ROTATING EQUIPMENT FOR CARBON DIOXIDE CAPTURE AND STORAGE

Report: 2010/07

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ROTATING MACHINERY FOR CO_2 COMPRESSION IN CCS SYSTEMS

Introduction

Carbon Dioxide capture plants need to have CO_2 compression facilities since all of the typical processes capture CO_2 at pressures well below those needed for transport and underground injection. The power required for this compression represents a significant part of the parasitic energy consumption of the CCS process. The projected compression power requirements are expected to become significant in comparison with for example those for natural gas transmission in the coming decades if CCS is adopted on a large scale. This study was commissioned to examine the type of compression machinery currently available for this duty, to look at novel compression concepts and investigate the options for integrating the compression train into the overall process.

Approach

A contact for the study was awarded to Foster Wheeler Italiana on the basis of competitive tender. The work was divided into two stages. The first part was to define the process requirements for the compression for typical coal fired, pre, post and oxy-combustion processes as well as for a gas fired post combustion process. Thereafter a number of optimisations of the integration of the compression system with the rest of the process were examined. In the second part selected manufacturers of CO_2 compressors were asked to provide general information on their products and also to make selections and indicate budget prices for compressors to perform the 4 specific sets of compression process requirement. Two novel compression concepts were also reviewed, the most revolutionary being the supersonic compression technology being developed by Ramgen, the other being the use of a low pressure axial flow compressor for the first stage of compression. In addition a novel method to use heat of compression for regeneration of mole sieve dryers was investigated

Results & Discussion

Basic Compression Process Requirements

Basic process requirements for compression were based on flow schemes and heat and material balances from the most recent previous studies of pre, post and oxy combustion reported by IEAGHG. These were used to define compression duty requirements to prospective compressor manufacturers. The specification for final water content of the compressed CO_2 was found to be an important parameter as this affects the selection of a drying step additional to compressor after-cooling and water knock out. The referenced studies all use mole sieve dryers which require a recycle stream of CO_2 and a heat source for bed regeneration. This has effects on the compressor stage flow preceding drying and the heat integration. TEG or Glycerol drying is an alternative which does not have these effects. For oxy-combustion processes deep drying is essential and is imposed by the required cryogenic processing conditions. Here glycol cannot be used partly because of the tight water specification but mainly because of the presence of oxygen which causes glycol degradation. The pressure at which drying is required is fixed by the parameters of the oxy-combustion CO_2 clean up process. However for pre- and post combustion processes there is considerable flexibility as to the pressure at which the drying step is placed in the compression train.

Strategies to optimise compression system

Several strategies to optimise compression were evaluated. Some of these can be applied to all of the capture processes but their effect on overall process efficiency and economics varies. Hence the effects of the various strategies are summarised for each capture process. Each strategy leads to a difference in net power output of the complete plant and also a change in capex/opex. There are other effects such as increased complexity which are covered in the main report. In the overview just the effect on overall plant efficiency will be quantified.

Post-combustion CO_2 compression optimisation

The base-line option for post combustion capture for both coal and gas fired processes is to collect CO_2 from a single stream from the overheads of a stripper operating at slightly above atmospheric pressure (1.6bara). There is some heat integration between the compressor and the rest of the process because the compressor stage discharges are partly cooled by heating boiler feed water. The baseline coal fired capture plant has an output of 655 MWe with an efficiency, based on LHV, of 34.8%.

The most interesting option to improve overall efficiency was found to be to increase the operating pressure of the stripper thus reducing the compression power required. It was found that the savings in compression power more than compensated for reductions in power output from the steam turbine caused by extraction of steam at higher pressure to perform the solvent regeneration. Although positive the gains are relatively small compared to the total output of the plant - The extra 7.3 MW generated in this option represents an overall thermal efficiency increase of just 0.38%.. Whilst low, this is large enough to be of interest to power plant operators. However for this particular option the raised temperature in the stripper will result in accelerated degradation of today's typical solvents to the extent that overall power production costs were estimated to increase.

The other options found to provide some gain in output were to reduce compressor stage pressure ratio or to regenerate at two different pressure levels. Neither were very effective because in both strategies the amount of heat recoverable to preheat BFW is significantly reduced. Halving the compressor pressure ratio increased overall efficiency by only 0.1%. and applying split pressure regeneration 0.11%. It is largely because of this heat integration that use of vapour recompression in the stripping section was found to **reduce** overall efficiency by 0.15%.

Pre-combustion CO_2 compression optimisation

The base line option for pre-combustion capture is coal gasification in a Shell type gasifier. The resulting syngas is shifted to convert the bulk of the CO to CO_2 and hydrogen by reaction with steam. After cooling the CO_2 is removed in a proprietary Selexol unit configured to deliver two wet concentrated CO_2 streams at 1 bara and 4.8 bara. The overall plant has a net output of 750MWe and a net thermal efficiency of 43.1%. Adding an additional flash stage to the system, at a higher pressure of 11.5 bar, slightly increases efficiency by 0.11%. This is a very small gain and may not be worth the extra complexity. Vapour recompression was found to reduce overall efficiency by 0.33% and two other strategies investigated proved to be non-viable.

Oxy-combustion CO_2 compression optimisation

The base case plant had an electrical output of 530MWe and an efficiency of 35.4%. Part of the compressor duty is for auto-refrigeration in the CO_2 cryogenic purification section as a result of which the CO_2 compression duty is proportionately larger for oxy-combustion.

The option to replace the final stages of compression by cooling and then pumping the CO_2 proved to offer no change in efficiency. The gains in compression power are offset by loss of heat recovery into the feed water from compressor discharge cooling. The option to expand the purification vent gas to create cold for the cryogenic purification unit proved to be less efficient overall by 0.39%. Heating and then passing this stream through a turbo-expander before it is vented appears to be the most efficient strategy. The strategy of pumping liquefied CO_2 up to pressure once it is formed in the clean up unit and providing the refrigeration duty using external propane refrigeration proved to be similarly unattractive reducing efficiency by 0.51%

Selection of compressors

A number of manufacturers were provided with the basic compression requirements for the baseline plants described above. 4 companies (Rolls Royce, Man & Diesel, GE and Ramgen) responded with varying levels of detail covering three basic types of CO_2 compression equipment. For the capacities required in large scale CCS plants reciprocating compressors are too small. Information was thus received on barrel type centrifugal machines, integral gear machines and a novel 2 stage supersonic machine for the duty. Appended to this summary is a chart showing all of the basic compressor packages which were offered.

Number of compression stages

The post combustion process places the least restriction on staging since the optimum arrangement simply requires a straight compression from just above atmospheric pressure to the pipeline pressure. The main constraint is the need to find a suitable intermediate pressure to insert a drying step and typically this would be at a point somewhat below the critical pressure so that most water has been removed in compressor intercoolers and knock out drums and so that drying occurs in the gas phase. In all three schemes the pressure at which dehydration is conducted is approximately 33bara. This is a significant constraint for the Ramgen concept which is constrained to dry either at around 11bara or in the supercritical state at 111bara. However it is possible to conduct the dehydration at the lower pressure with little or no effect on the cost of the dehydration equipment thus removing this apparent constraint.

Pressure and flow in intermediate compression stages

Post combustion capture processes impose no particular constraints on choice of intermediate pressure given that there is a wide range of choice for the pressure at which dehydration is done. In the pre-combustion process there is an intermediate pressure level defined by the pressure of the first flash. In the base case this is at 4.8 bara. The large proportion (about 2/3) of the CO_2 is released in this flash so that any variations would have significant implications for flows and efficiency. However there may be scope for optimisation of this pressure although this has not been investigated in this study. Introduction of an additional flash stage at 11.5bara was investigated. This allowed about 19% of the total gas flow to require one stage less compression but did not as mentioned earlier result in any overall gain in efficiency.

In the oxy-combustion process there are very specific intermediate stage pressure requirements because in the initial stages the CO_2 has to be raised to the pressures required for reactors in the CO_2 clean up process. Thereafter the CO_2 is the working fluid for the auto-refrigeration process which places constraints on intermediate operating pressures.

Use of a desiccant bed drying system introduces a requirement for circulation of dry and hot regeneration gas. One option is to provide this using the previous stage of compression although this is wasteful of compression energy since the beds have a low pressure drop. The alternative would be to provide a separate recirculation blower. A further interesting alternative is to use “heat of compression” drying which is able to reduce the recycle flow for drying substantially. This is briefly described under the sections on novel concepts and a fuller description of this process is included in appendix 1 of the main report.

Options using in line centrifugal compressors

Rolls Royce has developed a range of horizontal split and barrel type centrifugal compressors suitable for CO_2 service. They provided basic selection information and budget price information on several arrangements to satisfy the base case requirements.

The duty is covered by two ranges of machines. The first are machines with horizontal split casing coming in 2 frame sizes and suitable for the low pressure part of the compression trajectory. These frames are undergoing upgrades to raise their maximum discharge pressure currently 25-28 bar to above 34 bar.

The high pressure part of the duty is covered by machines with a barrel casing with 5 barrel sizes. (designated RAB through REB). The smallest size is still in need of an upgrade for CO_2 duty. The two largest machines (RDB/REB) have been upgraded to accommodate higher maximum discharge pressures of 241/137 bara respectively. (The inlet flow capacities of the bigger machines are well in excess of the flows from the base case designs considered so that a single train would be possible for the higher pressure part of the compression trajectory up to capacities of coal fired plants approaching 4GW.

The duty with the largest inlet flow volume is that for the baseline coal fired oxy-combustion plant mainly because the CO_2 has to be compressed from the lowest pressure i.e atmospheric. By contrast the starting pressure for post combustion is at least 1.6 bar and only 33% of the CO_2 in the pre-combustion case is at atmospheric pressure, the rest is already at 4.8bara.

Thus the basic Rolls Royce offerings would be based on 3 parallel low pressure horizontally split compression trains for the oxy-combustion option, 2 such trains for post combustion and only a single train for pre-combustion. Thereafter the higher pressure stages are served by various combinations of the three smaller sizes of barrel compressor. For larger capacities the number of low pressure trains would have to be increased due to the limitation on inlet flow capacity. However the number of higher pressure trains running in parallel could be reduced by utilising larger barrel sizes. The compression system is driven by electric motors with variable speed drive. Because the higher pressure machines stages run at higher speeds a number of separate motors are needed as only machines which can run at the same speed are driven on one shaft.

GE made outline proposals for post and oxy-combustion processes only. For post combustion the proposed arrangement was to have 2 parallel trains with 4 stages in 2 separate casings with each driven by a single motor and two gearboxes. For the oxy-combustion process two parallel three stage machines with two casings driven by a motor and single gearbox followed by a third machine with two stages and a single casing driven by a motor and gearbox for the higher pressure part of the compression trajectory. They did not however provide any information about the full capacity range of their machines although in principle larger train sizes should be possible.

Options using integrally geared compressors

Man-Turbo & Diesel produce a range of integrally geared compressors which can have up to 5 shafts with two impellors on each shaft. The maximum flow is in general restricted by the size and number of the lowest pressure impellers. It is possible to connect a number of impellers in parallel to increase the capacity of the lowest pressure stage. The range currently runs from the smallest frame with a 25mm inlet diameter impeller rated for 4MW up to the largest with a 160 diameter inlet rated for up to

60MW. However for CO_2 service the current range is from 45mm – 140 mm. There is thus considerable flexibility in layout and the designs can easily incorporate variations in the flow rate though different stages and accommodate tightly specified intermediate pressure levels.
Man-Turbo & Diesel designate their machines and frame sizes with a coding of the form RG 140-8 in which the first number is the first stage inlet impeller diameter and he second is the number of impellers fitted.

Again the oxy-combustion process requires the largest inlet flow volume and Man Turbo & Diesel offered 2 separate compressor trains using their 125mm inlet diameter impellers with just 4 stages in series. A third train provides the compression for the cryogenic section of the process using a smaller machine of 56mm inlet diameter also with 4 stages in series.

For post combustion a single 140mm inlet diameter impeller machine with either 6 or 7 stages is offered. Total power is 48.8/45.6MW for these two options which is close to the stated maximum of 50MW for this frame size. In principle it would be possible to accommodate higher capacity coal fired plants if stripper pressure were higher towards 1GW. Thereafter multiple trains would probably be required as even the 160mm inlet diameter machine does not offer that much more maximum power. For pre-combustion a single machine with 100mm inlet diameter first stage impellers using parallel pairs of impellers for the first 3 stages and single impellers for the last two stages is possible. Alternatively the duty can be split between a machine of similar frame size with just 4 impellers, arranged in 2 parallel stages feeding a second smaller machine of 63mm inlet stage diameter with 3 stages in series.

GE offered integral gear options for pre and post combustion processes but not the oxy-fuel application. For post combustion 2 parallel machines of either 6 or 8 stages were offered. The 8 stage machine consumes marginally less power but lower heat recovery of compressor heat will reduce the value of this reduction on the overall power plant efficiency. No information on the range of machines available was provided. It is however likely that larger power plant capacities will require more trains.

Novel solutions-Uncooled axial compressors for first stage.

To avoid having to have multiple parallel trains the option of using a single axial machine for the lowest pressure part of the compressions was considered. This option was examined and would in principle enable reduction in number of trains and would make a small reduction in total power. However such a machine would have to be executed throughout in corrosion resistant materials making this a potentially expensive option. It would have poorer turndown than multiple trains.

Novel solutions – Supersonic high ratio compression

Ramgen has developed a high compression ratio compressor based on supersonic shockwave compression principles. The machine is compact and highly efficient. It would use only two stages so that despite its high efficiency it would use more power. But, because of the high compression ratio the stage outlet temperatures are high (around 240C) and hence a greater portion of the heat of compression is available for useful integration into the overall process. This is most applicable in the post combustion capture process. Simulations show that based on the quoted performance use of the concept should slightly **increase** overall power plant efficiency provided the heat of compression is used in the solvent regenerator reboiler. In fact the quality of the heat is higher than that of the LP steam which it backs from this application out but unless a higher temperature destination can be found this further potential advantage cannot be realised. Use in heat of compression regeneration of mole sieve driers is one potential application. Were glycol drying to be used the glycol regenerator reboiler would also be a potential destination and is a higher temperature application (around 205C) than amine solvent regeneration.(around 135C). However the duty of such a system is likely be to be only a few MW.

In practical terms RAMGEN offers a range of 7 frame sizes for each of the LP and HP stages. The LP sizes range from 22 to 46 inch rotors in steps of 4 inches and the HP from 12 to 26 inch in steps of 2 inches. For the post combustion duty a 38inch LP and a 26 inch HP frame size is selected. The total compressor power is estimated at 55.9MW compared to between 43.5 and 50.4 MW for the various options for other types of compressor. These figures do not allow for the effects of compression heat recovery which, counter-intuitively, means that less efficient compression can sometimes result in a more efficient power plant overall.

The RAMGEN compressor is at an early stage of development with first prototypes having operated. It will be some time before they are fully commercially proven.

Novel solutions – Heat of compression drying

This system for drying mole sieves in drying service has been developed by a company named SPX. It can be used when the un-cooled compressor outlet temperature prior to the dehydration step is high enough and is thus particularly applicable to the Ramgen alternative. The key principle is to use a part of the hot un-cooled feed gas to heat the bed requiring regeneration. Most of the hot gas bypasses this bed to go straight to the compressor after-cooler where is rejoined by the now wet stream from the bed under regeneration. A small pressure drop has to be created to get the slip stream to flow through the bed. The hot stream is cooled in the aftercooler to knock out free water as normal before passing in to the on line bed for drying. With a discharge temperature of 240C this is almost but not quite enough to complete the regeneration. There is still a requirement for a final purge of the regenerated bed with dried gas but this represents anly about

Turndown and sparing

Manufacturers provided some information on turndown. Combinations of variable speed electric drives and inlet guide vanes provide turndown to about 70% of capacity. There is a small loss of polytropic efficiency typically around 4% at maximum turndown. Thereafter it is necessary to resort to recycling and/or multiple parallel units. However as several options used only 2 x 50 % trains these have a turn down gap between 50 and 70%. Only with three parallel trains of 33% capacity is there a possibility to turn down continuously below 70% by turning trains off.

Reliability of integral gear machines is typically around 97% and for centrifugal machines 99%. Hence it is unlikely that for this non-critical duty the machines would be spared. Splitting into smaller units to gain turndown would significantly increase costs and maintenance. Hence recycling is most likely to be the best option for extending turndown beyond 70%.

Manufacturing capacity

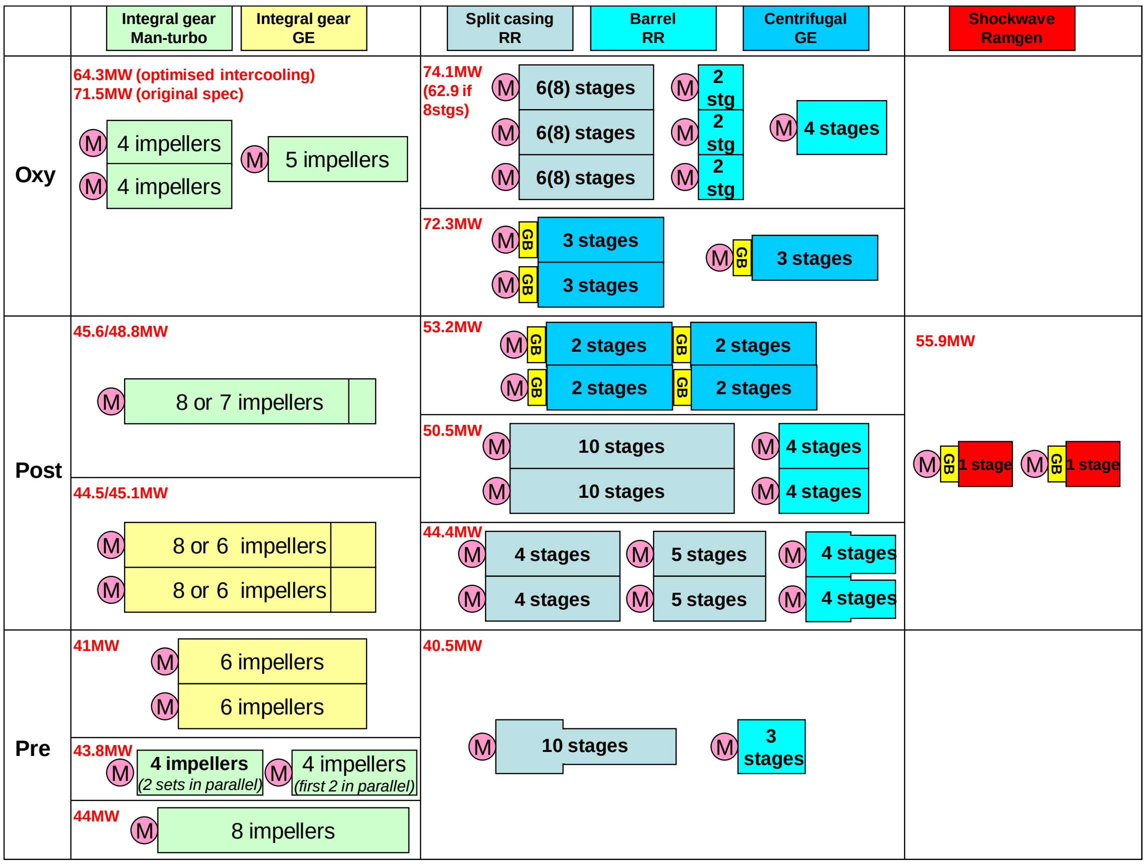
Based on the IEA bluemap scenario for CCS approximately 40 large CO_2 compression units would be required per year up to 2030 and thereafter as many as 100 per year. The two main suppliers who responded during this study indicated a joint capacity of around 40 units per year. This represents capacity for these specific type of rotating machinery and there are other manufacturers. There should be ample industry capacity to fabricate the required compression equipment if CCS gains a prominent role.

Prices

Manufacturers indicated budget prices for machines to meet the various duties supplied to them during enquiries. These prices need to be treated with care as the exact scopes of supply are not specified in detail and may vary between manufacturers. Hence only rather general ranges are mentioned in this report to prevent inappropriate conclusions being drawn. Based on budget prices and contractors own information the in line centrifugal compressors are expected to be slightly more expensive than integral gear compressors. The capital costs of the Ramgen compressor system in this duty is expected to be considerably less than either in line centrifugal or integral gear systems.

However the study highlighted the fact that the different types of compressor need to be integrated into the power plant in significantly different ways. The cost of the compressor alone is not necessarily indicative of lowest overall cost. It is not possible to prepare a single design specification which suits all the types of compressor available and to do so would inevitably bias the choice. Hence it is recommended that for the time being a more flexible and iterative approach is adopted when specifying and selecting CO_2 compressors for CCS power plant duty. A project contract strategy which allows the overall CO_2 capture plant efficiency and cost using the different types of CO_2 compressor available on the market to be compared on a competitive basis should be devised. Such a contract strategy would encourage process design variants to be generated tailored to the different performance characteristics offered by compressor suppliers. That said it may be that other considerations are overriding when contract strategy for a CCS system is formulated and trying to optimise this subsystem in this way may not always be appropriate.

DIAGRAM – Summary of compression packages offered for base case specifications.



This figure summarises the basic compression arrangements offered by manufacturers to satisfy the base case conditions. Where two trains are shown in parallel each is of 50% capacity and where 3 are shown in parallel each is 33.3% capacity. In some cases a slightly different arrangements were offered for the alternative strategies but these are not shown here. The total compressor shaft power is indicated but a higher number does not necessarily result in a less efficient power plant because some heat of compression is recovered in the processes.

Conclusions

The study confirmed the importance of CO_2 compression power in CCS overall plant efficiency. Manufacturer's current efficiency claims proved to be slightly higher than assumptions made in previous studies. Alternative strategies for compression were found to have only a small impact on overall plant efficiency.

Integration of heat of compression into the power plant is essential to maximise efficiency.

The Ramgen concept is well suited to the post combustion application, less so to pre-combustion and does not fit with current oxy combustion capture power plant designs. Despite it's higher compression power requirement when full heat integration is used it offers slightly higher overall power plant efficiency, greater simplicity and potentially lower capital cost.

Optimum plant design is influenced by compressor selection because of the way heat is recovered. To optimise a project effectively an iterative design and compressor selection process is needed.

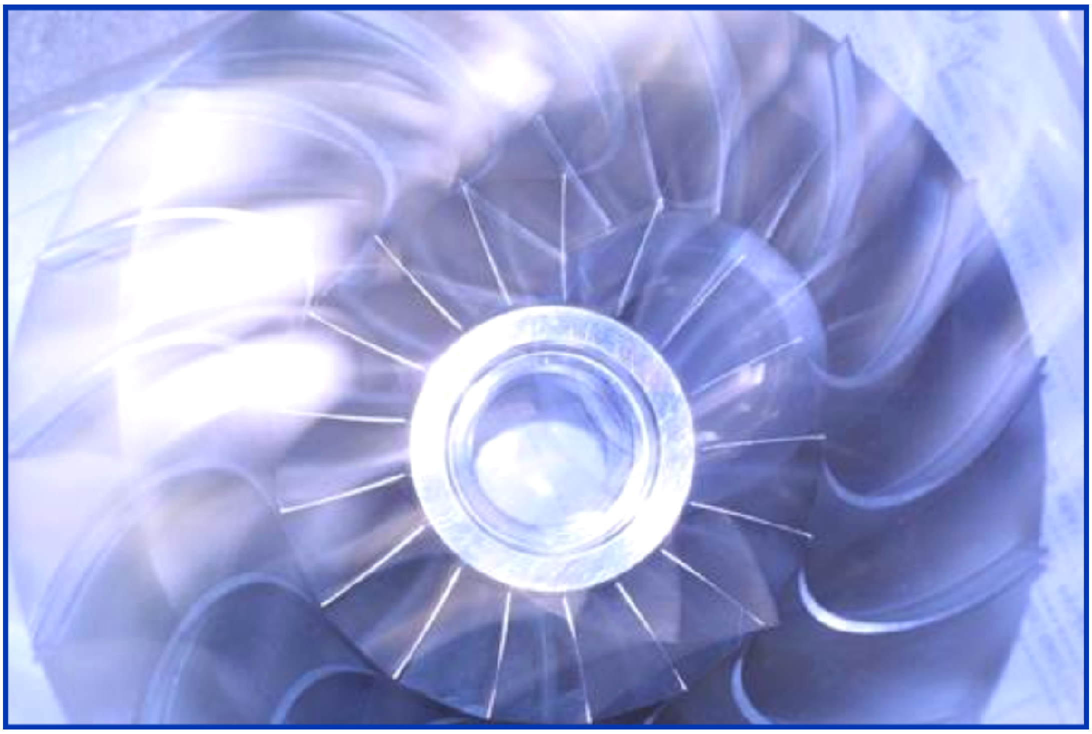
The process needed for drying CO_2 influences the compressor design and heat integration. Current assumptions for the drying process may not be optimum and CO_2 drying is suggested as being a process worth more detailed study.

Recommendations

It is recommended that further work is done on options for drying captured CO_2. This is a specialised area and could either be undertaken as a stand alone study or as part of further engineering studies on capture processes

The basic principles and options for integrating conventional compressors into CCS processes are well understood and broadly covered in this report but further work is needed to optimise the integration of the Ramgen type of compressor into post combustion CCS. IEAGHG should encourage this work to be done and consider offering further guidance to ensure that the potential of this new technology is realised. However Ramgen themselves and engineering organisations preparing detailed designs for post combustion CCS plant are in the best position to execute such development.

IEA Greenhouse Gas R&D Programme



Rotating machinery for CO_2 compression in CCS systems

FINAL REPORT

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GENERAL INDEX

SECTION C EVALUATION OF CO_2 COMPRESSION STRATEGIES

SECTION D COMPRESSION EQUIPMENT SURVEY

SECTION E NOVEL CONCEPTS FOR CO_2 COMPRESSION

APPENDIX 1 CO_2 DEHYDRATION

ABBREVIATIONS

AGR	Acid Gas Removal
ASU	Air Separation Unit
BFW	Boiler Feed Water
BL	Battery Limits
BOP	Balance Of Plant
CC	Combined Cycle
COE	Cost Of Electricity
GT	Gas Turbine
HP	High Pressure
HRSG	Heat Recovery Steam Generator
IP	Intermediate Pressure
LHV	Low Heating Value
LP	Low Pressure
MP	Medium Pressure
MWe	Mega Watt electrical
MWth	Mega Watt thermal
NG	Natural Gas
PC	Pulverised Coal
RH	Re-Heated
SH	Superheated
ST	Steam Turbine
TIC	Total Investment Cost
USC	Ultra Super Critical
VLP	Very Low Pressure
FW	Foster Wheeler
CCS	Carbon Capture and Storage
EOR	Enhanced Oil Recovery
SRU	Sulphur Removal Unit
FGD	Flue Gas Desulphurization
MEA	Mono Ethanol Amine
COP	Coefficient Of Performance
RR	Rolls-Royce
PFD	Process Flow Diagram
H&MB	Heat and Material Balance
IGCC	Integrated Gasification Combined Cycle
IGV	Inlet Guide Vanes
IRR	Internal Rate of Return
NPV	Net Present Value
CW	Cooling Water
WGS	Water Gas Shift

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SECTION A
EXECUTIVE SUMMARY

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1 Background and objectives of the study

In the scientific community it is generally recognized that, by year 2030, the world energy demand will increase by 50%, while fossil fuels, mainly coal and natural gas, will continue to supply most of the energy demands. This reality will continue for many years, until the use of renewable energies will increase significantly. On the other hand, the use of fossil fuels is necessarily correlated to the production of carbon dioxide (CO_2), which contributes to global warming. In this scenario, Carbon Capture and Storage (CCS) represents one of the most effective responses to partially reduce CO_2 emissions in the next few years.

For industrial applications with CCS, the power demand of the CO_2 compression unit and the process units that are thermally related to this system contribute significantly to the energy penalties of the plant, thus reducing its overall efficiency. Therefore, any reduction of the electrical consumption of this system may result in an important overall net plant efficiency improvement.

This report summarizes the outcomes of a study executed by Foster Wheeler for IEA-GHG R&D Programme, aimed at identifying the main types of compression equipment, available in the market for CCS applications, and assessing the key characteristics of different compression systems and machinery configurations. From an energy point of view, for a given final discharge pressure of the carbon dioxide, there are a number of different alternatives that can be considered for the capture and compression unit, corresponding to different power demands and investment cost requirements. This study has investigated different compression strategies, making a techno-economic assessment of various alternatives, applicable to the post, pre and oxy-fuel de-carbonisation processes. General strategies, i.e. valid for any compression type, have also been assessed in this work.

A generic overview of the implications of the identified strategies on compressor selection and design has also been performed, on the basis of the operating conditions of the compressors. Finally, the study has made a description of novel compression concepts that are expected to offer high-stage efficiency, identifying their state of development and the strategies for their use in a typical industrial plant with carbon capture and storage.

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- General Electric,
- Man Diesel & Turbo;
- QB Johnson;
- Ramgen;

- Rolls-Royce;
- SPX.

2 Base of the study

2.1 Base Cases

For each combustion capture type, a Base Case has been identified and used as reference to carry out the comparison with alternative CO_2 compression strategies, which have the potential for lower parasitic loads.

The base cases have been derived from previous studies undertaken for IEA-GHG R&D Programme in the past years on CCS-related topics. The following table provides a summary of the reference cases for each capture technology.

Table 2-1 Base cases summary

Technology	PRE-COMBUSTION	POST-COMBUSTION	OXY-FUEL COMBUSTION.
Type of plant	IGCC	USC-PC	USC-PC
IEA GHG Report ref.	Report PH4/19 [1] Case D4	Report PH4/33 [2] Case 4	Report PH2005/9 [3] Case 2
Gross Power output [MWe]	942.1	827.0	737.0
Net Power output [MWe]	705.0	666.0	532.0
Base Case Tag	A0	B0	C0
CO_2 capture rate	85%	85%	90%
CO_2 compression parasitic load [MW]	47.4 (1)	57.7 (1)	79.3 (1)

Note 1: Re-calculated for the present study through process simulation of the compression unit

Although the above listed studies are not of recent publication, this has not affected the considerations made in this work, as they refer mainly to differences in performance and cost of a limited number of units.

2.2 CO_2 Dehydration system design basis

For the purpose of the study, the moisture specification of the final CO_2 product has been set to 50 ppmv. This figure has been considered as typical for the relatively low ambient conditions used in the study.

The two well-known dehydration technologies applicable to carbon dioxide drying are TEG absorption and solid desiccant. For this study the reference configuration of Dehydration unit is based on solid bed adsorption for the following main reasons:

- Solid bed adsorption units generally have a lower whole life cost than TEG units and provide higher flexibility in terms of dew point depression, being capable to achieve lower moisture in the export gas (down to 1 ppm vs 30 ppm of the TEG process); this is a key factor as the debate on the moisture spec to be used in the different applications associated to CO_2 capture is still open.

- Even though it is recognised that the need of recycling a portion of the CO_2 for bed regeneration causes an energy penalty, it is appreciable that, including this configuration as reference for the study, it is possible to investigate the impacts which the recycle may have on machinery performance and selection.

Furthermore, regarding the potential application of the TEG technology to CO_2 dehydration there were some uncertainties related to the impacts in terms of carbon capture rate, since a minor part of the CO_2 in the wet gas is absorbed in the glycol stream and released from the glycol regeneration section. For this reason, Foster Wheeler have carried out further investigation on this subject with drying systems Vendors and performed simplified process simulations. The outcome is that only a minor portion of the incoming CO_2 (approx. 0.3%) is expected to be absorbed by the glycol, therefore the overall carbon capture rate is not significantly affected if the vent from the glycol regeneration is routed to atmosphere. Hence, it has been concluded that TEG process can be regarded as a technically viable option of CO_2 dehydration in CCS applications for moisture specification above the limit of 30 ppmv.

It has to be noted that there is yet no consensus on a widely recognised pipeline specification, as far as water dew point is concerned. Foster Wheeler and IEA GHG have investigated this topic further and have found recent works reporting that, in many applications, it is likely that the specification can be relaxed with respect to the 50 ppmv used for the study. For instance, values as high as 500 ppmv may be acceptable in the pipeline and even in colder climates 200-300 ppmv should be sufficient to prevent hydrates formation even in the gas phase.

2.3 Basic criteria for techno-economic comparison

One of the main objectives of this study has been to make a technical comparison between the Base Case configurations and alternative CO_2 compression strategies. This evaluation takes into account that the CO_2 compression scheme modifications may lead to a utility requirement, mainly steam and cooling water, which is different from that of the Base Cases. Nevertheless, different utility consumptions also correspond to a different power demand, which then affects the overall net electrical efficiency of the power plant.

For each steam pressure level used in the plant, as well as for the cooling water, the figures shown in Table 2-2 have been used to convert each utility requirement into an equivalent power demand.

Table 2-2 Equivalent electrical consumption of different utilities

Utility	Case involved	Specific equivalent electrical consumption
LP steam		
at 7.5 bara	*Pre-combustion*	191 kWe/t/h
at 3.3 bara	*Post-combustion*	172 kWe/t/h
at 2.5 bara	*Oxy-combustion*	145 kWe/t/h
IP steam		
at 61 bara	*Oxy-combustion*	375 kWe/t/h
Cooling Water	*All*	0.102 kW/m^3/h

With reference to the economic assessment, the general IEA GHG guidelines have been applied to the present analysis, for the evaluation of the various compression strategies, in terms of differential figures with respect to the plant configurations taken as reference (Base Case).

The main factors applicable to this type of analysis are defined as follows:

- Discount rate: 10%.
- Cost of consumed Electricity: 3.8 €c/kWh (to cover lost export electricity revenue, rather than generation costs).
- Cost of coal: 3.0 €/GJ (LHV basis).
- Maintenance costs - IGCC: 3.4% of Differential Investment Cost
- Maintenance costs -USC PC: 3.1% of Differential Investment Cost

3 Technical assessment of CO_2 compression strategies

Table 3-1 shows the main outcomes of the technical assessment, while the following sections provide further details for each CO_2 compression strategy investigated in the study.

Table 3-1 Technical analysis outcome summary

Case	Strategy description	Δ Power (1) [MWe]	Δ efficiency (2) [% points]	Main remarks
A1	Vapour recompression (pre-combustion)	+ 6.0	- 0.3 % pts	Not effective due to low temperature constraint for Drying System operation
A2	Increase of number of flash stages in the AGR	- 2.1	+ 0.1 % pts	Integrated approach with respect to compressor design
B1	Vapour recompression (post-combustion)	+ 2.9	- 0.2 % pts	Not effective due to low temperature constraint for Drying System operation
B2a	Increase of stripper pressure in CO_2 capture unit	- 4.4	+ 0.2 % pts	Good results but higher solvent degradation rates are expected to penalise OPEX
B2b		- 7.3	+ 0.4 % pts	
B3	Staging of solvent regen. in CO_2 capture unit	- 1.7	+ 0.1 % pts	Improvement limited by the high thermal integration in the base CO_2 capture Unit
C1	Expansion of flue gases	+ 5.9	- 0.4 % pts	Mechanical energy recovery from flue gases is more beneficial than cold thermal energy recovery
C2	Refrigeration of compressed CO_2	+ 7.7	- 0.5 % pts	Even an optimised refrigeration system does not overcome auto-refrigerated scheme
C3	CO_2 Liquefaction with CW	- 0.2	~ 0.0 % pts	Partially off-set by the reduction of recoverable compression heat. Sensitive to CW conditions.
D1	Increasing number of stages	- 2.0	+ 0.1 % pts	Partially off-set by the reduction of recoverable compression heat
D2a	Early CO_2 liquefaction (post-combustion)	- 0.2	~ 0.0 % pts	Conventional Chiller power demand off-sets energy savings
D2b	Early CO_2 liquefaction (pre-combustion)	- 7.6	+ 0.3 % pts	Key feature is the use of an absorption chiller for CO_2 liquefaction

Case	Strategy description	Δ Power (1) [MWe]	Δ efficiency (2) [% points]	Main remarks
D2c	CO_2 Liquefaction with CW (post-combustion).	- 3.3	+ 0.2 % pts	Sensitive to CW conditions.
D3a	Deeper inter-cooling (post-combustion)	- 2.6	+ 0.1 % pts	Improvement limited by hydrate formation and CO_2 dew point
D3b	Deeper inter-cooling (pre-combustion)	- 2.0	+ 0.1 % pts	Improvement limited by hydrate formation and CO_2 dew point

(1) Negative values indicates a net consumption reduction with respect to the reference case.
(2) Positive figures indicate an overall efficiency improvement.

3.1 General strategies

The study has assessed some general compression strategies, as summarized in Table 3-2, which could be applied to any of the CO_2 capture processes (pre, post or oxy).

Table 3-2 General – Summary of compression strategies

CASE TAG	Description
Case D1	Increased number of compression stages
Case D2	CO_2 liquefaction instead of gaseous compression
Case D3	Deeper inter-cooling

Case D1: this compression strategy has been evaluated for the post-combustion capture and consists in doubling the number of compression stages (i.e. 8 vs. 4) with respect to the Base Case (B0). Overall, the estimated energy saving has been approximately 2.0 MWe. This net figure includes the adverse effects of higher steam consumption in the Power Island. In fact, due to the higher number of stages, the waste heat from the CO_2 compression is lower and the consequent ST condensate preheating is reduced.

Case D2 has been analysed for both the post-combustion (D2A and D2C) and the pre-combustion capture (D2B).
Case D2A (post-combustion): the early liquefaction of the CO_2 has been estimated by considering the application of a conventional chiller, using propane as working fluid. Overall, the strategy has not led to a significant optimisation of the energy demand, as the estimated net equivalent consumption reduction is approximately 0.2 MW_e. In fact, the significant reduction of the compression energy is off-set by:

- the electrical consumption of the chiller;

- the reduction of the CO_2 compression waste heat available for condensate preheating (increase of the steam demand);
- the higher electrical consumption associated to the increase of Cooling water usage, mainly due to the introduction of the chiller in the system.

Case D2B (pre-combustion): the early liquefaction of the CO_2 has been evaluated with the application of an absorption chiller, due to the large amount of low-grade heat available in the process. The estimated net compression energy reduction has been 7.6 MWe. Also, this strategy has allowed a marginal reduction of the coal thermal input to the gasification plant, since 0.3 % of the Gas Turbines thermal demand is fulfilled by the hydrogen rich gas separated in the liquefaction process.

Case D2C (post-combustion): an assessment has been made for the early liquefaction by using the available cooling water. Overall, this alternative has led to a net equivalent consumption reduction of 3.3 MWe with respect to the Base Case.

Case D3: deeper inter-cooling has been estimated for both the post-combustion (Case D3A) and the pre-combustion alternatives (Case D3B). These options have been evaluated with respect to the possible hydrate formation in the CO_2 stream and the necessity to keep the temperature above the CO_2 dew point. The net equivalent consumption deltas with respect to their respective Base Cases have been estimated respectively equal to 2.6 MWe and 2.0 MWe.

3.2 Pre-combustion strategies

Table 3-3 lists the most relevant compression strategies that have been assessed in the study for this capture type, while Table 3-4 shows the performance delta, in terms of equivalent power demand of each equipment or sub-unit, between the compression strategy and the Base Case (A0). Other two compression strategies, namely the “AGR stripper pressure increase, Case A3” and the “Re-use of waste heat from CO_2 compression, Case A4” have not been fully investigated as they are not technically convenient.

Table 3-3 Pre-combustion – Summary of compression strategies

CASE TAG	Description
Case A1	Vapour recompression in the AGR stripping column
Case A2	Increase of number of flash stages in the AGR

Case A1: The concept behind the vapour recompression strategy is the increase of the CO_2 compression discharge temperature, so to use the higher-grade heat available in the process (e.g. AGR reboiler). With respect to the Base Case, the compression work is higher (+12.9 MWe), due to the increase of the CO_2 average temperature in the compression path, while the LP steam demand is lower, thus leading to an increase of the ST output (6.8 MWe), since a portion of the reboiler heat requirement is supplied by the CO_2 compression.

Case A2: In the Base Case A0, the liquid phase at the bottom of the CO_2 absorber column in the AGR passes through three sequential flash stages: CO_2 Recycle flash, MP flash and LP flash. The vapour phase from the CO_2 Recycle flash flows back to the CO_2 absorber column, while the liquid phase is expanded successively in the MP flash and then in the LP flash. The CO_2-lean solution after the LP flash is recycled back to the CO_2 absorber column. In this way, high-purity CO_2 is recovered and delivered to the CO_2 Compression unit at two pressure levels: 4.8 bara (MP) and 1.2 bara (LP).
This compression strategy consists in considering an additional CO_2 flash stage, located between the CO_2 Recycle flash and the MP flash. Therefore, the CO_2 Compression unit receives three CO_2 streams respectively at 11.5 bara (HP), 4.8 bara (MP) and 1.2 bara (LP). As a consequence, the duty required by the CO_2 compressors is lower (-2.1) than the Base Case, because part of the CO_2 is already available at higher pressure (11.5 bara), which is similar to the discharge pressure of the reference configuration, at the second compression stage discharge.

Table 3-4 Pre-combustion cases –Equivalent Electrical Consumption delta with Case A0

CASE TAG	CASE A1	CASE A2
Cooling water		
CW consumption [MW_e]	- 0.1	~ 0
Thermal integration with AGR		
Solvent regeneration [MW_e]	-6.8	N/A
Compressor Electrical Consumption		
Overall electrical consumption difference [MW_e]	+ 12.9	- 2.1
Overall Plant Electrical Consumption Gap		
TOTAL [MW_e]	**+ 6.0**	**- 2.1**

Note: Negative value indicates lower consumption with respect to the base case

From the figures in the table, it can be drawn that Case A1 strategy is not attractive, while for Case A2 there is a net power consumption decrease of 2.1 MWe.

3.3 Post-combustion strategies

Table 3-5 lists the most relevant compression strategies that have been assessed in the study for this capture type, while Table 3-6 shows the performance delta, in terms of equivalent power demand of each equipment or sub-unit, between the compression strategy and the Base Case (B0). Another compression strategy, namely the "Re-use of waste heat from CO_2 compression, Case B4" has not been fully investigated as it are not technically convenient.

Table 3-5 Post-combustion – Summary of compression strategies

CASE TAG	Description
B1	Vapour recompression in the AGR stripping column
B2	Increase of stripper pressure in the CO_2 capture unit
B3	Staging of solvent regeneration in the CO_2 capture unit

Case B1: The concept behind the vapour recompression strategy is same as the post-combustion capture (Case A1). However, in this Base Case a portion of the CO_2 compression waste heat is already recovered to preheat the Steam Turbine condensate at condensate pump discharge, thus limiting the amount of heat available for the vapour recompression.

Case B2A: This compression strategy consists in increasing the operating pressure of the stripper (210 kPa), so to increase the CO_2 pressure released from the CO_2 capture unit and reduce the overall pressure ratio for the compression unit (-5.5 MWe). The higher stripper operating pressure also induces a lower specific heat requirement for the solvent stripping in the reboiler, since at high pressure (and therefore high temperature) the CO_2 mass transfer rate, throughout the stripper column, is positively affected via the increased driving force. However, higher amine degradation rates is expected and considered in the economic assessment.
Further, the higher stripper operating temperature will require a higher steam pressure at ST extraction. This has a negative impact on the overall performance of the plant and partially off-sets the benefits highlighted above (+1.2 MWe).

Case B2B: same as Case B2A, with a higher stripper pressure (i.e. 260 vs. 210 kPa).

Case B3: The Base Case for post combustion capture already includes a flash of the preheated rich amine to produce a semi-lean amine stream, which is recycled back to the absorber at an intermediate height in the beds packing. Therefore, the concept of staging of solvent regeneration is introduced in this compression strategy as a multi-pressure stripper. The multi pressure stripper operates at three different pressure levels (160 kPa, 230 kPa and 330 kPa), with two additional compressors installed to take the stripping vapour from the bottom pressure level to the top one. The increase of the parasitic power associated to the additional compressors is in part off-set by:

- a significant reduction of the reboiler heat requirement, as part of the stripping is carried out at higher pressure (-13.2 MWe);
- a lower parasitic consumption of the conventional Compression Unit, due to the higher pressure at which the CO_2 is released from the stripper.

Table 3-6 Post-combustion case – Equivalent Electrical Consumption delta with Case B0

CASE TAG	CASE B1	CASE B2A	CASE B2B	CASE B3
Thermal Integration with the Power Plant/CO_2 capture unit				
Steam cons. for Condensate Pre-heating [MW_e]	0	+ 1.2 (2)	+ 2.3 (2)	+ 5.3
Steam cons. for MEA Reboiling [MW_e]	- 9.3			- 18.5
Cooling water				
CW consumption [MW_e]	- 0.4	-0.1	- 0.2	- 0.9
Compressor/Turbine Electrical Consumption				
Overall electrical consumption difference [MW_e]	+ 12.5	-5.5	-9.4	+ 12.4
Overall Plant Electrical Power Gap				
TOTAL [MW_e]	**+ 2.9**	**-4.4**	**-7.3**	**-1.7**

Note 1: Negative value indicates lower consumption with respect to the base case
Note 2: Beneficial effects of reduced ST extraction flow rate overlaps with adverse effects of the different steam conditions. Therefore the approach of the equivalent electrical consumption gap is not applicable to this particular case. The resulting net differential ST output is reported for cases B2A and B2B.

From the figures in the table, it can be drawn that Case B1 is not attractive, whereas for Case B2A, B2B and B3 there is an overall net power consumption decrease.

3.4 Oxy-combustion strategies

The compression strategies assessed for this capture type are listed in Table 3-7, while Table 3-8 shows the performance delta, in terms of equivalent power demand of each equipment or sub-unit, between the compression strategy and the Base Case (C0).

Table 3-7 Oxy combustion – Summary of compression strategies

CASE TAG	Description
Case C1	Expansion of incondensable
Case C2	Refrigeration of compressed CO_2
Case C3	CO_2 Liquefaction

Case C1: In this compression strategy, with respect to the Base Case C0, the incondensable coming from the last CO_2 cooling, at -53°C, are expanded. The expansion to atmospheric pressure reduces the temperature of this stream. The "cold" energy is recovered in the cold box and so the CO_2 expansion request for the auto-refrigeration in the cold box is reduced. As a consequence, the CO_2 exiting the auto-refrigeration system shows a pressure higher than the Base Case, leading to a lower power demand for the last two CO_2 compression stages. On the other hand, the Flue Gas Expander is not required in this configuration, leading to an overall consumption increase of +9.8 MWe.

