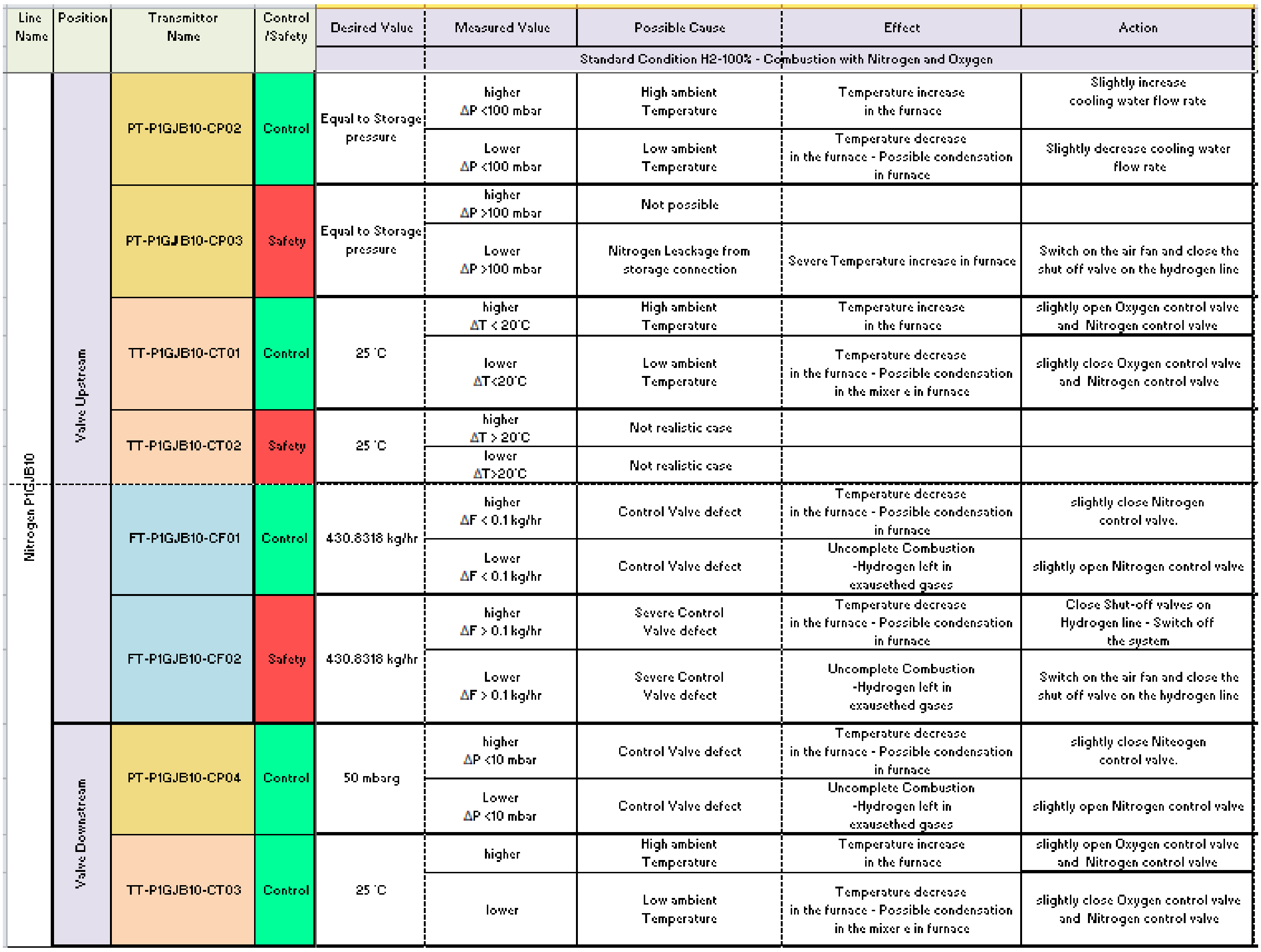
Fig. 70: HAZOP – static mixer downstream line - case: " 100% load combustion of hydrogen with air"