Case C2: In this compression strategy the CO_2 is refrigerated with an external chiller system, instead of using an auto-refrigeration cycle. After the first two steps of compression and after the Dehydration system, the CO_2 stream enters a train of exchangers, where four chillers cool the CO_2 down to -53°C. After each chiller, liquid CO_2 is separated and collected. Finally, liquid CO_2, with a purity of 96.1% by volume, is pumped up at 111 bara.
The chiller's duty is provided by a conventional cycle refrigeration circuit based on a cascade system. The "warmer" circuit is composed by a two-stage propane compression/expansion system; the propane expansion provides the required duty to the condenser of the "cooler" circuit. On the other hand, the "cooler" circuit is composed by a three-stage ethane compression/expansion system; each expansion stage provides, at different temperature, the chilling power to the CO_2 stream.

Case C3: With respect to the base case C0, in this compression strategy CO_2 is compressed in the last compression stage at 73 bara and firstly cooled against the cold incondensable stream from the cold box, secondly against condensate from the power island and finally condensed with cooling water. The resulting liquid stream is at 19°C and can be pumped up to 111 bara.

Table 3-8 Oxy-combustion cases – Equivalent Electrical Consumption delta with Case C0

CASE TAG	CASE C1	CASE C2	CASE C3
Thermal Integration with the Power Plant			
Condensate Pre-heating [MW_e]	- 2.7	+ 6.0	+ 3.1
BFW heating [MW_e]	0	0	0
IP steam consumption [MW_e]	- 1.2	+ 3.2	+ 1.1
Cooling water			
CW consumption [MW_e]	0	~ 0	+ 0.2
Compressor/Turbine Electrical Consumption			
Overall electrical consumption difference [MW_e]	+ 9.8	- 1.5	- 4.6
Overall Plant Electrical Power Gap			
TOTAL [MW_e]	**+ 5.9**	**+ 7.7**	**- 0.2**

Note: Negative value indicates lower consumption with respect to the base case

From the figures in the table, it can be drawn that Case C1 and C2 are not attractive, while for Case C3 there is a net power consumption decrease of -0.2 MWe.

4 Economic assessment of CO_2 compression strategies

An economic evaluation has been carried out for all the compression strategies, discussed in Section 5, which present a reduction of the electrical consumption associated to the CO_2 compression system. The evaluation is made to assess the economic convenience of each strategy in terms of differential figures with respect to the Base Cases.

Generally, the reduction of the parasitic consumption due to the CO_2 compression system leads to an increase of the electricity export revenue. For each case, the economic convenience of the strategy is evaluated through the calculation of the NPV and IRR, for a given Cost of the Electricity (C.O.E.).

The overall power reduction of each strategy and the main outputs of the economic assessment are summarised in Table 4-1.

Table 4-1 Economic analysis outcome summary

Case tag	Strategy description	Δ POWER [1] [MWe]	Δ CAPEX [M €]	NPV[2]	IRR[2]
A2	Increase of number of flash stages in the AGR	- 2.1	- 3.6	> 0	N/A
B2a	Increase of stripper pressure in CO_2 capture unit	- 4.4	- 5.2	< 0	N/A
B2b		- 7.3	- 10.8	< 0	N/A
B3	Staging of solvent regeneration in CO_2 capture unit	- 1.7	+ 6.8	< 0	0.3 %
C3	CO_2 Liquefaction with CW	- 0.2	- 1.6	> 0	N/A
D1	Increasing number of stages	- 2.0	- 2.1	> 0	N/A
D2a	Early CO_2 liquefaction post comb.	- 0.2	- 8.2	> 0	N/A
D2b	Early CO_2 liquefaction pre comb.	- 7.6	+ 5.2	> 0	30.4 %
D2c	CO_2 Liquefaction with CW post comb.	- 3.3	- 1.7	> 0	N/A
D3a	Deeper inter-cooling post comb.	- 2.6	+ 4.7	> 0	10.6 %
D3b	Deeper inter-cooling pre comb.	- 2.0	+ 4.9	< 0	5.9 %

(1) Negative values indicates a net consumption reduction with respect to the reference case.
(2) Based on a lost export electricity revenue of 3.8 €c /kWh

All the strategies that present a Net Present Value greater than zero (highlighted in green) may be considered techno and economically attractive.

It is noted that most of the compression strategies show an investment cost lower than the Base Case. This is mainly due to a more compact compressor design, which results in a significant reduction of the overall CAPEX requirement.

For some specific cases it is worth to draw the following comments:

- Case A2: the good economics shown by the strategy of increasing the flash stages number in the AGR are essentially driven by an integrated approach, as far as AGR and Compression Unit designs are concerned. In fact, the additional CO_2 flash stage is introduced at a pressure that is very close to the second compressor stage discharge condition, thus avoiding design complications to the compressor itself.
- Case B2A and B2B: from the technical point of view the strategy of increasing the stripper operating pressure is one of the most promising alternatives, whereas its economics are not attractive. This is explained through the significant impact that the higher solvent degradation has on the overall OPEX of the plant. However, it is noted that these solvent degradation rates have been taken from literature data, so figures should be confirmed by referenced Licensors of the technology.
- Case D1: the increase of compression stage numbers show both CAPEX and OPEX improvements. Further increase of the stages number would theoretically lead to improved economics; however, the resulting further drop of the single stage compression ratio may not be acceptable for centrifugal machines, thus making this strategy not technically viable.
- Case C3 and D2: All the CO_2 liquefaction strategies have NPVs greater than zero, showing that these solutions are economically attractive. However, the convenience of this strategy needs to be evaluated in conjunction with the cost of the pipeline, especially in warmer climates and for long transport distances, where either proper insulation/burying are required to keep the CO_2 below its critical temperature or the pipeline design needs to take into account drastic physical properties changes as the dense phase CO_2 is heated while it travels along the line.
- Case D3a: the deeper inter-cooling in the post-combustion capture show a positive NPV. However, this represents a border line situation as indicated by the estimated IRR (10.6 %), which is close to the discount rate (10 %), assumed as basis for the study. Either uncertainties on the cost estimate or slight changes to the basic economic factors (i.e. cost of the consumed electricity) may affect the attractiveness of this strategy from a techno-economic point of view.

5 Compression equipment survey

5.1 Types and sizes

In the past years, reciprocating compressors have been conventionally used for the compression of CO_2. Nevertheless, this technology has revealed several limits, mainly due to the low range of capacities that such machines can handle, typically from 25,000 kg/h to 40,000 kg/h, i.e. an order of magnitude less than the capacity required by the large-scale industrial plants assessed in this study. Nowadays, the technology of reciprocating compressors is leaving the space to centrifugal compressors, which by far represent the current state of the art for CCS applications.

For this specific study, Foster Wheeler has contacted the following Vendors: MAN Turbo, Rolls-Royce, General Electric, Elliott and Siemens. MAN Turbo and Rolls-Royce have provided full cooperation on the study, while partial reply only has been received from GE. On the other hand, Elliott and Siemens have decided of not supporting the study.

Usually, the centrifugal compressors can be categorized into two main branches, namely called "single shaft in-line between bearings" and "multi-shaft integral gear type". In both cases; machines are basically designed according to the International code API 617(Axial and centrifugal compressors for petroleum, chemical and gas industry). By comparison with reciprocating machines, the centrifugal compressors offer:

- Higher efficiency.
- Greater reliability (typically in the range of about 97% for integral gear compressor and 99% for in-line compressors).
- Extended intervals between overhauls.
- Direct couple to the high speed driver, via either steam turbine or electric motor.

Typically, the design of the centrifugal compressors is such that the maximum inlet flow is driven by the inlet Mach number limit (about 0.9), which is imposed to avoid aerodynamic issues. This parameter, in conjunction with the molecular weight of the gas (about 44 for CO_2), defines the max allowable relative inlet velocity to the impeller and thus the maximum axial inlet velocity, assuming that the maximum peripheral speed is also limited by the mechanical strength and deformations, due to the centrifugal force.

Within the family of the centrifugal compressors, the configuration and the characteristics of the "in-line" machines are somehow different from the "integral-gear" machines, due to their intrinsic design. In a traditional in-line compressor, all the impellers are shrunk-on the shaft and consequently once the shaft speed is defined, it is the same for all the impellers enclosed in the casing. Nevertheless, the compressor manufacturers have standardised the size of casing and the maximum numbers of impellers for each casing and within a pre-defined range of pressure.

The main advantage of the “in-line” compressors over the “integral-gear” is related to maintenance access, since the “in-line” machine configuration allows the inner bundle of barrel casing or upper casing of axially split machine to be easily inspected from the end (barrel casing) or from top (horizontal split casing), generally without disturbing the process gas piping. Other technical advantages of the in-line type are as follows:

- Higher operating flexibility, due to the multiple parallel trains configuration, being the turndown capability of the single train very similar for both types. Also, the VFD provided with the in-line machines ensures better efficiency (i.e. lower parasitic consumption) when the plant operates at partial load. This is a very important feature since it is expected that CCS power plants will be required to operate in the actual electricity market, responding to the normal daily and seasonal variability of electricity demand.
- Higher reliability, typically by 2% with respect to the integral-gear compressors.
- Lower mesh losses, since the bull gear in the integrally geared solution introduces additional losses.
- Generally lower power demand.
- Reduced impact on the electrical system design. The impact of using large motors is mainly represented by the necessity for a significant over design of the electrical systems equipment (transformers, cables, etc.) to support the peak current demand at motor starting. For the in-line compressor, smaller size (roughly half) for the largest motors has been proposed with respect to the integral–gear compressors. Also, VFD’s have been included for in-line machines capacity control, which are expected to perform better in smoothing the peak demand at motor starting than the soft starters proposed with the integral gear type.
- Higher flexibility in dealing with uncertainties (e.g. process upset, changes in operating conditions with time) once the machine is built. However it is noted that, in the integral gear concept, reducing the number of drivers and modifying the design of impeller/volute or pinion speed are viable modifications to face changes to the operating conditions.

On the other hand, the construction of an “integral gear” compressor is based on a single bull gear coupled to driver which rotates up to five shafts at the end of which are shrunk-on the impellers; each shaft has its own speed that is defined by the number of teeth of the pinion.
The main advantages of the integral gear compressor over the “in-line” can be summarised as follows:

- Optimum impeller flow coefficient and volute, due to the fact that optimum speed can be selected for each pair of impellers.
- Design facilities of impellers such as small hub/tip ratio, shrouded or non-shrouded version available.
- Inter-cooling connection facilitated after each stage (impeller), being each impeller enclosed in its own casing.
- External connection after each stage offers the possibility to define the level of pressure with minor impact on the compressor arrangement (an in-line compressor may need to change the configuration, i.e. number of impellers or casings).
- The general arrangement is such that the compressor rotors and impellers are located all around the bull gear, making the machine design compact and taking space in vertical / radial directions rather than in axial direction. However this advantage may be smoothed as integrally geared concepts typically have more coolers and scrubbers than inline systems.
- Typically lower investment cost.

Among the leading manufacturers on the market, Rolls-Royce, GE and MAN Diesel & Turbo have demonstrated interest in CCS applications. Rolls-Royce is one of the most well-known manufacturers of traditional "in-line" centrifugal compressors, whereas MAN Diesel & Turbo and GE can offer both types of centrifugal compressors. However, for most CO_2 applications MAN Diesel & Turbo propose the multi-shaft integral-gear design.

5.2 Machinery selection for the Base Cases

Vendors demonstrating interest in this study have proposed their machinery selection for the specified base cases (ref. 2.1).
It is generally noted that the Vendors have shown the willingness to increase the number of inter-cooling steps for either avoiding technical issues related to high discharge temperatures or improving the compressor performance. On the other hand, the Oxy-fuel Combustion and the Post Combustion baseline cases include a strong thermal integration with the Power Island, as the heat available at the CO_2 compression stages outlet is recovered into the Steam Condensate / Boiler Feed Water systems and the Steam Turbine Island, which requires relatively high discharge temperatures. For this reason, the manufacturers have been requested to propose for each of the two processes, two options as far as inter-cooling arrangements are concerned:

1. Configuration as close as possible to the original specification, to allow the thermal integration as implemented in the base case configuration.
2. Configuration with an optimised number of inter-cooling steps, to best suit the selection to the standard machine frames available. In this case the

resulting compressor power demand reduction is partially off-set by the consequent decrease of the waste heat available for recovery in the thermal cycle.

For the post-combustion process, an additional case with an increased number of stages and inter-cooling steps has been also requested to the Vendors in order to get a feedback regarding the associated compression strategy (ref. 3.4).
Table 2-2 provides a summary of the cases considered for machinery selection in the market survey. Reference is generally made to 2.1 for the definition of the operating envelopes.

Table 5-1 Cases summary for machinery selection

Case	Description
Pre-combustion	Operating envelope as defined by base case A0. No particular restrictions on inter-cooling steps, since compression heat is not recovered in the process.
Post-combustion inter-cooling as specified	Operating envelope as defined by base case B0. Regarding inter-cooling, Vendors are requested to keep as close as possible to the original specification.
Post-combustion optimised inter-cooling	Operating envelope as defined by base case B0. Regarding inter-cooling steps, Vendors are free to optimise the selection.
Post-combustion Increased stages	Operating envelope as defined by case D1 (ref. 3.1).
Oxy-fuel inter-cooling as specified	Operating envelope as defined by base case C0. Regarding inter-cooling, Vendors are requested to keep as close as possible to the original specification.
Oxy-fuel optimised inter-cooling	Operating envelope as defined by base case C0. Regarding inter-cooling steps, Vendors are free to optimise the selection.

5.2.1 Proposed configurations and absorbed power figures

Table 5-2 provides main information on train arrangement and machine selection undertaken by the different manufacturers involved in the study, and summarizes the absorbed powers for each case. It is noted that the performance figures provided by the compressor manufacturers are for the purposes of the study only and do not represent performance guarantees. All manufacturers have included flange to flange losses, indicating the shaft power at the driver.

Table 5-2 Configuration proposed and shaft power (kW)

Case	Rolls-Royce	MAN Diesel & Turbo	GE
Pre-combustion	40,540 kW Trains: 1x100% 4 in-line machines 13 compression stages, 6 inter-cooling steps	43,960 kW Trains: 1x100% 1 integral gear machine 8 compression stages, 7 inter-cooling steps	41,000 kW Trains: 2x50% 2 integral gear machine #.stages not avail. 5 inter-cooling steps
Post-combustion inter-cooling as specified	50,460 kW Trains: 2x50% 4 in-line machines/train 13 compression stages, 3 inter-cooling steps T range: 80÷170 °C (1)	53,160 kW Trains: 1x100% 1 integral gear machine 6 compression stages, 3 inter-cooling steps T range: 85÷180 °C (1)	53,160 kW Trains: 2x50% 4 in-line machines #.stages not avail. 3 inter-cooling steps T range: 60÷180 °C (1)
Post-combustion optimised inter-cooling	44,410 kW Trains: 2x50% 4 in-line machines/train 12 compression stages, 5 inter-cooling steps T range: 60÷115 °C (1)	48,810 kW Trains: 1x100% 1 integral gear machine 6 compression stages, 5 inter-cooling steps T range: 80÷100 °C (1)	–
Post-combustion Increased stages	43,490 kW Trains: 2x50% 4 in-line machines/train 13 compression stages, 6 inter-cooling steps T range: 50÷150 °C (1)	45,650 kW Trains: 1x100% 1 integral gear machine 7 compression stages, 6 inter-cooling steps T range: 65÷100 °C (1)	44,480 kW Trains: 2x50% 2 integral gear machine #.stages not avail., 7 inter-cooling steps T range: not avail. (1)
Oxy-fuel inter-cooling as specified	74,100 kW Trains: 3x33%+1x100% 7 in-line machines 12 compression stages, 3 inter-cooling steps T range: 65÷280 °C (1)	71,540 kW Trains: 2x50%+1x100% 3 integral gear machines 9 compression stages, 4 inter-cooling steps T range: 65÷185 °C (1)	72,280 kW Trains: 2x50%+1x100% 5 in-line machines #.stages not avail., 4 inter-cooling steps T range: not avail. (1)
Oxy-fuel optimised inter-cooling	62860 kW Trains: 3x33%+1x100% 7 in-line machines 14 compression stages, 7 inter-cooling steps T range: 70÷135 °C (1)	64340 kW Trains: 2x50%+1x100% 3 integral gear machines 9 compression stages, 8 inter-cooling steps T range: 65÷95 °C (1)	–

Notes:

1) The stage discharge temperatures range is indicated only for oxy-fuel and post-combustion processes, in which the compression heat is recovered through thermal integration with the Power Island.

Generally, for the Base Cases the table shows absorbed powers lower than the figures estimated in the study (ref. 2.1), as summarized in the following:

- Oxy-fuel: (from 7% to 21% lower)
- Post-combustion: (from 8% to 23% lower)
- Pre-combustion: (from 7% to 14% lower).

This is mainly due to both the higher stage efficiencies proposed by the Vendors with respect to the assumptions made in the reference studies and, in most cases, the use of additional inter-cooling steps.

For the pre-combustion case, the reduction in terms of power consumption would be entirely reflected into a net power output improvement, since the CO_2 compression waste heat is disposed to the cooling water only, i.e. there is no recovery of waste heat from the CO_2 compression.

As far as Oxy-fuel combustion and Post Combustion cases are concerned, the reduction of power consumption due to the higher efficiency or increased inter-cooling would be partially off-set by the consequent decrease of compression heat available at the CO_2 compression stage outlet, a fraction of which is recovered into the Steam Condensate / Boiler Feed Water systems of the Boiler / Steam Turbine Island. This is particularly true for the machinery selections in which the Suppliers have included additional inter-cooling steps with respect to the original design, in order to optimise the selection and minimise compressor electrical consumption.

The feedback from MAN Diesel & Turbo for the oxy-fuel combustion baseline case indicates that integral gear machines may have some technical limitations in handling the first un-cooled section of the CO_2 compression process, due to the high discharge temperature. Hence, with the integral gear compressors proposed by MAN Diesel & Turbo, the deep thermal integration with the Power Island can not be implemented as foreseen in the reference case. The compressor itself would have lower parasitic load but elsewhere in the plant (e.g. Steam Turbine Island) there will be an increased thermal energy demand.

Generally, in terms of performance evaluation, it would be a mistake to draw conclusions based just on the compressor electrical consumption. The performance of the machines has to be considered in the context of overall plant performance, which is influenced by the different integration into the overall process depending on the proposed configuration.

For the post combustion capture, the comparison between 4 stages and 8 stages from Vendors data confirms the beneficial effects in terms of electrical consumption reduction, as expected in case D1 (ref. 3.1)

5.2.2 <u>Budget costs</u>

An indication of the expected specific cost range for each machinery class is included in this report, based on Foster Wheeler in-house information and specific judgements originated from a range of sources. The cost ranges, reported in the following table, cover the selections for the operating envelope given in the different applications investigated (pre-combustion, post-combustion and oxy-fuel combustion).

Table 5-3 Specific investment cost range for each machine type.

Type	Specific cost range
In-line centrifugal (Rolls Royce)	600÷900 €/kW
Integral-gear (MAN Diesel & Turbo)	300÷600 €/kW

Table 5.2 generally shows a higher investment cost for in-line centrifugal machines than integral-gear, the delta cost being mainly justified by the following reasons:

- In-line centrifugal compressors need multiple train solutions;
- More compact design of the integrally geared centrifugal compressors, which results in a lower machine investment cost;
- Variable Speed Drivers are required for capacity control at partial load in the in-line machines, whereas the integral-gear type is supplied with Inlet Guide Vanes.

Despite the generally higher investment cost, the “in-line” machines offer the following technical advantages over the integral-gear type: better maintainability, higher operating flexibility, improved reliability, reduced impact on the electrical system design, as reported in section 5.1.

Therefore, from the indications reflected in this report, it is not possible to draw any definitive conclusion on the economics of the different machine types. The selection of the machine, as usual, is case-specific and not driven by machine investment cost only. Other features like reliability, flexible operation, easy maintainability and associated impacts on other systems in the plant shall be also accurately assessed. Also, the cost of the machines has to be considered in the context of overall plant cost and performance, which may differ as each requires slightly different integration into the overall process.

6 Novel concepts for CO_2 compression

Two novel CO_2 compression concepts, which may find potential application in CCS plants in the next few years, have been assessed in the study.

6.1 Ramgen technology

Ramgen is based on the supersonic shock wave compression, using the same principle as a supersonic aircraft, where the engine forward motion is used to compress the air. In the Ramgen compressor, a rotating disc simulates forward motion of the aircraft: it spins at high speed to create a supersonic effect, like the centre-body in a supersonic aircraft. The fluid enters through a common inlet, flows into the annular space between the disc and the casing, where the three raised sections create shock waves. The shock waves generate a pressure increase and compress the fluid.

The compression is developed into two stages, with a pressure ratio of approximately 10:1 and can be fitted with an inter-cooler and after-cooler, depending on the application. Ramgen claimed high efficiency for their compressors (about 87%), because of the relatively simple design, with a low number of leading edges that reduce the drag and, therefore, minimize the losses. Other advantages over other compressor types are: the high pressure ratio per stage, which reduces the footprint, and the possibility to use the high-grade waste heat in the other process units, due to the elevated discharge temperature (typically 240°C).

The main performance parameters of this novel compressor are shown in Table 6-1. An evaluation of the overall performance impact of the Ramgen compressor in the plant (see Table 6-2) has been performed through a comparison in the post-combustion case with the integrally geared centrifugal compressor, proposed by MAN Diesel & Turbo, and with the in-line machines configuration with optimised inter-cooling, proposed by Rolls Royce, the latter being the minimum power demanding scheme among the centrifugal options (ref. 5.2.1).

Table 6-1: Ramgen performance

Item	LP	HP	Total
Inlet pressure – bara	1.62	12.90	
Outlet pressure – bara	14	111.5	111
Pressure ratio	8.6	8.6	68.5
Stage efficiency – isentropic%	86.5	86.5	
Volume flow – m3/hr	71161	7690	
Discharge temperature - °C	236.5	225.3	
Total power – KW	29988	25892	**55880**
Motor power – KW	33000	28500	
Polytropic efficiency - %	89.3	89.8	

Table 6-2: Ramgen performance delta with respect to centrifugal compressors

Ramgen Thermal Integration with the Power Plant / CO_2 capture unit		
	Comparison with integral-gear machine (inter-cooling as specified)	Comparison with in-line machine (optimised inter-cooling)
Steam cons. for Condensate Pre-heating	+ 9.4 MW_{th} = + 2.5 MW_e	-7.1 MW_{th} = - 1.9 MW_e
Steam cons. for MEA Reboiling	- 35.3 MW_{th} = - 9.3 MW_e	- 35.3 MW_{th} = - 9.3 MW_e
Cooling water		
CW consumption	~ 0.0 t/h = ~ 0.0 MW_e	- 1905 t/h = - 0.2 MW_e
Compressor Electrical Consumption		
Overall electrical consumption difference	+ 5.4 MW_e	+ 1.1 MW_e
Overall Plant Electrical Power Gap		
TOTAL	**- 1.4 MW_e**	**- 0.3 MW_e**

The comparative analysis has been undertaken on the basis of a moisture spec of 50 ppmv, achieved through the absorption process of a TEG unit. In fact, it is recognised that, with the 10% recycle assumed for the basic Dryer configuration (i.e. adsorption in desiccant beds, ref. 2.2), Ramgen power penalty would be aggravated by the added mass flow compressed from lower pressures, the result of fewer discrete

stages to work with. Having assessed that, for the given basic moisture specification, a TEG system can be successfully applied to both Ramgen compressor and conventional machines, requiring no CO_2 recycle (ref. 2.2), it is acceptable to deviate from the basic configuration selected for the study, as far as Ramgen performance evaluation is regarded.
Furthermore, Ramgen compressor shows unique potential for combination with the HOC (Heat Of Compression) Drying system, developed by SPX, to achieve moisture content even lower than 10 ppmv in the dried stream. The HOC Dryer uses the available heat at LP compressor stage discharge as main source for the regeneration of the soild bed with no need for recycling part of the dried CO_2. This is an important feature, as Ramgen concept has the potential to work with low moisture spec (e.g. in the CO_2 purification process of the oxy-fuel combustion technology) without major energetic penalties.

Table 6-2 shows that the integration of the Ramgen concept in the post combustion scheme has the potential to improve the overall plant performance with respect to the centrifugal compressors. It has to be noted that the net reduction of the equivalent compression parasitic load is diminished but not overtaken even when compared to the in-line machines configuration with optimised inter-cooling, the minimum power demanding scheme among the centrifugal options.

Regarding this comparison with more conventional machines, the two following factors should be taken into consideration:

- The low cooling water temperature assumed for this study encourages compression staging rather than the de-staging approach proposed by Ramgen. As a matter of fact, Ramgen provided performance figures also with cooling water at 30°C, showing a parasitic load increase of 3.5% only with respect to the 12°C case, whereas for centrifugal compressor the expected penalty would be approx 5÷6%.
- The significant potential for high grade heat recovery offered by Ramgen is not fully exploited in the present analysis for the post combustion process, as the MEA stripper reboiler operates at about 120 °C, i.e. over 100°C less than compression discharge temperature

Ramgen have provided also budget cost of the proposed selection, which shall be deemed as preliminary only. As per the previous cost information, only an indication of the expected specific cost range is included in this report. Based on the budgetary information received, the specific cost for the Ramgen compressor is expected to be in the range 170 ÷ 280 €/kW.

From the quoted figures, it can be drawn that the Ramgen compression strategy has potential to offer not only lower cost and higher simplicity than the other compressors type, but also a lower power demand of the whole system. However, it is noted that the Ramgen compressor concept is not yet a proven technology. Further development and testing are required to demonstrate its capability at commercial scale.

Ramgen have recently developed a test program on the frame HP-16, which can support a CCS power plant in the capacity range of 200 - 250MW$_e$. These tests are scheduled to start on 2nd quarter 2011.
On that machine Ramgen expect to be able to offer commercial performance guarantees and terms by 1st quarter 2012.

Further development and test activities will then be needed on HP-32 and LP-48, which are the largest anticipated size for approx. 800MW$_e$ CCS applications, for HP and LP stages respectively.

6.2 Axial machine at the front end of CO_2 compression

In general, axial compressors can handle a much higher flowrate than the centrifugal compressors and with a higher efficiency. Therefore, a possible novel compression concept is represented by the use of a single train axial machine for the initial compression of the CO_2, generating a lower volumetric flowrate at compressor discharge and allowing the use of a single train, integrally geared machine, for the final compression step.

The same Vendors that supported the study have been also asked to provide some feedback regarding the feasibility of this concept for the post combustion capture case.

Rolls Royce stated that, though they used axial compressors extensively in Aero Engines, these machines are not available at the moment and there are no plans for their development. Rolls Royce believes the design flexibility given by their current, pre-customised centrifugal approach is preferable. If the volume flow rates are higher than the maximum capacity of this system, then parallel trains would be preferred, also showing improved turndown capability/redundancy.

With reference to MAN Diesel & Turbo, it is noted that they already selected a single train machine (integrally geared compressor) for the post combustion case, thus making this axial-based configuration loose the main potential advantage. However, in the past they offered axial flow compressor solutions for CO_2 applications, especially where the duty was too high for centrifugal designs, or the adiabatic heat of compression was required by the process.

Based on the information from MAN Diesel & Turbo, it has been possible to estimate the impacts on the overall performance of the unit with respect to their integrally geared centrifugal solution. The expected overall consumption reduction is approximately 2.5 MWe, the major contribution being the equivalent gain from the Steam Turbine output in the Power Island and not from the compressor shaft power itself. In fact, the use of an un-cooled compressor at the front end makes more compression heat available, thus reducing the steam requirement for the ST condensate preheating in the Power Island.

On the other hand, in terms of investment cost, MAN Diesel & Turbo stated this is a much more expensive solution as the complete axial compressor must be manufactured in acid-resistant materials (i.e. similar to the MDT axial machines for nitric acid service), whereas for the integrally geared compressor only the impellers and volutes are in acid resistant materials. This much higher CAPEX leads to the expectation that benefits in terms of lower consumption are more than off-set by the additional investment cost of the system.

7 Summary findings

From the considerations made in this study, the main conclusions that can be drawn are the following:

- In the pre-combustion capture, the overall economics of the plant improve by increasing the number of solvent flash stages in the AGR unit. However, the selection of the number and operating conditions of each flash stage shall be carefully made in relation to the characteristics of the compressor. In particular, it is recommended to introduce additional solvent flash stages at a pressure as much as possible close to the compressor stage discharge conditions, thus avoiding complications in the design of the compressor.

- The strategy of increasing the compression stage numbers show both CAPEX and OPEX improvements. Further increase of the stages number would theoretically lead to improved economics; however, the resulting further drop of the single stage compression ratio may not be acceptable for centrifugal machines, thus making this attempt not technically viable.

- All the CO_2 liquefaction strategies are economically attractive. However, especially in warmer climates, the convenience of this strategy shall be assessed in conjunction with the cost of the pipeline, which needs to be designed for the transportation of either a sub-cooled liquid CO_2 (kept below its critical temperature of 31 °C) or a dense fluid whose physical properties rapidly change as it is heated up along the line.

- Early liquefaction is particularly promising for the pre-combustion CCS, where the compression parasitic load is reduced by approx 16 % with respect to the reference case. In this application, the large amount of relatively low temperature waste heat from the syngas cooling unit is recovered in an absorption refrigeration system, allowing pumping the CO_2 from a pressure of about 40 bara.

- The vapour recompression strategy is not effective in both the pre and post combustion captures. The necessity of cooling down the CO_2 upstream of the drying process implies that, along the compression path, there must be a low temperature point in correspondence of the dryer operating pressure, which prevents the vapour recompression mechanism from being fully effective. In this sense, a performance benefit would be expected for the pre-combustion capture in case of a solvent washing process, based on methanol, which generates CO_2 streams completely free of water.

- The strategy of increasing the stripper operating pressure is one of the most promising alternatives from the technical point of view (i.e. electrical consumption is reduced by up to 13% in the pressure range considered), whereas its economics are not attractive. This is explained through the significant impact that the higher solvent degradation rate (literature data) has on the overall OPEX of the plant.

- With respect to the reference cases, the most promising compression strategies lead to an overall net plant efficiency improvement in the range of 0.1 – 0.4 percentage points for the pre-combustion and 0.1 – 0.2 points for the post combustion. On the other hand, strategies investigated for the oxy-fuel combustion process do not show significant improvements.

- Centrifugal compressors represent the current state of the art for large-scale CCS applications. By comparison with reciprocating machines, which have been conventionally used for the compression of CO_2 in the past years, centrifugal compressors generally offer higher efficiency, improved reliability (typically in the range of about 97% for integral gear compressor and 99% for in-line compressors) and easier maintainability (extended intervals between overhauls).

- The specific investment cost of “in-line” machines is higher than “integral-gear” types, based on Foster Wheeler judgements from a range of sources. However, “in-line” machines offer technical advantages over the “integral-gear” type: better maintainability, higher operating flexibility, improved reliability, reduced impact on the electrical system design. Therefore it is not possible to draw any definitive conclusion on the economics of the different machine types. The selection of the machine, as usual, is case-specific and not driven by machine investment cost only. Also the cost of the machines has to be considered in the context of overall plant cost and performance which may differ as each requires slightly different integration into the overall process.

- The base cases electrical consumptions provided by the Vendors who have supported the study are generally lower than the figures shown in the former IEA GHG reports (from 7 to 23%, depending on the capture type), confirming the developments made in the field of CCS in the last years This is mainly due to both the higher stage efficiencies proposed by the Vendors with respect to the assumptions made in the reference studies and, in most cases, the use of additional inter-cooling steps. For Oxy-fuel combustion and Post Combustion cases, the reduction of power consumption is not entirely reflected into a net power output improvement, due to the decrease of waste heat available at the CO_2 compression stage outlet, a fraction of which is recovered into the Steam

Condensate / Boiler Feed Water preheating systems of the Boiler / Steam Turbine Island.

- The novel compression concept developed by Ramgen has potential to offer not only lower cost and higher simplicity than the other compressors type, but also a lower overall equivalent power demand of the system. However, the Ramgen compressor is not yet a proven technology; further development and testing are required to demonstrate its capability at large commercial scale.

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SECTION B

GENERAL INFORMATION

INDEX

1 Introduction

In the scientific community it is generally recognized that, by year 2030, the world energy demand will increase by 50%, while fossil fuels, mainly coal and natural gas, will continue to supply most of the energy demands. This reality will continue for many years, until the use of renewable energies will increase significantly. On the other hand, the use of fossil fuels is necessarily correlated to the production of carbon dioxide (CO_2), which contributes to global warming. In this scenario, Carbon Capture and Storage (CCS) represents one of the most effective responses to partially reduce CO_2 emissions in the next few years.

For industrial applications with CCS, the power demand of the CO_2 compression unit and the process units that are thermally related to this system contribute significantly to the energy penalties of the plant, thus reducing its overall efficiency. Therefore, any reduction of the electrical consumption of this system may result in an important overall net plant efficiency improvement.

This report summarizes the outcomes of a study executed by Foster Wheeler for IEA-GHG R&D Programme, aimed at identifying the main types of compression equipment, available in the market for CCS applications, and assessing the key characteristics of different compression systems and machinery configurations.

From an energy point of view, for a given final discharge pressure of the carbon dioxide, there are a number of different alternatives that can be considered for the capture and compression unit, corresponding to different power demands and investment cost requirements. This study investigated different compression strategies, making a techno-economic assessment of various alternatives, applicable to the post, pre and oxy-fuel de-carbonisation processes.

A generic overview of the implications of the identified strategies on compressor selection and design has also been performed, on the basis of the operating conditions of the compressors.

Finally, the study made a description of novel compression concepts that are expected to offer high-stage efficiency, identifying their state of development and the strategies for their use in a typical industrial plant with carbon capture and storage.

To show the results of the analyses carried out in the study, this report has been arranged as follows:

- Section A - Executive Summary: provides a summary of the main contents of the study.

- Section B - General Information: shows the main assumptions adopted for the study development and the reference CO_2 compression strategy in power plants with CCS.
- Section C - Evaluation of CO_2 Compression Strategies: presents the alternative strategies investigated in the study, analysing the main characteristics and performance and costs of each configuration.
- Section D - Compression Equipment Survey: illustrates and compares the CO_2 compression machines, available in the market and proposed by specialized Vendors.
- Section E - Novel Concepts for CO_2 Compression: introduces novel and alternative concepts for the compression of CO_2, highlighting their peculiarities and future development.

2 Base of the study

The following sections describe the general design and cost estimating criteria used as a common basis for the development of the study.

2.1 Location and reference ambient conditions

Ambient temperature	:	9 °C
Cooling Water inlet temperature	:	12 °C
Cooling Water outlet temperature	:	19 °C
Location	:	NE coast of the Netherlands (Greenfield)

2.2 Feedstock characteristics

The main fuel of the different alternatives is a bituminous coal, whose main characteristics are listed hereinafter.

	Eastern Australian Coal
	Proximate Analysis, wt%
Inherent moisture	9.50
Ash	12.20
Coal (dry, ash free)	78.30
Total	100.00
	Ultimate Analysis, wt% (dry, ash free)
Carbon	82.50
Hydrogen	5.60
Nitrogen	1.77
Oxygen	9.00
Sulphur	1.10
Chlorine	0.03
Total	100.00
Ash Fluid Temperature at reduced atm., °C	1350
HHV (Air Dried Basis), kcal/kg (*)	6464

LHV (Air Dried Basis), kcal/kg (*)	6180
Grindability, Hardgrove Index	45

(*) based on Ultimate Analysis, but including inherent moisture and ash.

2.3 Carbon dioxide characteristics

The characteristics of the produced carbon dioxide at plant Battery Limits (B.L.) are the following:

Status	:	Supercritical
Pressure (reference)	:	110 bar g
Temperature	:	(1)
Purity		
H_2S content	:	0.1% wt (max)
CO content	:	0.1 % wt (max)
Moisture	:	< 50 ppmv
N_2 content	:	to be minimized (2)
Flowrate	:	corresponding to approximately 85% CO_2 capture from large power plants, ranging from 500 to 750 MWe and according to the reference plant design.

Notes:

(1) Depending on the alternative of the study. Refer to the case-specific report in section C.

(2) High N_2 concentration in the CO_2 product stream has a negative impact for CO_2 storage, particularly if CO_2 is used for Enhanced Oil Recovery (EOR). N_2 seriously degrades the performance of CO_2 in EOR, unlike H_2S, which enhances it.

2.4 Reference Reports

For each combustion capture type, a Base Case has been identified and used as reference to carry out the comparison with alternative CO_2 compression strategies, which have the potential for lower parasitic loads.

The base cases have been derived from previous studies undertaken for the IEA-GHG R&D Programme in the past years on CCS-related topics. The following table provides a summary of the reference cases for each capture technology.

Table 2-1 Base cases summary

Technology	**PRE-COMBUSTION**	**POST-COMBUSTION**	**OXY-FUEL COMBUSTION.**
Type of plant	IGCC	USC-PC	USC-PC
IEA GHG Report ref.	Report PH4/19 [1] Case D4	Report PH4/33 [2] Case 4	Report PH2005/9 [3] Case 2
Gross Power output [MWe]	942.1	827.0	737.0
Net Power output [MWe]	705.0	666.0	532.0
Base Case Tag	A0	B0	C0
CO_2 capture rate	85%	85%	90%
CO_2 compression parasitic load [MW]	47.4 (1)	57.7 (1)	79.3 (1)

Note 1: Re-calculated for the present study through process simulation of the compression unit

3 Base Cases

De-carbonisation processes in fossil fuel-based power plants fall in one of the following three main categories (Figure 3-1):

1) **Post-combustion**: CO_2 separation from flue gases, through a capture process at boiler back-end, with minor modifications of the conventional plants.
2) **Pre-combustion**: CO_2 separation from a synthesis gas, downstream a water gas shift reactor that converts CO and H_2O to CO_2 and H_2. This solution implies the re-allocation of the heating value contained in the original feedstock in a "de-carbonized" fuel (hydrogen) that feeds the power cycle, after carbon removal.
3) **Oxy-combustion**: CO_2 concentration in the exhaust gases. In this case, the energy conversion process is modified by using oxygen combustion instead of air and suitable techniques are applied, so that CO_2 can be removed at a convenient stage of the process with a high purity degree.

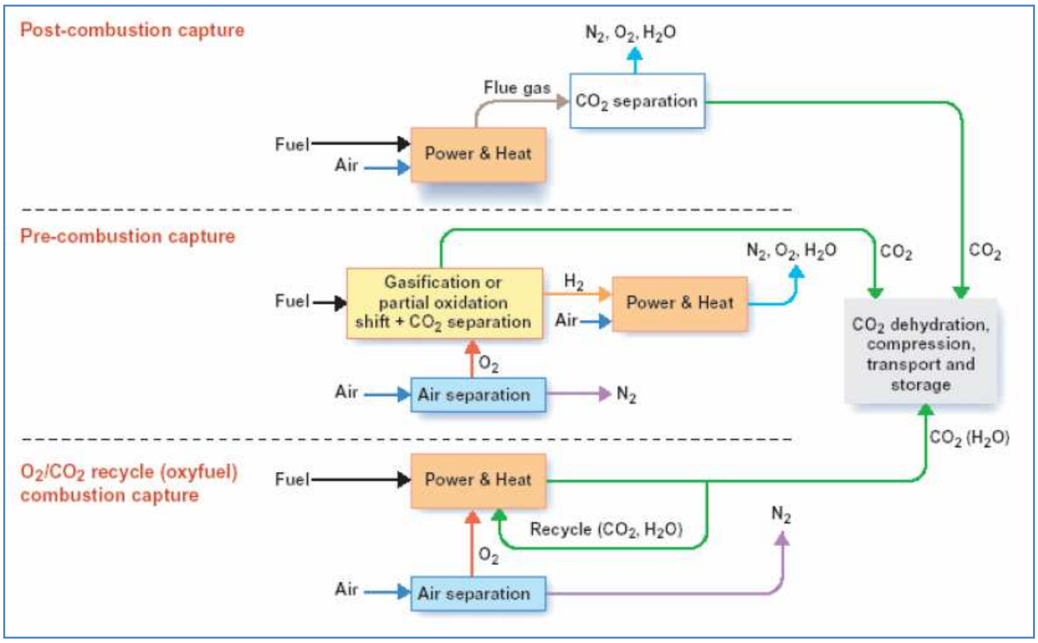

Figure 3-1 De-carbonization schemes in industrial plants

For each of the above CO_2 capture schemes, the following sections show the main technical features of a "Base Case" configuration, as taken from the technical reports published by IEA GHG in the past year.

3.1 Pre-combustion: Base Case (A0)

3.1.1 Overall power plant process description

The IGCC plant is a power production facility that converts coal to electric energy with a minimum impact to the environment. The key process step of the IGCC plant is coal gasification. Gasification is the partial oxidation of coal, or any other heavy feedstock, to a gas, often identified as syngas, in which the major components are hydrogen and carbon monoxide. The fuel also contains other elements such as nitrogen (N_2) and carbon dioxide (CO_2) and small quantities of H_2S (hydrogen sulfide), chlorine, NH_3, HCN that are removed from the syngas in the treatment section before combustion in the gas turbine.

The IGCC Complex is a combination of several process units. The main process blocks of the plant are the following:

- Coal milling and gasifier feed preparation;
- Air Separation Unit;
- Gasification Island;
- Syngas treatment and conditioning;
- Acid Gas Removal / CO_2 capture unit;
- Sulphur recovery and Tail gas treatment;
- Combined Cycle power generation.

These basic blocks are supported by other ancillary units and a number of utility and offsite units, such as cooling water, flare, plant/instrument air, machinery cooling water, demineralised water, auxiliary fuels, etc.

The oxygen required by the gasifier comes from the Air Separation Unit (ASU), which performs cryogenic separation of ambient air into high purity oxygen and nitrogen streams. Air is compressed and cooled to low temperature such that oxygen and nitrogen, which have different boiling points (respectively –183°C vs. –195.8), can then easily be separated from each other by fractional distillation. The nitrogen is mainly used to dilute the syngas before combustion in the gas turbine.

The syngas generated by the gasification enters the shift reactors. In the water-shift process, syngas and water are mixed in the presence of a catalyst to convert CO into CO_2 and H_2 in an exothermic reaction, accordingly to the following:

$$CO + H_2O \rightarrow CO_2 + H_2$$

A hydrogen-rich stream is then produced, cooled by means of steam generation at four pressure levels and delivered to the AGR where CO_2 is captured by using a physical solvent washing process (in this case Selexol licensed by UOP). From the AGR, the following main streams are produced at the unit battery limits:

- H_2S rich gas: this stream is sent to the Sulphur Removal Unit (SRU) for production of sulphur.
- De-carbonized fuel: this stream, mainly H_2, re-enters the syngas treatment and conditioning unit for final heating and mixing with either nitrogen or water, before combustion in the gas turbine of the combined cycle.
- CO_2: two CO_2 streams are produced and sent to the CO_2 compression unit.

Inside the AGR, the CO_2 loaded solvent is flashed into two consecutive steps to recover CO_2. The two flashes are disposed sequentially and expand the loaded solvent at two different pressure levels. The gas streams exiting the flash drums are mainly composed of CO_2, while the liquid regenerated solvent exiting the last flash drum is recirculated to the CO_2 absorber.

For the present study, as far as the AGR configuration and performance are concerned, the data of the reference report have been updated with a new set of latest information.

3.1.2 CO_2 compression unit description

The following description makes reference to the simplified process flow diagram shown in Figure 3.1 and to the Heat and Material Balance in Table 3.1, Table 3.2 and Table 3.3.

Two CO_2 streams exit the AGR unit and enter the CO_2 compression unit, where they are dehydrated and compressed before transportation and storage into the final destination.

The first stream at low pressure is at 1.2 bara, -5°C and presents a CO_2 concentration of 99.8% by volume LP CO_2 stream flows to a first KO drum that avoids water droplets, possibly condensed in the pipeline between the two units, entering the compressor. LP CO_2 stream enters the first compression stage where it is compressed up to 5 bara and then cooled with CW to 19°C.

At the exit of the inter-cooler, the compressed CO_2 stream is mixed with the second CO_2 stream coming from the AGR. This latter stream, MP CO_2, is at 4.8 bara, 1°C and presents a CO_2 concentration of 97.3% by volume.

The combined LP and MP CO_2 streams flow to a second KO drum, where condensated water is separated, while the gaseous phase flows to the second and third stages of the compression, intercooled with CW. After this latter compression stage, the stream at 34 bara is again cooled to 19°C. Water condensate is separated in a KO drum, while the gaseous phase enters the CO_2 Dehydration System.

The CO_2 Dehydration System is sized to reduce the water content of the CO_2 stream to 50 ppmv to meet the typical transmission network specification. The system is composed by two (or more) solid-bed dessicants (typically molecular sieves or activated alumina). While one bed adsorbs water from the wet CO_2 stream, the other bed is in regeneration mode. Regeneration of the saturated bed is carried out using a portion (typically around 10%) of the dried gas product. The water in the bed is desorbed by thermal swing (heating up the regeneration stream to 250°C). The stream used for regeneration is then flashed to recover water and the remaining wet CO_2 stream is recycled back at the outlet of the first stage compression.

The dry CO_2 is further compressed in the fourth compression stage to 70 bara and cooled with CW to 30°C. Before leaving the unit, the stream is finally compressed in the fifth compression stage to 111 bara and cooled with CW to 40°C.

Figure 3-2 Pre-combustion - Base case A0: CO_2 Compression Unit scheme.

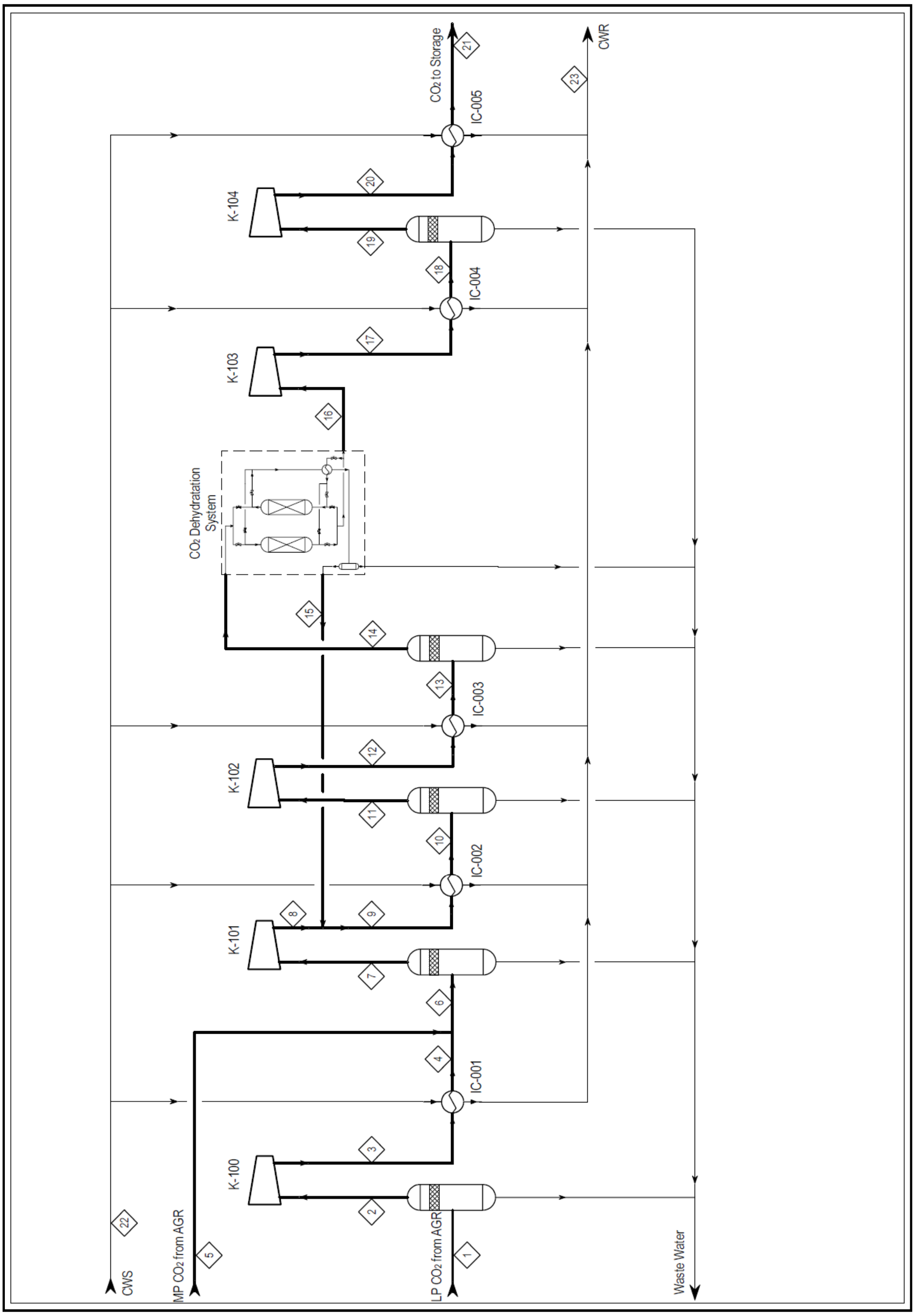

Table 3-1 Pre-combustion – Base case A0: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Pre-Combustion - Case A0										
STREAM	1	2	3	4	5	6	7	8	9	10
Temperature (°C)	-5	-5	121	19	1	7	7	88	82	19
Pressure (bar)	1.2	1.2	5.0	4.8	4.8	4.8	4.8	12.0	12.0	11.8
TOTAL FLOW										
Mass flow (kg/h)	208775	208775	208775	208775	411177	619951	619951	619951	689122	689122
Molar flow (kgmole/h)	4751	4751	4751	4751	9569	14320	14320	14320	15920	15920
LIQUID PHASE										
Mass flow (kg/h)										
GASEOUS PHASE										
Mass flow (kg/h)	208775	208775	208775	208775	411177	619951	619951	619951	689122	689122
Molar flow (kgmole/h)	4751	4751	4751	4751	9569	14320	14320	14320	15920	15920
Molecular Weight	43.95	43.95	43.95	43.95	42.97	43.29	43.29	43.29	43.29	43.29
Composition (vol %)										
CO_2	99.77	99.77	99.77	99.77	97.30	98.12	98.12	98.12	98.09	98.09
CO	0.01	0.01	0.01	0.01	0.20	0.14	0.14	0.14	0.14	0.14
H_2S+COS	0.02	0.02	0.02	0.02	0.01	0.01	0.01	0.01	0.01	0.01
H_2	0.04	0.04	0.04	0.04	2.34	1.57	1.57	1.57	1.57	1.57
N_2	0.00	0.00	0.00	0.00	0.03	0.02	0.02	0.02	0.02	0.02
Ar	0.00	0.00	0.00	0.00	0.06	0.04	0.04	0.04	0.04	0.04
H_2O	0.17	0.17	0.17	0.17	0.07	0.11	0.11	0.11	0.13	0.13

Table 3-2 Pre-combustion – Base case A0: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Pre-Combustion - Case A0										
STREAM	11	12	13	14	15	16	17	18	19	20
Temperature (°C)	19	118	19	19	28	24	94	40	40	80
Pressure (bar)	11.8	34.0	33.8	33.8	12.0	32.9	70.0	69.8	69.8	111.2
TOTAL FLOW										
Mass flow (kg/h)	689122	689122	689122	689090	69171	619775	619775	619775	619775	619775
Molar flow (kgmole/h)	15920	15920	15920	15918	1600	14307	14307	14307	14307	14307
LIQUID PHASE										
Mass flow (kg/h)			32							
GASEOUS PHASE										
Mass flow (kg/h)	689122	689122	689090	689090	69171	619775	619775	619775	619775	619775
Molar flow (kgmole/h)	15920	15920	15920	15918	1600	14307	14307	14307	14307	14307
Molecular Weight	43.29	43.29	43.29	43.29	43.22	43.32	43.32	43.32	43.32	43.32
Composition (vol %)										
CO_2	98.09	98.09	98.09	98.10	97.85	98.22	98.22	98.22	98.22	98.22
CO	0.14	0.14	0.14	0.14	0.14	0.14	0.14	0.14	0.14	0.14
H_2S+COS	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01
H_2	1.57	1.57	1.57	1.57	1.57	1.58	1.58	1.58	1.58	1.58
N_2	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02
Ar	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04
H_2O	0.13	0.13	0.13	0.12	0.38	0.00	0.00	0.00	0.00	0.00

Table 3-3 Pre-combustion – Base case A0: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Pre-Combustion - Case A0										
STREAM	21	22	23							
Temperature (°C)	40	12	19							
Pressure (bar)	111.0	6.0	5.8							
TOTAL FLOW										
Mass flow (kg/h)	619775	8929725	8929725							
Molar flow (kgmole/h)	14307	495655	495655							
LIQUID PHASE										
Mass flow (kg/h)		8929725	8929725							
GASEOUS PHASE										
Mass flow (kg/h)	619775	0	0							
Molar flow (kgmole/h)	14307	0	0							
Molecular Weight	43.32	18.02	18.02							
Composition (vol %)										
CO_2	98.22	0.00	0.00							
CO	0.14	0.00	0.00							
H_2S+COS	0.01	0.00	0.00							
H_2	1.58	0.00	0.00							
N_2	0.02	0.00	0.00							
Ar	0.04	0.00	0.00							
H_2O	0.00	100.00	100.00							

3.1.3 CO_2 compression unit performance

The main battery limit streams conditions, the electrical and cooling water consumptions of the CO_2 Compression unit for Case A0 are summarized in Table 3-4.

Table 3-4 Pre-combustion - Base case A0: CO_2 Compression Unit consumption.

PRE-COMBUSTION Base case A0: CO_2 compression consumption			
CO_2 Inlet Streams	LP	MP	
Flowrate	106,480	214,482	Nm^3/h
Temperature	-5	1	°C
Pressure	1.2	4.8	bar a
CO_2 Outlet Stream			
Flowrate		320,673	Nm^3/h
Temperature		40	°C
Pressure		111	bar a
CO_2 purity		98.2	% v/v
Overall Plant Carbon Capture			
Carbon Capture		85.0	%
Cooling Water			
CW consumption		8,930	t/h
Compressor/Turbine Electrical Consumption			
1^{st} stage		6.6	MW_e
2^{nd} stage		12.2	MW_e
3^{rd} stage		16.0	MW_e
4^{th} stage		8.5	MW_e
5^{th} stage		4.1	MW_e
TOTAL		**47.4**	**MW_e**

3.2 Post-combustion: Base Case (B0)

3.2.1 Overall power plant process description

3.2.1.1 Power Plant

The plant configuration of the reference Ultra Supercritical (USC) pulverized coal (PC) fired power plant with postcombustion CO_2 capture is described here below.

The boiler is "once-through" type, capable to generate steam at ultra supercritical conditions and to reheat exhaust steam from the HP steam turbine module.

The coal is first pulverized by dedicated mills. The pulverized coal exits each mill via the coal piping and is distributed to the coal nozzles in the furnace walls, using air supplied by the primary air fans. Primary air for conveying pulverized coal and secondary air for the burners windboxes are blown by a dedicated set of fans. Prior to enter the pulverizer mills/coal burners, a portion of primary air and secondary air streams are pre-heated into the rotating regenerative exchangers (Ljungstrom), counter current with hot flue gases exiting the SCR deNOx described below. The primary air preheating allows drying the pulverized coal; a portion of the primary air is bypassed in order to control the air/coal temperature leaving the mills.

The pulverized coal and air mixture flows to the coal nozzles at various elevations of the furnace for NOx reduction through controlled staged combustion. Gases exiting the boiler combustion chamber flow through the superheater, re-heater and economizer coils, then enter the catalyst modules of the SCR deNOx system, downstream the ammonia injection grids. The regenerative air pre-heaters described above, further cool flue gases, which then pass through the fabric filter, the flue gas de-sulphurization unit (FGD) and are finally routed to the CO_2 capture unit located upstream the stack. The induced draft fan, installed at the FGD unit inlet, balances the boiler draft.

The Steam Turbine is fully reheated, condensing type, fed by ultra-supercritical steam at one pressure level, generated in the USC PC boiler. The ultra-supercritical steam produced by the boiler is admitted in the HP module of the Steam Turbine (ST). Most of the HP module exhaust steam, named Cold Re- Heat (RH), is sent to the boiler for re-heating, while the remaining part is routed to the final exchanger of the BFW preheating line. The reheated steam coming from the boiler is admitted to the MP section of the steam turbine. Some amount of steam is extracted from the MP turbine section to meet the steam demand of the deaerator and the steam turbine driver of the BFW pump; the remainder amount is admitted to the LP section of the steam turbine. The exhaust wet steam from the LP module outlet is discharged into a water-cooled condenser.

3.2.1.2 Acid Gas Removal

Absorption in chemical solvents, such as amines, is the most mature technology, already commercially available for CO_2 capture, though it has not been proven at a large scale yet.

The flue gases, after deep sulphur removal and further cooling, are fed to the absorption tower by a flue gas blower. A lean amine solution, typically Mono-Ethanol-Amine (MEA), counter-currently interacts with the flue gases to absorb the CO_2. The clean flue gases continue to the stack.

Some of the heat reaction of the solvent with CO_2 is removed by the pump around coolers, located at different heights of the column. Before leaving the absorber, the sweet gas is scrubbed with make-up water to remove the entrained solvent and avoid any dispersion to the atmosphere.

From the bottom of the absorption columns, the rich solvent is split into two streams: the first is heated in a cross exchanger against the hot stripper bottom and routed to the regeneration column; the remainder is flashed to produce steam which is used in the top rectification section of the stripper, thus reducing the amount of steam needed from the reboiler.. The flash partially desorbs CO_2 creating a liquid semi-lean amine stream, which is recycled back to the absorber at a intermediate height. Prior to be flashed, the rich amine is heated in the semi-lean amine cooler (where it is cross exchanged with the hot flashed amine) and in the Flash Preheater (where it is heated by the stripper bottom).

The steam necessary for solvent regeneration comes from the steam turbine IP/LP cross-over , while saturated condensate is pumped back to the deaerator.

The vapour at the top of the column passes through the overhead stripper condenser, where it is cooled versus cold condensate from the steam turbine condenser. The remaining condensing duty is achieved with cooling water. At the overhead stripper condenser outlet, water vapor is separated generating the rich CO_2 stream, which flows to the CO_2 compression unit, while condensed water is partially returned to the column as reflux.

The lean solvent at the bottom of the stripping column is pumped back to the absorption, after final cooling against cooling water.

3.2.2 CO_2 compression unit description

The following process description makes reference to the simplified process flow diagram shown in Figure 3-2 and to the Heat Material Balance in Table 3-5 and Table 3-6.

The vapour stream from the stripper condenser is mainly composed by CO_2 (95.88% vol) and H_2O. From the AGR unit the wet carbon dioxide flows to the CO_2 compression unit, where it is dehydrated and compressed to be further stored.

A first KO drum avoids water droplets, possibly condensed in the pipeline between the two units, to enter the first compression stage. The stream at 38°C and 1.6 bara is compressed up to 7 bara and cooled firstly with Steam Turbine condensate and then with CW to 19°C. The condensate leaves the CO_2 compression unit at a temperature of approximately 100 °C.

At the exit of the inter-coolers, condensed water is separated in a KO drum, while the gaseous phase flows to the second stage of the compression. After this latter compression stage, the stream at 34 bara is again cooled firstly with condensate and then with CW to 19°C. Water condensate is separated in a KO drum, while the gaseous phase enters the CO_2 Dehydration System.

The CO_2 Dehydration System is sized to reduce the water content of the CO_2 stream to 50 ppmv to meet the typical transmission network specification. The system is composed by two (or more) solid-bed desiccants (typically molecular sieves or activated alumina). While one bed adsorbs water from the wet CO_2 stream, the other bed is in regeneration mode. Regeneration of the saturated bed is carried out using a portion (typically around 10%) of the dried gas product. The water in the bed is desorbed by thermal swing (heating up the regeneration stream to 250°C). The stream used for regeneration is then flashed to recover water and the remaining wet CO_2 stream is recycled back at the outlet of the first stage compression.

The dry CO_2 is further compressed in the third compression stage to 70 bara, cooled with condensate and with CW to 30°C. Before leaving the unit, the stream is finally compressed in the fourth compression stage to 111 bara, cooled with condensate first and then with CW to 40°C.

Figure 3-3 Post-combustion - Base case B0: CO_2 Compression Unit scheme.

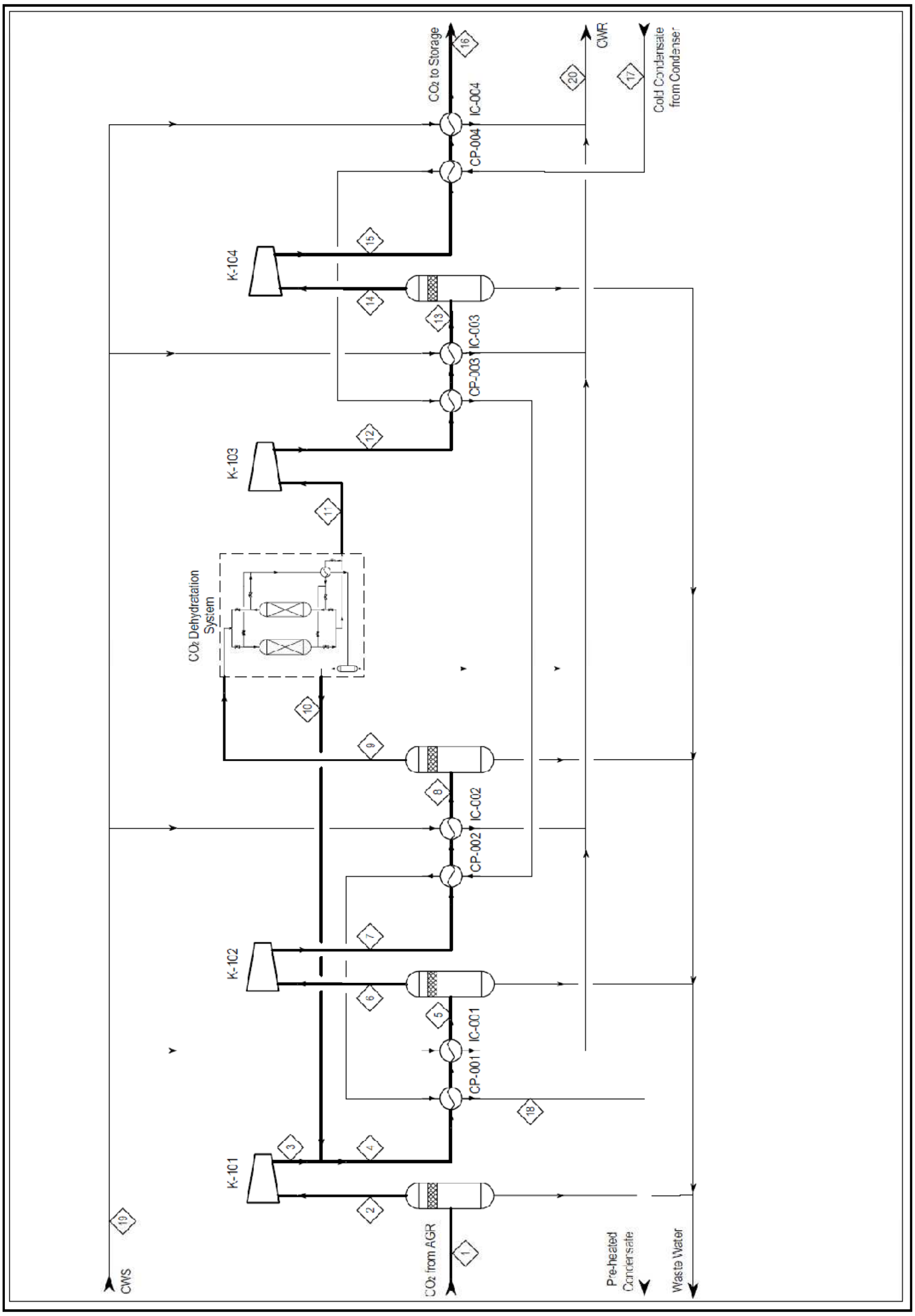

Table 3-5 Post-combustion - Base case B0: CO_2 Compression Unit Heat and Material Balance.

HEAT AND MATERIAL BALANCE										
Post-Combustion - Case B0										
STREAM	1	2	3	4	5	6	7	8	9	10
Temperature (°C)	38	38	184	168	19	19	176	19	19	15
Pressure (bar)	1.6	1.6	7.0	7.0	6.6	6.6	34.0	33.6	33.6	7.0
TOTAL FLOW										
Mass flow (kg/h)	556451	556451	556451	617374	617374	608630	608630	608630	607976	60923
Molar flow (kgmole/h)	12959	12959	12959	14346	14346	13860	13860	13860	13824	1387
LIQUID PHASE										
Mass flow (kg/h)					8744			654		
GASEOUS PHASE										
Mass flow (kg/h)	556451	556451	556451	617374	608630	608630	608630	607976	607976	60923
Molar flow (kgmole/h)	12959	12959	12959	14346	13860	13860	13860	13824	13824	1387
Molecular Weight	42.94	42.94	42.94	43.04	43.04	43.91	43.91	43.91	43.98	43.94
Composition (vol %)										
CO_2	95.88	95.88	95.88	96.25	96.25	99.62	99.62	99.62	99.88	99.71
N_2	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01
O_2	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Ar	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
H_2O	4.11	4.11	4.11	3.74	3.74	0.37	0.37	0.37	0.11	0.28

Table 3-6 Post-combustion - Base case B0: CO_2 Compression Unit Heat and Material Balanace.

HEAT AND MATERIAL BALANCE										
Post-Combustion - Case B0										
STREAM	11	12	13	14	15	16	17	18	19	20
Temperature (°C)	24	97	40	40	81	73	70	101	12	19
Pressure (bar)	32.7	70.0	69.6	69.6	111.2	111.0	10.0	9.2	6.0	5.8
TOTAL FLOW										
Mass flow (kg/h)	546855	546855	546855	546855	546855	546855	956880	956880	5392000	5392000
Molar flow (kgmole/h)	12426	12426	12426	12426	12426	12426	53113	53113	299290	299290
LIQUID PHASE										
Mass flow (kg/h)							956880	956880	5392000	5392000
GASEOUS PHASE										
Mass flow (kg/h)	546855	546855	546855	546855	546855	546855	0	0	0	0
Molar flow (kgmole/h)	12426	12426	12426	12426	12426	12426	0	0	0	0
Molecular Weight	44.01	44.01	44.01	44.01	44.01	44.01	18.02	18.02	18.02	18.02
Composition (vol %)										
CO_2	99.99	99.99	99.99	99.99	99.99	99.99	0.00	0.00	0.00	0.00
N_2	0.01	0.01	0.01	0.01	0.01	0.01	0.00	0.00	0.00	0.00
O_2	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Ar	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
H_2O	0.00	0.00	0.00	0.00	0.00	0.00	100.00	100.00	100.00	100.00

3.2.3 CO_2 compression unit performance

The main battery limit streams conditions, the thermal integration with the power plant, the electrical and cooling water consumptions of the CO_2 Compression unit are summarized in Table 3-7.

Table 3-7 Post-combustion - Base case B0: CO_2 Capture/Compression Unit consumption.

POST-COMBUSTION Base case B0: CO_2 capture/compression consumption		
CO_2 Inlet Stream		
Flowrate	290461	Nm^3/h
Temperature	38	°C
Pressure	1.6	bar a
CO_2 Outlet Stream		
Flowrate	278518	Nm^3/h
Temperature	73	°C
Pressure	111	bar a
CO_2 purity	99.99	% v/v
Overall Plant Carbon Capture		
Carbon Capture	86.7	%
Cooling Water		
CW cons. CO_2 compression	5392	t/h
CW cons. Stripper Overhead Cond.	8676	t/h
Thermal Integration with the Power Plant		
Condensate pre-heating	34.0	MW_{th}
Amine Stripping		
Reboiler Thermal Duty	490.0	MW_{th}
Compressor Electrical Consumption		
1^{st} stage	21.7	MW_e
2^{nd} stage	24.1	MW_e
3^{rd} stage	8.0	MW_e
4^{th} stage	3.7	MW_e
TOTAL	**57.5**	**MW_e**

3.3 Oxy-combustion: Base Case (C0)

3.3.1 Overall power plant process description

In an oxy-fuel process (Figure 3-4), the fuel combustion is made by utilising almost pure oxygen as oxidising medium. As a consequence, the flue gases are mainly composed of carbon dioxide and other components like water and inerts (excess O_2, and N_2 and Ar entrained in the oxygen stream delivered from the ASU). Therefore, the carbon dioxide capture process mainly consists of a purification of the flue gases for the removal of these components. The higher is the oxygen purity, the lower is the content of inerts in the flue gases.

To moderate the peak temperature in the combustion chamber and avoid an increase of the radiant heat pick-up, part of the flue gas leaving the boiler, around 67% of the original flue gas leaving the economiser, needs to be recirculated back to the burners. Recycled gases are mixed with oxygen from the ASU and then supplied to the boiler into two streams:

- Primary recycle: it passes through the coal mills and transports the pulverised coal to the burners. The volumetric flow rate of the primary recycle gas is maintained at a value required for the air firing.
- Secondary recycle: it provides the additional inert gases to the fuel burners in order to keep the furnace temperatures at levels similar to those of the air fired boilers.

The flue gas exiting the boiler is used to heat the primary and secondary recycle flue gas streams via a regenerative gas/gas heater. The flue gas is de-dusted via the ESP. The clean flue gas is then split into two streams, with one stream forming the secondary recycle and returning back through the gas/gas heater to the burners. The remaining stream is cooled, dried and split again to generate the primary recycle and the CO_2 product streams. The primary recycle passes through the gas/gas heater and is then delivered to the coal mills.

The steam turbine and the BFW heating are essentially the same as the conventional USC-PC case.

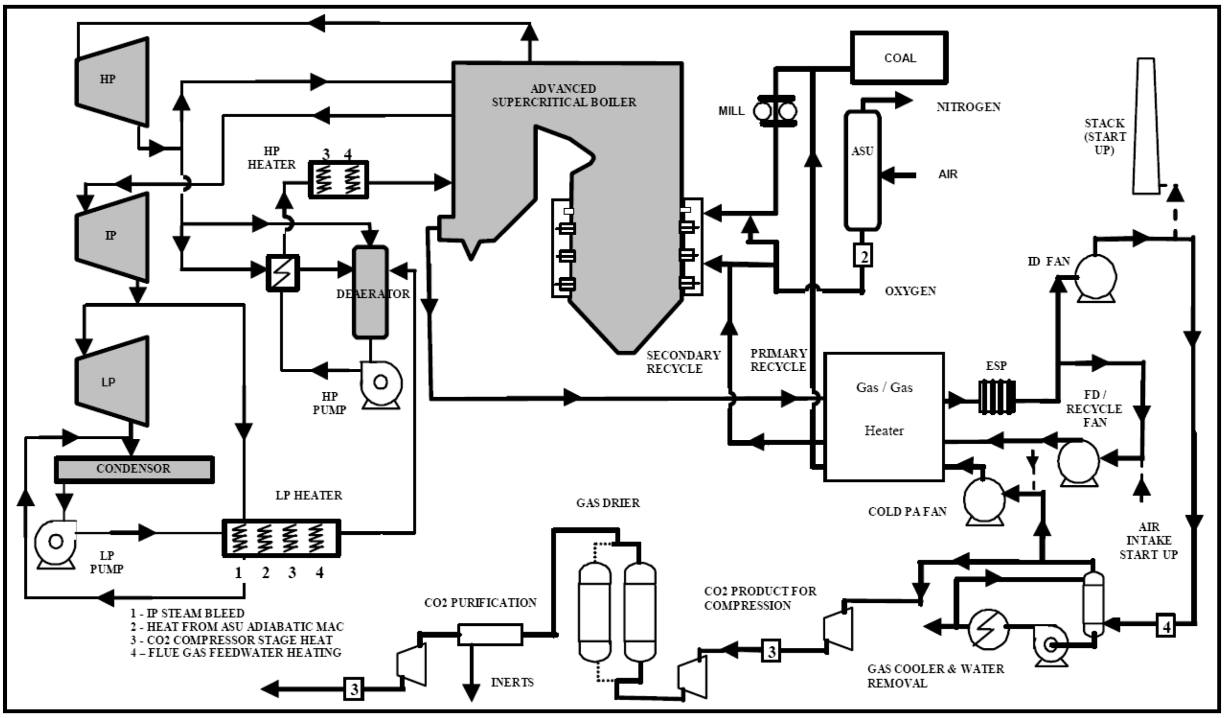

Figure 3-4 Oxy combusted USC-PC typical scheme with cryogenic CO_2 purification

3.3.2 CO_2 compression unit description

The following description refers to the simplified process flow diagram shown in figure 3-4 and to the Heat & Material Balance in Table 3-8, Table 3-9 and Table 3-10.

The net flue gas from the boiler island is passed through the CO_2 cryogenic purification, which is the most efficient technique to remove incondensable contaminants from a highly concentrated CO_2 stream.

The process considered in the reference work is an "auto-refrigerated cycle" (Figure 3-5), which uses the same cold CO_2 separated in the plant as working fluid of a refrigerating cycle that provides flue gases cooling. Although there are not yet many industrial applications, this process has been preferred with respect to conventional refrigeration cycles, because it significantly improves the economics of the project.

The flue gas entering the unit is initially cooled and compressed into an inter-cooled two stages compressor to about 30 bar. Compressed carbon dioxide flows through the swing dual bed desiccant dryer, to remove the last traces of water before entering the cold box. The dry gas is fed to the cold box and initially cooled by heat exchange to approximately -28°C in the “warm exchanger” against the evaporating, superheating CO_2 streams and the waste streams from the cold exchanger.

The cooled feed is sent to a knock-out drum, which separates liquid and vapour phase; the liquid contains part of the CO_2 product, while the vapour from the separator still contains a significant fraction of CO_2 and almost all the other lighter components present in the flue gas. To further recover the carbon dioxide, the vapour phase is cooled to about –54°C in the “cold exchanger”, very close to the triple point, then flowing to a second knock-out drum. The vapour from the second separator, containing the separated inerts and part of the CO_2, is sent back through the two main heat exchangers, where it is heated by cooling the rich CO_2 stream entering the unit. After pre-heating and expansion, this stream is finally released to the atmosphere.
Both the warm and the cold heat exchangers are made of multi-stream plate-fin aluminium blocks.

The liquid phase from the first separator, containing part of the CO_2, is throttled through a valve and then heated. The liquid phase from the second separator is heated, throttled through a valve and then separated in a third flash drum. The resulting liquid stream is carbon dioxide at high purity. Because of throttling, the last liquid stream is at a temperature of about –55°C, thus being used as refrigerator in the cold exchanger.

The high-purity CO_2 vapour stream leaving the warm exchanger is compressed and mixed with the CO_2 stream from the first separator. The two streams are combined and finally compressed for CO_2 transportation and storage.

In an oxy-fuel process, the low pressure oxygen is provided by a dedicated Air Separation Unit (ASU), which is based on an industry standard method of cryogenic air separation, using a double column distillation cycle. In accordance to the boiler requirements, oxygen is delivered at low pressure, generally slightly higher than the ambient pressure.

Figure 3-5 Oxy-combustion - Base case C0: CO_2 Compression Unit scheme.

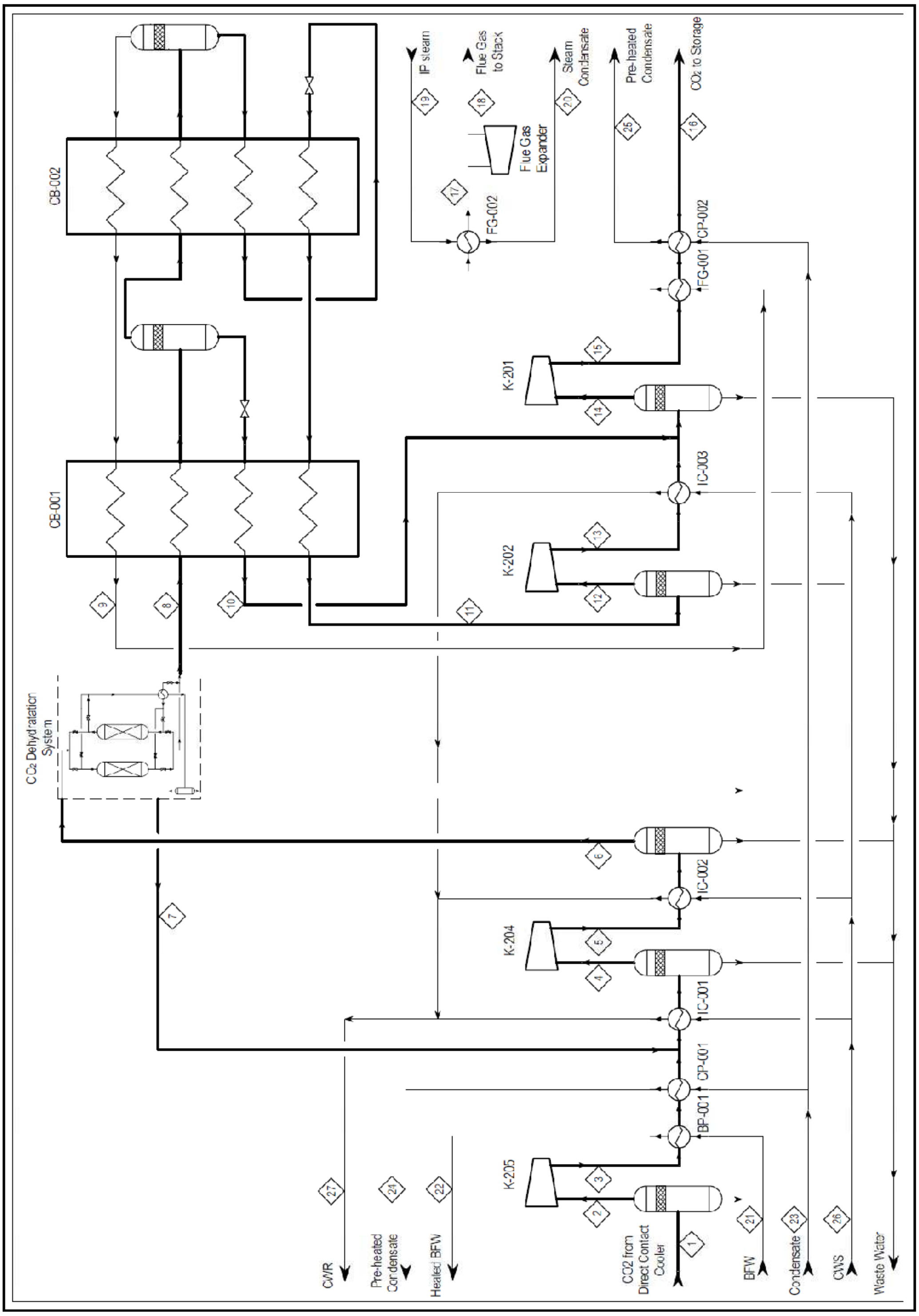

Table 3-8 Oxy-combustion – Base case C0: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Oxy-Combustion - Case C0										
STREAM	1	2	3	4	5	6	7	8	9	10
Temperature (°C)	12	12	281	19	19	19	84	22	18	7
Pressure (bar)	1.0	1.0	15.0	14.4	14.4	30.9	34.0	30.0	28.9	18.6
TOTAL FLOW										
Mass flow (kg/h)	602055	602055	602055	664617	664617	664388	66246	597575	137173	228563
Molar flow (kgmole/h)	14882	14882	14882	16313	16313	16300	1635	14652	4092	5216
LIQUID PHASE										
Mass flow (kg/h)										
GASEOUS PHASE										
Mass flow (kg/h)	602055	602055	602055	664617	664617	664388	66246	597575	137173	228563
Molar flow (kgmole/h)	14882	14882	14882	16313	16313	16300	1635	14652	4092	5216
Molecular Weight	40.45	40.45	40.45	40.74	40.74	40.76	40.53	40.78	33.52	43.82
Composition (vol %)										
CO_2	74.66	74.66	74.66	75.62	75.62	75.67	74.90	75.76	24.24	96.52
N_2	14.98	14.98	14.98	15.18	15.18	15.19	15.03	15.20	49.15	1.46
O_2	6.15	6.15	6.15	6.23	6.23	6.23	6.17	6.24	19.33	0.80
Ar	2.41	2.41	2.41	2.44	2.44	2.45	2.42	2.45	7.15	0.41
SO_2	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.00	0.79
$NO + NO_2$	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.12	0.01
H_2O	1.45	1.45	1.45	0.19	0.19	0.11	1.13	0.00	0.00	0.00

Table 3-9 Oxy-combustion – Base case C0: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Oxy-Combustion - Case C0										
STREAM	11	12	13	14	15	16	17	18	19	20
Temperature (°C)	7	7	66	13	188	43	300	20	620	277
Pressure (bar)	9.3	9.3	18.7	18.6	111.4	111.0	28.5	1.1	61.1	60.9
TOTAL FLOW										
Mass flow (kg/h)	231838	231838	231838	460402	460402	460402	137173	137173	7359	7359
Molar flow (kgmole/h)	5344	5344	5344	10560	10560	10560	4092	4092	408	408
LIQUID PHASE										
Mass flow (kg/h)										7359
GASEOUS PHASE										
Mass flow (kg/h)	231838	231838	231838	460402	460402	460402	137173	137173	7359	0
Molar flow (kgmole/h)	5344	5344	5344	10560	10560	10560	4092	4092	408	0
Molecular Weight	43.39	43.39	43.39	43.60	43.60	43.60	33.52	33.52	18.02	18.02
Composition (vol %)										
CO_2	94.95	94.95	94.95	95.73	95.73	95.73	24.24	24.24	0.00	0.00
N_2	2.62	2.62	2.62	2.05	2.05	2.05	49.15	49.15	0.00	0.00
O_2	1.51	1.51	1.51	1.16	1.16	1.16	19.33	19.33	0.00	0.00
Ar	0.83	0.83	0.83	0.63	0.63	0.63	7.15	7.15	0.00	0.00
SO_2	0.07	0.07	0.07	0.42	0.42	0.42	0.00	0.00	0.00	0.00
$NO + NO_2$	0.02	0.02	0.02	0.02	0.02	0.02	0.12	0.12	0.00	0.00
H_2O	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	100.00	100.00

Table 3-10 Oxy-combustion – Base case C0: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Oxy-Combustion - Case C0										
STREAM	21	22	23	24	25	26	27			
Temperature (°C)	165	206	33	95	97	12	19			
Pressure (bar)	21.0	21.0	6.0	5.8	5.8	6.0	5.8			
TOTAL FLOW										
Mass flow (kg/h)	329940	329940	709113	330635	378478	3524494	3524494			
Molar flow (kgmole/h)	18314	18314	39360	18352	21008	195631	195631			
LIQUID PHASE										
Mass flow (kg/h)	329940	329940	709113	330635	378478	3524494	3524494			
GASEOUS PHASE										
Mass flow (kg/h)	0	0	0	0	0	0	0			
Molar flow (kgmole/h)	0	0	0	0	0	0	0			
Molecular Weight	18.02	18.02	18.02	18.02	18.02	18.02	18.02			
Composition (vol %)										
CO_2	0.00	0.00	0.00	0.00	0.00	0.00	0.00			
N_2	0.00	0.00	0.00	0.00	0.00	0.00	0.00			
O_2	0.00	0.00	0.00	0.00	0.00	0.00	0.00			
Ar	0.00	0.00	0.00	0.00	0.00	0.00	0.00			
SO_2	0.00	0.00	0.00	0.00	0.00	0.00	0.00			
$NO + NO_2$	0.00	0.00	0.00	0.00	0.00	0.00	0.00			
H_2O	100.00	100.00	100.00	100.00	100.00	100.00	100.00			

3.3.3 CO_2 compression unit performance

The main battery limit streams conditions, the thermal integration with the power plant, the electrical and cooling water consumption of the CO_2 Compression unit for Case C0 are summarized in Table 3-11.

Table 3-11 Oxy-combustion - Base case C0: CO_2 Compression Unit consumption.

OXY-COMBUSTION Base case C0: CO_2 compression consumption		
CO_2 Inlet Stream		
Flowrate	333,568	Nm^3/h
Temperature	12	°C
Pressure	1.0	bar a
CO_2 Outlet Stream		
Flowrate	236,684	Nm^3/h
Temperature	43	°C
Pressure	111	bar a
CO_2 purity	95.7	% v/v
Overall Plant Carbon Capture		
Carbon Capture	91.1	%
Cooling Water		
CW consumption	3,524	t/h
Thermal Integration with the Power Plant		
Condensate pre-heating	51.8	MW_{th}
BFW heating	16.4	MW_{th}
IP steam consumption	5.1	MW_{th}
Compressor/Turbine Electrical Consumption		
1^{st} stage	43.9	MW_e
2^{nd} stage	14.7	MW_e
3^{rd} stage	3.1	MW_e
4^{th} stage	17.6	MW_e
Flue Gas Expander	-9.8	MW_e
TOTAL	**69.5**	**MW_e**

4 Basic criteria for technical comparison

One of the main objectives of this study is to make a technical comparison between the Base Case configurations described in the previous sections for each CO_2 capture plant and alternative CO_2 compression strategies, in order to find solutions with lower parasitic loads.

This evaluation takes into account that CO_2 compression scheme modifications may lead to a utility requirement, mainly steam and cooling water, which is different from that of the base cases. But different utility consumption also corresponds to a different power demand, which then affects the overall net electrical efficiency of the power plant.

For each steam pressure level used in the plant, as well as for the cooling water, the following sections describe the conceptual criteria used to convert the utility requirement into an equivalent power demand.

4.1 Steam

For each compression strategy, a different utility steam demand leads to a different steam turbine electrical power production. To estimate the delta electrical production, a reference steam turbine adiabatic efficiency of 90% has been generally assumed.

With this efficiency, it has been possible to estimate the specific equivalent electrical consumption of the different steam pressure levels in the plant, as shown in Table 4-1.

Table 4-1 Equivalent electrical consumption of different steam pressure levels

Utility	Case involved	Specific equivalent electrical consumption
LP steam		
at 7.5 bara	*Pre-combustion*	191 kWe/t/h
at 3.3 bara	*Post-combustion*	172 kWe/t/h
at 2.5 bara	*Oxy-combustion*	145 kWe/t/h
IP steam		
at 61 bara	*Oxy-combustion*	375 kWe/t/h

The following sections provide additional information on the main modifications found in the various compression strategies analyzed in this study.

4.1.1 LP steam for condensate pre-heating

In post-combustion and oxy-combustion base case configurations, the heat available at each CO_2 compressor stage outlet is partially recovered by pre-heating the condensate coming from the steam cycle condenser. Then, the pre-heated condensate flows to the deaerator, after final heating with steam extracted from the steam turbine.

In an alternative compression strategy, if the condensate does not recover the same level of heat from the CO_2 compression unit, more steam is required for heating the condensate, before entering the deaerator. In other words, the thermal power given to the condensate in the CO_2 compression unit, if it is not provided by the exchangers in this unit, it has to be provided by the condensate pre-heater in the power plant.

Different steam requirement by the condensate pre-heater in the power plant varies the steam expanding in the steam turbine, while also changing the condensate flowing to the CO_2 compression unit, where heat is recovered.

4.1.2 LP steam to AGR reboiler

Any modification in the Acid Gas Removal Unit (for post-combustion and pre-combustion configuration) leads to changes in the stripper for solvent regeneration. As a consequence, the duty of the stripper reboiler changes and so the LP steam requirements from the LP steam header in the plant.

Variation in the LP steam extraction from the steam turbine is so reflected in the amount of steam entering the last module of the steam turbine, as well as the condensate flowing from the steam condenser to the CO_2 compression unit, where heat may be recovered.

4.1.3 IP steam for flue gas heating

In the oxy-combustion configuration, before expanding, the flue gases are heated up to 300°C with IP steam from the power plant. Modifications in the exchanger scheme of the auto-refrigerated CO_2 compression/purification unit vary the amount of steam required by the flue gas heater.

These changes are also reflected in the amount of steam entering the medium pressure module of the steam turbine and the condensate flowing from the steam condenser to the CO_2 compression/purification unit, where heat is recovered.

4.2 Cooling water

The primary cooling system of the base cases is seawater in once through system, mainly used for the steam turbine condenser, ASU exchangers, CO_2 compression and drying exchangers, fresh cooling water-cooling. On the other hand, a secondary system in closed circuit, cooled by the primary system, is used for machinery cooling and for all other users not listed before.

In the alternative compression strategies, the different cooling water requirement corresponds to a different cooling water circulation pump and seawater pump electrical demand.

Based on this consideration, it has been estimated that the equivalent electrical consumption of the cooling water is 0.102 kW for each m^3/h of cooling water demand.

5 Basic criteria for economic comparison

General IEA GHG economic assessment guidelines are applied to the present analysis for the evaluation of the various compression strategies, in terms of differential figures with respect to the plant configurations taken as reference (Base Case).

The main factors that are applicable to this type of analysis are defined in the following sections.

5.1 Capital Charges

Discounted cash flow calculations have been expressed at a discount rate of 10%.

5.2 Inflation

No inflation have been applied to the economic analysis.

5.3 Maintenance Costs

Differential maintenance costs have been estimated as percentage of the differential investment cost. The following factors have been used:

IGCC plant:	3.4% of Differential Investment Cost
USC PC boiler plant:	3.1% of Differential Investment Cost

5.4 Cost of consumed Electricity

A cost of 0.05 $/kWh, corresponding to 3.8 €c/kWh has been defined to cover lost export electricity revenue (associated to reduced electrical consumption), rather than generation costs.

5.5 Fuel Costs

Cost of coal delivered to site is 3.0 €/GJ (LHV basis).

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SECTION C

EVALUATION OF CO_2 COMPRESSION STRATEGIES

INDEX

1 Introduction

Scope of this Section C is the technical assessment of CO_2 compression strategies, alternative to those shown in the above mentioned reports, with the main objective of reducing the parasitic power consumption of the overall plant for each carbon dioxide capture type.

For each identified compression strategy, a generic overview of the “envelope” of flow conditions, which the CO_2 compressors will be required to handle, has been prepared to identify the range of possible stage flow, temperature, pressure and compositions and the implications for the compressor selection and design.

An economic assessment of the most promising compression strategies is also made in this action, in order to verify if the energy saving affects the delta cost of the alternative, so to assess the economic convenience of the strategy with respect to the Base Case.

2 Pre-combustion strategies

With respect to the Base Case (A0) shown in Section B, the CO_2 compression and the process units integrated with this system have been modified in order to investigate alternative CO_2 compression strategies. The technical assessment is made for the strategies listed in Table 2-1, while an economic assessment of the most advantageous alternatives is shown in Section 6.

Table 2-1 Pre combustion – Summary of compression strategies.