The next standard situation is the one in which hydrogen is burned using the mixture of oxygen and nitrogen; for this situation only the oxygen, nitrogen and static mixer downstream lines will vary. Hence only those are reported:



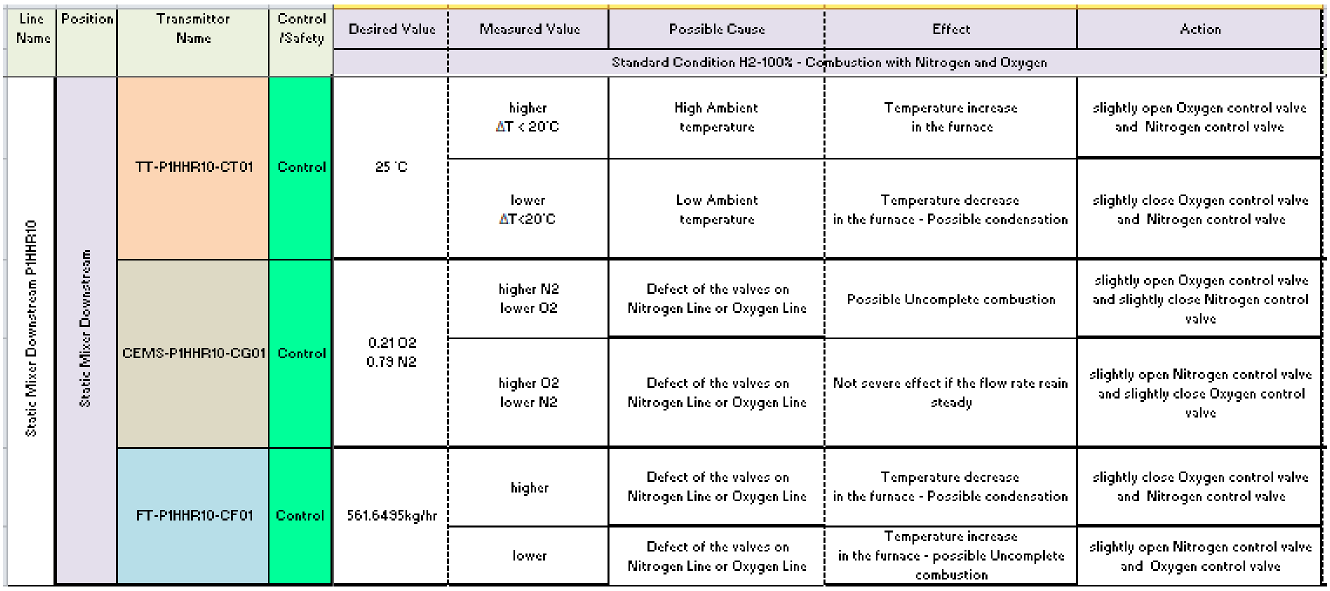
Line Name	Position	Transmittor Name	Control /Safety	Desired Value	Measured Value	Possible Cause	Effect	Action
				Standard Condition H2-100% - Combustion with Nitrogen and Oxygen				
Oxygen P1QFB10	Valve Upstream	PT-P1QFB10-CP02	Control	Equal to Storage pressure	higher ΔP <100 mbar	High ambient Temperature	Temperature increase in the furnace	slightly open Oxygen control valve and Nitrogen control valve
					Lower ΔP <100 mbar	Low ambient Temperature	Temperature decrease in the furnace - Possible condensation in furnace	slightly close Oxygen control valve and Nitrogen control valve
		PT-P1QFB10-CP03	Safety	Equal to Storage pressure	higher ΔP >100 mbar	Not possible		
					Lower ΔP >100 mbar	Oxygen Leackage from storage connection	Uncomplete Combustion -Hydrogen left in exausethed gases - Temperature increase in furnace	Switch on the air fan and close the shut off valve on the hydrogen line
		TT-P1QFB10-CT01	Control	25 °C	higher ΔT < 20°C	High ambient Temperature	Temperature increase in the furnace	slightly open Oxygen control valve and Nitrogen control valve
					lower ΔT<20°C	Low ambient Temperature	Temperature decrease in the furnace - Possible condensation in the mixer e in furnace	slightly close Oxygen control valve and Nitrogen control valve
		TT-P1QFB10-CT02	Safety	25 °C	higher ΔT > 20°C	Not realistic case		
					lower ΔT>20°C	Not realistic case		
		FT-P1QFB10-CF01	Control	130.8177kg/hr	higher ΔF < 0.1 kg/hr	Control Valve defect	Temperature decrease in the furnace - Possible condensation in furnace	slightly close OXygen control valve.
					Lower ΔF < 0.1 kg/hr	Control Valve defect	Uncomplete Combustion -Hydrogen left in exausethed gases	slightly open Oxygen control valve
		FT-P1QFB10-CF02	Safety	130.8177kg/hr	higher ΔF > 0.1 kg/hr	Severe Control Valve defect	Temperature decrease in the furnace - Possible condensation in furnace	Close Shut-off valves on Hydrogen line - Switch off the system
					Lower ΔF > 0.1 kg/hr	Severe Control Valve defect	Uncomplete Combustion -Hydrogen left in exausethed gases	Switch on the air fan and close the shut off valve on the hydrogen line
	Valve Downstream	PT-P1QFB10-CP04	Control	50 mbarg	higher ΔP <10 mbar	Control Valve defect	Temperature decrease in the furnace - Possible condensation in furnace	slightly close OXygen control valve.
					Lower ΔP <10 mbar	Control Valve defect	Uncomplete Combustion -Hydrogen left in exausethed gases	slightly open Oxygen control valve
		TT-P1QFB10-CT03	Control	25 °C	higher	High ambient Temperature	Temperature increase in the furnace	slightly open Oxygen control valve and Nitrogen control valve
					lower	Low ambient Temperature	Temperature decrease in the furnace - Possible condensation in the mixer e in furnace	slightly close Oxygen control valve and Nitrogen control valve