Case tag	Description
Case A1	Vapour recompression in the AGR stripping column
Case A2	Increase ofnumber of flash stages in the AGR
Case A3	AGR stripper pressure increase
Case A4	Re-use of waste heat from CO_2 compression

2.1 Case A1 - Vapour recompression in the AGR stripping column

2.1.1 CO_2 compression unit description

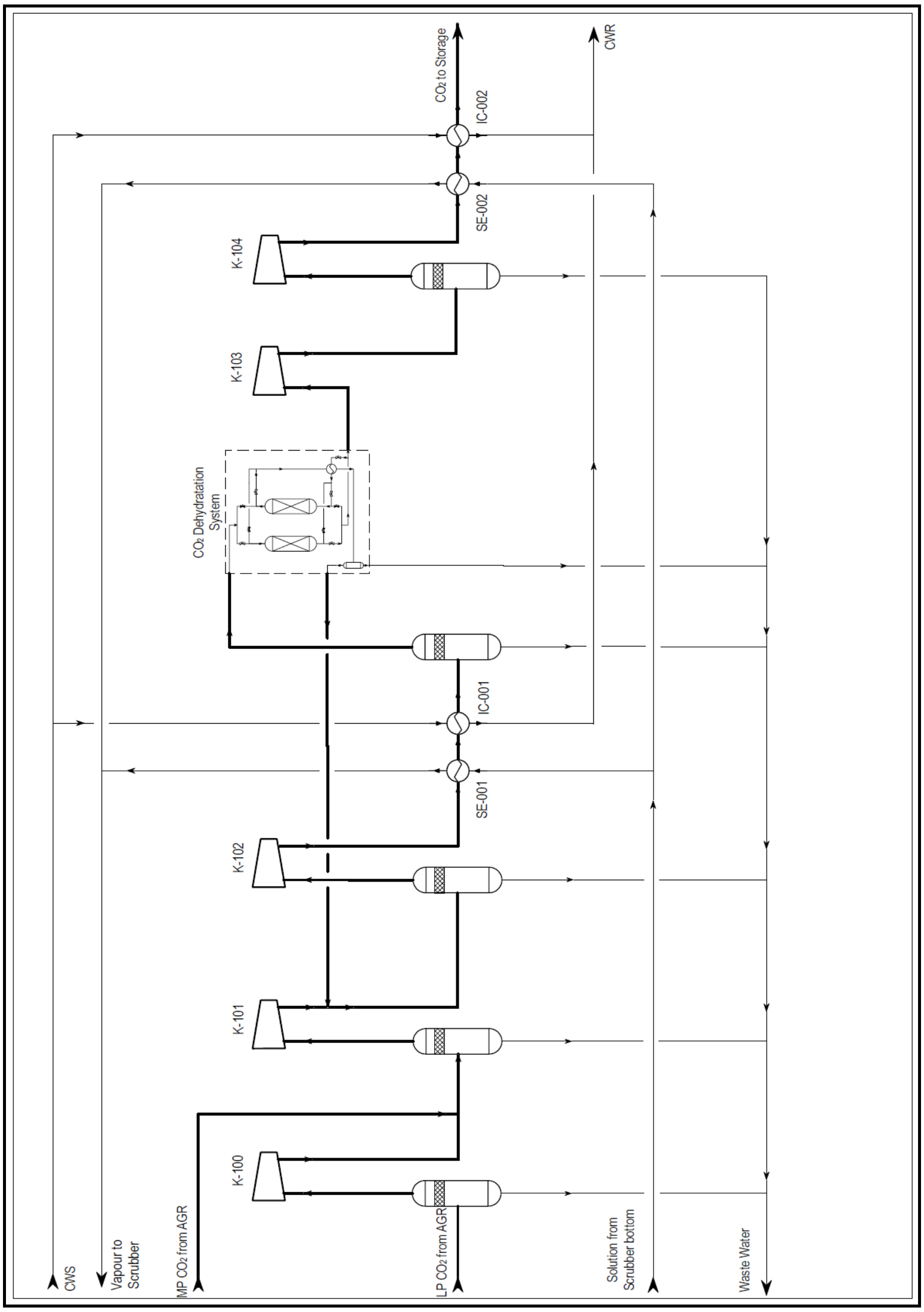
The reference process flow scheme for this compression strategy is shown in Figure 2-1.

The concept behind the vapour recompression strategy is the maximisation of the heat available from the CO_2 compression discharge and its potential utilisation at higher temperatures in the process (e.g. for the AGR solvent regeneration). In the scheme applied to the pre-combustion capture plant, the inter-cooling of the compressor would be ideally recovered by the adiabatic compression heat into the AGR Stripper Reboiler. With respect to the Base Case, the compression work is higher, due to the increase of the CO_2 average temperature in the compression path, while the LP steam (6.5 barg) demand is lower, thus leading to an increase of the ST output, since a portion of the reboiler heat requirement is supplied by the CO_2 compression.

A constraint for the implementation of the vapour recompression concept in the CO_2 capture process is represented by the necessity to cool down the CO_2 for a proper operation of the CO_2 dehydration system. In fact, both the desiccant solid beds and the TEG systems require a maximum inlet temperature of 50 °C. For this reason, a CW intercooler upstream of the dehydration unit is still required.

The necessity of cooling down the CO_2 upstream of the drying process implies that, along the compression path, there must be a low temperature point in correspondence of the dryer operating pressure, which prevents the vapour recompression mechanism from being fully effective. In this sense, a performance benefit would be expected in case of a solvent washing processes, like the methanol, which generates CO_2 streams completely free of water.

Figure 2-1 Pre-combustion – Case A1: CO_2 Compression Unit scheme with Vapour Recompression.



2.1.2 CO_2 compression unit performance

The main battery limit streams conditions, the thermal integration with the AGR unit, the electrical and cooling water consumption of the CO_2 Compression unit for Case A1 are summarized in Table 2-2.

Table 2-2 Pre-combustion - Case A2: CO_2 Compression Unit consumption.

PRE-COMBUSTION Case A1: CO_2 compression consumption			
CO_2 Inlet Streams	LP	MP	
Flowrate	106,480	214,482	Nm^3/h
Temperature	-5	1	°C
Pressure	1.2	4.8	bar a
CO_2 Outlet Stream			
Flowrate		320,673	Nm^3/h
Temperature		40	°C
Pressure		111	bar a
CO_2 purity		98.2	% v/v
Overall Plant Carbon Capture			
Carbon Capture		85.0	%
Cooling Water			
CW consumption		8,157	t/h
Compressor/Turbine Electrical Consumption			
1st stage		6.6	MW_e
2nd stage		13.8	MW_e
3rd stage		22.2	MW_e
4th stage		10.3	MW_e
5th stage		7.4	MW_e
TOTAL		**60.3**	**MW_e**

The following Table 2-2 shows the performance delta between this compression strategy (A1) and the Base Case (A0). From the figures in the Table, it can be drawn that this strategy is not advantageous, as it leads to an equivalent electrical consumption increase of about 6.0 MWe.

Table 2-3 Pre-combustion – Case A1: Performance delta with respect to the base case.

PRE-COMBUSTION Case A1: Performance delta with respect to the base case A0					
Cooling water					
CW consumption	- 773	t/h	⟷	- 0.1	MW_e
Thermal integration with AGR					
Solvent regeneration (2)	-21.0	MW_{th}	⟷	-6.8	MW_e
Compressor Electrical Consumption					
Overall electrical consumption difference				+ 12.9	MW_e
Overall Plant Electrical Consumption Gap					
TOTAL				**+ 6.0**	**MW_e**

Note 1: Negative value indicates a lower consumption with respect to the base case. Conversion factor, see section B.

Note 2: Heat recovered from CO_2 compression represents the 70% of the total reboiler duty.

2.2 Case A2 – Increase ofnumber of flash stages in the AGR

2.2.1 CO_2 compression unit description

The reference process flow diagram of this compression strategy is shown in Figure 2-2 and the related Heat and Material Balance is given in Table 2-4,
Table 2-5 and
Table 2-6.

In the Base Case A0, the liquid phase at the bottom of the CO_2 absorber column in the AGR passes through three sequential flash stages: CO_2 Recycle flash, MP flash and LP flash. The vapour phase from the CO_2 Recycle flash flows back to the CO_2 absorber column, while the liquid phase is expanded successively in the MP flash and then in the LP flash. The CO_2-lean solution after the LP flash is recycled back to the

CO_2 absorber column. In this way, high-purity CO_2 is recovered and delivered to the CO_2 Compression unit at two pressure levels: 4.8 bara (MP) and 1.2 bara (LP).

Case A2 considers an additional CO_2 flash stage, located between the CO_2 Recycle flash and the MP flash. Therefore, the CO_2 Compression unit receives three CO_2 streams respectively at 11.5 bara (HP), 4.8 bara (MP) and 1.2 bara (LP). As a consequence, the duty required by the CO_2 compressors is lower than the Base Case, because part of the CO_2 is already available at higher pressure (11.5 bara), which is similar to the discharge pressure of the in the reference configuration at the second compression stage discharge.

Figure 2-2 Pre-combustion - Case A2: CO_2 Compression Unit scheme.

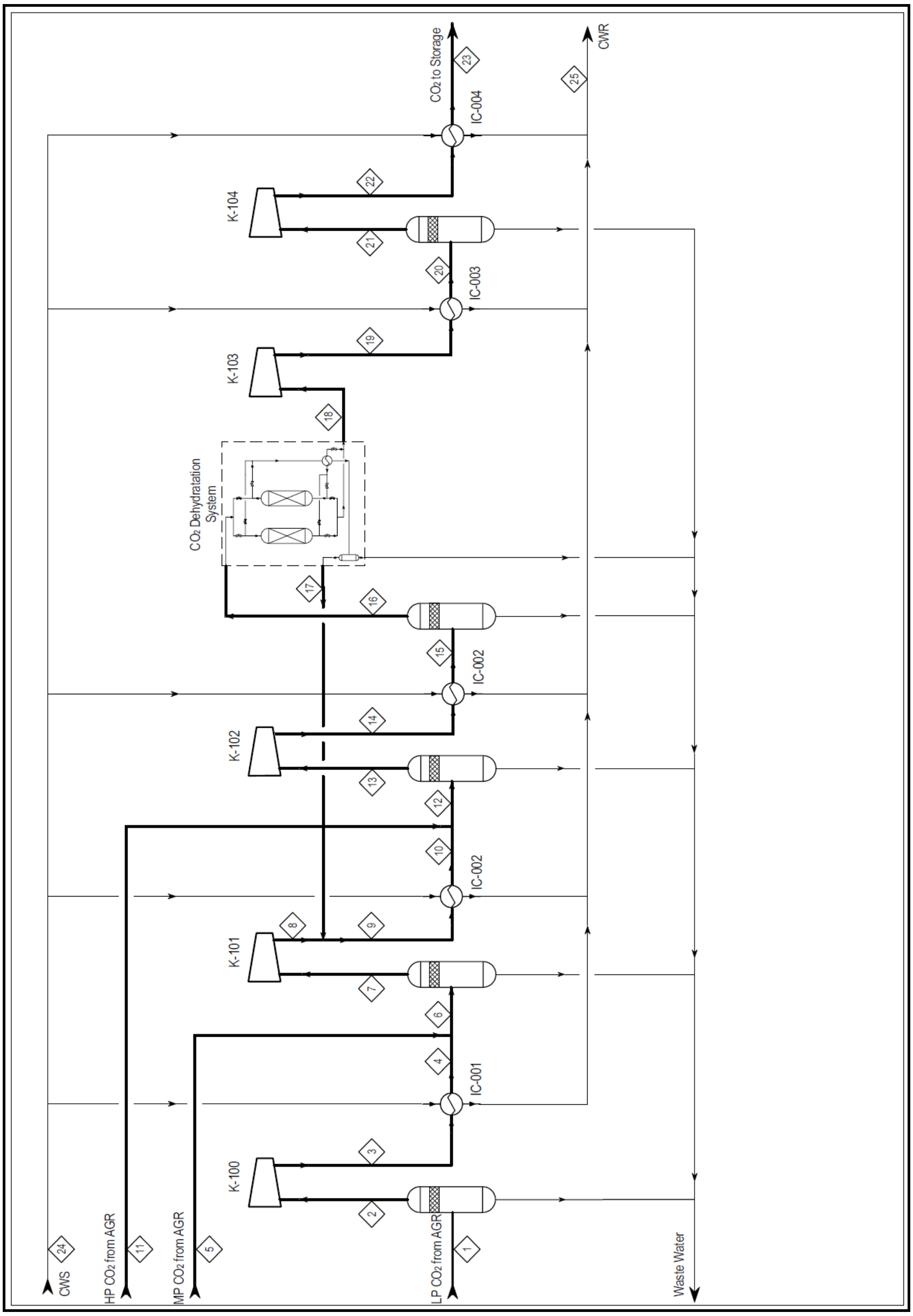


Table 2-4 Pre-combustion – Case A2: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Pre-Combustion - Case A2										
STREAM	1	2	3	4	5	6	7	8	9	10
Temperature (°C)	-5	-5	121	19	1	9	7	87	79	19
Pressure (bar)	1.2	1.2	5.0	4.8	4.8	4.8	4.8	11.7	11.7	11.5
TOTAL FLOW										
Mass flow (kg/h)	212162	212162	212162	212162	297757	509919	509919	509919	579126	579126
Molar flow (kgmole/h)	4826	4826	4826	4826	6780	11606	11606	11606	13207	13207
LIQUID PHASE										
Mass flow (kg/h)										
GASEOUS PHASE										
Mass flow (kg/h)	212162	212162	212162	212162	297757	509919	509919	509919	579126	579126
Molar flow (kgmole/h)	4826	4826	4826	4826	6780	11606	11606	11606	13207	13207
Molecular Weight	43.96	43.96	43.96	43.96	43.92	43.94	43.94	43.94	43.85	43.85
Composition (vol %)										
CO_2	99.81	99.81	99.81	99.81	99.71	99.75	99.75	99.75	99.52	99.52
CO	0.00	0.00	0.00	0.00	0.03	0.02	0.02	0.02	0.03	0.03
H_2S+COS	0.02	0.02	0.02	0.02	0.01	0.01	0.01	0.01	0.01	0.01
H_2	0.00	0.00	0.00	0.00	0.16	0.09	0.09	0.09	0.27	0.27
N_2	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Ar	0.00	0.00	0.00	0.00	0.01	0.01	0.01	0.01	0.01	0.01
H_2O	0.17	0.17	0.17	0.17	0.08	0.11	0.11	0.11	0.15	0.15

Table 2-5 Pre-combustion – Case A2: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Pre-Combustion - Case A2										
STREAM	11	12	13	14	15	16	17	18	19	20
Temperature (°C)	12	18	18	118.9	19	24	27	24	94	40
Pressure (bar)	11.5	11.5	11.5	34.0	33.8	32.9	11.7	32.9	70.0	69.8
TOTAL FLOW										
Mass flow (kg/h)	109982	689108	689108	689108	689108	689082	69207	619769	619769	619769
Molar flow (kgmole/h)	2712	15919	15919	15919	15919	15918	1601	14307	14307	14307
LIQUID PHASE										
Mass flow (kg/h)					26					
GASEOUS PHASE										
Mass flow (kg/h)	109982	689108	689108	689108	689082	689082	69207	619769	619769	619769
Molar flow (kgmole/h)	2712	15919	15919	15919	15919	15918	1601	14307	14307	14307
Molecular Weight	40.55	43.29	43.29	43.29	43.29	43.29	43.23	43.32	43.32	43.32
Composition (vol %)										
CO_2	91.12	98.09	98.09	98.09	98.09	98.10	97.86	98.22	98.22	98.22
CO	0.64	0.14	0.14	0.14	0.14	0.14	0.14	0.14	0.14	0.14
H_2S+COS	0.00	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01
H_2	7.91	1.57	1.57	1.57	1.57	1.57	1.57	1.58	1.58	1.58
N_2	0.09	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02
Ar	0.17	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04
H_2O	0.06	0.13	0.13	0.13	0.13	0.12	0.37	0.00	0.00	0.00

Table 2-6 Pre-combustion – Case A2: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Pre-Combustion - Case A2										
STREAM	21	22	23	24	25					
Temperature (°C)	40	80	40	12	19					
Pressure (bar)	69.8	111.2	111.0	6.0	5.8					
TOTAL FLOW										
Mass flow (kg/h)	619769	619769	619769	8710439	8710439					
Molar flow (kgmole/h)	14307	14307	14307	483484	483484					
LIQUID PHASE										
Mass flow (kg/h)				8710439	8710439					
GASEOUS PHASE										
Mass flow (kg/h)	619769	619769	619769	0	0					
Molar flow (kgmole/h)	14307	14307	14307	0	0					
Molecular Weight	43.32	43.32	43.32	18.02	18.02					
Composition (vol %)										
CO_2	98.22	98.22	98.22	0.00	0.00					
CO	0.14	0.14	0.14	0.00	0.00					
H_2S+COS	0.01	0.01	0.01	0.00	0.00					
H_2	1.58	1.58	1.58	0.00	0.00					
N_2	0.02	0.02	0.02	0.00	0.00					
Ar	0.04	0.04	0.04	0.00	0.00					
H_2O	0.00	0.00	0.00	100.00	100.00					

2.2.2 CO_2 compression unit performance

The main battery limit streams conditions, the thermal integration with the power plant, the electrical and cooling water consumption of the CO_2 Compression unit for Case A2 are summarized in Table 2-7.

Table 2-7 Pre-combustion - Case A2: CO_2 Compression Unit consumption.

PRE-COMBUSTION Case A2: CO_2 compression consumption				
CO_2 Inlet Streams	LP	MP	HP	
Flowrate	108,167	151,969	60,798	Nm^3/h
Temperature	-5	1	12	°C
Pressure	1.2	4.8	11.5	bar a
CO_2 Outlet Stream				
Flowrate			320,673	Nm^3/h
Temperature			40	°C
Pressure			111	bar a
CO_2 purity			98.2	% v/v
Overall Plant Carbon Capture				
Carbon Capture			85.0	%
Cooling Water				
CW consumption			8,710	t/h
Compressor Electrical Consumption				
1st stage			6.7	MW_e
2nd stage			9.6	MW_e
3rd stage			16.4	MW_e
4th stage			8.5	MW_e
5th stage			4.1	MW_e
TOTAL			**45.3**	$\mathbf{MW_e}$

No appreciable loss in terms of CO_2 purity is noted for this strategy. In fact, the purity of the final CO_2 product is mainly affected by the operating pressure of the CO_2 Recycle flash, rather than the number of flash stages between the CO_2 recycle and the stripper section.

Table 2-7 shows the performance delta between this compression strategy (A2) and the Base Case (A0). Overall, there is a net power consumption decrease of 2.1 MWe for this strategy.

Table 2-8 Pre-combustion – Case A2: Performance delta with respect to the base case.

PRE-COMBUSTION Case A2: Performance delta with respect to the base case A0					
Cooling water					
CW consumption	+ 220	t/h	⟷	~ 0	MW_e
Compressor Electrical Consumption					
Overall electrical consumption difference				- 2.1	MW_e
Overall Plant Electrical Power Gap					
TOTAL				**- 2.1**	**MW_e**

Note: Negative value indicates a lower consumption with respect to the base case. Conversion factors, see Section B.

2.3 Case A3 - AGR stripper pressure increase

As the reference case for the pre-combustion capture is based on the separate removal of H_2S and CO_2 through a physical absorption process, the stripper operating pressure does not impact on the overall CO_2 compression strategy. In fact, the CO_2 in the AGR is released from a multi-flash system, located upstream of the stripping section. For this reason, this compression strategy is not further investigated in this study.

2.4 Case A4 - Re-use of waste heat from CO_2 compression

The discharge temperatures of the CO_2 compressor stages in the Base Case do not allow the use of compression waste heat in the AGR stripping section, which requires heat at approximately 165 °C.
For this reason, this compression strategy is not further investigated in this study.

3 Post-combustion strategies

For the post-combustion carbon dioxide capture, a coal fired plant is selected over a natural gas fired one, due to its higher ratio between carbon dioxide emission and electrical output, which makes this alternative more relevant with respect to the objectives of the present study.

With respect to the Base Case configuration (B0) shown in Section B, the CO_2 compression and the process units integrated with this system have been modified in order to investigate alternative CO_2 compression strategies. The technical assessment is made for the strategies listed in Table 3-1, while an economic assessment of the most advantageous alternatives is shown in Section 6.

Table 3-1 Post-combustion – Summary of compression strategies.

Case tag	Description
Case B1	Vapour recompression
Case B2	Increase of stripper pressure in CO2 capture unit
Case B3	Staging of solvent regeneration in CO2 capture unit
Case B4	Re-use of waste heat from CO2 compression

3.1 Case B1 - Vapour recompression in the AGR stripping column

3.1.1 CO_2 compression unit description

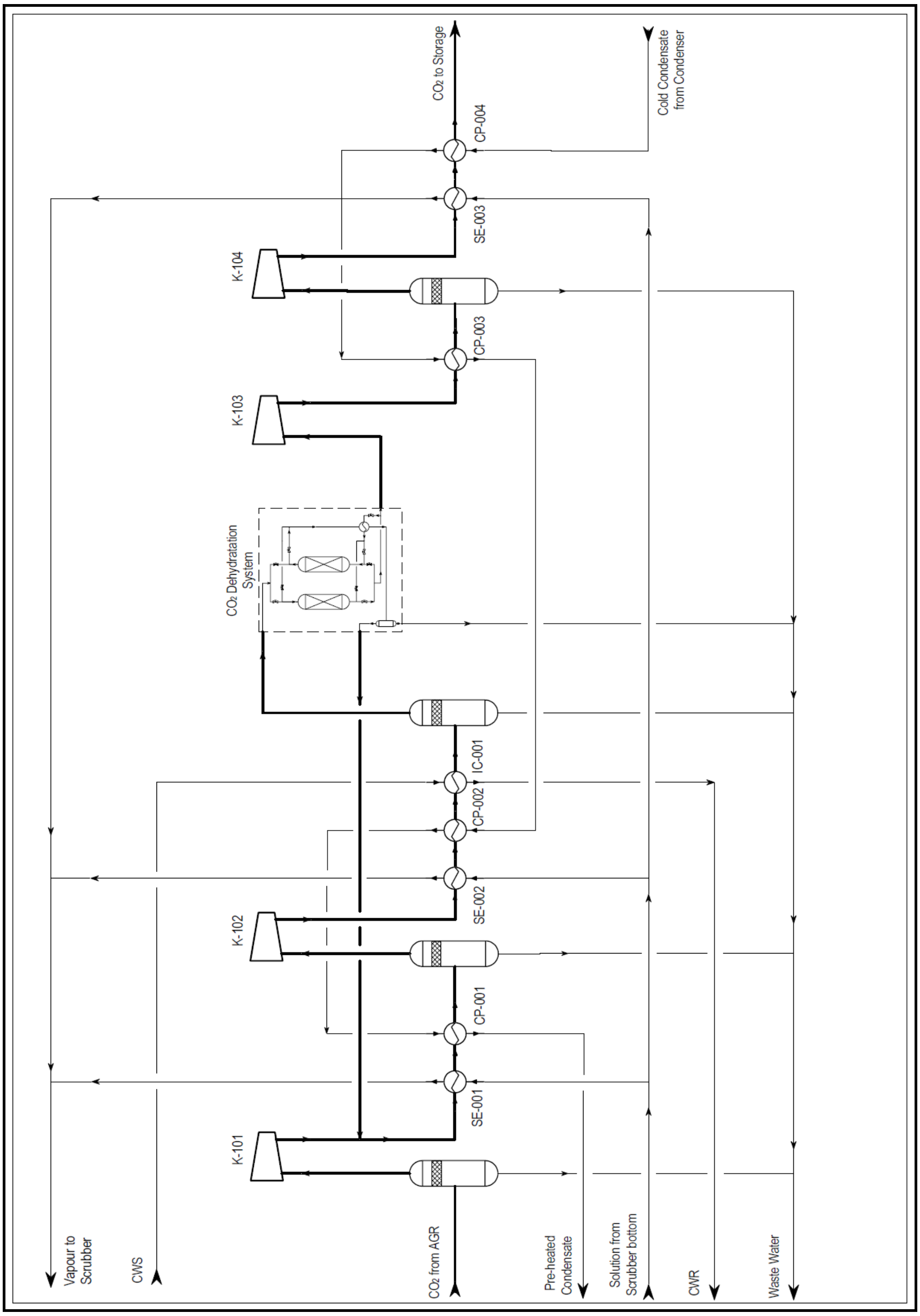
The reference process flow scheme for this compression strategy is shown in Figure 3-1.

As written for the pre-combustion capture (Section 2.1.1), the concept behind the vapour recompression strategy is the maximisation of the heat available from the CO_2 compression discharge and its potential utilization at higher temperatures in the process (e.g. for the MEA regeneration). In the scheme applied to the post-combustion capture plant, the inter-cooling of the compressor would be ideally recovered by the adiabatic compression heat into the MEA Stripper Reboiler. With respect to the Base Case, the compression work is higher, due to the increase of the CO_2 average temperature in the compression path, while the LP Steam (6.5 barg) demand is lower, leading to an increase of the ST power output, since a portion of the reboiler heat requirement is supplied by the CO_2 compression.

However, in the Base Case for post-combustion capture, a portion of the CO_2 compression waste heat is already recovered to preheat the Steam Turbine condensate at condensate pump discharge, as described in section B. This thermal integration allows limiting the steam requirement for the condensate preheating in the Steam Turbine Island. In other terms, by eliminating the CO_2 waste heat recovery into the ST condensate system to make vapour recompression, then the steam extraction from the ST would increase to supply the same steam as used in the reboiler. For this reason, the thermal integration between ST condensate preheating system and the CO_2 compression is kept in the vapour recompression configuration as well.

Also, a constraint for the implementation of the vapour recompression concept in the post-combustion capture is represented by the necessity to cool down the CO_2 for a proper operation of the CO_2 dehydration system. In fact, both the desiccant solid beds and the TEG systems require a maximum inlet temperature of 50 °C.

Figure 3-1 Post-combustion - Case B1: CO_2 Compression Unit scheme.



3.1.2 CO_2 compression unit performance

The main battery limit streams conditions, the thermal integration with the power plant and the CO_2 capture unit, the electrical and cooling water consumption of the CO_2 Compression unit for Case B1 are summarized in Table 3-2.

The necessity of cooling down the CO_2 upstream of the dehydration process implies that, along the compression path, there must be a low temperature point in correspondence of the dryer operating pressure, which prevents the vapour recompression mechanism from being fully effective.

The potential benefits associated to the vapour recompression are also smoothed by the thermal integration with the ST condensate preheating system, as mentioned in the previous section.

Table 3-2 Post-combustion – Case B1: Vapour Recompression.

POST-COMBUSTION **Base case B1: CO_2 compression consumption**		
CO_2 Inlet Stream		
Flowrate	290461	Nm^3/h
Temperature	38	°C
Pressure	1.6	bar a
CO_2 Outlet Stream		
Flowrate	278518	Nm^3/h
Temperature	83	°C
Pressure	111	bar a
CO_2 purity	99.99	% v/v
Overall Plant Carbon Capture		
Carbon Capture	86.7	%
Cooling Water		
CW consumption	1675	t/h
Thermal Integration with the Power Plant		
Condensate pre-heating	36.0	MW_{th}
Thermal Integration with the CO_2 Capture Unit		
MEA Reboiling	35.1	MW_{th}
Compressor Electrical Consumption		
1^{st} stage	21.7	MW_e
2^{nd} stage	32.0	MW_e
3^{rd} stage	9.6	MW_e
4^{th} stage	6.1	MW_e
TOTAL	**69.4**	**MW_e**

The following Table 3.3 shows the performance delta between this compression strategy (B1) and the Base Case (B0). From the figures in the table, it can be drawn that this strategy is not advantageous, as it leads to an equivalent consumption increase of approximately 2.9 MWe.

Table 3-3 Post-combustion – Case B1: Performance delta with respect to the Base Case.

POST-COMBUSTION Case B1: Performance delta with respect to the base case B0					
Thermal Integration with the Power Plant / CO_2 capture unit					
Steam cons. for Condensate Pre-heating	0	MW_{th}	⟷	0	MW_e
Steam cons. for MEA Reboiling	- 35.1	MW_{th}	⟷	- 9.3	MW_e
Cooling water					
CW consumption	- 3717	t/h	⟷	- 0.4	MW_e
Compressor/Turbine Electrical Consumption					
Overall electrical consumption difference				+ 12.5	MW_e
Overall Plant Electrical Power Gap					
TOTAL				**+ 2.9**	**MW_e**

Note: Negative value indicates a lower consumption with respect to the base case. Conversion factors, see Section B.

3.2 Case B2 - Solvent stripper pressure increase

3.2.1 CO_2 compression unit description

The reference process flow diagram for this compression strategy is shown in Figure 3-2 and the related Heat and Material Balance is shown in Table 3-4, Table 3-5, Table 3-6 and Table 3-7 for different alternatives.

The compression strategy associated to this case B2 is the increase of the CO_2 pressure as released from the CO_2 capture unit, to reduce the overall pressure ratio for the compression unit and, therefore, its parasitic electrical consumption. The higher stripper operating pressure is also expected to induce a lower specific heat requirement for the solvent stripping in the reboiler, since at high pressure (and therefore high temperature) the CO_2 mass transfer rate, throughout the stripper column, is positively affected via the increased driving force [1], [2].

However, it might be realistic to expect that higher amine degradation rates and corrosion problems will occur at these elevated pressures and temperatures. This has to be taken into account in the evaluation of the strategy. In the present study, the effect of a higher MEA degradation is preliminary estimated and included in the differential OPEX with respect to the Base Case (refer to Section 6).

Also, the higher stripper operating temperature will require a higher steam pressure at ST extraction. This has a negative impact on the overall performance of the plant and partially off-sets the benefits highlighted above.

It is noted that the considerations made in this section shall be deemed as preliminary only; they are the results of technical simulations made by dedicated software, so validity of these results should be checked and confirmed by the solvent Licensors of this technology.

The impacts of this compression strategy are evaluated at two different stripper overhead pressure levels, as shown in the next sections:

- Alternative A: 210 kPa;
- Alternative B: 260 KPa.

Figure 3-2 Post-combustion - Case B2: CO_2 Compression Unit scheme.

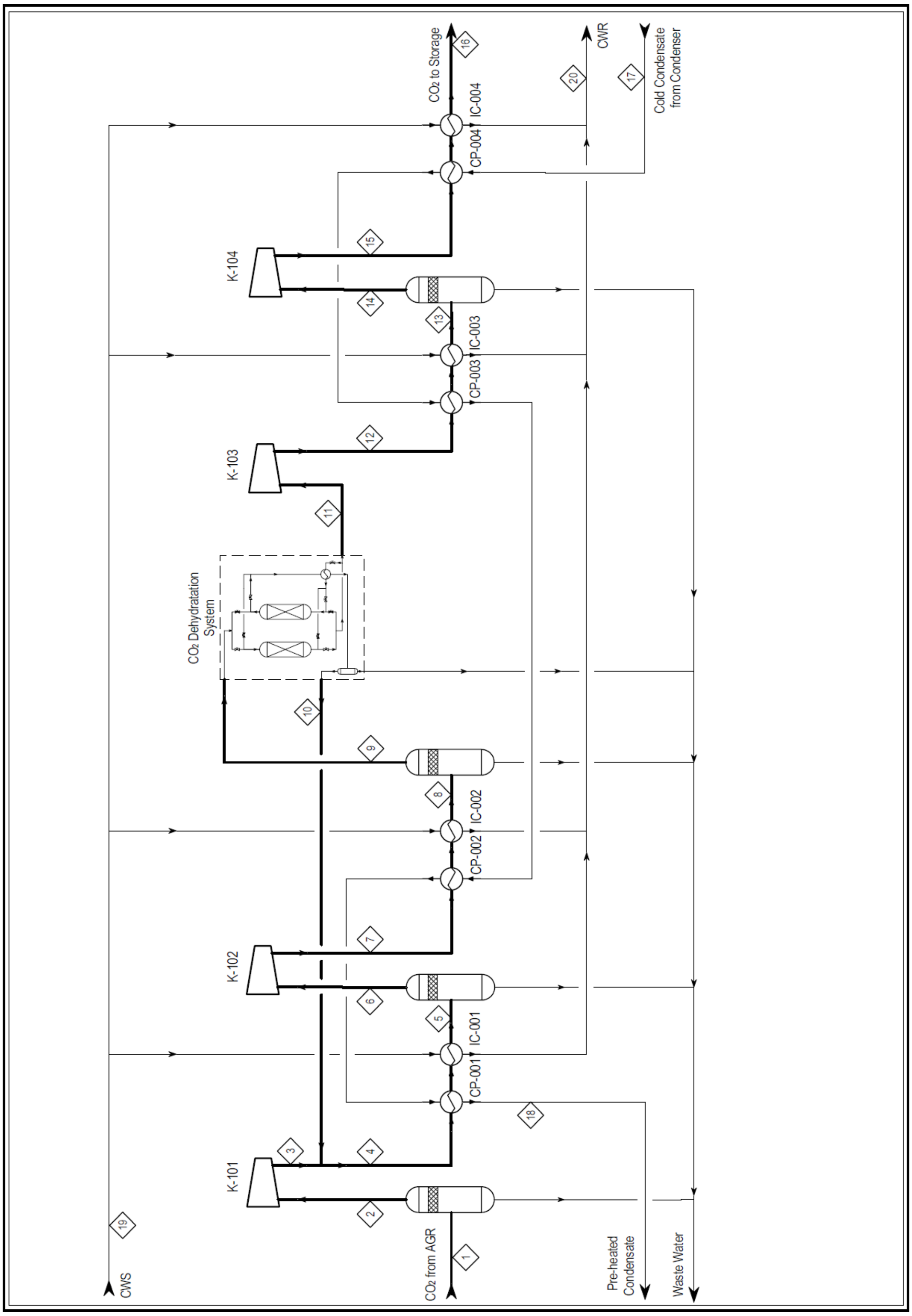


Table 3-4 Post-combustion – Case B2A: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Post-Combustion - Case B2a										
STREAM	1	2	3	4	5	6	7	8	9	10
Temperature (°C)	38	38	176.8	161.7	19	19	155	19	19	17
Pressure (bar)	2.1	2.1	8.5	8.5	8.3	8.3	34.0	33.6	33.6	8.5
TOTAL FLOW										
Mass flow (kg/h)	553878	553878	553878	614923	614923	608262	608262	608262	607783	61044
Molar flow (kgmole/h)	12827	12827	12827	14217	14217	13847	13847	13847	13821	1389
LIQUID PHASE										
Mass flow (kg/h)					6661			479		
GASEOUS PHASE										
Mass flow (kg/h)	553878	553878	553878	614923	608262	608262	608262	607783	607783	61044
Molar flow (kgmole/h)	12827	12827	12827	14217	13847	13847	13847	13821	13821	1389
Molecular Weight	43.18	43.18	43.18	43.25	43.25	43.93	43.93	43.93	43.98	43.94
Composition (vol %)										
CO_2	96.79	96.79	96.79	97.08	97.08	99.67	99.67	99.67	99.86	99.71
N_2	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02
O_2	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Ar	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
H_2O	3.18	3.18	3.18	2.90	2.90	0.30	0.30	0.30	0.11	0.27

Table 3-5 Post-combustion – Case B2A: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Post-Combustion - Case B2a										
STREAM	11	12	13	14	15	16	17	18	19	20
Temperature (°C)	24	97	40	40	81	76	75	99	12	19
Pressure (bar)	32.7	70.0	69.6	69.6	111.2	111.0	10.0	9.2	6.0	5.8
TOTAL FLOW										
Mass flow (kg/h)	546681	546681	546681	546681	546681	546681	1001214	1001214	5103000	5103000
Molar flow (kgmole/h)	12423	12423	12423	12423	12423	12423	55574	55574	283248	283248
LIQUID PHASE										
Mass flow (kg/h)							1001214	1001214	5103000	5103000
GASEOUS PHASE										
Mass flow (kg/h)	546681	546681	546681	546681	546681	546681	0	0	0	0
Molar flow (kgmole/h)	12423	12423	12423	12423	12423	12423	0	0	0	0
Molecular Weight	44.01	44.01	44.01	44.01	44.01	44.01	18.02	18.02	18.02	18.02
Composition (vol %)										
CO_2	99.97	99.97	99.97	99.97	99.97	99.97	0.00	0.00	0.00	0.00
N_2	0.02	0.02	0.02	0.02	0.02	0.02	0.00	0.00	0.00	0.00
O_2	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Ar	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
H_2O	0.00	0.00	0.00	0.00	0.00	0.00	100.00	100.00	100.00	100.00

Table 3-6 Post-combustion – Case B2B: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Post-Combustion - Case B2b										
STREAM	1	2	3	4	5	6	7	8	9	10
Temperature (℃)	38	38	164.5	150.6	19	19	155	19	19	18
Pressure (bar)	2.6	2.6	9.4	9.4	9.2	9.2	34.0	33.6	33.6	9.4
TOTAL FLOW										
Mass flow (kg/h)	552285	552285	552285	613313	613313	608015	608015	608015	608015	61028
Molar flow (kgmole/h)	12744	12744	12744	14133	14133	13839	13839	13839	13826	1389
LIQUID PHASE										
Mass flow (kg/h)					5298			413		
GASEOUS PHASE										
Mass flow (kg/h)	552285	552285	552285	613313	608015	608015	608015	607601	608015	61028
Molar flow (kgmole/h)	12744	12744	12744	14133	13839	13839	13839	13826	13826	1389
Molecular Weight	43.34	43.34	43.34	43.39	43.39	43.93	43.93	43.93	43.98	43.94
Composition (vol %)										
CO_2	97.39	97.39	97.39	97.62	97.62	99.69	99.69	99.69	99.86	99.71
N_2	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02
O_2	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Ar	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
H_2O	2.58	2.58	2.58	2.35	2.35	0.28	0.28	0.28	0.11	0.27

Table 3-7 Post-combustion – Case B2B: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE

Post-Combustion - Case B2b

STREAM	11	12	13	14	15	16	17	18	19	20
Temperature (°C)	24	97	40	40	81	76	75	96	12	19
Pressure (bar)	32.7	70.0	69.6	69.6	111.2	111.0	10.0	9.2	6.0	5.8
TOTAL FLOW										
Mass flow (kg/h)	546518	546518	546518	546518	546518	546518	1027882	1027882	4873000	4873000
Molar flow (kgmole/h)	12419	12419	12419	12419	12419	12419	57054	57054	270482	270482
LIQUID PHASE										
Mass flow (kg/h)							1027882	1027882	4873000	4873000
GASEOUS PHASE										
Mass flow (kg/h)	546518	546518	546518	546518	546518	546518	0	0	0	0
Molar flow (kgmole/h)	12419	12419	12419	12419	12419	12419	0	0	0	0
Molecular Weight	44.01	44.01	44.01	44.01	44.01	44.01	18.02	18.02	18.02	18.02
Composition (vol %)										
CO_2	99.97	99.97	99.97	99.97	99.97	99.97	0.00	0.00	0.00	0.00
N_2	0.02	0.02	0.02	0.02	0.02	0.02	0.00	0.00	0.00	0.00
O_2	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Ar	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
H_2O	0.00	0.00	0.00	0.00	0.00	0.00	100.00	100.00	100.00	100.00

3.2.2 CO_2 compression unit performance

The main battery limit streams conditions, the thermal integration with the power plant, the electrical and the cooling water consumption of the CO_2 Compression unit for Case B2 – alternative A are summarized in Table 3-8.

Table 3-8 Post-combustion – Case B2A: Higher Stripper pressure (210 kPa).

POST-COMBUSTION Case B2A: CO_2 capture/compression consumption		
CO_2 Inlet Stream		
Flowrate	287514	Nm^3/h
Temperature	38	°C
Pressure	2.1	bar a
CO_2 Outlet Stream		
Flowrate	278448	Nm^3/h
Temperature	76	°C
Pressure	111	bar a
CO_2 purity	99.99	% v/v
Overall Plant Carbon Capture		
Carbon Capture	86.7	%
Cooling Water		
CW consumption	5103	t/h
CW cons. Stripper Overhead Cond.	8388	t/h
Thermal Integration with the Power Plant		
Condensate pre-heating	28.0	MW_{th}
Compressor Electrical Consumption		
1st stage	20.3	MW_e
2nd stage	20.0	MW_e
3rd stage	8.0	MW_e
4th stage	3.6	MW_e

POST-COMBUSTION Case B2A: CO_2 capture/compression consumption	
TOTAL	**51.9 MW_e**

Table 3-9 shows the performance delta between this compression strategy (B2-A) and the Base Case (B0). Overall, there is a net equivalent power consumption reduction of 4.4 MWe. The equivalent consumption for the ST condensate preheating has slightly increased with respect to the Base Case, since the waste heat recoverable from the CO_2 compression is lower, due to the reduction of the overall pressure ratio. For the same reason, the Cooling Water demand of the compression unit decreases. Another contribution to the CW demand reduction is given by the lower duty of the MEA stripper overhead condenser, which is driven by the higher operating pressure.

It is noted that the overall beneficial effects of reduced steam consumption overlaps with the adverse effects of the different steam conditions. Therefore the approach of the equivalent electrical consumption gap is not applicable to this particular case. Table 3-9 simply reports the resulting net differential ST output.

Table 3-9 Post-combustion – Case B2A: Performance delta with respect to the base case.

POST-COMBUSTION Case B2A: Performance delta with respect to the base case B0				
Steam turbine output				
Net differential ST output				- 1.2 MW_e
Cooling water				
CW consumption (1), (2)	- 577	t/h	⟷	- 0.1 MW_e
Compressor/Turbine Electrical Consumption				
Overall electrical consumption difference (1)				- 5.5 MW_e
Overall Plant Electrical Power Gap				
TOTAL (1)				**- 4.4 MW_e**

Note 1: Negative value indicates a lower consumption with respect to the base case. Conversion factors, Section B.
Note 2: Including differential duty of the MEA Stripper Overhead Condenser.

As far as alternative B is concerned, the main results are reported in Table 3-10 and Table 3-11, the latter showing a net equivalent power consumption reduction of 7.3

MWe. Table 3-11 simply reports the resulting net differential ST output, for the same reason explained for alternative A.

Generally, the benefits on the overall consumption reported for alternative A are amplified in alternative B, due to the further increase of the stripper operating pressure. On the other hand, higher amine degradation rates and corrosion problems are expected in the CO_2 capture unit. As already stated, it would be recommended to further investigate this topic with the referenced solvent Licensors.

Table 3-10 Post-combustion – Case B2B: Higher Stripper pressure (260 kPa).

POST-COMBUSTION Case B2B: CO_2 capture/compression consumption		
CO_2 Inlet Stream		
Flowrate	285655	Nm^3/h
Temperature	38	°C
Pressure	2.1	bar a
CO_2 Outlet Stream		
Flowrate	278366	Nm^3/h
Temperature	76	°C
Pressure	111	bar a
CO_2 purity	99.99	% v/v
Overall Plant Carbon Capture		
Carbon Capture	86.7	%
Cooling Water		
CW consumption	4872	t/h
CW cons. Stripper Overhead Cond.	6711	t/h
Thermal Integration with the Power Plant		
Condensate pre-heating	24813	kW_{th}
Compressor Electrical Consumption		
1^{st} stage	18308	kW_e
2^{nd} stage	18205	kW_e
3^{rd} stage	7972	kW_e
4^{th} stage	3615	kW_e
TOTAL	**48100**	**kW_e**

Table 3-11 Post-combustion – Case B2B: Performance delta with respect to the base case.

POST-COMBUSTION Case B2A: Performance delta with respect to the base case B0				
Steam turbine output				
Net differential ST output			- 2.3	MW_e
Cooling water				
CW consumption (1), (2)	- 2484	t/h	⟷ - 0.2	MW_e
Compressor/Turbine Electrical Consumption				
Overall electrical consumption difference (1)			- 9.4	MW_e
Overall Plant Electrical Power Gap				
TOTAL (1)			**- 7.3**	**MW_e**

Note 1: Negative value indicates a lower consumption with respect to the base case. Conversion factors, Section B.
Note 2: Including differential duty of the MEA Stripper Overhead Condenser.

3.3 Case B3 - Staging of solvent regeneration

3.3.1 CO_2 compression unit description

The reference process flow diagrams of this compression strategy is shown in Figure 3-4.

The Base Case for post-combustion capture already includes a flash of the preheated rich amine to produce a semi-lean amine stream, which is recycled back to the absorber at an intermediate height in the beds packing. Therefore, the concept of staging of solvent regeneration is introduced into the post-combustion capture process as a multi-pressure stripper (Figure 3-3), reflecting some schemes already analysed in the literature ([3], [4]) and avoiding the complication associated to the generation of further semi-lean streams in the regeneration staging. Although this represents a deviation with respect to some alternative configurations proposed in the literature [5], it has to be noticed that those analyses start from a base case in which the heat integration scheme between rich and lean amine is simple and there is no semi-lean solvent production as in the Base Case (B0) of the reference study.

Figure 3-3 Post-combustion - Case B3: multi-pressure stripper.

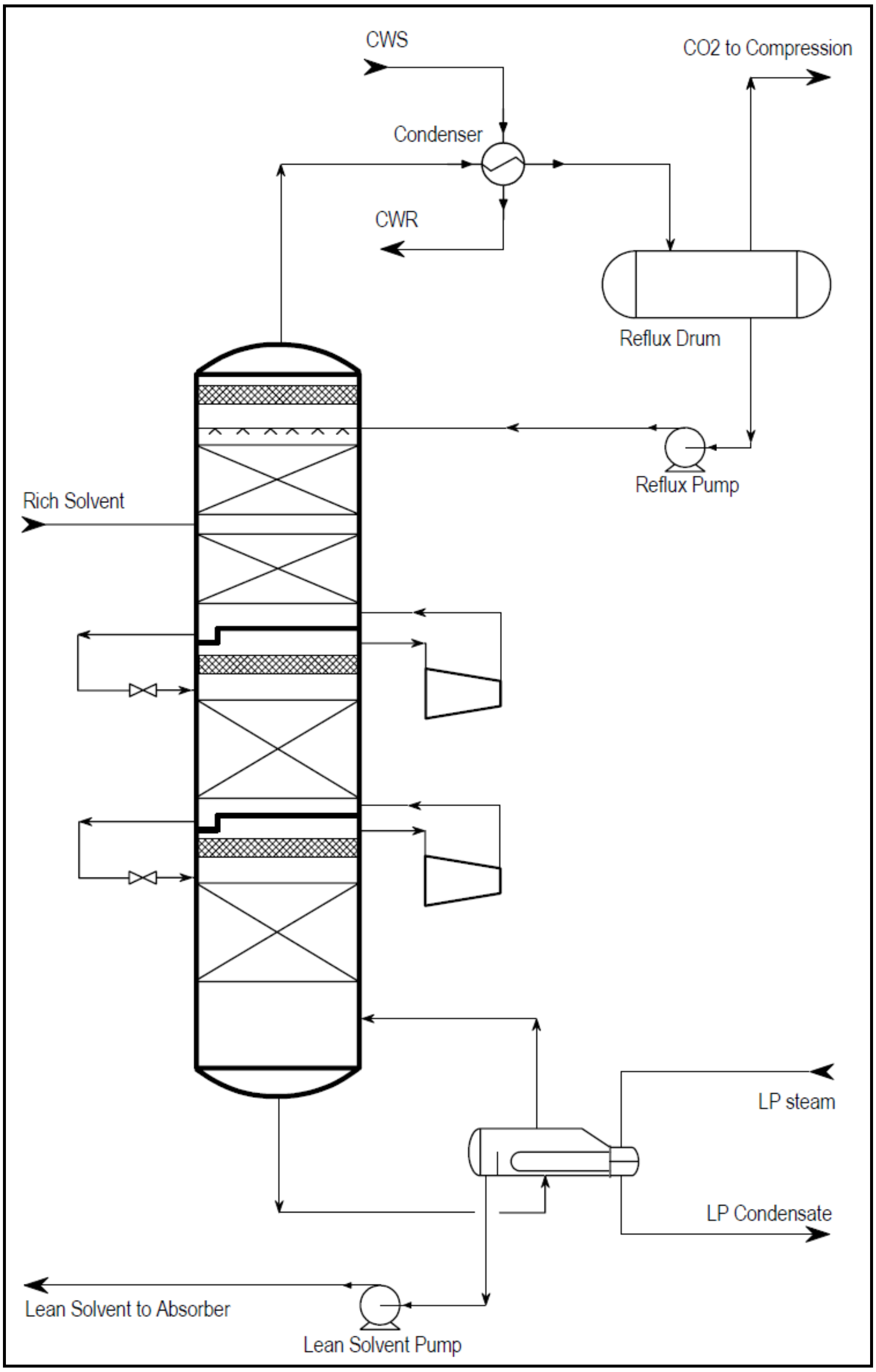
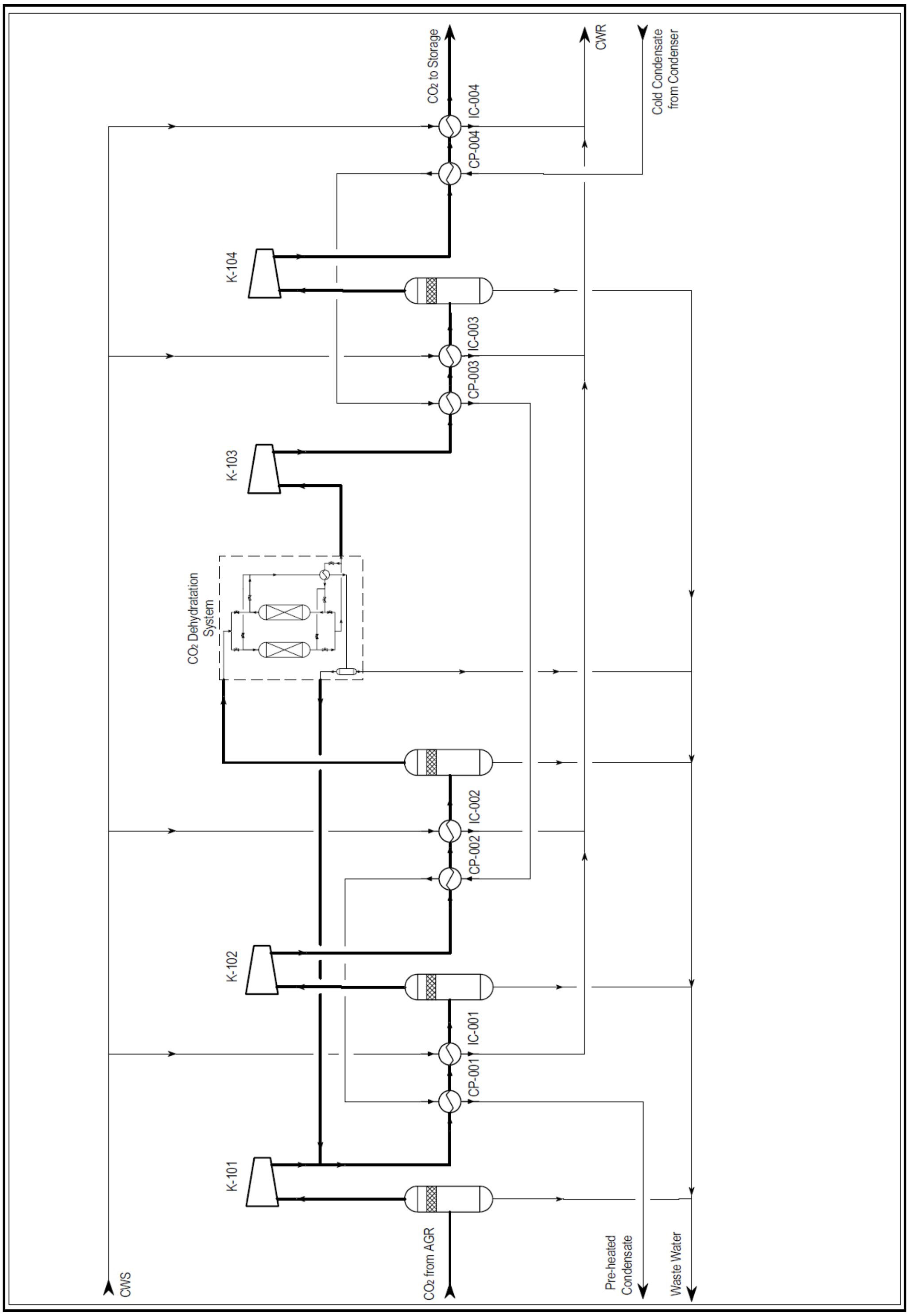


Figure 3-4 Post-combustion - Case B3: CO_2 Compression Unit scheme.



The multi pressure stripper operates at three different pressure levels (160 kPa, 230 kPa and 330 kPa), with two additional compressors installed to take the stripping vapour from the bottom pressure level to the top one. The increase of the parasitic power associated to the additional compressors is potentially off-set by:

- a significant reduction of the reboiler heat requirement, as part of the stripping is carried out at higher pressure;
- a lower parasitic consumption of the conventional Compression Unit, due to the higher pressure at which the CO_2 is released from the stripper.

The operating pressure of the reboiler is the same as the Base Case; hence the steam extraction pressure from the Steam Turbine is not affected.

3.3.2 CO_2 compression unit performance

The main battery limit streams conditions, the heat, the electrical and cooling water consumption of the CO_2 Capture/Compression units for Case B3 are summarized in Table 3-12.

Table 3-12 Post-combustion – Case B3: Staging of Solvent regeneration.

POST-COMBUSTION Case B2: CO_2 capture/compression consumption		
CO_2 Inlet Stream		
Flowrate	283292	Nm^3/h
Temperature	38	°C
Pressure	3.3	bar a
CO_2 Outlet Stream		
Flowrate	278545	Nm^3/h
Temperature	62	°C
Pressure	111	bar a
CO_2 purity	99.99	% v/v
Overall Plant Carbon Capture		
Carbon Capture	86.7	%
Cooling Water		
CW cons. CO_2 compression	3909	t/h
CW cons. Stripper Overhead Cond.	1504	t/h
Amine Stripping		
Reboiler thermal duty	420.0	MW_{th}
Compressor Electrical Consumption		
Capture unit compressors	25.4	MW_e
1^{st} stage	18.2	MW_e
2^{nd} stage	14.1	MW_e
3^{rd} stage	8.0	MW_e
4^{th} stage	3.6	MW_e
TOTAL	**69.3**	**MW_e**

Table 3-13 shows the performance delta between this compression strategy (B3) and the Base Case (B0). Overall, there is a net equivalent power consumption reduction of 1.7 MWe. The equivalent consumption for ST condensate preheating has increased with respect to the base case for the following reasons:

- the waste heat recoverable from the CO_2 capture/compression is lower due to the reduction of the overall pressure ratio;
- the higher operating pressure of the MEA Stripper Overhead Condenser reduces the total Condensing Duty, part of which is recovered into the ST condensate preheating.

For the same reasons, the Cooling Water demand decreases significantly.

Table 3-13 Post-combustion – Case B3: Performance delta with respect to the base case.

POST-COMBUSTION Case B3: performance delta with respect to the base case B0					
Thermal Integration with the Power Plant					
Condensate pre-heating	+ 19.9	MW_{th}	⟷	+ 5.3	MW_e
MEA stripping Heat Requirement					
Reboiler Duty	- 70.0	MW_{th}	⟷	- 18.5	MW_e
Cooling water					
CW consumption	- 8655	t/h	⟷	- 0.9	MW_e
Compressor/Turbine Electrical Consumption					
Overall electrical consumption difference				+ 12.4	MW_e
Overall Plant Electrical Power Gap					
TOTAL				**- 1.7**	**MW_e**

Note 1: Negative value indicates a lower consumption with respect to the base case. Conversion factors, see Section B.

The calculated overall performance improvement is not as high as shown in the literature [3], [4]. This is because in the present analysis the Base Case scheme to evaluate the alternative configurations presents a deep thermal integration between the absorber and the stripper sections, contrary to what shown in the reference studies. This optimisation allows minimising the heat demand of the reboiler across all the cases and therefore smoothes the reduction of the reboiler thermal duty associated to the multi-pressure stripper configuration.

3.4 Case B4 - Re-use of waste heat from the CO_2 compression

The discharge temperature of the first two stages of the CO_2 compression in the Base Case would allow a re-use in the amine stripping process. However, a portion of the CO_2 compression waste heat is recovered to preheat the Steam Turbine condensate at condensate pump discharge, as described in section B. The heat source for both ST condensate preheating and MEA stripping is the LP steam extraction at 3.25 bara from the ST. Therefore, moving the CO_2 compression heat recovery from the ST preheating to the Amine stripping would not lead to the improvement of the overall steam balance across the Steam Turbine.

For this reason, this compression strategy is not further investigated in this study.

4 Oxy-fuel combustion strategies

With respect to the Base Case (C0) shown in Section B, the CO_2 compression and the process units integrated with this system have been modified in order to investigate alternative CO_2 compression strategies. The technical assessment is made for the strategies listed in Table 4-1, while on economic assessment of the most advantageous alternatives is shown in Section 6.

Table 4-1 Oxy combustion – Summary of compression strategies.

Case tag	Description
Case C1	Expansion of incondensable
Case C2	Refrigeration of compressed CO_2
Case C3	CO_2 Liquefaction

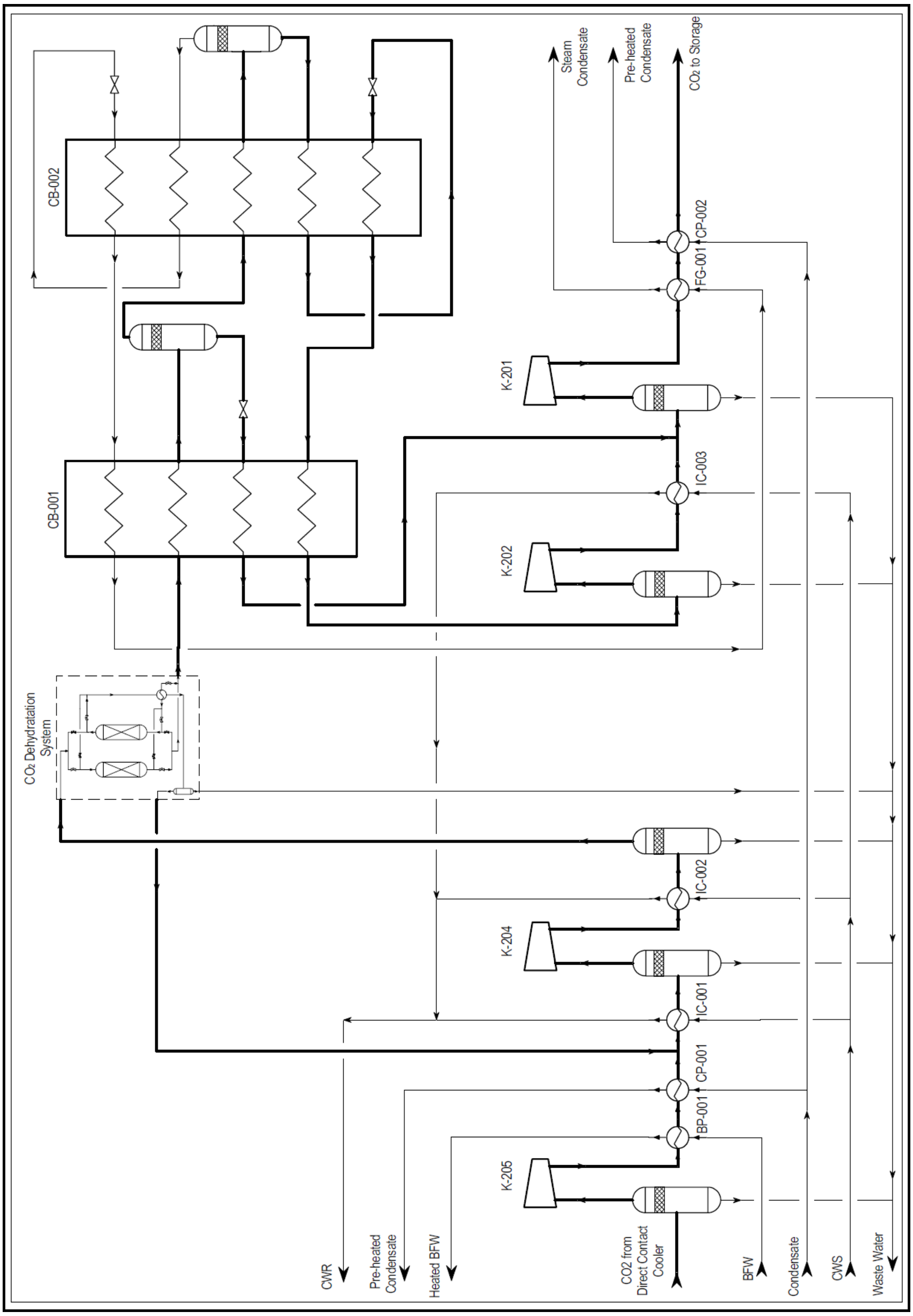
4.1 Case C1 - Expansion of incondensable

4.1.1 CO_2 compression unit description

The reference process flow diagram of this compression strategy is shown in Figure 4-1.

With respect to the Base Case C0, the incondensable coming from the last CO_2 cooling at -53°C are expanded. The expansion to atmospheric pressure reduces the temperature of this stream. The “cold” energy is recovered in the cold box and so the CO_2 expansion request for the auto-refrigeration in the cold box is reduced. As a consequence, the CO_2 exiting the auto-refrigeration system shows a pressure higher than the Base Case, leading to a lower power demand for the last two CO_2 compression stages. On the other hand, the Flue Gas Expander is not required in this configuration.

Figure 4-1 Oxy-combustion – Case C1: CO_2 Compression Unit scheme.



4.1.2 CO_2 compression unit performance

The main battery limit streams conditions, the thermal integration with the power plant, the electrical and cooling water consumption of the CO_2 Compression unit for Case C1 are summarized in Table 4-2.

Table 4-2 Oxy-combustion – Case C1: CO_2 Compression Unit consumption.

OXY-COMBUSTION Case C1: CO_2 compression consumption		
CO_2 Inlet Stream		
Flowrate	333,568	Nm^3/h
Temperature	12	°C
Pressure	1.0	bar a
CO_2 Outlet Stream		
Flowrate	236,684	Nm^3/h
Temperature	43	°C
Pressure	111	bar a
CO_2 purity	95.7	% v/v
Overall Plant Carbon Capture		
Carbon Capture	91.1	%
Cooling Water		
CW consumption	3,524	t/h
Thermal Integration with the Power Plant		
Condensate pre-heating	57.6	MW_{th}
BFW heating	16.4	MW_{th}
IP steam consumption	0	MW_{th}
Compressor/Turbine Electrical Consumption		
1st stage	43.9	MW_e
2nd stage	14.7	MW_e

OXY-COMBUSTION Case C1: CO_2 compression consumption		
3rd stage	3.1	MW_e
4th stage	17.6	MW_e
Flue Gas Expander	0	MW_e
TOTAL	**79.3**	**MW_e**

Table 4-2 shows the performance delta between this compression strategy (C1) and the Base Case (C0). Overall, there is a net power consumption increase of 5.9 MWe. It may be noted from the above table that Case C1 presents a more efficient thermal integration with the rest of the plant. This is due firstly to the fact that the flue gas are heated in the last flue gas exchanger up to 20°C (stack exit temperature), instead of 170°C as per the Base case C0; this leads to a higher heat in the CO_2 that is recovered with the condensate pre-heating. In addition to that, the flue gas does not require to be preheated before the flue gas expander, thus saving 5.1 MWth of IP steam.

On the other hand, the absence of the flue gas expander (approximately 9.8 MWe) is not compensated by the gain in the thermal integration mentioned before. For this reason, the summary electrical consumption of the CO_2 Compression unit is higher than the Base Case, leading to the conclusion that this alternative is not technically attractive.

Table 4-3 Oxy-combustion – Case C1: Consumption gap respect to the base case.

OXY-COMBUSTION Case C1: Consumption gap respect to the base case C0					
Thermal Integration with the Power Plant					
Condensate pre-heating	- 5.8	MW_{th}	⟷	- 2.7	MW_e
BFW heating	0	MW_{th}	⟷	0	MW_e
IP steam consumption	- 5.1	MW_{th}	⟷	- 1.2	MW_e
Cooling water					
CW consumption	0	t/h		0	MW_e
Compressor/Turbine Electrical Consumption					
Overall electrical consumption difference				+ 9.8	MW_e

Overall Plant Electrical Power Gap	
TOTAL	**+ 5.9 MW_e**

Note: Negative value indicates a lower consumption with respect to the base case. Conversion factors, see Section B.

4.2 Case C2 - Refrigeration of compressed CO_2

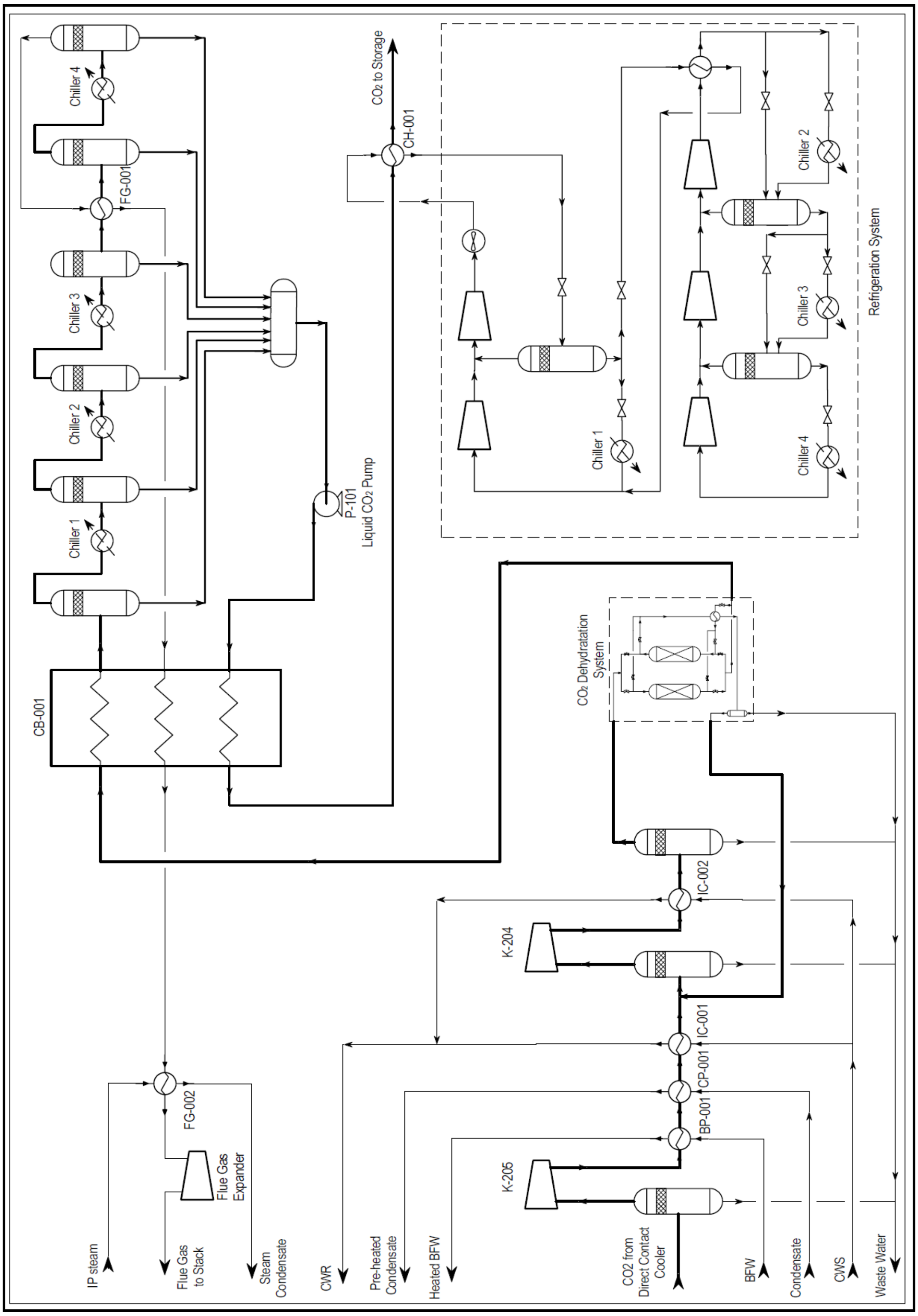
4.2.1 CO_2 compression unit description

The reference process flow diagram of this compression strategy is shown in Figure 4-2.

In this case C2, the CO_2 is refrigerated with an external chiller system, instead of using an auto-refrigeration cycle as described in the Base Case C0. After the first two steps of compression and after the Dehydration system, the CO_2 stream enters a train of exchangers, where four chillers cool the CO_2 down to -53°C. After each chiller, liquid CO_2 is separated and collected. Finally, liquid CO_2, with a purity of 96.1% by volume, is pumped up at 111 bara.

The chiller's duty is provided by a conventional cycle refrigeration circuit based on a cascade system. The "warmer" circuit is composed by a two-stage propane compression/expansion system; the propane expansion provides the required duty to the condenser of the "cooler" circuit. On the other hand, the "cooler" circuit is composed by a three-stage ethane compression/expansion system; each expansion stage provides, at different temperature, the chilling power to the CO_2 stream.

Figure 4-2 Oxy-combustion – Case C2: CO_2 Compression Unit scheme.



4.2.2 CO_2 compression unit performance

The main battery limit streams conditions, the thermal integration with the power plant, the electrical and cooling water consumption of the CO_2 Compression unit for Case C2 are summarized in Table 4-4.

Table 4-4 Oxy-combustion – Case C2: CO_2 Compression Unit consumption.

OXY-COMBUSTION Case C2: CO_2 compression consumption		
CO_2 Inlet Stream		
Flowrate	333,568	Nm^3/h
Temperature	12	°C
Pressure	1.0	bar a
CO_2 Outlet Stream		
Flowrate	235,636	Nm^3/h
Temperature	10	°C
Pressure	111	bar a
CO_2 purity	96.1	% v/v
Overall Plant Carbon Capture		
Carbon Capture	90.9	%
Cooling Water		
CW consumption	3,254	t/h
Thermal Integration with the Power Plant		
Condensate pre-heating	23.8	MW_{th}
BFW heating	16.4	MW_{th}
IP steam consumption	11.1	MW_{th}
Compressor/Turbine Electrical Consumption		
1st stage	43.9	MW_e
2nd stage	14.7	MW_e

OXY-COMBUSTION Case C2: CO_2 compression consumption		
CO_2 Pump	1.3	MW_e
Refrigeration package	18.0	MW_e
Flue Gas Expander	-9.9	MW_e
TOTAL	**68.0**	**MW_e**

Table 4-4 shows the performance delta between this compensation strategy (C2) and the Base Case (C0). Overall, there is a net power consumption increase of 7.7 MWe. The advantage of using pump, instead of compressor, and a moderate consumption of the selected refrigeration cycle (COP of about 2) leads to a saving of 1.5 MWe. However, the lack of low temperature heat recovery, from the compressors exit decreases substantially the condensate pre-heating. As a consequence, the condensate needs to be pre-heated inside the thermal cycle through steam extraction from the steam turbine, thus leading to an electric power production loss.

Table 4-5 Oxy-combustion – Case C2: Performance delta with respect to the base case.

OXY-COMBUSTION Case C2: Performance delta with respect to the base case C0					
Thermal Integration with the Power Plant					
Condensate pre-heating	+ 28.0	MW_{th}	⟷	+ 6.0	MW_e
BFW heating	0	MW_{th}	⟷	0	MW_e
IP steam consumption	+ 6.0	MW_{th}	⟷	+ 3.2	MW_e
Cooling water					
CW consumption	- 270	t/h	⟷	~ 0	MW_e
Compressor/Turbine Electrical Consumption					
Overall electrical consumption difference				- 1.5	MW_e
Overall Plant Electrical Power Gap					
TOTAL				**+ 7.7**	**MW_e**

Note: Negative value indicates a lower consumption respect to the base case. Conversion factors, see Section B.

4.3 Case C3 - CO_2 Liquefaction

4.3.1 CO_2 compression unit description

The reference process flow diagram of this compression strategy is shown in Figure 4-3 and the related Heat and Material Balance is shown in Table 4-6, Table 4-7 and Table 4-8.

With respect to the base case C0, CO_2 is compressed in the last compression stage at 73 bara and firstly cooled against the cold incondensable stream from the cold box, secondly against condensate from the power island and finally condensed with cooling water. The resulting liquid stream is at 19°C and can be pumped up to 111 bara.

Figure 4-3 Oxy-combustion – Case C3: CO_2 Compression Unit scheme.

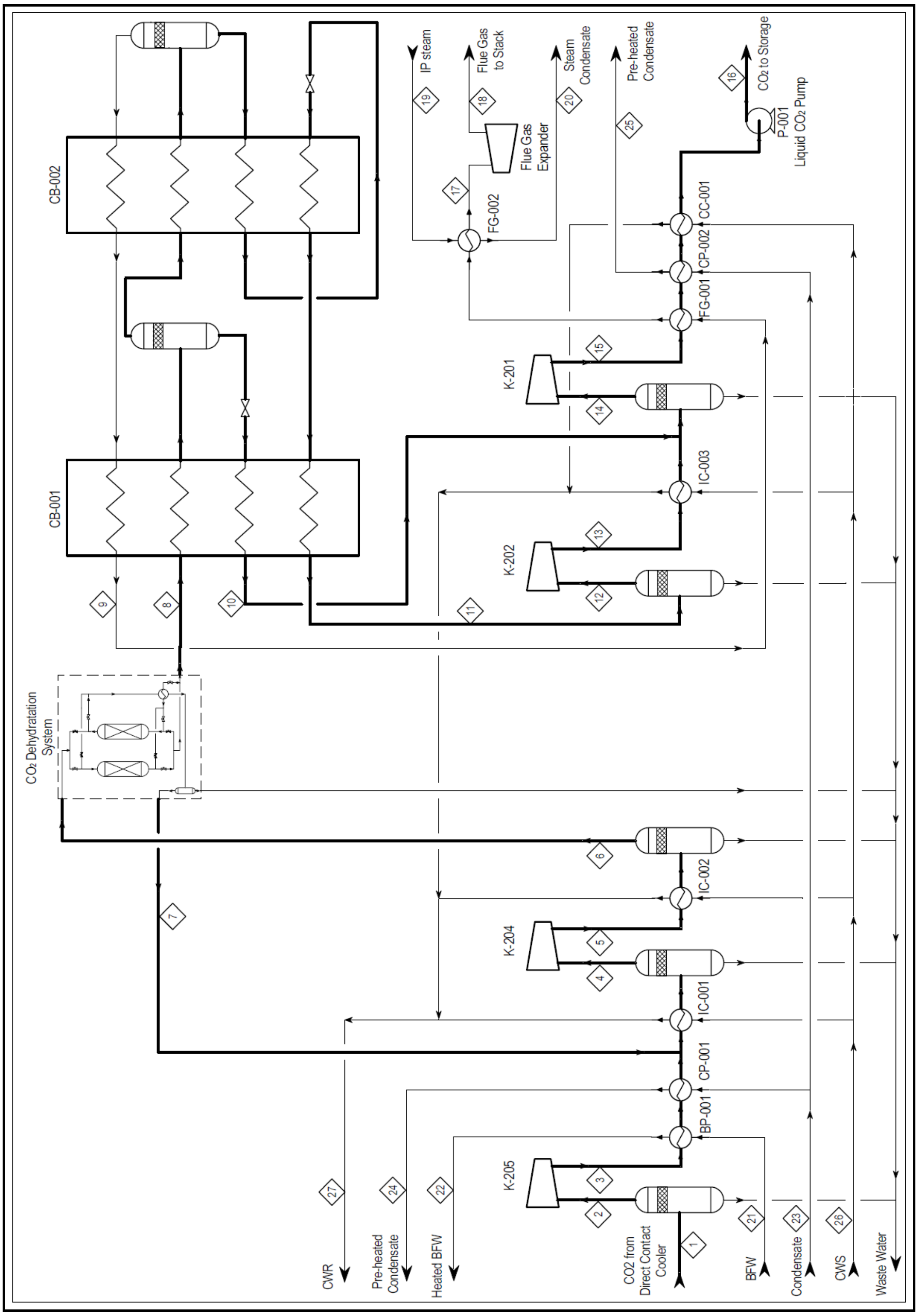


Table 4-6 Oxy-combustion – Case C3: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Oxy-Combustion - Case C3										
STREAM	1	2	3	4	5	6	7	8	9	10
Temperature (°C)	12	12	281	19	19	19	84	22	18	7
Pressure (bar)	1.0	1.0	15.0	14.4	14.4	30.9	34.0	30.0	28.9	18.6
TOTAL FLOW										
Mass flow (kg/h)	602055	602055	602055	664617	664617	664388	66246	597575	137173	228563
Molar flow (kgmole/h)	14882	14882	14882	16313	16313	16300	1635	14652	4092	5216
LIQUID PHASE										
Mass flow (kg/h)										
GASEOUS PHASE										
Mass flow (kg/h)	602055	602055	602055	664617	664617	664388	66246	597575	137173	228563
Molar flow (kgmole/h)	14882	14882	14882	16313	16313	16300	1635	14652	4092	5216
Molecular Weight	40.45	40.45	40.45	40.74	40.74	40.76	40.53	40.78	33.52	43.82
Composition (vol %)										
CO_2	74.66	74.66	74.66	75.62	75.62	75.67	74.90	75.76	24.24	96.52
N_2	14.98	14.98	14.98	15.18	15.18	15.19	15.03	15.20	49.15	1.46
O_2	6.15	6.15	6.15	6.23	6.23	6.23	6.17	6.24	19.33	0.80
Ar	2.41	2.41	2.41	2.44	2.44	2.45	2.42	2.45	7.15	0.41
SO_2	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.00	0.79
NO + NO_2	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.12	0.01
H_2O	1.45	1.45	1.45	0.19	0.19	0.11	1.13	0.00	0.00	0.00

Table 4-7 Oxy-combustion – Case C3: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Oxy-Combustion - Case C3										
STREAM	11	12	13	14	15	16	17	18	19	20
Temperature (°C)	7	7	66	13	188	43	300	20	620	277
Pressure (bar)	9.3	9.3	18.7	18.6	111.4	111.0	28.5	1.1	61.1	60.9
TOTAL FLOW										
Mass flow (kg/h)	231838	231838	231838	460402	460402	460402	137173	137173	10197	10197
Molar flow (kgmole/h)	5344	5344	5344	10560	10560	10560	4092	4092	566	566
LIQUID PHASE										
Mass flow (kg/h)						460402				10197
GASEOUS PHASE										
Mass flow (kg/h)	231838	231838	231838	460402	460402	0	137173	137173	10197	0
Molar flow (kgmole/h)	5344	5344	5344	10560	10560	0	4092	4092	566	0
Molecular Weight	43.39	43.39	43.39	43.60	43.60	43.60	33.52	33.52	18.02	18.02
Composition (vol %)										
CO_2	94.95	94.95	94.95	95.73	95.73	95.73	24.24	24.24	0.00	0.00
N_2	2.62	2.62	2.62	2.05	2.05	2.05	49.15	49.15	0.00	0.00
O_2	1.51	1.51	1.51	1.16	1.16	1.16	19.33	19.33	0.00	0.00
Ar	0.83	0.83	0.83	0.63	0.63	0.63	7.15	7.15	0.00	0.00
SO_2	0.07	0.07	0.07	0.42	0.42	0.42	0.00	0.00	0.00	0.00
$NO + NO_2$	0.02	0.02	0.02	0.02	0.02	0.02	0.12	0.12	0.00	0.00
H_2O	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	100.00	100.00

Table 4-8 Oxy-combustion – Case C3: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Oxy-Combustion - Case C3										
STREAM	21	22	23	24	25	26	27			
Temperature (°C)	165	206	33	95	67	12	19			
Pressure (bar)	21.0	21.0	6.0	5.8	5.8	6.0	5.8			
TOTAL FLOW										
Mass flow (kg/h)	329940	329940	684970	330635	354335	6061204	6061204			
Molar flow (kgmole/h)	18314	18314	38020	18352	19668	336434	336434			
LIQUID PHASE										
Mass flow (kg/h)	329940	329940	684970	330635	354335	6061204	6061204			
GASEOUS PHASE										
Mass flow (kg/h)	0	0	0	0	0	0	0			
Molar flow (kgmole/h)	0	0	0	0	0	0	0			
Molecular Weight	18.02	18.02	18.02	18.02	18.02	18.02	18.02			
Composition (vol %)										
CO_2	0.00	0.00	0.00	0.00	0.00	0.00	0.00			
N_2	0.00	0.00	0.00	0.00	0.00	0.00	0.00			
O_2	0.00	0.00	0.00	0.00	0.00	0.00	0.00			
Ar	0.00	0.00	0.00	0.00	0.00	0.00	0.00			
SO_2	0.00	0.00	0.00	0.00	0.00	0.00	0.00			
$NO + NO_2$	0.00	0.00	0.00	0.00	0.00	0.00	0.00			
H_2O	100.00	100.00	100.00	100.00	100.00	100.00	100.00			

4.3.2 CO_2 compression unit performance

The main battery limit streams conditions, the thermal integration with the power plant, the electrical and cooling water consumption of the CO_2 Compression unit for Case C3 are summarized in Table 4-9.

Table 4-9 Oxy-combustion – Case C3: CO_2 Compression Unit consumption.

OXY-COMBUSTION Case C3: CO_2 compression consumption		
CO_2 Inlet Stream		
Flowrate	333,568	Nm^3/h
Temperature	12	°C
Pressure	1.0	bar a
CO_2 Outlet Stream		
Flowrate	236,684	Nm^3/h
Temperature	43	°C
Pressure	111	bar a
CO_2 purity	95.7	% v/v
Overall Plant Carbon Capture		
Carbon Capture	91.1	%
Cooling Water		
CW consumption	6,061	t/h
Thermal Integration with the Power Plant		
Condensate pre-heating	37.2	MW_{th}
BFW heating	16.4	MW_{th}
IP steam consumption	7.0	MW_{th}
Compressor/Turbine Electrical Consumption		
1^{st} stage	43.9	MW_e
2^{nd} stage	14.7	MW_e
3^{rd} stage	3.1	MW_e
4^{th} stage	12.1	MW_e
CO_2 Pump	0.9	MW_e
Flue Gas Expander	-9.8	MW_e
TOTAL	**64.9**	**MW_e**

Table 4-10 shows the performance delta between this compression strategy (C3) and the Base Case (C0). Overall, there is a net power consumption decrease of 0.2 MWe. Because of the lower exit pressure of the last CO_2 compression stage (73.0 bar versus 111.2 bara of the Base Case), the heat available for the flue gas heating and condensate pre-heating is lower. Therefore, the heat integration with the rest of the plant is lower and consequently more heat is required from the steam cycle to pre-heat the condensate and the flue gas upstream of the expander. Steam cycle modifications with respect to the Base case are shown in Figure 4-4.

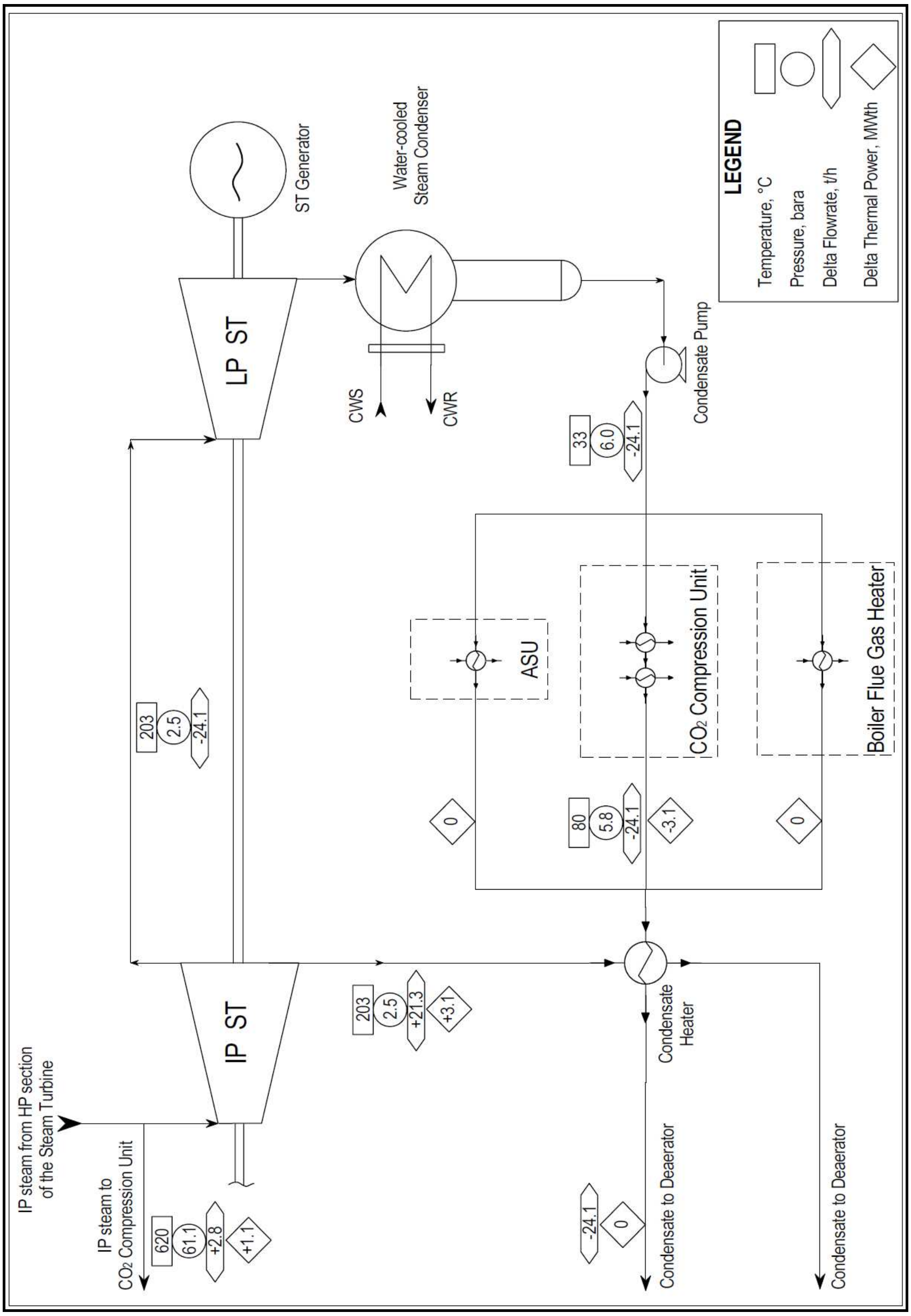
In addition, to condense the CO_2 stream a much higher amount of cooling water is required, leading to a slightly increase of the auxiliary unit consumption.
Anyway, the use of the pump shows a moderate electrical consumption saving that leads to a lower overall electrical demand of this compression strategy.

Table 4-10 Oxy-combustion – Case C3: Performance delta with respect to the base case.

OXY-COMBUSTION Case C3: performance delta with respect to the base case C0						
Thermal Integration with the Power Plant						
Condensate pre-heating	+ 14.6	MW_{th}	⟷	+ 3.1	MW_e
BFW heating	0	MW_{th}	⟷	0	MW_e
IP steam consumption	+ 1.9	MW_{th}	⟷	+ 1.1	MW_e
Cooling water					
CW consumption	+ 2,537	t/h	⟷	+ 0.2	MW_e
Compressor/Turbine Electrical Consumption					
Overall electrical consumption difference				- 4.6	MW_e
Overall Plant Electrical Power Gap					
TOTAL				**- 0.2**	**MW_e**

Note: Negative value indicates a lower consumption with respect to the base case. Conversion factors, see Section B.

Figure 4-4 Oxy-combustion – Case C3: CO_2 Compression Unit scheme.



5 General strategies for CO_2 compression

This section makes a technical assessment of general CO_2 compression strategies, aimed at minimising parasitic loads of the system, which could be applied to any type of CO_2 capture process is used (i.e. pre-, post- or oxy-fuel).

The technical assessment is made for the strategies summarised in Table 5-1. If deemed appropriate, the strategies are tailored for a specific capture technology. As for the other capture types, the economic assessment of the most advantageous alternatives is shown in Section 6.

Table 5-1: Summary of general compression strategies.

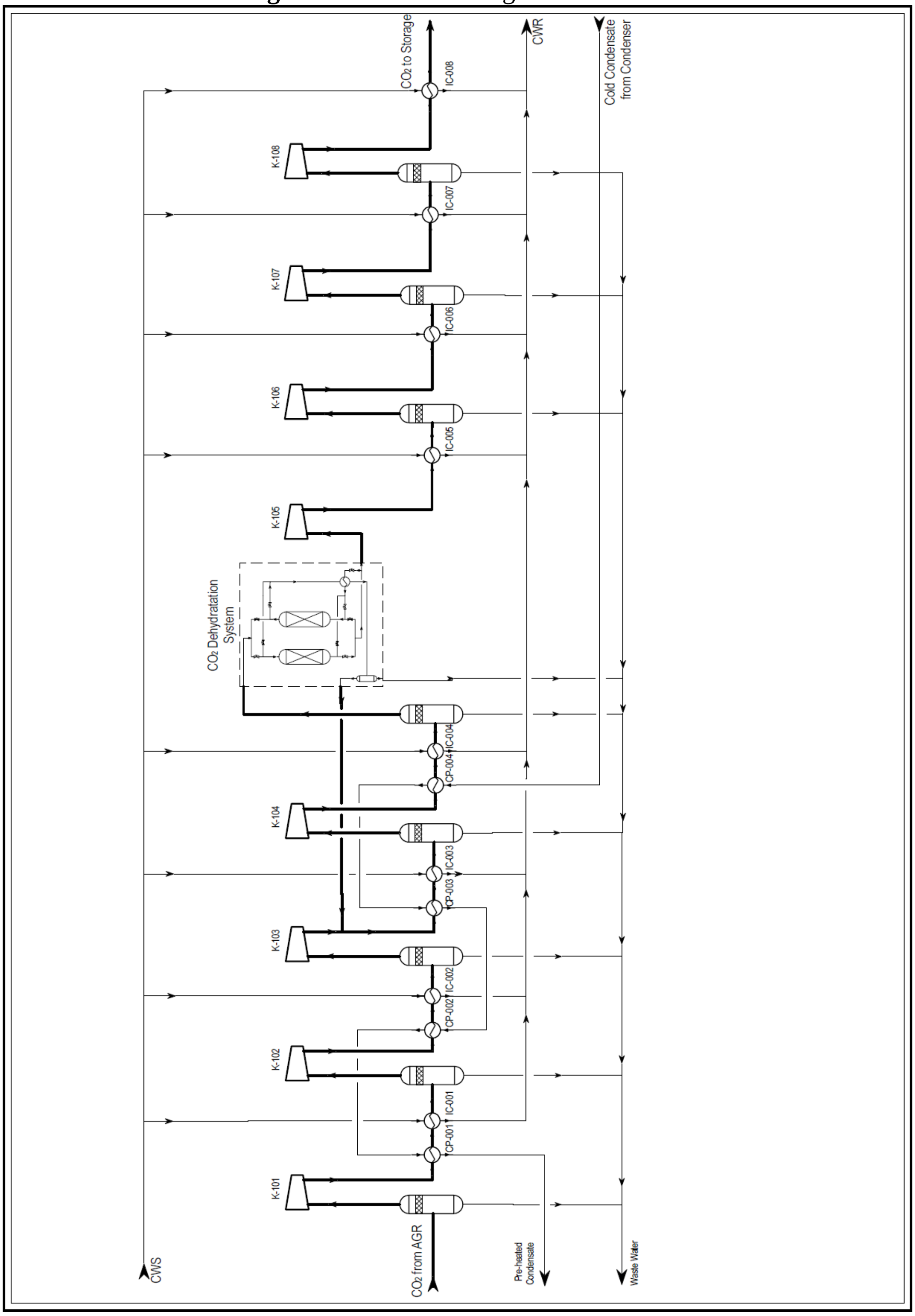
Case tag	Description
Case D1	Increasing number of stages
Case D2	CO_2 liquefaction
Case D3	Deeper inter-cooling

5.1 Case D1 – Increasing number of stages

The reference process flow scheme of this compression strategy is shown in Figure 5-1

The CO_2 capture type best suited to this compression strategy is the post-combustion, as there is a single stream entering the CO_2 compression unit and there are no process constraints for the definition of the inter-stage pressures. With respect to the Base Case (B0), the number of compression stages has been doubled in this strategy (i.e. 8 vs. 4).

Figure 5-1 Case D1: Stages number increased.



The main battery limit streams conditions, the thermal integration with the power plant, the electrical and cooling water consumption of the CO_2 Compression unit for Case D1 are summarized in Table 5-2.

Table 5-2: Case D1: CO_2 Compression Unit consumption.

CO_2 compression in general Case D1: CO_2 compression consumption		
CO_2 Inlet Stream		
Flowrate	290461	Nm^3/h
Temperature	38	°C
Pressure	1.6	bar a
CO_2 Outlet Stream		
Flowrate (liquid)	546.8	t/h
Temperature	35	°C
Pressure	111	bar a
CO_2 purity	99.99	% mol/mol
Overall Plant Carbon Capture		
Carbon Capture	86.7	%
Cooling Water		
CW cons. CO_2 compression	9752	t/h
Thermal Integration with the Power Plant		
Condensate pre-heating	13.5	MW_{th}
Compressor/Pump Electrical Consumption		
1^{st} stage	10.7	MW_e
2^{nd} stage	10.6	MW_e
3^{rd} stage	9.0	MW_e
4^{th} stage	10.3	MW_e
5^{th} stage	2.8	MW_e

CO_2 compression in general Case D1: CO_2 compression consumption	
6th stage	2.3 MW_e
7th stage	2.5 MW_e
8th stage	1.7 MW_e
TOTAL	**49.9 MW_e**

Table 5-3 shows the performance delta between this compression strategy (D1) and the Base Case (B0). From the figures in the table, it can be drawn that the increase of number of compression stages leads to a reduction of the compression energy, due to the presence of intercoolers at each inter-stage point. For this reason, the waste heat from the CO_2 compression is lower. The ST condensate pre-heaters are located only where the stage discharge temperatures are adequate for this service (ref. Figure 5-1). As a consequence, the lower ST condensate preheating effect achievable in the CO_2 compression unit leads to a higher steam consumption in the Power Island, thus partially off-setting the reduction of compression energy, as shown in Table 5-3. Overall, the total expected energy saving of this compression strategy is approximately 2.0 MWe.