Fig. 71: HAZOP – Oxygen line - case: " 100% load combustion of hydrogen with oxygen and nitrogen"



Line Name	Position	Transmitter Name	Control /Safety	Desired Value	Measured Value	Possible Cause	Effect	Action
					Standard Condition H2-100% - Combustion with Nitrogen and Oxygen			
Nitrogen P1GJB10	Valve Upstream	PT-P1GJB10-CP02	Control	Equal to Storage pressure	higher $\Delta P <100$ mbar	High ambient Temperature	Temperature increase in the furnace	Slightly increase cooling water flow rate
					Lower $\Delta P <100$ mbar	Low ambient Temperature	Temperature decrease in the furnace - Possible condensation in furnace	Slightly decrease cooling water flow rate
		PT-P1GJB10-CP03	Safety	Equal to Storage pressure	higher $\Delta P >100$ mbar	Not possible		
					Lower $\Delta P >100$ mbar	Nitrogen Leackage from storage connection	Severe Temperature increase in furnace	Switch on the air fan and close the shut off valve on the hydrogen line
		TT-P1GJB10-CT01	Control	25 °C	higher $\Delta T < 20°C$	High ambient Temperature	Temperature increase in the furnace	slightly open Oxygen control valve and Nitrogen control valve
					lower $\Delta T<20°C$	Low ambient Temperature	Temperature decrease in the furnace - Possible condensation in the mixer e in furnace	slightly close Oxygen control valve and Nitrogen control valve
		TT-P1GJB10-CT02	Safety	25 °C	higher $\Delta T > 20°C$	Not realistic case		
					lower $\Delta T>20°C$	Not realistic case		
		FT-P1GJB10-CF01	Control	430.8318 kg/hr	higher $\Delta F < 0.1$ kg/hr	Control Valve defect	Temperature decrease in the furnace - Possible condensation in furnace	slightly close Nitrogen control valve.
					Lower $\Delta F < 0.1$ kg/hr	Control Valve defect	Uncomplete Combustion -Hydrogen left in exausethed gases	slightly open Nitrogen control valve
		FT-P1GJB10-CF02	Safety	430.8318 kg/hr	higher $\Delta F > 0.1$ kg/hr	Severe Control Valve defect	Temperature decrease in the furnace - Possible condensation in furnace	Close Shut-off valves on Hydrogen line - Switch off the system
					Lower $\Delta F > 0.1$ kg/hr	Severe Control Valve defect	Uncomplete Combustion -Hydrogen left in exausethed gases	Switch on the air fan and close the shut off valve on the hydrogen line
	Valve Downstream	PT-P1GJB10-CP04	Control	50 mbarg	higher $\Delta P <10$ mbar	Control Valve defect	Temperature decrease in the furnace - Possible condensation in furnace	slightly close Niteogen control valve.
					Lower $\Delta P <10$ mbar	Control Valve defect	Uncomplete Combustion -Hydrogen left in exausethed gases	slightly open Nitrogen control valve
		TT-P1GJB10-CT03	Control	25 °C	higher	High ambient Temperature	Temperature increase in the furnace	slightly open Oxygen control valve and Nitrogen control valve
					lower	Low ambient Temperature	Temperature decrease in the furnace - Possible condensation in the mixer e in furnace	slightly close Oxygen control valve and Nitrogen control valve