Table 5-3: Case D1: Performance delta with respect to the base case.

CO_2 compression in general Case D1: Performance delta with respect to the base case B0			
Thermal Integration with the Power Plant			
Steam cons. for Condensate Pre-heating	+ 19.5 MW_{th}	⟷	+ 5.2 MW_e
Cooling water			
CW consumption	+ 4360 t/h	⟷	+ 0.4 MW_e
Compression/Pumping Electrical Consumption			
Overall electrical consumption difference			- 7.6 MW_e
Overall Plant Electrical Power Gap			
TOTAL			**- 2.0 MW_e**

Note: Negative value indicates a lower consumption with respect to the base case. Conversion factor, see section B.

5.2 Case D2 - CO_2 liquefaction

Early liquefaction of CO_2 allows the use of pumps for final pressure boosting, leading to a reduction of the compression energy. The analysis has been made for both the pre-combustion and the post-combustion cases, since the liquefaction options depend on the overall process configuration. For instance, the potential application of absorption refrigeration for CO_2 liquefaction has different implications in the two processes, as described in the following sections.

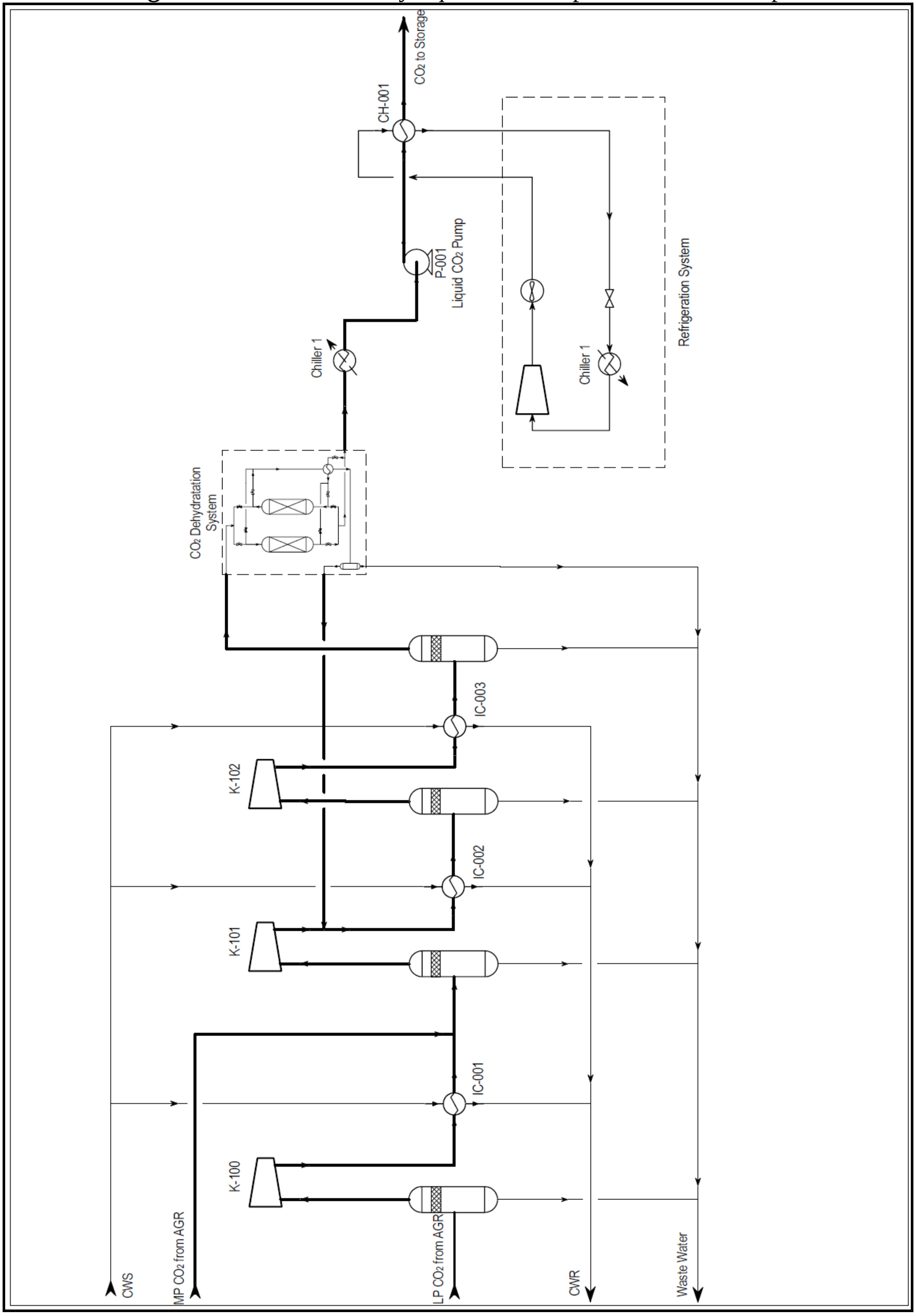
5.2.1 Case D2A: Early CO_2 liquefaction in the post-combustion capture

The relevant process flow scheme of this compression strategy is shown in Figure 5-2.

Early liquefaction of the CO_2 has been evaluated with the application of a conventional chiller, using propane as working fluid. The CO_2 is liquefied at -20°C, which allows pumping the CO_2 from a pressure of 20.5 bara.

In the proposed configuration, the CO_2 stream is compressed in a two stage compressor to the liquefaction pressure. At compressor discharge, the CO_2 is dehydrated before being liquefied.

Figure 5-2 Case D2a: Early liquefaction in post-combustion capture.



The Chiller is based on a single effect propane refrigeration cycle, which is optimised by thermal integration with the cold liquid CO_2 at Liquefier outlet. The cold CO_2 is used to sub-cool the liquid propane at chiller condenser outlet, thus reducing the propane circulation rate required to liquefy the carbon dioxide stream and consequently the chiller electrical consumption. The overall C.O.P. estimated for this refrigeration cycle is 3.8.

The main battery limit streams conditions, the thermal integration with the power plant, the electrical and cooling water consumption of the CO_2 Compression unit for Case D2A are summarized in Table 5-4.

Table 5-4: Case D2A: CO_2 Compression Unit consumption.

CO_2 compression in general Case D2A: CO_2 compression consumption		
CO_2 Inlet Stream		
Flowrate	290461	Nm^3/h
Temperature	38	°C
Pressure	1.6	bar a
CO_2 Outlet Stream		
Flowrate (liquid)	546.8	t/h
Temperature	6	°C
Pressure	111	bar a
CO_2 purity	99.99	% mol/mol
Overall Plant Carbon Capture		
Carbon Capture	86.7	%
Cooling Water		
CW cons. CO_2 compression	10399	t/h
Thermal Integration with the Power Plant		
Condensate pre-heating	23.9	MW_{th}

CO_2 compression in general Case D2A: CO_2 compression consumption	
Compressor/Pump Electrical Consumption	
1st stage	19.1 MW_e
2nd stage	20.0 MW_e
CO_2 pump	1.8 MW_e
TOTAL	**40.9 MW_e**
Chiller Electrical Consumption	
Conventional Chiller	13.5 MW_e

It is noted that the process could be slightly further optimised as the CO_2 outlet temperature is relatively low, i.e. some further refrigeration energy recovery may be possible.

With respect to the base case, the significant reduction of the compression energy is potentially off-set by the following factors:

- the electrical consumption of the chiller;
- the reduction of the CO_2 compression waste heat available for condensate preheating with a consequent increase of the steam consumption in the power island feed water heater;
- the higher electrical consumption associated to the increase of Cooling water usage, mainly due to the introduction of the chiller in the system.

With respect to the Base Case (B0), Table 5-5 shows a net equivalent consumption reduction of 0.2 MW. Therefore, the strategy does not lead to a significant optimisation of the energy demand.

Table 5-5: Case D2A: Performance delta with respect to base case.

CO_2 compression in general Case D2A: Performance delta respect to the base case B0					
Thermal Integration with the Power Plant					
Steam cons. for Condensate Pre-heating	+ 9.4	MW_{th}	⟷	+ 2.5	MW_e
Chiller					
Chiller electrical consumption				+ 13.4	MW_e
Cooling water					
CW consumption	+ 5007	t/h	⟷	+ 0.5	MW_e
Compression/Pumping Electrical Consumption					
Overall electrical consumption difference				- 16.6	MW_e
Overall Plant Electrical Power Gap					
TOTAL				**- 0.2**	**MW_e**

Note: Negative value indicates a lower consumption with respect to the base case. Conversion factors, see Section B.

The application of an ammonia absorption refrigeration system has been evaluated as well. Being no waste heat source available in the post-combustion capture process at the temperatures required for this application, the steam extracted at IP/LP cross over is used as heat source for the refrigeration cycle.

The main operating parameters of the absorption chiller are based on preliminary data available from Suppliers. A Coefficient Of Performance (defined as ratio between the chilling duty and absorbed heat) of 0.58 has been estimated with the heat source and the available CW, i.e. for each MW_{th} of chilling duty approx 1.72 MW_{th} are required as steam extraction. Considering the electrical power loss associated to the ST extraction (ref. Section B), the resulting equivalent electrical consumption is approx 0.45 MW_e per each MW_{th} of chilling duty.

The corresponding C.O.P. is 2.2, which is worse than the conventional chiller C.O.P and makes the absorption refrigeration option not advantageous for early CO_2 liquefaction in the post-combustion capture case.

5.2.2 Case D2B: Early CO_2 liquefaction in the pre-combustion capture

The reference process flow diagrams of this compression strategy are shown in Figure 5-3, while the relevant Heat & Mass Balance is reported in, Table 5-6, Table 5-7 and Table 5-8.

In an IGCC with CCS, the CO shift reaction in the Syngas treatment Unit typically makes a considerable amount of heat available at lower temperatures than the case without CCS. For this reason, in the syngas cooling unit of the reference case, the syngas heat recovery in the VLP generator is not maximised since the possible further production of VLP steam would not be re-usable either in the Process Units or in the Power Island. The syngas temperature at VLP generator outlet is 164 °C and final cooling is made either by using ST condensate or cooling water. On the other hand, this temperature is suitable for application in an ammonia absorption refrigeration cycle.

An assessment of the maximum heat available for the Absorption Chiller (without affecting the ST Condensate Pre-heating) has been carried out in this study. It was concluded that maximum 83 MW_{th} can be absorbed by the chiller, keeping the preheated ST condensate temperature at 85 °C as per the Base Case. This option implies a tighter design of the ST condensate pre-heater, which is considered in the techno-economic assessment in Section 6.

The operating parameters of the Absorption refrigeration system considered for this specific case are based on preliminary figures available from Suppliers and summarised in the following:

- CO_2 liquefaction temperature = -25°C;
- C.O.P. (defined as ratio between the chilling duty and absorbed heat) = 0.56;
- Heating medium is hot water produced at 125 °C in the syngas cooling and returned to the hot water generator at 115 °C.

As far as CO_2 compression unit is concerned, the CO_2 is liquefied at -25°C, which allows pumping the CO_2 from a pressure of 40.7 bara. The CO_2 streams are compressed in a three stage compressor to the liquefaction pressure. At compressor discharge, the CO_2 is dehydrated before being liquefied.

In the pre-combustion process, the captured CO_2 stream contains some incondensable gases (mainly Hydrogen and Carbon Monoxide), which cannot be liquefied at the conditions achievable with the industrial absorption chillers. Some CO_2 (approx 50%

concentration in the gas stream) is also entrained in the gaseous stream, thus affecting the overall CO_2 capture rate of the plant.

Downstream liquefaction, incondensable gases and gaseous CO_2 are separated in a drum and mixed with the syngas fed to the Gas Turbine. It is assumed that the small amount of Hydrogen rich gas produced does not affect either design or operation of the GT.

The "cold" available from separated liquid CO_2 product is used in a cross exchanger to cool down the gaseous CO_2 at the front end of the Liquefaction, in order to reduce the chiller duty.

In conclusion, the selected pressure/temperature conditions for CO_2 liquefaction allow:

- The application of an ammonia refrigeration cycle to re-use the maximum amount of waste heat recoverable form the syngas cooling;
- A significant reduction of the overall compression energy.
- Releasing the incondensable gases at a pressure suitable for combustion in the GT without recompression.
- Limiting the adverse impact on carbon capture rate to a reasonable value (<0.5%).

Figure 5-3 Case D2B: CO_2 Compression Unit scheme.

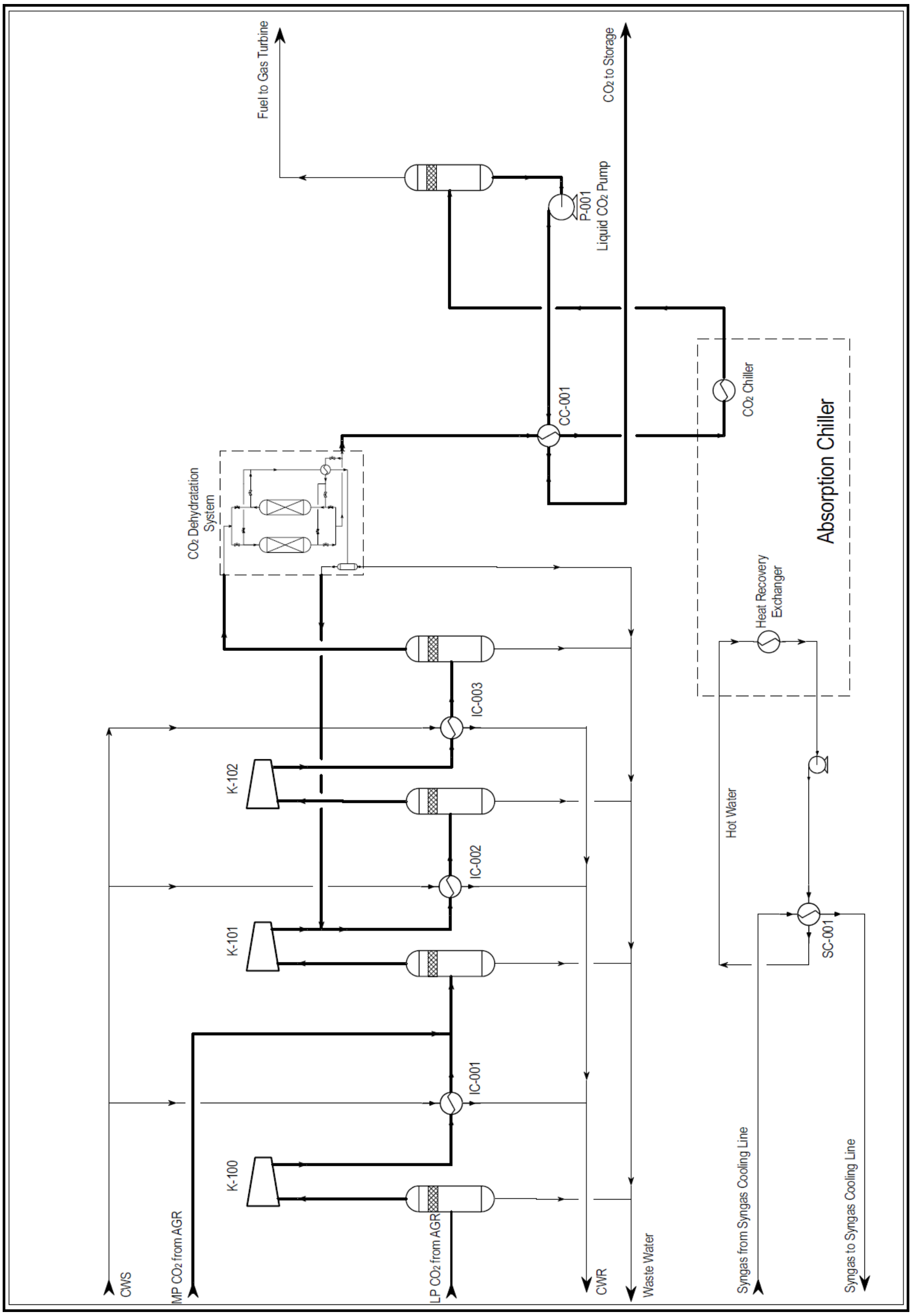


Table 5-6: Case D2B: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Pre-Combustion - Case D2b										
STREAM	1	2	3	4	5	6	7	8	9	10
Temperature (°C)	-5	-5	121	19	1	7	7	102	94	19
Pressure (bar)	1.2	1.2	5.0	4.8	4.8	4.8	4.8	14.0	14.0	13.8
TOTAL FLOW										
Mass flow (kg/h)	208775	208775	208775	208775	411177	619951	619951	619951	689262	689262
Molar flow (kgmole/h)	4751	4751	4751	4751	9569	14320	14320	14320	15923	15930
LIQUID PHASE										
Mass flow (kg/h)										17
GASEOUS PHASE										
Mass flow (kg/h)	208775	208775	208775	208775	411177	619951	619951	619951	689262	689245
Molar flow (kgmole/h)	4751	4751	4751	4751	9569	14320	14320	14320	15923	15929
Molecular Weight	43.95	43.95	43.95	43.95	42.97	43.29	43.29	43.29	43.29	43.27
Composition (vol %)										
CO_2	99.77	99.77	99.77	99.77	97.30	98.12	98.12	98.12	98.09	98.02
CO	0.01	0.01	0.01	0.01	0.20	0.14	0.14	0.14	0.14	0.14
H_2S+COS	0.02	0.02	0.02	0.02	0.01	0.01	0.01	0.01	0.01	0.01
H_2	0.04	0.04	0.04	0.04	2.34	1.57	1.57	1.57	1.57	1.57
N_2	0.00	0.00	0.00	0.00	0.03	0.02	0.02	0.02	0.02	0.02
Ar	0.00	0.00	0.00	0.00	0.06	0.04	0.04	0.04	0.04	0.04
H_2O	0.17	0.17	0.17	0.17	0.07	0.11	0.11	0.11	0.13	0.20

Table 5-7: Case D2B: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Pre-Combustion - Case D2b										
STREAM	11	12	13	14	15	16	17	18	19	20
Temperature (℃)	19	124	19	19	30	24	5	-25	-20	-25
Pressure (bar)	13.8	42.0	41.8	33.8	14.0	40.9	40.7	40.2	111.5	40.2
TOTAL FLOW										
Mass flow (kg/h)	689244	689244	689244	689010	69096	619671	619671	619671	616565	3105
Molar flow (kgmole/h)	15929	15929	15929	15916	1598	14304	14304	14304	14170	135
LIQUID PHASE										
Mass flow (kg/h)			234					616565	616565	
GASEOUS PHASE										
Mass flow (kg/h)	689244	689244	689010	689010	69096	619671	619671	3105	0	3105
Molar flow (kgmole/h)	15929	15929	15916	15916	1598	14304	14304	135	0	135
Molecular Weight	43.27	43.27	43.27	43.29	43.23	43.32	43.32	43.32	43.51	23.07
Composition (vol %)										
CO_2	98.02	98.02	98.02	98.10	97.86	98.22	98.22	98.22	98.69	48.79
CO	0.14	0.14	0.14	0.14	0.14	0.14	0.14	0.14	0.12	1.55
H_2S+COS	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.00
H_2	1.57	1.57	1.57	1.57	1.57	1.58	1.58	1.58	1.12	49.16
N_2	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.22
Ar	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.26
H_2O	0.20	0.20	0.20	0.12	0.36	0.00	0.00	0.00	0.00	0.00

Table 5-8: Case D2B: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE

Pre-Combustion - Case D2b

STREAM	21	22	23							
Temperature (℃)	0	12	19							
Pressure (bar)	111.0	6.0	5.8							
TOTAL FLOW										
Mass flow (kg/h)	619775	5116000	5116000							
Molar flow (kgmole/h)	14307	283970	283970							
LIQUID PHASE										
Mass flow (kg/h)		5116000	5116000							
GASEOUS PHASE										
Mass flow (kg/h)	619775	0	0							
Molar flow (kgmole/h)	14307	0	0							
Molecular Weight	43.32	18.02	18.02							
Composition (vol %)										
CO_2	98.69	0.00	0.00							
CO	0.12	0.00	0.00							
H_2S+COS	0.01	0.00	0.00							
H_2	1.12	0.00	0.00							
N_2	0.02	0.00	0.00							
Ar	0.04	0.00	0.00							
H_2O	0.00	100.00	100.00							

The main battery limit streams conditions, the thermal integration with the power plant, the electrical and cooling water consumption of the CO_2 Compression unit for Case D2B are summarized in Table 5-4.

Table 5-9: Case D2B: CO_2 Compression Unit consumption.

CO_2 compression in general Case D2B: CO_2 compression consumption			
CO_2 Inlet Streams	LP	MP	
Flowrate	106,480	214,482	Nm^3/h
Temperature	-5	1	°C
Pressure	1.2	4.8	bar a
CO_2 Outlet Stream			
Flowrate		616.6	t/h
Temperature		0	°C
Pressure		111	bar a
CO_2 purity		98.7	% mol/mol
Overall Plant Carbon Capture			
Carbon Capture		84.6	%
Cooling Water			
CW consumption (1)		21188	t/h
Compressor/Turbine Electrical Consumption			
1^{st} stage		6.6	MW_e
2^{nd} stage		14.6	MW_e
3^{rd} stage		16.8	MW_e
CO_2 pump		1.6	MW_e
TOTAL		**39.6**	**MW_e**

Note 1: Quoted figure includes absorption chiller consumption.

Table 5-9 shows the performance delta between this compression strategy (D2B) and the Base Case (D0). Overall, a net compression energy reduction equal to 7.6 MWe has been estimated. The significant performance improvement is mainly due to the introduction of a refrigeration system, whose energy input is taken recovering unused low temperature waste heat from the syngas cooling unit. It is noted that the process could be slightly further optimised as the CO_2 outlet temperature is relatively low, i.e. some further refrigeration energy recovery may be possible.

Also, this strategy allows a slight reduction of the coal thermal input to the gasification plant, since the 0.3 % of the Gas Turbines thermal demand is fulfilled by the hydrogen rich gas separated downstream in CO_2 liquefaction process.

Table 5-10 Case D2B: Performance delta with respect to the base case.

CO_2 compression in general Case D2b: Performance delta with respect to the base case A0					
Cooling water					
CW consumption (1)	+ 2066	t/h	⟷	+ 0.2	MW_e
Compressor Electrical Consumption					
Overall electrical consumption difference				- 7.8	MW_e
Overall Plant Electrical Consumption Gap					
TOTAL				**- 7.6**	**MW_e**

Note 1: Quoted figure include differential CW consumption in the Syngas Cooling Unit due to the modification introduced with the present strategy.
Note 2: Negative value indicates a lower consumption with respect to the base case. Conversion factor, see section B.

5.2.3 Case D2c: CO_2 liquefaction with CW in the post-combustion capture

The process flow scheme of this compression strategy is shown in Figure 5-4 and the relevant Heat & Mass Balance data are indicated in Table 5-11 and Table 5-12.

In the proposed configuration, the CO_2 liquefaction is carried out using the cooling water available in the plant. Therefore, CO_2 is liquefied at 20°C, which allows pumping the CO_2 from a pressure of 65.6 bara. With respect to the reference Base Case, the fourth compressor stage is then replaced by a pump.

Figure 5-4 Case D2C: CO_2 Compression Unit scheme.

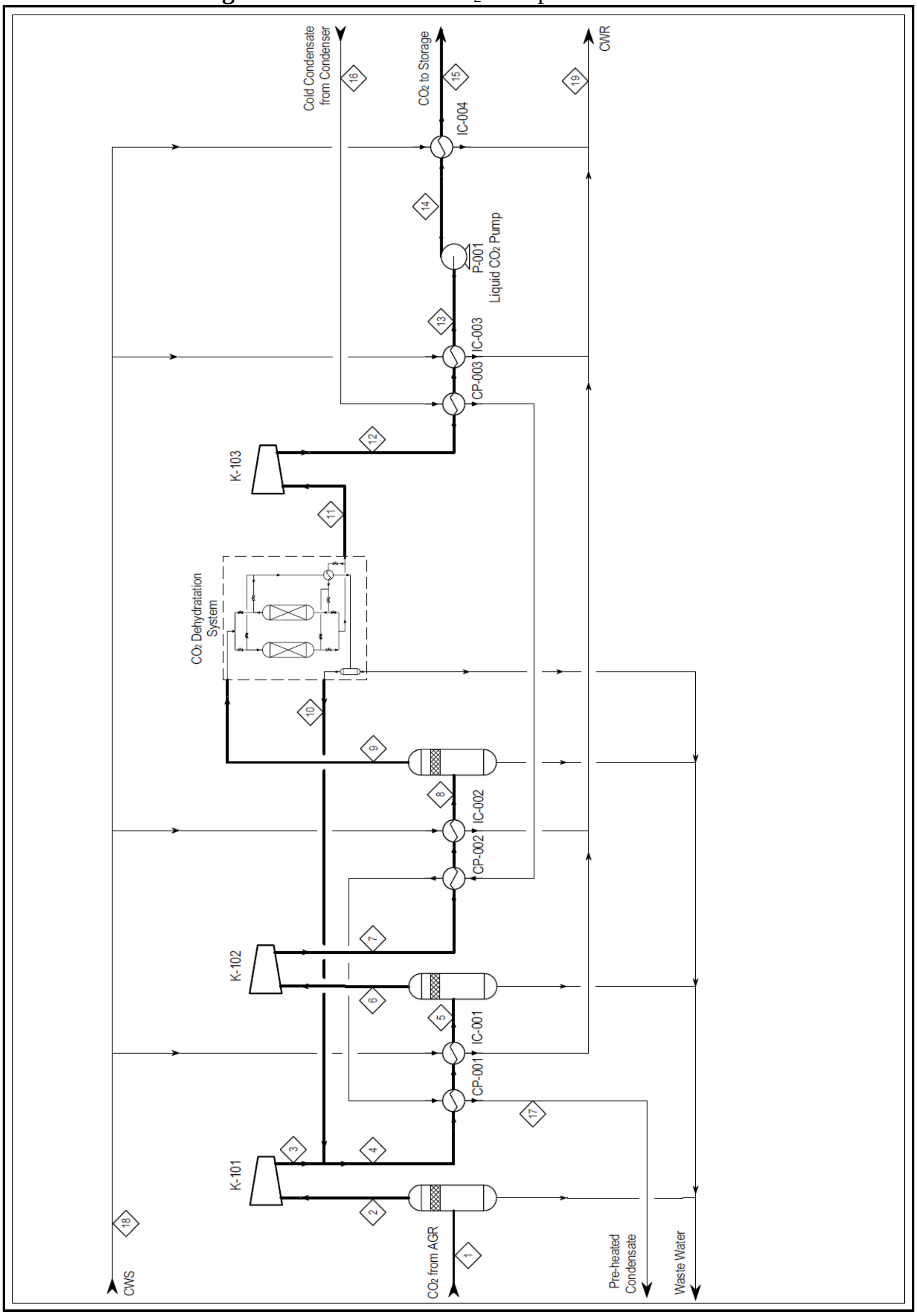


Table 5-11 Case D2C: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Post-Combustion - Case D2c										
STREAM	1	2	3	4	5	6	7	8	9	10
Temperature (°C)	38	38	184	168	19	19	176	19	19	15
Pressure (bar)	1.6	1.6	7.0	7.0	6.6	6.6	34.0	33.6	33.6	7.0
TOTAL FLOW										
Mass flow (kg/h)	556451	556451	556451	617374	617374	608630	608630	608630	607976	60923
Molar flow (kgmole/h)	12959	12959	12959	14346	14346	13860	13860	13860	13824	1387
LIQUID PHASE										
Mass flow (kg/h)					8744			654		
GASEOUS PHASE										
Mass flow (kg/h)	556451	556451	556451	617374	608630	608630	608630	607976	607976	60923
Molar flow (kgmole/h)	12959	12959	12959	14346	13860	13860	13860	13824	13824	1387
Molecular Weight	42.94	42.94	42.94	43.04	43.04	43.91	43.91	43.91	43.98	43.94
Composition (vol %)										
CO_2	95.88	95.88	95.88	96.25	96.25	99.62	99.62	99.62	99.88	99.71
N_2	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01
O_2	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Ar	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
H_2O	4.11	4.11	4.11	3.74	3.74	0.37	0.37	0.37	0.11	0.28

Table 5-12 Case D2C: CO_2 Compression Unit Heat and Material balance.

HEAT AND MATERIAL BALANCE										
Post-Combustion - Case D2c										
STREAM	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	
Temperature (°C)	24	91	20	28	20	70	98	12	19	
Pressure (bar)	32.7	66.0	65.6	111.2	111.0	10.0	9.2	6.0	5.8	
TOTAL FLOW										
Mass flow (kg/h)	546855	546855	546855	546855	546855	952700	952700	8973000	8973000	
Molar flow (kgmole/h)	12426	12426	12426	12426	12426	52881	52881	498057	498057	
LIQUID PHASE										
Mass flow (kg/h)			546855	546855	546855	952700	952700	8973000	8973000	
GASEOUS PHASE										
Mass flow (kg/h)	546855	546855	0	0	0	0	0	0	0	
Molar flow (kgmole/h)	12426	12426	0	0	0	0	0	0	0	
Molecular Weight	44.01	44.01	44.01	44.01	44.01	18.02	18.02	18.02	18.02	
Composition (vol %)										
CO_2	99.99	99.99	99.99	99.99	99.99	0.00	0.00	0.00	0.00	
N_2	0.01	0.01	0.01	0.01	0.01	0.00	0.00	0.00	0.00	
O_2	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
Ar	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
H_2O	0.00	0.00	0.00	0.00	0.00	100.00	100.00	100.00	100.00	

The main battery limit streams conditions, the thermal integration with the power plant, the electrical and cooling water consumption of the CO_2 Compression unit for Case D2C are summarized in Table 5-13.

Table 5-13: Case D2C: CO_2 Compression Unit consumption.

CO_2 compression in general Case D2C: CO_2 compression consumption		
CO_2 Inlet Stream		
Flowrate	290461	Nm^3/h
Temperature	38	°C
Pressure	1.6	bar a
CO_2 Outlet Stream		
Flowrate (liquid)	546.8	t/h
Temperature	20	°C
Pressure	111	bar a
CO_2 purity	99.99	% mol/mol
Overall Plant Carbon Capture		
Carbon Capture	86.7	%
Cooling Water		
CW cons. CO_2 compression	8932	t/h
Thermal Integration with the Power Plant		
Condensate pre-heating	31.4	MW_{th}
Compressor/Pump Electrical Consumption		
1st stage	21.7	MW_e
2nd stage	23.6	MW_e
3rd stage	7.3	MW_e
CO_2 pump	1.2	MW_e
TOTAL	**53.8**	**MW_e**

With respect to the Base Case, the reduction of the compression energy is partially off-set by the following factors:

- the electrical consumption of the chiller;
- the slight reduction of the CO_2 compression waste heat, available for condensate preheating with a consequent increase of the steam consumption in the power island feed water heater;
- the higher electrical consumption associated to the increase of Cooling water usage, mainly due to the liquefaction duty.

Overall, as shown in Table 5-14, the CO_2 liquefaction at conditions achievable with the cooling water leads to a net equivalent consumption reduction of 3.3 MWe.

Table 5-14: Case D2C: Performance delta with respect to base case.

CO_2 compression in general Case D2C: Performance delta respect to the base case B0					
Thermal Integration with the Power Plant					
Steam cons. for Condensate Pre-heating	+ 2.5	MW_{th}	⟷	+ 0.5	MW_e
Cooling water					
CW consumption	+ 3599	t/h	⟷	+ 0.4	MW_e
Compression/Pumping Electrical Consumption					
Overall electrical consumption difference				- 3.7	MW_e
Overall Plant Electrical Power Gap					
TOTAL				**- 3.3**	**MW_e**

Note: Negative value indicates a lower consumption with respect to the base case. Conversion factor, see section B.

5.3 Case D3 – Deeper inter-cooling

5.3.1 Case D3A: Deeper inter-cooling in the post-combustion capture

The reference process flow scheme for this compression strategy is shown in Figure 5-6.

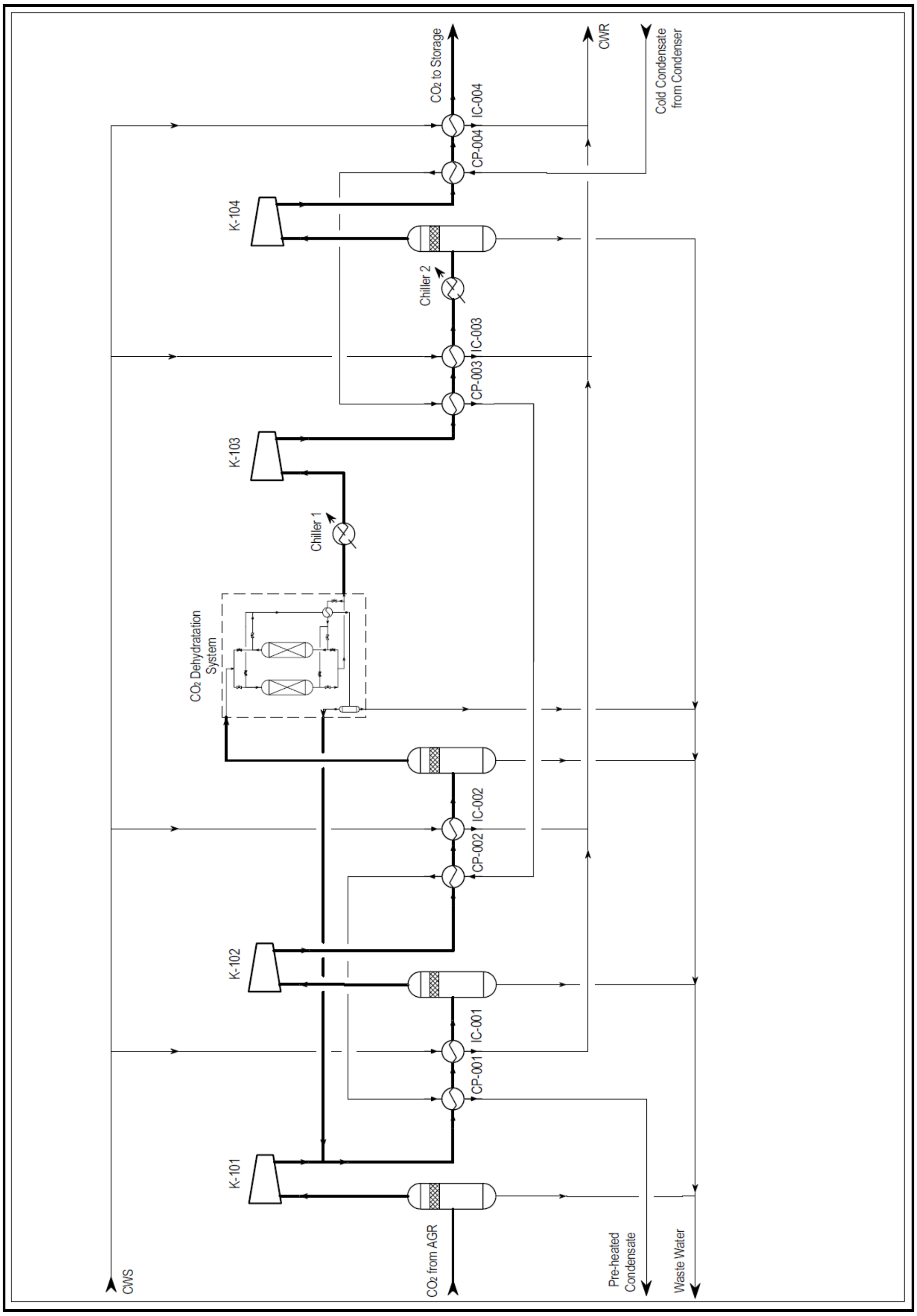
The deeper inter-cooling strategy has been evaluated taking into account the following possible limitations:

- Avoiding hydrate formation in the CO_2 stream;

- Keeping the compression trajectory reasonably far from the CO_2 vapour-liquid phase boundary.

The hydrate formation temperature for the CO_2 gas at the first inter-stage operating pressure is predicted to be not far below 0°C. It is noticed that some uncertainty influences this figure because the hydrate formation kinetics of CCS streams are not so well known as for natural gas. Taking a safety margin on the quoted figure, there is not much space for deeper inter-cooling upstream of the CO_2 dehydration unit, where the water content is still enough to cause issues with hydrate formation. For this reason, the strategy has been implemented moving the dehydration at lower pressure than the base case, to allow downstream deeper inter-cooling. Lower operating pressure drives additional investment cost for this unit, due to the higher volumetric flow rates.

Figure 5-5 Case D3A: CO_2 Compression Unit scheme.



Downstream the dehydration, the limitation for a deeper inter-cooling is represented by the necessity of keeping the temperature above the CO_2 dew point. Therefore, taking a margin of 15 °C above the dew point, the CO_2 is cooled to -35°C downstream the dryer and to 10 °C downstream the fourth stage.

The main battery limit streams conditions, the thermal integration with the power plant, the electrical and cooling water consumption of the CO_2 Compression unit for Case D3A are summarized in Table 5-15.

Table 5-15: Case D3A: CO_2 Compression Unit consumption.

CO_2 compression in general Case D3A: CO_2 compression consumption		
CO_2 Inlet Stream		
Flowrate	290461	Nm^3/h
Temperature	38	°C
Pressure	1.6	bar a
CO_2 Outlet Stream		
Flowrate (liquid)	278518	Nm^3/h
Temperature	72	°C
Pressure	111	bar a
CO_2 purity	99.99	% mol/mol
Overall Plant Carbon Capture		
Carbon Capture	86.7	%
Cooling Water		
CW cons. CO_2 compression (1)	6015	t/h
Thermal Integration with the Power Plant		
Condensate pre-heating	25.0	MW_{th}
Compressor/Pump Electrical Consumption		

CO_2 compression in general Case D3A: CO_2 compression consumption	
1st stage	25.4 MW_e
2nd stage	16.8 MW_e
3rd stage	8.7 MW_e
4th stage	3.7 MW_e
TOTAL	**54.6 MW_e**
Chiller	
Chiller electrical consumption (2)	3.2 MW_e

Note 1: Including Chiller consumtpion.
Note 2: Assumed C.O.P. = 3.0.

Table 5-16 shows that, with respect to the Base Case, the reduction of the compression energy is off-set by the following factors:

- the electrical consumption of the chiller;
- the reduction of the CO_2 compression waste heat available for condensate preheating with a consequent increase of the steam consumption in the power island feed water heater;
- higher electrical consumption associated to the increase of Cooling water usage, mainly due to the chiller.

From the figures in the table, it can be drawn that the use of external refrigeration for deeper intercooling in the post-combustion CCS is not attractive.

Table 5-16: Case D3A: Performance delta with respect to base case.

CO_2 compression in general Case D3A: Performance delta respect to the base case B0					
Thermal Integration with the Power Plant					
Steam cons. for Condensate Pre-heating	+ 8.6	MW_{th}	⟷	+ 2.3	MW_e
Cooling water					
CW consumption	+ 623	t/h	⟷	+ 0.1	MW_e
Compression Electrical Consumption					
Overall electrical consumption difference				- 2.9	MW_e
Chiller					
Chiller electrical consumption				+3.2	MW_e
Overall Plant Electrical Power Gap					
TOTAL				**- 2.6**	**MW_e**

Note: Negative value indicates a lower consumption with respect to the base case. Conversion factor, see section B.

5.3.2 Case D3B: Deeper inter-cooling in the pre-combustion capture

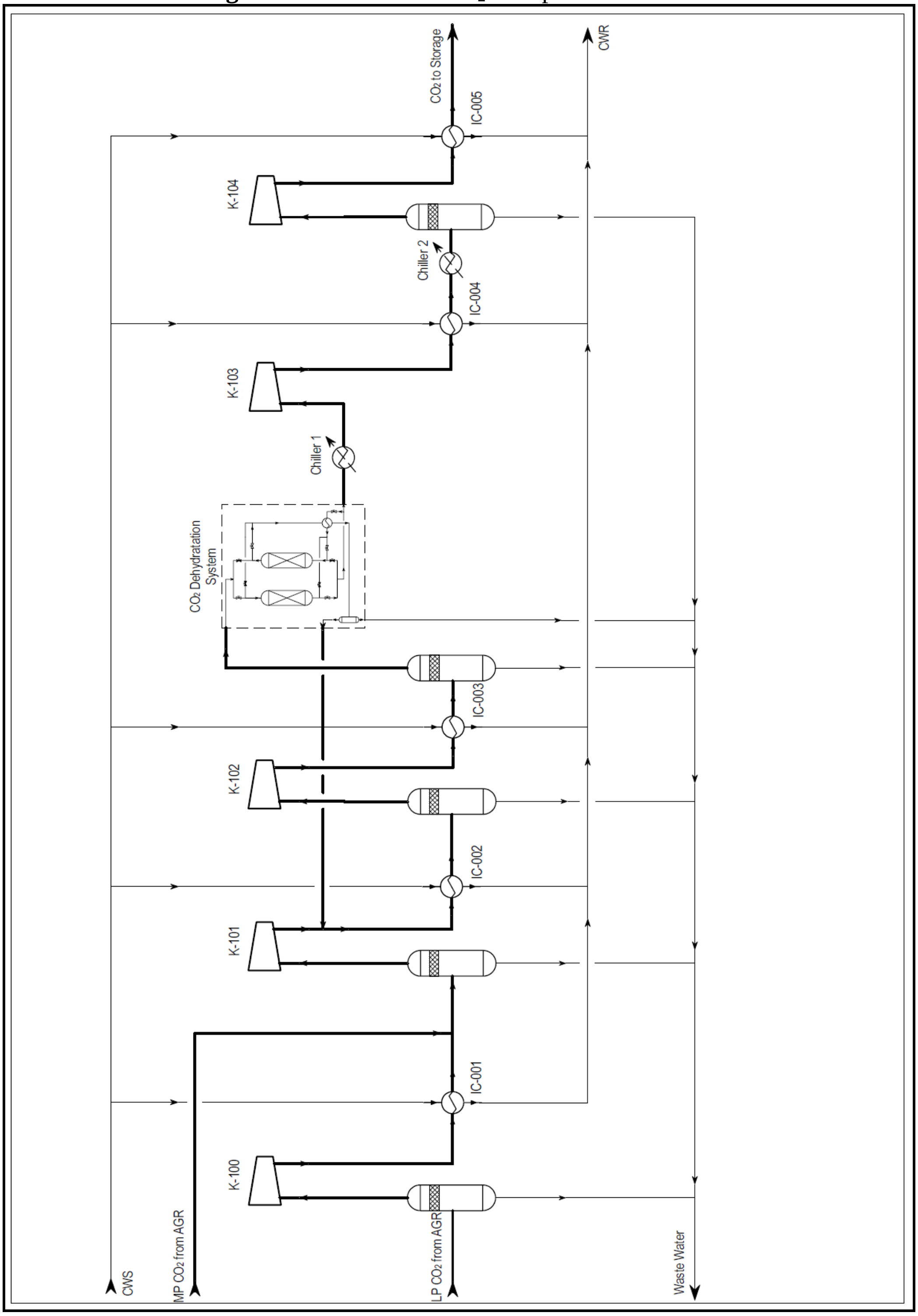
The reference process flow scheme for this compression strategy is shown in Figure 5-6.

As assessed for the early liquefaction option (ref. 5.2.1), a significant amount of waste heat is available from the syngas for re-use in an absorption chiller. The absorption chiller can be applied to achieve a deeper inte-rcooling in the CO_2 compression Unit.

Like in the post-combustion case, the deeper inter-cooling option has been evaluated with respect to the possible hydrate formation in the CO_2 stream and the necessity to keep the temperature above the CO_2 dew point.

Not much variation in the hydrate formation temperature for the CO_2 gas is expected in the pressure range between 2 bara and 30 bara, with a formation temperature of approximately 10°C at 10 bara. With respect to the post-combustion case, the formation temperature is increased by the possible presence of H_2S, which is generally subject to early formation, as recorded for natural gas compression applications. Taking a safety margin on the quoted figure, there is not much space for deeper inter-cooling upstream the CO_2 dehydration unit, where the water content is still enough to cause issues with hydrate formation. For this reason the strategy has been implemented moving the dehydration at lower pressure than the base case, to allow deeper inter-cooling downstream. Lower operating pressure drives additional investment cost for this unit, due to the higher volumetric flow rates.

Figure 5-6 Case D3B: CO_2 Compression Unit scheme.



Downstream the dehydration, the limitation to a deeper inter-cooling is represented by the necessity of keeping the temperature above the CO_2 dew point. Therefore, taking a margin of 15 °C above the dew point, the CO_2 is cooled to -20°C downstream 3rd compressor stage and to 2 °C downstream the fourth stage.

The main battery limit streams conditions, the electrical and cooling water consumption of the CO_2 Compression unit for Case D3b are summarized in Table 5-17.

Table 5-17: Case D3B: CO_2 Compression Unit consumption.

CO_2 compression in general Case D3b: CO_2 compression consumption			
CO_2 Inlet Streams	LP	MP	
Flowrate	106,480	214,482	Nm^3/h
Temperature	-5	1	°C
Pressure	1.2	4.8	bar a
CO_2 Outlet Stream			
Flowrate		616.6	t/h
Temperature		50	°C
Pressure		111	bar a
CO_2 purity		98.7	% mol/mol
Overall Plant Carbon Capture			
Carbon Capture		84.6	%
Cooling Water			
CW consumption (1)		9829	t/h
Compressor/Turbine Electrical Consumption			
1st stage		6.6	MW_e
2nd stage		14.5	MW_e
3rd stage		8.8	MW_e
4th stage		8.5	MW_e

CO_2 compression in general Case D3b: CO_2 compression consumption		
5^{th} stage	6.9	MW_e
TOTAL	**45.3**	**MW_e**

Note 1: Quoted figure includes absorption chiller consumption.

The net equivalent consumption deltas with respect to Base Case are reported in Table 5-18, which shows a net compression energy reduction equal to 2.0 MWe.
No consumption delta has been associated to the absorption chiller as the relevant heat input (approx 18 MW_{th}) is recovered from the low temperature heat, available in syngas cooling unit and not used in the Base Case.

Table 5-18 Case D3B: Performance delta with respect to the base case.

CO_2 compression in general Case D3B: Performance delta with respect to the base case A0					
Cooling water					
CW consumption	+ 899	t/h	⟷	+ 0.1	MW_e
Compressor Electrical Consumption					
Overall electrical consumption difference				- 2.1	MW_e
Overall Plant Electrical Consumption Gap					
TOTAL				**- 2.0**	**MW_e**

Note 1: Negative value indicates a lower with consumption respect to the base case. Conversion factor, see section B.

6 Sensitivities to ambient conditions

Purpose of this section is to indicate the sensitivity of the effectiveness of the compression strategies to higher ambient conditions.

Table 6-1 shows the qualitative effects of increasing the ambient condition and ultimately the Cooling Water (CW) temperature, for each of the compression strategies investigated in the study.

Generally CW is used for CO_2 inter-cooling between one compression stage and the following one. An increase in CW temperature leads to a lower CO_2 density at compressor suction, because of the higher temperature. The consequent increased electrical consumption of CO_2 compressors represents a cross effect for all the cases at the same extent, with some exceptions (e.g. the vapour recompression cases, as reported in Table 6-1). Therefore this is not generally included in the present sensitivity analysis.

Table 6-1 Effect of increase in temperature of cooling water.

CASE TAG	Description	Influence of higher ambient conditions	Effect / Consequence
Case A1	Vapour recompression in the AGR stripping column	Low	No remarkable effects on this case, whereas the base case consumption would tend to increase. Consequence: reduction of the additional consumption with respect to the base case.
Case A2	Increase of number of flash stages in the AGR	None	No remarkable effects, since ambient and CW conditions do not represent a key factor for the effectiveness of the strategy.
Case B1	Vapour recompression	Low	No remarkable effects on this case, whereas the base case consumption would tend to increase. Consequence: reduction of the additional consumption with respect to the base case.

CASE TAG	Description	Influence of higher ambient conditions	Effect / Consequence
Case B2	Increase of stripper pressure in CO_2 capture unit	None	No remarkable effects, since ambient and CW conditions do not represent a key factor for the effectiveness of the strategy.
Case B3	Staging of solvent regeneration in CO_2 capture unit	None	Lower adverse effect expected in this case than in the base case, being the AGR compressors uncooled, i.e. part of the overall compression duty is not affected by higher ambient conditions. Consequence: slightly greater effectiveness of the strategy.
Case C1	Expansion of incondensable	None	No remarkable effects, since ambient and CW conditions do not represent a key factor for the effectiveness of the strategy.
Case C2	Refrigeration of compressed CO_2	Low	Worse performance of the external chiller, leading to higher electrical consumption for this case. Consequence: further increase of the additional consumption with respect to the base case.
Case C3	CO_2 Liquefaction with CW	High	Impossibility to achieve full sub-critical liquefaction or, if CW temperature is higher than CO_2 critical temperature, impossibility to pump CO_2 as a sub-cooled liquid. Consequence: effectiveness of the strategy significantly affected with potential for impracticability.
Case D1	Increasing number of stages	None	No remarkable effects

CASE TAG	Description	Influence of higher ambient conditions	Effect / Consequence
Case D2a	Early CO_2 liquefaction (post-combustion)	Low	Worse performance of the external chiller, leading to higher electrical consumption for this case. Consequence: lower effectiveness of the strategy.
Case D2b	Early CO_2 liquefaction (pre-combustion)	Low	Worse performance of the absorption chiller, leading to higher liquefaction pressure at a given amount of waste heat available for the asbsorption refrigeration. Consequence: lower effectiveness of the strategy.
Case D2c	CO_2 liquefaction with CW (post-combustion)	High	Impossibility to achieve full sub-critical liquefaction or, if CW temperature is higher than CO_2 critical temperature, impossibility to pump CO_2 as a sub-cooled liquid. Consequence: effectiveness of the strategy significantly affected with potential for impracticability.
Case D3a	Deeper inter-cooling (post-combustion)	Low	Worse performance of the external chiller, leading to higher electrical consumption for this case. Consequence: lower effectiveness of the strategy.
Case D3b	Deeper inter-cooling (pre-combustion)	Low	Worse performance of the absorption chiller, having no significant impact on the performance as the waste heat avialable is much higher than heat input of the absorption refrigerator. No remarkable consequences.

7 Economic analysis

An economic evaluation has been carried out for all the discussed compression strategies, which present a reduction of the electrical consumption associated to CO_2 compression system.

The evaluation is made to assess the economic convenience of each strategy in terms of differential figures with respect to the base cases.

Generally, the reduction of the parasitic consumption due to the CO_2 compression system leads to an increase of the electricity export revenue. For each case, the economic convenience of the strategy is evaluated through the calculation of the NPV and IRR, for a given Cost of the Electricity (C.O.E.).

The economic calculations are performed using the standard IEA GHG spreadsheet, already used in all the reference studies of this report.

The major contribution to differential CAPEX, revenue and OPEX are summarised in the following sections.

7.1 Differential Investment Cost with respect to base cases

The differential investment cost evaluation of each process unit involved in the compression strategies takes into consideration the impacts on the following items:

- Direct Materials, including equipment and bulk materials;
- Construction, including mechanical erection, instrument and electrical installation, civil works and, where applicable, buildings and site preparation;
- Other Costs, including temporary facilities, solvents, catalysts, chemicals, training, commissioning and start-up costs, spare parts etc.;
- EPC Services including Contractor's home office services and construction supervision.

CAPEX figures are based on IV Q 2010 cost level.

For each of the three main CCS technologies considered in the present study the differential CAPEX results are reported and briefly commented in the next section.

7.1.1 Pre-combustion

CASE A2

The compression strategy of case A2 consists in the increase of the number of flash stages in the AGR.

In the AGR the incremental contribution is given by the inclusion of HP Flash Vessel, through the duty of the MP flash Vessel decreases, thus corresponding to reduction of the size.

In the CO_2 Compression Unit, there is a reduction in CAPEX as the 2nd compressor stage and following intercooler capacities are lower than the base case.

The figures for each unit and the resulting overall differential CAPEX are included in Table 7-1, attached to the end of the section.

CASE D2B

The connession strategy of case D2B consists in the early liquefaction of CO_2 using an absorption refrigeration package.

An incremental contribution to CAPEX is given by:

- The addition of an Absorption chiller (included in the CO_2 Compression Unit);
- The addition of the Hot Water Generator in the Syngas Cooling Unit and the Hot Water circulation Pump.
- The tighter design of the ST condensate preheater in the Syngas Cooling Unit;
- Additional Equipment in the CO_2 Compression Unit (incondensables separator, CO_2 pump).

The above additional costs are partially offset by a saving in the installed equipment in the compression unit, due to the centrifugal compressor (only the first three stages out of five are required) and the intercoolers.

The figures for each unit and the resulting overall differential CAPEX are included in Table 7-1, attached to the end of the section.

CASE D3B

The same considerations as Case D2B can be reported for the deeper inter-cooling strategy, since the two cases are very similar in terms of process modifications, as far as impacts on the syngas cooling are concerned. The effects are generally smoothed by the smaller size of the chiller.

A saving for CO_2 compressor (due to reduced volumetric flow rate for some stages) and intercoolers is estimated. On the other hand, the operating pressure of the Dehydartion Unit is decreased (ref.Table 7-1), thus requiring the installation of larger equipment and leading to a significant CAPEX increase.

The figures for each unit and the resulting overall differential CAPEX are included in Table 7-1, attached to the end of the section.

Table 7-1 Differential CAPEX for compression strategies related to pre-combustion capture.

FOSTER WHEELER ITALIANA

DIFFERENTIAL CAPEX SUMMARY

PRE-COMBUSTION CASES

Refer : Rotating machinery for CO2 compression in CCS systems
Client : IEA GHG
Location : NETHERLANDS
Date : Jan 11

FIGURE IN MM EURO

DESCRIPTION	A2 MM €	D2b MM €	D3b			
CO_2 Compression and Drying Unit (1)	-3.4	-2.3	2.5			
AGR	-0.2	_	_			
Syngas Cooling Unit	_	7.5	2.4			
TOTAL DIFFERENTIAL INVESTMENT COST	-3.6	5.2	4.9			

Notes: 1. Including absorption chiller for cases D2b and D3b.

7.1.2 Post-combustion

CASE B2

The compression strategy of case B2 consists in the increase of the stripper operating pressure to release the captured CO_2 to the compression unit at higher pressure.

Regarding CAPEX, the following main impacts are estimated with respect to the base case:

- general reduction of the cost for the CO_2 capture unit, primarily due to the reduced capacity of the reboilers and reduced diameter of the stripper column, the operating pressure increase is assumed to have no impacts on the thickness of the column as the design pressure is set to 3.5 barg for all cases;
- reduction of CO_2 compressor cost, due to lower compression ratio of the first stages.
- negligible increase of the Power Island investment cost, due to the additional duty for the ST condensate preheater.

The resulting overall impact is a reduction of the total investment cost, as shown in Table 7-2, attached to the end of this Section.
Case B2A and B2B show the same trend with respect to CAPEX, however the effects related to case B2B are amplified due to the higher increase of the stripper operating pressure.

CASE B3

The compression strategy of Case 3B consists in the use of a multi pressure stripper in the CO_2 Capture Unit.

In the capture unit, additional investment costs are associated to the installation of two compressors to route the vapour form one pressure level to the other in the multi pressure column. Also, the complication added to the stripper column causes a further CAPEX increase. These effects are partially off-set by the reduction of the reboilers cost, due to their lower thermal duty.

In the CO_2 compressor a reduction of cost is estimated due to the lower compression ratio of the first stages.

A negligible cost increase for the Power Island is associated to the additional thermal duty for the ST condensate preheater.

The figures for each Unit and the resulting overall differential CAPEX are included in Table 7-2, attached to the end of this section.

CASE D1

The compression strategy of Case D1 consists in the increased number of stages in the compressor (from 4 to 8).

In terms of CAPEX, this strategy leads to an overall reduction of the investment required for the machinery itself, as confirmed by Vendor's feedback (ref. Section D). On the other hand, a higher cost is estimated for the intercoolers.

In the Power Island the additional thermal duty for the ST condensate pre-heater leads to a negligible increase of the CAPEX.

The resulting overall impact is a reduction of the total investment cost, as shown in Table 7-2, attached to the end of this section.

CASE D2A

The compression strategy of Case D2A consists in the early liquefaction of CO_2, using an external conventional refrigeration package.

An incremental contribution to CAPEX is given by:
- The addition of the chiller (included in CO_2 Compression Unit);
- Additional Equipment in the CO_2 Compression Unit (i.e. CO_2 pump).

The above additional costs are offset by a significant saving associated to the compressor (only the first two stages off a total number of four are required) and the intercoolers.

The figures for each Unit and the resulting overall differential CAPEX are included in Table 7-2, attached to the end of this section-

CASE D2C

The compression strategy of case D2C consists in the liquefaction of CO_2 using the available Cooling Water.

An incremental contribution to CAPEX is given by:
- additional Equipment in the CO_2 Compression Unit (i.e. CO_2 pump, CO_2 liquefier).

- additional thermal duty for the ST condensate pre-heater in the Power Island (negligible)

The above additional costs are offset by the savings associated to the compressor (only three stages off a total number of four are required) and the intercoolers.

The figures for each Unit and the resulting overall differential CAPEX are included in Table 7-2, attached to the end of this section.

CASE D3A

The compression strategy of Case D2A consists in a deeper inter-cooling, which is achieved by the use of an external refrigeration package.

An incremental contribution to the CAPEX is given by the addition of the Chiller.

Savings for CO_2 compressor (due to reduced volumetric flow rate for some stages) and intercoolers are estimated. On the other hand, the operating pressure for Dehydration Unit is decreased, thus requiring the installation of larger equipment and leading to a significant CAPEX increase.

The effect of additional thermal duty for the ST condensate pre-heater in the Power Island is negligible.

The figures for each Unit and the resulting overall differential CAPEX are included in Table 7-2, attached to the end of this section.

Table 7-2 Differential CAPEX for compression strategies related to post-combustion capture.

FOSTER WHEELER ITALIANA

DIFFERENTIAL CAPEX SUMMARY

POST-COMBUSTION CASES

Refer : Rotating machinery for CO2 compression in CCS systems
Client : IEA GHG
Location : NETHERLANDS
Date : Jan 11

FIGURE IN MM EURO

DESCRIPTION	B2a	B2b	B3	D1	D2a	D2c	D3a
	MM €	MM €	MM €	MM €	MM €	MM €	MM €
CO_2 Compression and Drying Unit (1)	-4.2	-7.1	-11.1	-2.2	-8.2	-1.7	4.7
CO_2 Capture Unit (2)	-1.0	-3.7	18.0	_	_	_	_
Power Island	0.0	0.0	0.0	0.1	0.0	0.0	0.0
TOTAL DIFFERENTIAL INVESTMENT COST	-5.2	-10.8	6.8	-2.1	-8.2	-1.7	4.7

Notes:
1. Including refrigeration package for cases D2a and D3a.
2. Including multipressure stripper compressors for case B3.

7.1.3 Oxy-fuel combustion

CASE C3

The compression strategy of case C3 consists in the liquefaction of CO_2 using the available Cooling Water.

An incremental contribution to CAPEX is given by:

- additional Equipment in the CO_2 Compression Unit (i.e. CO_2 pump, CO_2 liquefier).
- additional thermal duty for the ST condensate pre-heater in the Power Island (almost negligible)

The above additional costs are offset by the savings associated to the compressor (due to the reduced pressure ratio for the fourth stage) and the intercoolers.

The figures for each Unit and the resulting overall differential CAPEX are included in Table 7-3, attached to the end of this section.

Table 7-3 Differential CAPEX for compression strategies related to oxy-fuel combustion capture.

FOSTER WHEELER ITALIANA

Refer : Rotating machinery for CO2 compression in CCS systems
Client : IEA GHG
Location: NETHERLANDS
Date : Jan 11

DIFFERENTIAL CAPEX SUMMARY

OXY-FUEL COMBUSTION CASES

FIGURE IN MM EURO

DESCRIPTION	C3
	MM €
CO_2 Compression and Drying Unit (1)	-1.7
Power Island	0.1
TOTAL DIFFERENTIAL INVESTMENT COST	-1.6

7.2 Differential Operating Costs with respect to base cases

As far as OPEX are concerned, the following contributions are generally considered in the differential analysis:

- Maintenance costs, estimated as a percentage of the differential CAPEX.
- Chemicals & Consumables, only in cases where a differential consumption is calculated with respect to the base case (i.e. case B2A and B2B).
- Fuel costs, only for strategies leading to different feedstock flow rate than the base case (i.e. case D2B).
- Carbon Tax, only for cases where a differential carbon capture rate is expected with respect to the base case (i.e. case D2B).

As far as Chemicals consumption are concerned, case B2A and B2B present increased operating costs due to the higher degradation rate expected for the solvent in the CO_2 Capture Unit (ref. 3.2).
MEA thermal degradation rate is estimated to be significantly higher when stripper bottom temperature increases, as reported in literature [6]. For the specific cases considered in the present analysis, the following factors were used:

- Approximately 220% of degradation rate for case B2A (stripper pressure = 2.1 bara, bottom temperature = 129 °C)
- Approximately 320% of degradation rate for case B2B (stripper pressure = 2.6 bara, bottom temperature = 134 °C).

Specific considerations regarding OPEX have to be made for case D2B. This strategy is associated to a mentioned reduction of the coal feed rate and a minor increase of CO_2 emissions (ref. 5.2.2). These factors have been taken into account for the economics evaluation of the corresponding strategies.

For each compression strategy evaluated, reference is made to the calculation spreadsheets attached in Attachment 9.1.

7.3 Differential Revenues with respect to base cases

The economic evaluation has been carried out for all the cases presenting a reduction of the electrical consumption associated to the CO_2 compression strategy.

The reduction of compression parasitic consumption corresponds to an increase of the electricity export and therefore of the yearly revenues

The basis to evaluate the additional revenue is a cost of $ct 5/kWh, which effectively cover export electricity revenue rather than generation cost.

For each strategy evaluated, reference is made to the calculation spreadsheets attached in Attachment 9.1.

7.4 Results

The output of the economic assessment is summarised in Table 7-4.

The Internal Rate of Return (based on a lost export electricity revenue of 0.038 €c/kWh) is calculated for the cases showing an increased investment cost.

Table 7-4 Economic analysis outcome summary

Case tag	Strategy description	NPV (*)	IRR (*)
A2	Increase ofnumber of flash stages in the AGR	> 0	N/A
B2a	Increase of stripper pressure in CO2 capture unit	< 0	N/A
B2b		< 0	N/A
B3	Staging of solvent regeneration in CO_2 capture unit	< 0	0.3 %
C3	CO_2 Liquefaction with CW	> 0	N/A
D1	Increasing number of stages	> 0	N/A
D2a	Early CO_2 liquefaction post comb.	> 0	N/A
D2b	Early CO_2 liquefaction pre comb.	> 0	30.4 %
D2c	CO_2 Liquefaction with CW post comb.	> 0	N/A
D3a	Deeper inter-cooling post comb.	> 0	10.6 %
D3b	Deeper inter-cooling pre comb.	< 0	5.9 %

(*) based on an export electricity revenue of 3.8 €c /kWh

All the strategies that present a Net Present Value greater than zero (highlighted in green) may be considered techno and economically attractive.

It is noted that most of the compression strategies show an investment cost lower than the Base Case. This is mainly due to a more compact compressor design, which results in a significant reduction of the overall CAPEX requirement.

For some specific cases it is worth to draw the following comments:

- Case A2: the good economics shown by the strategy of increasing the flash stages number in the AGR are essentially driven by an integrated approach, as far as AGR and Compression Unit designs are concerned. In fact, the additional CO2 flash stage is introduced at a pressure that is very close to the second compressor stage discharge condition, thus avoiding design complications to the compressor itself.
- Case B2A and B2B: from the technical point of view the strategy of increasing the stripper operating pressure is one of the most promising alternatives, whereas its economics are not attractive. This is explained through the significant impact that the higher solvent degradation has on the overall OPEX of the plant. However, it is noted that these solvent degradation rates have been taken from literature data, so data should be confirmed by referenced Licensors of the technology.
- Case D1: the increase of compression stage numbers show both CAPEX and OPEX improvements. Further increase of the stages number would theoretically lead to improved economics; however, the resulting further drop of the single stage compression ratio may not be acceptable for centrifugal machines, thus making this strategy not feasible.
- Case C3 and D2: All the CO_2 liquefaction strategies have NPVs greater than zero, showing that these solutions are economically attractive. However, the convenience of this strategy needs to be evaluated in conjunction with the cost of the pipeline, especially in warmer climates and for long transport distances, where either proper insulation/burying are required to keep the CO_2 below its critical temperature or the pipeline design needs to take into account drastic physical properties changes as the dense phase CO_2 is heated while it travels along the line. Case D3A: the deeper inter-cooling in the post-combustion capture show a positive NPV. However this represents a border line situation as indicated by the IRR (10.6 %), which is close to the discount rate (10 %). Either uncertainties on the cost estimate or slight changes to the basic economic factors (i.e. cost of the consumed electricity) may affect the attractiveness of this strategy from the techno-economic point of view.

8 Bibliography

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9 Attachments

9.1 Economic calculation spreadsheets

Case A2 - Differential Cost Evaluation wrt base case - Discount Rate = 10%

Production

Net Power Output 2.1 MW

Capital Expenditures	**MM Euro**
Total Investment Cost	-3.6

Operating Costs [MM Euro/year] (90% availability)	
Fuel Cost	0.0
Maintenance	-0.1
Miscellanea	0.0
Chemicals + Consumable	0.0
Insurance and local taxes	0.0
Carbon tax	0.0

Working Capital		**MM Euro**
30 days Chemical Storage		0.0
30 days Coal Storage		0.0
Total Working capital		0.0
Labour Cost		**MM Euro/year**
# operators		0
Salary		0.06
Direct Labour Cost		0.0
Administration	30% L.C.	0.0
Total Labour Cost		0.0

Electricity Production Cost	0.038	Euro/kWh
Inflation	0.00	%
Taxes	0.00	%
Discount rate	10.00	%
Revenues / year	0.6	MM Euro/year
NPV	**7.70**	
IRR	**N/A**	

CASH FLOW ANALYSYS Millions Euro	**2010**	**2011**	**2012**	**2013**	**2014**	**2015**	**2016**	**2017**	**2018**	**2019**	**2020**	**2021**	**2022**	**2023**	**2024**
	000	**00**	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**
Load Factor				75%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%
Equivalent yearly hours				6570	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446
Expediture Factor	20%	45%	35%												
Revenues															
Electric Energy				0.5	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
Operating Costs															
Fuel Cost				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Maintenance				0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1
Labour				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Chemicals & Consumables				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Miscellanea				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Insurance				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Carbon tax				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Working Capital Cost				0.0											
Fixed Capital Expenditures	0.7	1.6	1.3												
Total Cash flow (yearly)	0.7	1.6	1.3	0.6	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7
Total Cash flow (cumulated)	0.7	2.3	3.6	4.2	4.9	5.6	6.3	7.0	7.7	8.4	9.2	9.9	10.6	11.3	12.0
Discounted Cash Flow (Yearly)	0.7	1.3	0.9	0.4	0.4	0.4	0.4	0.3	0.3	0.3	0.2	0.2	0.2	0.2	0.2
Discounted Cash Flow (Cumul.)	0.7	2.0	2.9	3.3	3.8	4.2	4.5	4.9	5.2	5.5	5.7	5.9	6.1	6.3	6.5

CASH FLOW ANALYSYS Millions Euro	**2025**	**2026**	**2027**	**2028**	**2029**	**2030**	**2031**	**2032**	**2033**	**2034**	**2035**	**2036**	**2037**	**2038**
	13	**14**	**15**	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**	**25**	**26**
Load Factor	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	
Equivalent yearly hours	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	
Expediture Factor														
Revenues														
Electric Energy	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	
Operating Costs														
Fuel Cost	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Maintenance	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	
Labour	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Chemicals & Consumables	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Miscellanea	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Insurance	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Carbon tax	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Working Capital Cost														0.0
Fixed Capital Expenditures														
Total Cash flow (yearly)	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.0
Total Cash flow (cumulated)	12.7	13.4	14.1	14.8	15.5	16.3	17.0	17.7	18.4	19.1	19.8	20.5	21.2	21.2
Discounted Cash Flow (Yearly)	0.2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.0	0.0
Discounted Cash Flow (Cumul.)	6.6	6.8	6.9	7.0	7.1	7.2	7.3	7.4	7.5	7.5	7.6	7.6	7.7	7.7

Case B2a - Differential Cost Evaluation wrt base case - Discount Rate = 10%

Production

Net Power Output 4.4 MW

Capital Expenditures	**MM Euro**
Total Investment Cost	-5.2

Operating Costs [MM Euro/year] (90% availability)	
Fuel Cost	0.0
Maintenance	-0.2
Miscellanea	0.0
Chemicals + Consumable	4.9
Insurance and local taxes	0.0
Carbon tax	0.0

Working Capital	**MM Euro**
30 days Chemical Storage	0.0
30 days Coal Storage	0.0
Total Working capital	0.0

Labour Cost		**MM Euro/year**
# operators		0
Salary		0.06
Direct Labour Cost		0.0
Administration	30% L.C.	0.0
Total Labour Cost		0.0

Electricity Production Cost	0.038	Euro/kWh
Inflation	0.00	%
Taxes	0.00	%
Discount rate	10.00	%
Revenues / year	1.3	MM Euro/year
NPV	**-15.05**	
IRR	**N/A**	

CASH FLOW ANALYSYS Millions Euro	**2010**	**2011**	**2012**	**2013**	**2014**	**2015**	**2016**	**2017**	**2018**	**2019**	**2020**	**2021**	**2022**	**2023**	**2024**	**2025**	**2026**	**2027**	**2028**	**2029**	**2030**	**2031**	**2032**	**2033**	**2034**	**2035**	**2036**	**2037**	**2038**
	000	**00**	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**	**25**	**26**
Load Factor				75%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	
Equivalent yearly hours				6570	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	
Expediture Factor	20%	45%	35%																										
Revenues																													
Electric Energy				1.1	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	
Operating Costs																													
Fuel Cost				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Maintenance				0.1	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	
Labour				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Chemicals & Consumables				-3.7	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	-4.3	
Miscellanea				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Insurance				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Carbon tax				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Working Capital Cost				0.0																									0.0
Fixed Capital Expenditures	1.0	2.3	1.8																										
Total Cash flow (yearly)	1.0	2.3	1.8	-2.5	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	-2.9	0.0
Total Cash flow (cumulated)	1.0	3.4	5.2	2.7	-0.2	-3.0	-5.9	-8.8	-11.6	-14.5	-17.4	-20.2	-23.1	-26.0	-28.8	-31.7	-34.5	-37.4	-40.3	-43.1	-46.0	-48.9	-51.7	-54.6	-57.5	-60.3	-63.2	-66.1	-66.1
Discounted Cash Flow (Yearly)	0.9	1.9	1.4	-1.7	-1.8	-1.6	-1.5	-1.3	-1.2	-1.1	-1.0	-0.9	-0.8	-0.8	-0.7	-0.6	-0.6	-0.5	-0.5	-0.4	-0.4	-0.4	-0.3	-0.3	-0.3	-0.2	-0.2	-0.2	0.0
Discounted Cash Flow (Cumul.)	0.9	2.9	4.2	2.5	0.8	-0.9	-2.3	-3.7	-4.9	-6.0	-7.0	-7.9	-8.7	-9.5	-10.2	-10.8	-11.4	-11.9	-12.3	-12.8	-13.2	-13.5	-13.8	-14.1	-14.4	-14.6	-14.8	-15.0	-15.0

Production

Net Power Output 1.7 MW

Capital Expenditures **MM Euro**

Total Investment Cost	-10.8

Operating Costs [MM Euro/year]
(90% availability)

Fuel Cost	0.0
Maintenance	-0.3
Miscellanea	0.0
Chemicals + Consumable	9.2
Insurance and local taxes	0.0
Carbon tax	0.0

Working Capital **MM Euro**

30 days Chemical Storage	0.0
30 days Coal Storage	0.0
Total Working capital	0.0

Labour Cost **MM Euro/year**

# operators		0
Salary		0.06
Direct Labour Cost		0.0
Administration	30% L.C.	0.0
Total Labour Cost		0.0

Electricity Production Cost	0.038	Euro/kWh
Inflation	0.00	%
Taxes	0.00	%
Discount rate	10.00	%
Revenues / year	0.5	MM Euro/year

NPV **-39.68**
IRR **N/A**

CASH FLOW ANALYSYS Millions Euro	2010 000	2011 00	2012 0	2013 1	2014 2	2015 3	2016 4	2017 5	2018 6	2019 7	2020 8	2021 9	2022 10	2023 11
Load Factor				75%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%
Equivalent yearly hours				6570	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621
Expediture Factor	20%	45%	35%											
Revenues														
Electric Energy				0.4	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5
Operating Costs														
Fuel Cost				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Maintenance				0.2	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3
Labour				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Chemicals & Consumables				-6.9	-8.0	-8.0	-8.0	-8.0	-8.0	-8.0	-8.0	-8.0	-8.0	-8.0
Miscellanea				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Insurance				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Carbon tax				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Working Capital Cost				0.0										
Fixed Capital Expenditures	2.2	4.9	3.8											
Total Cash flow (yearly)	2.2	4.9	3.8	-6.3	-7.2	-7.2	-7.2	-7.2	-7.2	-7.2	-7.2	-7.2	-7.2	-7.2
Total Cash flow (cumulated)	2.2	7.0	10.8	4.5	-2.7	-9.9	-17.1	-24.3	-31.5	-38.7	-45.9	-53.1	-60.3	-67.5
Discounted Cash Flow (Yearly)	2.0	4.0	2.8	-4.3	-4.5	-4.1	-3.7	-3.4	-3.1	-2.8	-2.5	-2.3	-2.1	-1.9
Discounted Cash Flow (Cumul.)	2.0	6.0	8.8	4.5	0.1	-4.0	-7.7	-11.1	-14.1	-16.9	-19.4	-21.7	-23.8	-25.7

CASH FLOW ANALYSYS Millions Euro	2024 12	2025 13	2026 14	2027 15	2028 16	2029 17	2030 18	2031 19	2032 20	2033 21	2034 22	2035 23	2036 24	2037 25	2038 26
Load Factor	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	
Equivalent yearly hours	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	
Expediture Factor															
Revenues															
Electric Energy	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	
Operating Costs															
Fuel Cost	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Maintenance	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	
Labour	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Chemicals & Consumables	-8.0	-8.0	-8.0	-8.0	-8.0	-8.0	-8.0	-8.0	-8.0	-8.0	-8.0	-8.0	-8.0	-8.0	
Miscellanea	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Insurance	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Carbon tax	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Working Capital Cost															0.0
Fixed Capital Expenditures															
Total Cash flow (yearly)	-7.2	-7.2	-7.2	-7.2	-7.2	-7.2	-7.2	-7.2	-7.2	-7.2	-7.2	-7.2	-7.2	-7.2	0.0
Total Cash flow (cumulated)	-74.7	-81.9	-89.1	-96.4	-103.6	-110.8	-118.0	-125.2	-132.4	-139.6	-146.8	-154.0	-161.2	-168.4	-168.4
Discounted Cash Flow (Yearly)	-1.7	-1.6	-1.4	-1.3	-1.2	-1.1	-1.0	-0.9	-0.8	-0.7	-0.7	-0.6	-0.5	-0.5	0.0
Discounted Cash Flow (Cumul.)	-27.4	-29.0	-30.4	-31.7	-32.9	-34.0	-34.9	-35.8	-36.6	-37.4	-38.0	-38.6	-39.2	-39.7	-39.7

Production

Net Power Output 1.7 MW

Capital Expenditures **MM Euro**
Total Investment Cost 6.8

Operating Costs [MM Euro/year]
(90% availability)

Fuel Cost	0.0
Maintenance	0.2
Miscellanea	0.0
Chemicals + Consumable	0.0
Insurance and local taxes	0.0
Carbon tax	0.0

Working Capital **MM Euro**

30 days Chemical Storage	0.0
30 days Coal Storage	0.0
Total Working capital	0.0

Labour Cost **MM Euro/year**

# operators		0
Salary		0.06
Direct Labour Cost		0.0
Administration	30% L.C.	0.0
Total Labour Cost		0.0

Electricity Production Cost	0.038	Euro/kWh
Inflation	0.00	%
Taxes	0.00	%
Discount rate	10.00	%
Revenues / year	0.5	MM Euro/year

NPV **-3.63**
IRR **0.25%**

CASH FLOW ANALYSYS Millions Euro	2010	2011	2012	2013	2014	2015	2016	2017	2018	2019	2020	2021	2022	2023	2024	2025	2026	2027	2028	2029	2030	2031	2032	2033	2034	2035	2036	2037	2038
	000	**00**	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**	**25**	**26**
Load Factor				75%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	
Equivalent yearly hours				6570	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	
Expediture Factor	20%	45%	35%																										
Revenues																													
Electric Energy				0.4	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	
Operating Costs																													
Fuel Cost				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Maintenance				-0.1	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	
Labour				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Chemicals & Consumables				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Miscellanea				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Insurance				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Carbon tax				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Working Capital Cost				0.0																									0.0
Fixed Capital Expenditures	-1.4	-3.1	-2.4																										
Total Cash flow (yearly)	-1.4	-3.1	-2.4	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.0
Total Cash flow (cumulated)	-1.4	-4.4	-6.8	-6.5	-6.2	-6.0	-5.7	-5.4	-5.1	-4.8	-4.5	-4.3	-4.0	-3.7	-3.4	-3.1	-2.9	-2.6	-2.3	-2.0	-1.7	-1.4	-1.2	-0.9	-0.6	-0.3	0.0	0.2	0.2
Discounted Cash Flow (Yearly)	-1.2	-2.5	-1.8	0.2	0.2	0.2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Discounted Cash Flow (Cumul.)	-1.2	-3.8	-5.6	-5.4	-5.2	-5.0	-4.9	-4.7	-4.6	-4.5	-4.4	-4.3	-4.3	-4.2	-4.1	-4.0	-4.0	-3.9	-3.9	-3.9	-3.8	-3.8	-3.8	-3.7	-3.7	-3.7	-3.7	-3.6	-3.6

FOSTER WHEELER

Case C3 - Differential Cost Evaluation wrt base case - Discount Rate = 10%

Production

Net Power Output 0.2 MW

Capital Expenditures	**MM Euro**
Total Investment Cost	-1.6

Operating Costs [MM Euro/year] (90% availability)	
Fuel Cost	0.0
Maintenance	0.0
Miscellanea	0.0
Chemicals + Consumable	0.0
Insurance and local taxes	0.0
Carbon tax	0.0

Working Capital	**MM Euro**
30 days Chemical Storage	0.0
30 days Coal Storage	0.0
Total Working capital	0.0

Labour Cost		**MM Euro/year**
# operators		0
Salary		0.06
Direct Labour Cost		0.0
Administration	30% L.C.	0.0
Total Labour Cost		0.0

Electricity Production Cost	0.038	Euro/kWh
Inflation	0.00	%
Taxes	0.00	%
Discount rate	10.00	%
Revenues / year	0.1	MM Euro/year
NPV	**2.00**	
IRR	**N/A**	

CASH FLOW ANALYSYS Millions Euro	**2010**	**2011**	**2012**	**2013**	**2014**	**2015**	**2016**	**2017**	**2018**	**2019**	**2020**	**2021**	**2022**	**2023**	**2024**	**2025**	**2026**	**2027**	**2028**	**2029**	**2030**	**2031**	**2032**	**2033**	**2034**	**2035**	**2036**	**2037**	**2038**
	000	**00**	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**	**25**	**26**
Load Factor				75%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	
Equivalent yearly hours				6570	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	
Expediture Factor	20%	45%	35%																										
Revenues																													
Electric Energy				0.0	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	
Operating Costs																													
Fuel Cost				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Maintenance				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Labour				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Chemicals & Consumables				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Miscellanea				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Insurance				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Carbon tax				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Working Capital Cost				0.0																									0.0
Fixed Capital Expenditures	0.3	0.7	0.6																										
Total Cash flow (yearly)	0.3	0.7	0.6	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.0
Total Cash flow (cumulated)	0.3	1.0	1.6	1.7	1.8	1.9	2.0	2.1	2.2	2.3	2.4	2.5	2.6	2.7	2.8	2.9	3.0	3.1	3.3	3.4	3.5	3.6	3.7	3.8	3.9	4.0	4.1	4.2	4.2
Discounted Cash Flow (Yearly)	0.3	0.6	0.4	0.1	0.1	0.1	0.1	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Discounted Cash Flow (Cumul.)	0.3	0.9	1.3	1.4	1.4	1.5	1.5	1.6	1.6	1.7	1.7	1.7	1.8	1.8	1.8	1.8	1.9	1.9	1.9	1.9	1.9	1.9	2.0	2.0	2.0	2.0	2.0	2.0	2.0

FOSTER WHEELER

Case D1 - Differential Cost Evaluation wrt base case- Discount Rate = 10%

Rev. : 0
Date : Jan 2011

Production

Net Power Output 2.0 MW

Capital Expenditures **MM Euro**
Total Investment Cost -2.1

Operating Costs [MM Euro/year]
(90% availability)

Fuel Cost	0.0
Maintenance	-0.1
Miscellanea	0.0
Chemicals + Consumable	0.0
Insurance and local taxes	0.0
Carbon tax	0.0

Working Capital **MM Euro**

30 days Chemical Storage	0.0
30 days Coal Storage	0.0
Total Working capital	0.0

Labour Cost **MM Euro/year**

# operators		0
Salary		0.06
Direct Labour Cost		0.0
Administration	30% L.C.	0.0
Total Labour Cost		0.0

Electricity Production Cost	0.038	Euro/kWh
Inflation	0.00	%
Taxes	0.00	%
Discount rate	10.00	%
Revenues / year	0.6	MM Euro/year

NPV **6.01**
IRR **N/A**

CASH FLOW ANALYSYS Millions Euro	2010	2011	2012	2013	2014	2015	2016	2017	2018	2019	2020	2021	2022	2023
	000	00	0	1	2	3	4	5	6	7	8	9	10	11
Load Factor				75%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%
Equivalent yearly hours				6570	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621
Expediture Factor	20%	45%	35%											
Revenues														
Electric Energy				0.5	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
Operating Costs														
Fuel Cost				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Maintenance				0.0	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1
Labour				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Chemicals & Consumables				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Miscellanea				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Insurance				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Carbon tax				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Working Capital Cost				0.0										
Fixed Capital Expenditures	0.4	0.9	0.7											
Total Cash flow (yearly)	0.4	0.9	0.7	0.5	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
Total Cash flow (cumulated)	0.4	1.4	2.1	2.6	3.3	3.9	4.6	5.2	5.8	6.5	7.1	7.8	8.4	9.0
Discounted Cash Flow (Yearly)	0.4	0.8	0.6	0.4	0.4	0.4	0.3	0.3	0.3	0.2	0.2	0.2	0.2	0.2
Discounted Cash Flow (Cumul.)	0.4	1.2	1.7	2.1	2.5	2.8	3.2	3.5	3.7	4.0	4.2	4.4	4.6	4.8

CASH FLOW ANALYSYS Millions Euro	2024	2025	2026	2027	2028	2029	2030	2031	2032	2033	2034	2035	2036	2037	2038
	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26
Load Factor	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	
Equivalent yearly hours	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	
Expediture Factor															
Revenues															
Electric Energy	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	
Operating Costs															
Fuel Cost	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Maintenance	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	
Labour	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Chemicals & Consumables	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Miscellanea	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Insurance	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Carbon tax	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Working Capital Cost															0.0
Fixed Capital Expenditures															
Total Cash flow (yearly)	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.0
Total Cash flow (cumulated)	9.7	10.3	10.9	11.6	12.2	12.9	13.5	14.1	14.8	15.4	16.1	16.7	17.3	18.0	18.0
Discounted Cash Flow (Yearly)	0.2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.0	0.0	0.0
Discounted Cash Flow (Cumul.)	4.9	5.1	5.2	5.3	5.4	5.5	5.6	5.7	5.7	5.8	5.9	5.9	6.0	6.0	6.0

Production

Net Power Output 0.2 MW

Capital Expenditures	**MM Euro**
Total Investment Cost	-8.2

Operating Costs [MM Euro/year] (90% availability)	
Fuel Cost	0.0
Maintenance	-0.3
Miscellanea	0.0
Chemicals + Consumable	0.0
Insurance and local taxes	0.0
Carbon tax	0.0

Working Capital		**MM Euro**
30 days Chemical Storage		0.0
30 days Coal Storage		0.0
Total Working capital		0.0

Labour Cost		**MM Euro/year**
# operators		0
Salary		0.06
Direct Labour Cost		0.0
Administration	30% L.C.	0.0
Total Labour Cost		0.0

Electricity Production Cost	0.038	Euro/kWh
Inflation	0.00	%
Taxes	0.00	%
Discount rate	10.00	%
Revenues / year	0.1	MM Euro/year

NPV 8.75
IRR N/A

CASH FLOW ANALYSYS Millions Euro	2010	2011	2012	2013	2014	2015	2016	2017	2018	2019	2020	2021	2022	2023	2024	2025	2026	2027	2028	2029	2030	2031	2032	2033	2034	2035	2036	2037	2038
	000	00	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26
Load Factor				75%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	
Equivalent yearly hours				6570	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	
Expediture Factor	20%	45%	35%																										
Revenues																													
Electric Energy				0.0	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	
Operating Costs																													
Fuel Cost				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Maintenance				0.2	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	
Labour				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Chemicals & Consumables				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Miscellanea				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Insurance				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Carbon tax				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Working Capital Cost				0.0																									0.0
Fixed Capital Expenditures	1.6	3.7	2.9																										
Total Cash flow (yearly)	1.6	3.7	2.9	0.2	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.0
Total Cash flow (cumulated)	1.6	5.3	8.2	8.4	8.7	9.0	9.3	9.7	10.0	10.3	10.6	10.9	11.2	11.5	11.8	12.1	12.5	12.8	13.1	13.4	13.7	14.0	14.3	14.6	14.9	15.3	15.6	15.9	15.9
Discounted Cash Flow (Yearly)	1.5	3.0	2.2	0.1	0.2	0.2	0.2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Discounted Cash Flow (Cumul.)	1.5	4.5	6.7	6.8	7.0	7.2	7.4	7.5	7.6	7.8	7.9	8.0	8.1	8.1	8.2	8.3	8.3	8.4	8.5	8.5	8.5	8.6	8.6	8.6	8.7	8.7	8.7	8.7	8.7

Case D2b - Differential Cost Evaluation wrt base case - Discount Rate = 10%

Production

Coal Flow rate	-0.96	t/h
Fuel Price	3	Euro/GJ
Net Power Output	7.6	MW
CO2 flow rate	2.9	t/h
Carbon tax	20	€/t(CO2)

Capital Expenditures	**MM Euro**
Total Investment Cost	5.2

Operating Costs [MM Euro/year]
(90% availability)

Fuel Cost	-0.7
Maintenance	0.2
Miscellanea	0.0
Chemicals + Consumable	0.0
Insurance and local taxes	0.0
Carbon tax	0.5

Working Capital	**MM Euro**
30 days Chemical Storage	0.0
30 days Coal Storage	0.0
Total Working capital	0.0

Labour Cost		**MM Euro/year**
# operators		0
Salary		0.06
Direct Labour Cost		0.0
Administration	30% L.C.	0.0
Total Labour Cost		0.0

Electricity Production Cost	0.038	Euro/kWh
Inflation	0.00	%
Taxes	0.00	%
Discount rate	10.00	%
Revenues / year	2.1	MM Euro/year

NPV **9.75**
IRR **30.43%**

CASH FLOW ANALYSYS Millions Euro	**2010**	**2011**	**2012**	**2013**	**2014**	**2015**	**2016**	**2017**	**2018**	**2019**	**2020**	**2021**	**2022**	**2023**	**2024**	**2025**	**2026**	**2027**	**2028**	**2029**	**2030**	**2031**	**2032**	**2033**	**2034**	**2035**	**2036**	**2037**	**2038**
	000	**00**	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**	**25**	**26**
Load Factor				75%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	
Equivalent yearly hours				6570	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	
Expediture Factor	20%	45%	35%																										
Revenues																													
Electric Energy				1.9	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	
Operating Costs																													
Fuel Cost				0.5	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	
Maintenance				-0.1	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	
Labour				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Chemicals & Consumables				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Miscellanea				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Insurance				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Carbon tax				-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	-0.4	
Working Capital Cost				0.0																									0.0
Fixed Capital Expenditures	-1.0	-2.3	-1.8																										
Total Cash flow (yearly)	-1.0	-2.3	-1.8	1.9	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	0.0
Total Cash flow (cumulated)	-1.0	-3.4	-5.2	-3.3	-1.3	0.8	2.9	5.0	7.0	9.1	11.2	13.3	15.3	17.4	19.5	21.6	23.6	25.7	27.8	29.8	31.9	34.0	36.1	38.1	40.2	42.3	44.4	46.4	46.4
Discounted Cash Flow (Yearly)	-0.9	-1.9	-1.4	1.3	1.3	1.2	1.1	1.0	0.9	0.8	0.7	0.7	0.6	0.5	0.5	0.5	0.4	0.4	0.3	0.3	0.3	0.3	0.2	0.2	0.2	0.2	0.2	0.1	0.0
Discounted Cash Flow (Cumul.)	-0.9	-2.9	-4.3	-3.0	-1.7	-0.5	0.5	1.5	2.4	3.2	3.9	4.6	5.2	5.7	6.2	6.7	7.1	7.5	7.8	8.1	8.4	8.6	8.9	9.1	9.3	9.4	9.6	9.8	9.8

Case D2c - Differential Cost Evaluation wrt base case - Discount Rate = 10%

Production

Net Power Output 3.4 MW

Capital Expenditures	**MM Euro**
Total Investment Cost	-1.7

Operating Costs [MM Euro/year] (90% availability)	
Fuel Cost	0.0
Maintenance	-0.1
Miscellanea	0.0
Chemicals + Consumable	0.0
Insurance and local taxes	0.0
Carbon tax	0.0

Working Capital		**MM Euro**
30 days Chemical Storage		0.0
30 days Coal Storage		0.0
Total Working capital		0.0
Labour Cost		**MM Euro/year**
# operators		0
Salary		0.06
Direct Labour Cost		0.0
Administration	30% L.C.	0.0
Total Labour Cost		0.0

Electricity Production Cost	0.038	Euro/kWh
Inflation	0.00	%
Taxes	0.00	%
Discount rate	10.00	%
Revenues / year	1.0	MM Euro/year
NPV	**8.25**	
IRR	**N/A**	

CASH FLOW ANALYSYS Millions Euro	**2010**	**2011**	**2012**	**2013**	**2014**	**2015**	**2016**	**2017**	**2018**	**2019**	**2020**	**2021**	**2022**	**2023**	**2024**	**2025**	**2026**	**2027**	**2028**	**2029**	**2030**	**2031**	**2032**	**2033**	**2034**	**2035**	**2036**	**2037**	**2038**
	000	**00**	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**	**25**	**26**
Load Factor				75%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	
Equivalent yearly hours				6570	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	
Expediture Factor	20%	45%	35%																										
Revenues																													
Electric Energy				0.8	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	
Operating Costs																													
Fuel Cost				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Maintenance				0.0	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	
Labour				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Chemicals & Consumables				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Miscellanea				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Insurance				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Carbon tax				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Working Capital Cost				0.0																									0.0
Fixed Capital Expenditures	0.3	0.7	0.6																										
Total Cash flow (yearly)	0.3	0.7	0.6	0.9	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	0.0
Total Cash flow (cumulated)	0.3	1.1	1.7	2.5	3.6	4.6	5.6	6.6	7.7	8.7	9.7	10.7	11.8	12.8	13.8	14.8	15.9	16.9	17.9	18.9	20.0	21.0	22.0	23.0	24.1	25.1	26.1	27.1	27.1
Discounted Cash Flow (Yearly)	0.3	0.6	0.4	0.6	0.6	0.6	0.5	0.5	0.4	0.4	0.4	0.3	0.3	0.3	0.2	0.2	0.2	0.2	0.2	0.2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.0
Discounted Cash Flow (Cumul.)	0.3	0.9	1.4	2.0	2.6	3.2	3.7	4.2	4.6	5.0	5.4	5.7	6.0	6.3	6.5	6.7	6.9	7.1	7.3	7.4	7.6	7.7	7.8	7.9	8.0	8.1	8.2	8.2	8.2