Fig. 72: : HAZOP – nitrogen line - case: " 100% load combustion of hydrogen with oxygen and nitrogen"



Line Name	Position	Transmitter Name	Control /Safety	Desired Value	Measured Value	Possible Cause	Effect	Action
					Standard Condition H2-100% - Combustion with Nitrogen and Oxygen			
Static Mixer Downstream P1HHR10	Static Mixer Downstream	TT-P1HHR10-CT01	Control	25 °C	higher $\Delta T < 20°C$	High Ambient temperature	Temperature increase in the furnace	slightly open Oxygen control valve and Nitrogen control valve
					lower $\Delta T<20°C$	Low Ambient temperature	Temperature decrease in the furnace - Possible condensation	slightly close Oxygen control valve and Nitrogen control valve
		CEMS-P1HHR10-CG01	Control	0.21 O2 0.79 N2	higher N2 lower O2	Defect of the valves on Nitrogen Line or Oxygen Line	Possible Uncomplete combustion	slightly open Oxygen control valve and slightly close Nitrogen control valve
					higher O2 lower N2	Defect of the valves on Nitrogen Line or Oxygen Line	Not severe effect if the flow rate reain steady	slightly open Nitrogen control valve and slightly close Oxygen control valve
		FT-P1HHR10-CF01	Control	561.6495kg/hr	higher	Defect of the valves on Nitrogen Line or Oxygen Line	Temperature decrease in the furnace - Possible condensation	slightly close Oxygen control valve and Nitrogen control valve
					lower	Defect of the valves on Nitrogen Line or Oxygen Line	Temperature increase in the furnace - possible Uncomplete combustion	slightly open Nitrogen control valve and Oxygen control valve

Fig. 73: HAZOP – Static mixer downstream - case: " 100% load combustion of hydrogen with oxygen and nitrogen"

For all the cases in which Natural gas is burned instead of hydrogen, the DCS and BMS works exactly in the same way, in this case the natural gas line will behave has the hydrogen line for the cases analyzed preciously.

APPENDIX 15 : Evaluation of the Internship

Stork Thermeq B.V. is a Dutch manufacturing and service providing company. It provides fully integrated operation, maintenance, modification and asset integrity solution. It is a leading company in the design and production of boilers, burners, deaerators and other equipment for power plants.

My role during the internship was to help in the development of a pilot plant for hydrogen combustion. Thanks to the experience of the people working in the company and of the supervisor I learned how the planning of a plant is approached, I learned how to realize PFD's and P&ID's drawings, how to size and design some of the instrumentation of a power plant and I increased my skills in 3D drawing, becoming aware of the problems in the 3D printing. I have given to the company documentations which can be used as reference to make the plant operative and that can be used for further implementation. The results are presented to the company during a presentation that will take place on the 20th of December 2018. The internship has been of great value above all for the notions concerning the methodology of development of a power plant from the technical and operational point of view.