FOSTER WHEELER

Case D3a- Differential Cost Evaluation wrt base case- Discount Rate = 10%

Production

Net Power Output 2.6 MW

Capital Expenditures	**MM Euro**
Total Investment Cost	4.7

Operating Costs [MM Euro/year] **(90% availability)**	
Fuel Cost	0.0
Maintenance	0.1
Miscellanea	0.0
Chemicals + Consumable	0.0
Insurance and local taxes	0.0
Carbon tax	0.0

Working Capital	**MM Euro**
30 days Chemical Storage	0.0
30 days Coal Storage	0.0
Total Working capital	0.0

Labour Cost		**MM Euro/year**
# operators		0
Salary		0.06
Direct Labour Cost		0.0
Administration	30% L.C.	0.0
Total Labour Cost		0.0

Electricity Production Cost	0.038	Euro/kWh
Inflation	0.00	%
Taxes	0.00	%
Discount rate	10.00	%
Revenues / year	0.7	MM Euro/year
NPV	**0.21**	
IRR	**10.63%**	

CASH FLOW ANALYSYS Millions Euro	**2010**	**2011**	**2012**	**2013**	**2014**	**2015**	**2016**	**2017**	**2018**	**2019**	**2020**	**2021**	**2022**	**2023**	**2024**
	000	**00**	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**
Load Factor				75%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%
Equivalent yearly hours				6570	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621
Expediture Factor	20%	45%	35%												
Revenues															
Electric Energy				0.6	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7
Operating Costs															
Fuel Cost				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Maintenance				-0.1	-0.1	-0.1	-0.1	-0.1	-0.1	-0.1	-0.1	-0.1	-0.1	-0.1	-0.1
Labour				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Chemicals & Consumables				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Miscellanea				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Insurance				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Carbon tax				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Working Capital Cost				0.0											
Fixed Capital Expenditures	-0.9	-2.1	-1.6												
Total Cash flow (yearly)	-0.9	-2.1	-1.6	0.5	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
Total Cash flow (cumulated)	-0.9	-3.1	-4.7	-4.2	-3.6	-3.0	-2.4	-1.8	-1.2	-0.6	0.0	0.6	1.2	1.8	2.4
Discounted Cash Flow (Yearly)	-0.9	-1.7	-1.2	0.4	0.4	0.3	0.3	0.3	0.3	0.2	0.2	0.2	0.2	0.2	0.1
Discounted Cash Flow (Cumul.)	-0.9	-2.6	-3.8	-3.5	-3.1	-2.8	-2.4	-2.2	-1.9	-1.7	-1.5	-1.3	-1.1	-1.0	-0.8

CASH FLOW ANALYSYS Millions Euro	**2025**	**2026**	**2027**	**2028**	**2029**	**2030**	**2031**	**2032**	**2033**	**2034**	**2035**	**2036**	**2037**	**2038**
	13	**14**	**15**	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**	**25**	**26**
Load Factor	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	87%	
Equivalent yearly hours	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	7621	
Expediture Factor														
Revenues														
Electric Energy	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	
Operating Costs														
Fuel Cost	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Maintenance	-0.1	-0.1	-0.1	-0.1	-0.1	-0.1	-0.1	-0.1	-0.1	-0.1	-0.1	-0.1	-0.1	
Labour	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Chemicals & Consumables	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Miscellanea	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Insurance	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Carbon tax	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Working Capital Cost														0.0
Fixed Capital Expenditures														
Total Cash flow (yearly)	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.0
Total Cash flow (cumulated)	3.0	3.6	4.2	4.8	5.4	6.0	6.6	7.2	7.8	8.4	9.0	9.6	10.2	10.2
Discounted Cash Flow (Yearly)	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.0	0.0	0.0
Discounted Cash Flow (Cumul.)	-0.7	-0.6	-0.5	-0.4	-0.3	-0.2	-0.1	0.0	0.0	0.1	0.1	0.2	0.2	0.2

FOSTER WHEELER

Case D3b - Differential Cost Evaluation wrt base case- Discount Rate = 10%

Production

Net Power Output 2.0 MW

Capital Expenditures **MM Euro**

Total Investment Cost 4.9

Operating Costs [MM Euro/year]
(90% availability)

Fuel Cost	0.0
Maintenance	0.2
Miscellanea	0.0
Chemicals + Consumable	0.0
Insurance and local taxes	0.0
Carbon tax	0.0

Working Capital **MM Euro**

30 days Chemical Storage	0.0
30 days Coal Storage	0.0
Total Working capital	0.0

Labour Cost **MM Euro/year**

# operators		0
Salary		0.06
Direct Labour Cost		0.0
Administration	30% L.C.	0.0
Total Labour Cost		0.0

Electricity Production Cost	0.038	Euro/kWh
Inflation	0.00	%
Taxes	0.00	%
Discount rate	10.00	%
Revenues / year	0.6	MM Euro/year

NPV **-1.27**
IRR **5.94%**

CASH FLOW ANALYSYS Millions Euro	**2010**	**2011**	**2012**	**2013**	**2014**	**2015**	**2016**	**2017**	**2018**	**2019**	**2020**	**2021**	**2022**	**2023**	**2024**	**2025**	**2026**	**2027**	**2028**	**2029**	**2030**	**2031**	**2032**	**2033**	**2034**	**2035**	**2036**	**2037**	**2038**
	000	**00**	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**	**25**	**26**
Load Factor				75%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	85%	
Equivalent yearly hours				6570	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	7446	
Expediture Factor	20%	45%	35%																										
Revenues																													
Electric Energy				0.5	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	
Operating Costs																													
Fuel Cost				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Maintenance				-0.1	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	
Labour				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Chemicals & Consumables				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Miscellanea				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Insurance				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Carbon tax				0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
Working Capital Cost				0.0																									0.0
Fixed Capital Expenditures	-1.0	-2.2	-1.7																										
Total Cash flow (yearly)	-1.0	-2.2	-1.7	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.0
Total Cash flow (cumulated)	-1.0	-3.2	-4.9	-4.5	-4.1	-3.7	-3.3	-2.9	-2.5	-2.1	-1.7	-1.3	-0.9	-0.5	-0.1	0.3	0.7	1.1	1.5	1.9	2.3	2.7	3.1	3.5	3.9	4.3	4.7	5.1	5.1
Discounted Cash Flow (Yearly)	-0.9	-1.8	-1.3	0.3	0.2	0.2	0.2	0.2	0.2	0.2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Discounted Cash Flow (Cumul.)	-0.9	-2.7	-4.0	-3.7	-3.5	-3.2	-3.0	-2.9	-2.7	-2.5	-2.4	-2.3	-2.1	-2.0	-1.9	-1.9	-1.8	-1.7	-1.6	-1.6	-1.5	-1.5	-1.4	-1.4	-1.4	-1.3	-1.3	-1.3	-1.3

CLIENT : IEA GREENHOUSE GAS R&D PROGRAMME
PROJECT NAME : ROTATING MACHINERY FOR CO_2 COMPRESSION IN CCS SYSTEMS
DOCUMENT NAME: COMPRESSION EQUIPMENT SURVEY

ISSUED BY : V. TOTA
CHECKED BY : L. MANCUSO
APPROVED BY : C. FILASTOPPI

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SECTION D

COMPRESSION EQUIPMENT SURVEY

INDEX

1 Types and sizes of compression machinery

In the past years, reciprocating compressors have been conventionally used for the compression of CO_2. Nevertheless, this technology has revealed several limits, mainly due to the low range of capacities that such machines can handle, typically from 25,000 kg/h to 40,000 kg/h, i.e. an order of magnitude less than the capacity required by the large-scale industrial plants assessed in this study. Nowadays, the technology of reciprocating compressors is leaving the space to centrifugal compressors, which by far represent the current state of the art for CCS applications.

However the number of factors in favour of the reciprocating are:

- Familiarity of the field operators with these machines which are frequently used in refinery processes
- Flexibility as far as concern the pressure ratio and capacity control system by means of valve unloaders (step or stepless capacity control system).
- Short delivery time, since some manufacturers dispose of selection of cylinders and frames on stock so the package can be prefabricated shortly.

On the other hand, the main disadvantages of reciprocating machines can be summarized as follows:

- Multiple compressor units to handle the process capacity required and the scale of CO_2 capture plant is growing up over the time.
- Large plot area to accommodate all the compressor units
- Impact on the design of the plant due to the massive foundation necessary to keep the vibration of the foundation within acceptable ranges, as well as a piping analogue study being necessary to limit the vibrations produced by the pulsating gas flow in pipe and equipment
- Maintenance aspects. Generally speaking the maintenance is intensive and would be more frequent when high density gas such as CO_2 is used especially for compressor valves
- Cylinder size is affected by the maximum allowable piston rod loading for each throw which is generally limited by the frame size rather than the max dimensions available for the cylinder casts. Frame size also defines the maximum numbers of throws (typically 10 is the max number of cylinders installable in a single frame, depending on manufacturer's capability) and maximum power for each throw is in the range of 3600 KW, hence max total power available is about 36000 KW per compressor.

For these reasons, reciprocating compressors are being displaced in the market place by centrifugal compressors, which are by far the state of the art and will be the solution for future CO_2 projects.

The manufactures being contacted by FWI for the study associated with CO_2 sequestration have been the following: MAN Turbo, Rolls-Royce, GE, Elliott and Siemens. FWI received full cooperation from MAN Turbo and Rolls-Royce, partial reply from GE and no answer from Elliott and Siemens.

Usually the centrifugal compressors can be categorized into two main branches namely single shaft in-line between bearings and multi-shaft integral gear type. Both types of machines are basically designed according to the International code API 617 (Axial and centrifugal compressors for petroleum, chemical and gas industry).
By comparison with reciprocating machines, the centrifugal compressors offer:

- higher efficiency
- superior reliability (typically in the range of about 97% for integral gear compressor and 99% for in-line compressors)
- extended intervals between overhauls
- direct couple to the driver running at high speed, via steam turbine or electric motor.

Generally, the design of the centrifugal compressors at the maximum inlet flow is driven by the inlet Mach number limit (about 0.9), which is imposed to avoid aerodynamic issues. This parameter, in conjunction with the molecular weight of the gas (about 44 for CO_2), defines the max allowable relative inlet velocity to the impeller and thus the maximum axial inlet velocity, assuming that the maximum peripheral speed is also limited by the mechanical strength and deformations due to the centrifugal force.
All these design parameters dictate the maximum inlet gas flow.

Within the family of the centrifugal compressors, the configuration and the characteristics of the "in-line" machines are significantly different from the "integral-gear" machines, due to their intrinsic design.

In a traditional in-line compressor, all the impellers are shrunk-on the shaft and consequently once the shaft speed is defined, it is the same for all the impellers enclosed in the casing.
Nevertheless the compressor manufacturers have standardised the size of casing and the maximum numbers of impellers for each casing and within a pre-defined range of pressures.
The main advantage of the "in-line" compressors over the "integral-gear" is related to maintenance access, since the "in-line" machine configuration allows the inner bundle of barrel casing or upper casing of axially split machines to be easily

inspected from the end (barrel casing) or from top (horizontal split casing) generally without disturbing the process gas piping.

The following sections provide an overview of the key characteristics, availability on the market and vendors experience for both centrifugal compressors branches in relation to potential application to CCS.

1.1 Single shaft "in-line" centrifugal compressors for CCS application

One of the most qualified manufacturers of traditional "in-line" centrifugal compressors demonstrating interest in the CCS application is Rolls-Royce, having in their current production the following series of casings as reported in the table 1.1 and 1.2 below. The tables summarise the main characteristics of the in-line compressors.

Table 1.1 – RR Centrifugal Vertically Split Frame

MODEL	RBB	RCB	RDB	REB
Max working pressure [bara]	310	222	140	85
Number of stages	1 – 9	1 - 9	1 - 9	1 - 9
Design speed range [rpm]	8000-13800	5000-11500	4500-8650	3500-6500
Maximum design flow [m3/h]	10200	23000	37000	60300

Table 1.2 – RR Centrifugal Horizontally Split Frame

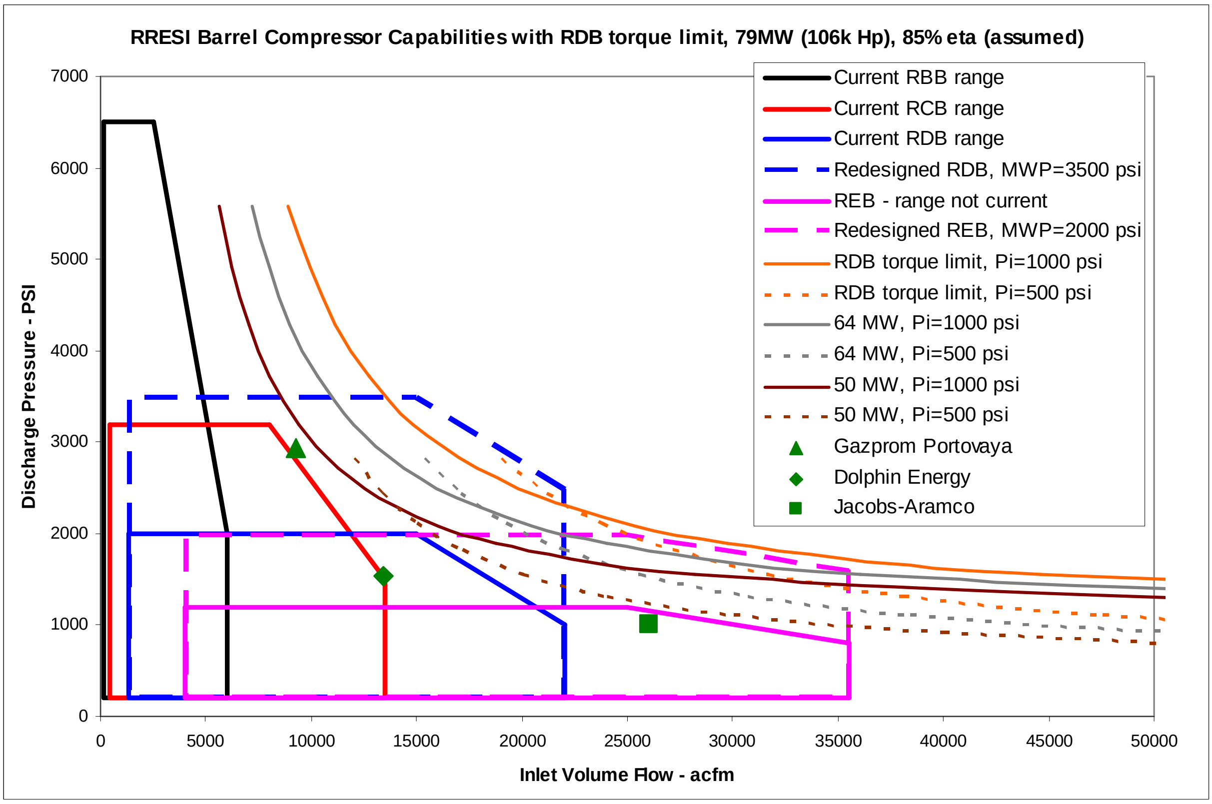
MODEL	RDS	RES	RFS
Max working pressure [bara]	34.5	28.6	25.1
Number of stages	1 - 9	1 - 9	1 - 9
Design speed range [rpm]	4500-8650	3000-6500	2500-5600
Maximum design flow [m3/h]	38200	68000	119000

The horizontal split compressors are legacy machines to be upgraded for CCS applications. Limits are not yet established for the upgraded versions of these frames but they would be above the limits for the legacy frames (i.e. the discharge pressure for RFS/RES would be above 34 bara).

Similarly the number of stages exhibited in the tables above is subject to modifications and likely increased as a result of the upgrading.

At the same time, even the barrel casings are under development (i.e. RBB, RCB, RDB) and the fig.1 shows the current and upgraded capabilities for R-R barrel compressors. The upgrades shown on the chart are currently being undertaken.
With the concurrent new 33 impellers family R-R expect to improve the efficiency of the compressors to 86%.

Fig 1.1 – RR Centrifugal barrel compressor: development work to extend frame size capabilities



The only further development that is currently proposed for barrel compressors is the upgrade of the RAB frame, the smallest in the product family [this model was not supposed to be used for CCS applications and requires a redevelopment].
This frame is expected to have the capability shown below

Tab 1.3 - RR Centrifugal barrel RAB compressor capabilities

MODEL	RAB
Max working pressure [bara]	500
Number of stages	1 - 9
Design speed range [rpm]	10000-18000
Maximum design flow [m3/h]	5700
Minimum design flow [m3/h]	150

Rolls Royce has limited experience with pure CO_2 compression as Cooper-Rolls exited the process compressor industry in the early '70s (In 1968 RR formed an oil and gas joint venture with Cooper Industries called Cooper Rolls and in 1999 acquired the remaining share of the rotating compression equipment interests of Cooper Energy Services).
However RR's experience of dealing with very varied applications has continued to build since then. This includes specific studies carried out by RR through its CO_2 Optimised Compression Project, COZOC , a significant number of compressors operating with high molar weight gases (around 90) and a number of recent applications (14 jobs) with varying CO_2 content up to 48% combined with some other "nasties" . RR's experience lists for pure CO_2 compressors is limited to a couple jobs:

1. First job in Mexico for Pemex, with train 'A' consisting of 1 machine RC7-6S (4,440 acfm, suction pressure 4 PSIA, discharge pressure 66 PSIA, compressor absorbed power 2,110 HP, steam turbine driver running at 9,500 rpm, ship year 1969) and train 'B' consisting of 1 machine RB8-7S (1,440 acfm, suction pressure 64 PSIA, discharge pressure 330 PSIA, year 1969)

2. Second job in India for Zuari Agro Chem consisting of 1 machine RC9-7S, 12,760 acfm, suction pressure 16 PSIA, discharge pressure 462 PSIA, steam turbine driver power 4820 HP.

Even if the applications with pure CO_2 are not extensive, all of these applications demonstrate R-R ability to safely handle a variety of gases (including the CO_2 mixtures expected for CCS applications). This is because, given the ability to determine the correct thermodynamic properties of the gas mixture (which RR has through its suite of specifically designed, well validated, in-house design software) and to address the corrosive potential of the gas with appropriate material selection, the project specific design of the compressors is part of everyday life for compressor designers.

The main advantage of the “in-line” compressors is related to maintenance access, since the “in-line” machine configuration allows the inner bundle of barrel casing or upper casing of axially split machine to be easily inspected from the end (barrel casing) or from top (horizontal split casing), generally without disturbing the process gas piping. Other technical advantages of the in-line type are as follows:

- Higher operating flexibility, due to the multiple parallel trains configuration, being the turndown capability of the single train very similar for both types. Also, the VFD provided with the in-line machines ensures better efficiency (i.e. lower parasitic consumption) when the plant operates at partial load. This is a very important feature since it is expected that CCS power plants will be required to operate in the actual electricity market, responding to the normal daily and seasonal variability of electricity demand.
- Higher reliability, typically by 2% with respect to the integral-gear compressors.
- Lower mesh losses, since the bull gear in the integrally geared solution introduces additional losses.
- Generally lower power demand.
- Reduced impact on the electrical system design. The impact of using large motors is mainly represented by the necessity for a significant over design of the electrical systems equipment (transformers, cables, etc.) to support the peak current demand at motor starting. For the in-line compressor, smaller size (roughly half) for the largest motors has been proposed with respect to the integral–gear compressors. Also, VFD’s have been included for in-line machines capacity control, which are expected to perform better in smoothing the peak demand at motor starting than the soft starters proposed with the integral gear type.
- Higher flexibility in dealing with uncertainties (e.g. process upset, changes in operating conditions with time) once the machine is built. However it is noted that, in the integral gear concept, reducing the number of drivers and modifying the design of impeller/volute or pinion speed are viable modifications to face changes to the operating conditions.

1.2 Multi-shaft integral gear centrifugal compressors for CCS application

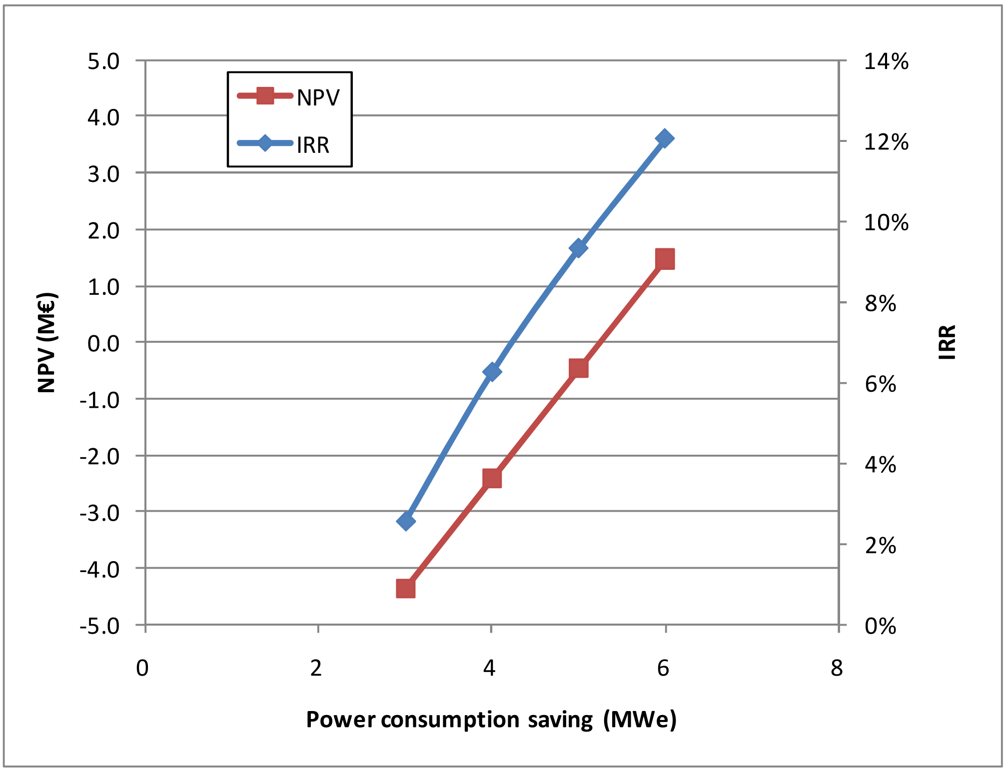
The construction of an integral gear compressor is based on a single bull gear coupled to driver which rotates up to five shafts at the end of which are shrunk-on the

impellers; each shaft has its own speed that is defined by the number of teeth of the pinion.
The main advantages of the integral gear compressor over the "in-line" can be summarised as follows:

- optimum impeller flow coefficient and volute, due to the fact that optimum speed can be selected for each pair of impellers
- design facilities of impellers such as small hub/tip ratio, shrouded or non-shrouded version available
- inter-cooling connection facilitated after each stage (impeller), being each impeller enclosed in its own casing.
- external connection after each stage offers the possibility to define the level of pressure with minor impact on the compressor arrangement (an in-line compressor may need to change the configuration, i.e. number of impellers or casings);
- The general arrangement is such that the compressor rotors and impellers are located all around the bull gear, making the machine design compact and taking space in vertical / radial directions rather than in axial direction. However this advantage may be smoothed as integrally geared concepts typically have more coolers and scrubbers than inline systems.
- Typically lower investment cost.

It is noted that the CAPEX tends to be lower for integral gear type; however, in the context of a general evaluation of the economics, this may be off-set by the typically higher electrical consumption and therefore OPEX. A differential Discounted Cash Flow (DCF) analysis can be used in order to appreciate how different CAPEX and OPEX are compared over the design life of a CCS plant. For example, assuming an incremental CAPEX of 10 MM€, Figure 1-1 shows the sensitivity of the DCF analysis results to power consumption saving figures varying in the range of 3 to 6 MW_e. With the assumed input data, the CAPEX increase is off-set when power saving is approximately above 5 MW_e.

Figure 1-1 Typical differential DCF analysis based on an incremental CAPEX of 10 M € – Sensitivity to power consumption saving figures from 3 to 6 MW_e



One of the manufacturer leaders on the market of CO_2 compression is MAN Diesel & Turbo. They can offer both types of centrifugal compressors but they came to the conclusion that for most CO_2 applications, the multi-shaft integral-gear design offers undeniable advantages.

The "standard range" of production of multi-shaft integral-gear compressors for general process gas applications showing the main characteristics is outlined in the table 1.4 below.

Table 1.4 – Overview of typical sizes, flows and generic data

MODEL	RG25	RG40	RG45	RG50	RG56
Flow [Am3/h]	10000	25000	30000	40000	50000
Power [MW]	4	15	18	20	20
Dimensions LxWxH [m]	2.7x3.6x2	3x3.6x2.5	3.4x3.6x3	3.7x3.6x3.5	4x3.6x3.5
Weight [t]	15	30	40	45	50

MODEL	RG80	RG100	RG140	RG160
Flow [Am3/h]	100000	200000	350000	500000
Power [MW]	20	35	50	60
Dimensions LxWxH [m]	4.5x3.6x4	5.5x3.6x5	- x>3.6x7	- x>7x>7
Weight [t]	60	>60	>60	>130

The driver can be electric motor or steam turbine and the number of pinions can be up to 5 with number of impellers up to 10.

Latest results of MAN development extend the product line for high volume flow and now they can offer the following compressor models for CO_2 applications, which use the existing gear, according to table 1.5.

Table 1.5 – Overview of typical sizes and flows

MODEL	RG45-8	RG80-8	RG100-8	2xRG80-8	RG140-8
Flow [Nm3/h]	20000	65000	120000	130000	205000
Flow [Am3/h]	27000	70000	130000	140000	245000
Mass flow [kg/s]	~ 12	~ 34	~ 66	~ 68	~ 110
Power [MW]	5	14	25	28	45
Suction Pressure [bara]	1.1	1.1	1.1	1.1	1.1
Discharge Pressure [bara]	140	200	215	200	215

MAN has delivered several integral-gear compressors for CO_2 service.
References are the following:

1. Eight (8) stages CO_2 compressor RG80-8 for coal gasification plant in North Dakota, where CO_2 is used for EOR in Weyburn oilfields:

-commissioned in 1998
-pressure: from 1.1 bara to 187 bara
-massflow: ~ 35 kg/s
-impeller diameters: 800 – 115 mm
-pinion speeds: 7350 – 26600 rpm
-driver: synchronous electric motor at fixed speed (power: 14700 kw)

2. Ten (10) stages CO_2 compressors RG 56-10 in Russia (Azot Nowomoskowsk):

-commissioned in 1992
-pressure: from 1 bara to 200 bara
-massflow: ~ 13 kg/s
-impeller diameters: 550 – 90 mm

-pinion speeds: 26600 - 48000 rpm
-driver: asynchronous electric motor at fixed speed

3. Eight (8) stages CO_2 compressor RG 40-8 for Duslo A.S. in Slovakia:
-commissioned in 2002
-pressure: from 1.1 bara to 150 bara
-massflow: ~ 8 kg/s
-impeller diameters: 400 – 95 mm
-pinion speeds: 8000 – 41000 rpm
-driver: synchronous electric motor at variable speed

4. Eight (8) stages CO_2 compressor RG 56-8 for Grodno Azot in Czech Republic:
-commissioned in 2006
-pressure: from 1.1 bara to 150 bara
-massflow: ~ 16 kg/s
-impeller diameters: 500 – 95 mm
-pinion speeds: 8000 – 36000 rpm
-driver: steam turbine

2 Machinery selection for the three carbon capture technologies

2.1 Base cases definition

For each of the three main carbon capture technologies a base case was identified in order to assess the operating envelopes as a basis for the rotating machinery market survey.

The CO_2 compression schemes for the base cases are described in section B, which includes also the PFD's and main H&MB data. From these sets of data, the process parameters necessary to define the operating envelope for the machines included in the schemes were extracted and passed to Vendors.

As reported in section B, the base cases are defined as follows.

PRECOMBUSTION: Coal IGCC, based on a quench type gasification with slurry feed; net output is 750 MWe, with carbon capture rate equal to 85%.

POST COMBUSTION: Ultra Super Critical PC Boiler with a MEA unit at boiler back end; net output is 655 MWe, with carbon capture rate equal to 85%.

OXYFUEL COMBUSTION: Advanced Super Critical PF Boiler, with an auto refrigerated scheme for the CPU at boiler back end; net output = 530 MWe, with a carbon capture rate equal to 90%.

For further details reference is made to section B.

2.2 Machinery selection by Vendors

The Vendors demonstrating interest in this study have proposed their machinery selection for the specified base cases (ref. Section B).
It is generally noted that the Vendors have shown the willingness to increase the number of inter-cooling steps for either avoiding technical issues related to high discharge temperatures or improving the compressor performance. On the other hand, the Oxy-fuel Combustion and the Post Combustion baseline cases include a strong thermal integration with the Power Island, as the heat available at the CO_2 compression stages outlet is recovered into the Steam Condensate / Boiler Feed Water systems and the Steam Turbine Island, which requires relatively high discharge temperatures. For this reason, the manufacturers have been requested to propose for each of the two processes two options as far as inter-cooling arrangements are concerned:

1. Configuration as close as possible to the original specification, to allow the thermal integration as implemented in the base case configuration.
2. Configuration with an optimised number of inter-cooling steps, to best suit the selection to the standard machine frames available. In this case the resulting compressor power demand reduction is partially off-set by the consequent decrease of the waste heat available for recovery in the thermal cycle.

For the post-combustion process, the compression unit configuration is based on 4 stage selection (i.e. 3 inter-cooling stages + 1 after cooler). Vendors have been request to provide an additional option with and increased number of stages (8 overall) as part of the investigation on compression strategies to reduce compression parasitic load.

Table 2-1 provides a summary of the cases considered for machinery selection in the market survey. Reference is generally made to sections B and C for the definition of the operating envelopes.

Table 2-1 Cases summary for machinery selection

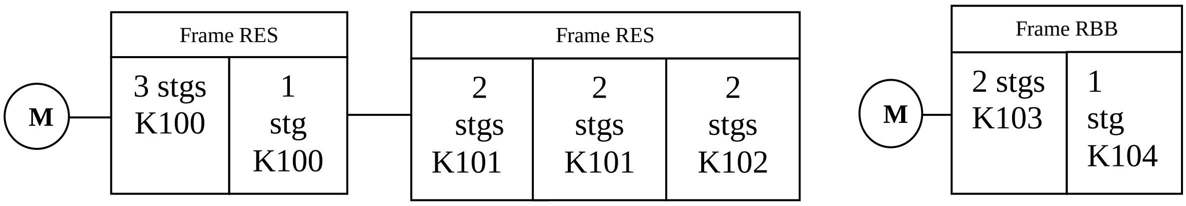
Case	Description
Pre-combustion	Operating envelope as defined by base case A0. No particular restrictions on inter-cooling steps, since compression heat is not recovered in the process.
Post-combustion inter-cooling as specified	Operating envelope as defined by base case B0. Regarding inter-cooling, Vendors are requested to keep as close as possible to the original specification.
Post-combustion optimised inter-cooling	Operating envelope as defined by base case B0. Regarding inter-cooling steps, Vendors are free to optimise the selection.
Post-combustion Increased stages	Operating envelope as defined bycase D1.
Oxy-fuel inter-cooling as specified	Operating envelope as defined by base case C0. Regarding inter-cooling, Vendors are requested to keep as close as possible to the original specification.
Oxy-fuel optimised inter-cooling	Operating envelope as defined by base case C0. Regarding inter-cooling steps, Vendors are free to optimise the selection.

The following paragraphs describe the machinery selected by the compressor manufacturers, which have demonstrated interest in supporting FWI for this study, namely Rolls-Royce and MAN Diesel & Turbo.
At the end of the section there is a brief presentation of GE proposal. GE demonstrated partial interest and did not submit all the data and information required by FWI.

2.2.1 Rolls Royce

PRE-COMBUSTION

Train n° 1 (two compressor packages running in series)



The key process and mechanical characteristics of the machines selected by Rolls Royce for this case are shown in the following table.

PRE-COMBUSTION PROCESS	
ROLLS-ROYCE	Conventional in-line compressor
Total n° of trains required	1
Composition Train 1:	
model	RES
casing n° 1	K100
Section n°1	
stages	3
flowrate (Nm^3/h)	111900
inlet temperature (°C)	-5
inlet pressure (bara)	1.2
compression ratio	3.0
speed	4800 RPM
brake power shaft (KW)	4347

poly- efficiency (%)	86.53
Section n°2	
stages	1
flowrate (Nm^3/h)	111900
inlet temperature (°C)	19
inlet pressure (bara)	3.55
compression ratio	1.4
speed	4800 RPM
brake power shaft (KW)	1282
poly- efficiency (%)	87.50
model	RES
casing n° 2	
section n° 1	K101
stages	2
flowrate (Nm^3/h)	337670
inlet temperature (°C)	6
inlet pressure (bara)	4.8
compression ratio	2.18
speed	4800 RPM
brake power shaft (KW)	9047
poly- efficiency (%)	86.02
section n° 2	K101
stages	2
flowrate (Nm^3/h)	337670
inlet temperature (°C)	19
inlet pressure (bara)	10.26
compression ratio	1.82
speed	4800 RPM
brake power shaft (KW)	6831
poly- efficiency (%)	87.20
section n° 3	K102
stages	2
flowrate (Nm^3/h)	375410
inlet temperature (°C)	19

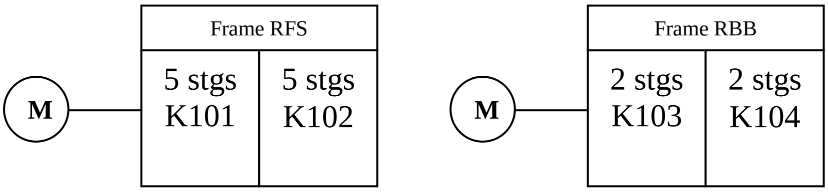
inlet pressure (bara)	18.55
compression ratio	1.83
speed	4800 RPM
brake power shaft (KW)	7229
poly- efficiency (%)	87.45
capacity control type	VFD (variable frequency drive)
motor driver	1
total brake power shaft (casing n° 1+2) (KW)	28736
model	RBB
casing n° 1	
section n° 1	K103
stages	2
flowrate (Nm^3/h)	320744
inlet temperature (°C)	24
inlet pressure (bara)	32.9
compression ratio	2.13
speed	10000 RPM
brake power shaft (KW)	7924
poly- efficiency (%)	85.18
section n° 2	K104
stages	1
flowrate (Nm^3/h)	320744
inlet temperature (°C)	40
inlet pressure (bara)	69.8
compression ratio	1.59
speed	10000 RPM
brake power shaft (KW)	3878
poly- efficiency (%)	85.5
capacity control type	VFD (variable frequency drive)
motor driver	1
total brake power shaft (sect. n° 1+2) (KW)	11802

Note: one intercooler is added on process K100 and one intercooler is added on process K101.

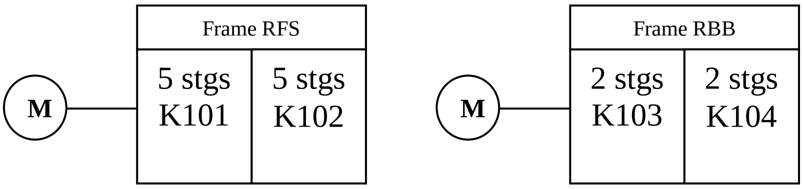
TOTAL BRAKE POWER (WHOLE PROCESS): **40538 KW**

POST-COMBUSTION – INTER-COOLING AS SPECIFIED

Train n° 1 (two compressor packages running in series)



Train n° 2 (two compressor packages running in series)



Train n°1 and train n° 2 run in parallel

The key process and mechanical characteristics of the machines selected by Rolls Royce for this case are shown in the following table.

POST-COMBUSTION PROCESS – INTER-COOLING AS SPECIFIED	
ROLLS-ROYCE	
Total n° of trains required	2
Composition train 1 / train 2:	
Model	RFS
casing n° 1	
section n° 1	K101
Stages	5
flowrate (Nm^3/h)	145243
inlet temperature (°C)	37.8
inlet pressure (bara)	1.6
compression ratio	4.32
Speed	4200 RPM
brake power shaft (KW)	9366

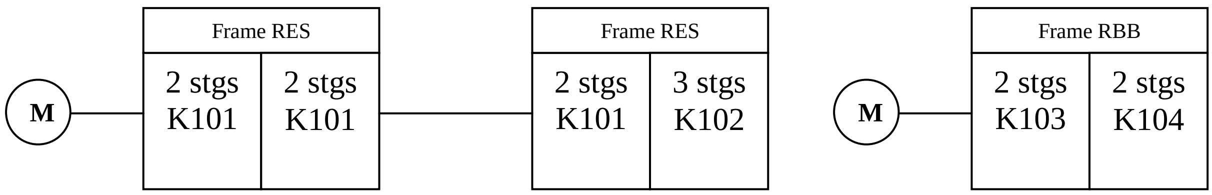
poly- efficiency (%)	87.44
turndown (%)	31.7 (estimated)
poly- efficiency at min flow (%)	84.8 (estimated)
section n° 2	K102
Stages	5
flowrate (Nm^3/h)	155329
inlet temperature (°C)	19
inlet pressure (bara)	6.6
compression ratio	5.15
Speed	4200 RPM
brake power shaft (KW)	10586
poly-efficiency (%)	84.7
turndown (%)	30.7 (estimated)
poly- efficiency at min flow (%)	81.8 (estimated)
capacity control type	VFD (variable frequency drive)
motor driver	1
total brake power shaft (sect. n° 1+2) (KW)	19952
Model	RBB
casing n° 1	
section n° 1	K103
stages	2
flowrate (Nm^3/h)	139303
inlet temperature (°C)	24
inlet pressure (bara)	32.7
compression ratio	2.14
speed	10000 RPM
brake power shaft (KW)	3433
poly-efficiency (%)	84.4
turndown (%)	30.8 (estimated)
poly- efficiency at min flow (%)	81 (estimated)
section n° 2	K104
stages	2
flowrate (Nm^3/h)	139303
inlet temperature (°C)	40
inlet pressure (bara)	69.6

compression ratio	1.6
speed	10000 RPM
brake power shaft (KW)	1844
poly-efficiency (%)	74.5
turndown (%)	30.6 (estimated)
poly- efficiency at min flow (%)	71.8 (estimated)
capacity control type	VFD (variable frequency drive)
motor driver	1
total brake power shaft (sect. n° 1+2) (KW)	5277
Total brake power shaft (each train) (KW)	25229

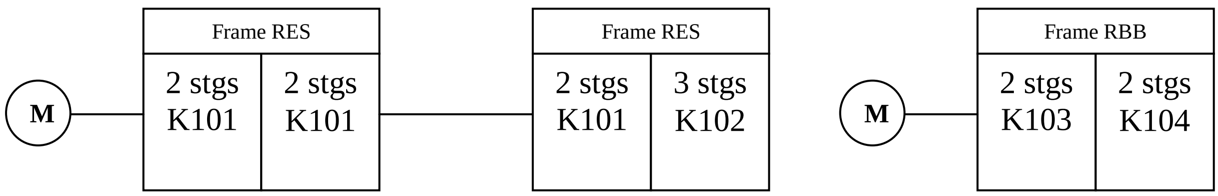
TOTAL BRAKE POWER (WHOLE PROCESS): **50458 KW**

POST-COMBUSTION – OPTIMISED INTER-COOLING

Train n° 1 (two compressor packages running in series)



Train n° 2 (two compressor packages running in series)



Train n°1 and train n° 2 run in parallel

The key process and mechanical characteristics of the machines selected by Rolls Royce for this case are shown in the following table.

POST-COMBUSTION PROCESS – OPTIMISED INTER-COOLING

ROLLS-ROYCE	
Total n° of trains required	2
Composition train 1 / train 2:	
model	RES
casing n° 1	
section n° 1	K101
stages	2
flowrate (Nm^3/h)	152570
inlet temperature (°C)	37.8
inlet pressure (bara)	1.6
compression ratio	2.35
speed	4900 RPM
brake power shaft (KW)	5297
poly- efficiency (%)	84.26
section n° 2	K101
stages	2
flowrate (Nm^3/h)	143990
inlet temperature (°C)	19
inlet pressure (bara)	3.71
compression ratio	2.1
speed	4900 RPM
brake power shaft (KW)	3812
poly- efficiency (%)	87.45
capacity control type	VFD (variable frequency drive)
motor driver	1
total brake power shaft (sect. n° 1+2) (KW)	9109
model	RES
casing n° 1	
section n° 1	K101
stages	2
flowrate (Nm^3/h)	143410
inlet temperature (°C)	19
inlet pressure (bara)	7.68

compression ratio	2.08
speed	4900 RPM
brake power shaft (KW)	3682
poly- efficiency (%)	87
section n° 2	K102
stages	3
flowrate (Nm^3/h)	163240
inlet temperature (°C)	19
inlet pressure (bara)	15.8
compression ratio	2.15
speed	4900 RPM
brake power shaft (KW)	4138
poly-efficiency (%)	86.80
capacity control type	VFD (variable frequency drive)
motor driver	1
total brake power shaft (sect. n° 1+2) (KW)	7820
model	RBB
casing n° 1	
section n° 1	K103
stages	2
flowrate (Nm^3/h)	139303
inlet temperature (°C)	24
inlet pressure (bara)	32.7
compression ratio	2.14
speed	10000 RPM
brake power shaft (KW)	3433
poly-efficiency (%)	84.4
section n° 2	K104
stages	2
flowrate (Nm^3/h)	139303
inlet temperature (°C)	40
inlet pressure (bara)	69.6
compression ratio	1.6

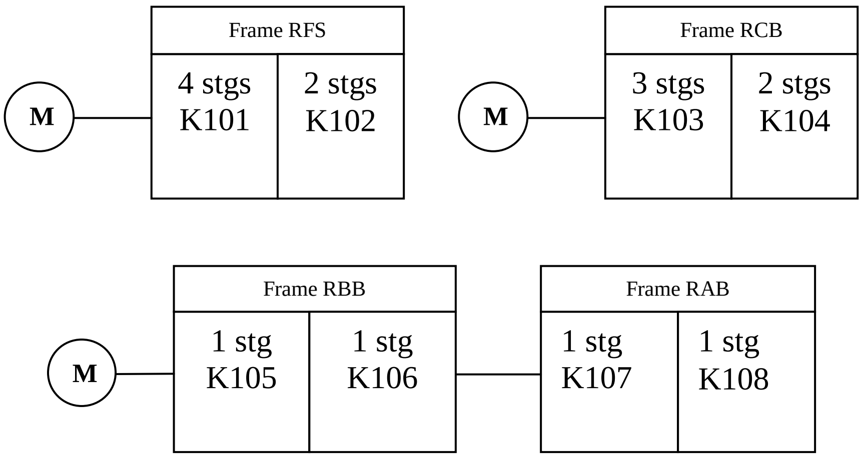
speed	10000 RPM
brake power shaft (KW)	1844
poly-efficiency (%)	74.5
capacity control type	VFD (variable frequency drive)
motor driver	1
total brake power shaft (sect. n° 1+2) (KW)	5277
Total brake power shaft (each train) (KW)	22206

Note: two intercoolers are added on process K101.

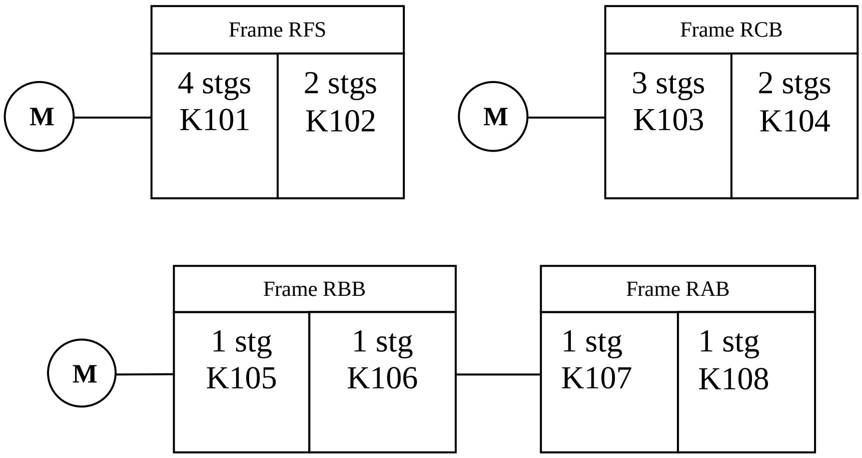
TOTAL BRAKE POWER (WHOLE PROCESS): **44412 KW**

POST-COMBUSTION - INCREASED STAGES

Train n° 1 (three compressor packages running in series)



Train n° 2 (three compressor packages running in series)



Train n°1 and train n° 2 run in parallel

The key process and mechanical characteristics of the machines selected by Rolls Royce for this case are shown in the following table.

POST-COMBUSTION PROCESS – INCREASED STAGES	
ROLLS-ROYCE	
Composition train 1 / train 2:	2
Ref. Train 1:	
model	RFS
casing n° 1	
section n° 1	K101
stages	4
flowrate (Nm^3/h)	145243
inlet temperature (°C)	37.8
inlet pressure (bara)	1.6
compression ratio	3.46
speed	4200 RPM
brake power shaft (KW)	7723
poly-efficiency (%)	87.61
turndown (%)	31.7 (estimated)
poly- efficiency at min flow (%)	85 (estimated)
section n° 2	K102
stages	2

flowrate (Nm^3/h)	139864
inlet temperature (°C)	19
inlet pressure (bara)	5.4
compression ratio	1.85
speed	4200 RPM
brake power shaft (KW)	3185
poly-efficiency (%)	86.44
turndown (%)	31 (estimated)
poly- efficiency at min flow (%)	83.8 (estimated)
capacity control type	VFD (variable frequency drive)
motor driver	1
total brake power shaft (sect. n° 1+2) (KW)	10908
model	RCB
casing n° 1	
section n° 1	K103
stages	3
flowrate (Nm^3/h)	139639
inlet temperature (°C)	19
inlet pressure (bara)	9.8
compression ratio	2.24
speed	6000 RPM
brake power shaft (KW)	4172
poly-efficiency (%)	86.45
turndown (%)	31.5 (estimated)
poly- efficiency at min flow (%)	83.8 (estimated)
section n° 2	K104
stages	2
flowrate (Nm^3/h)	154881
inlet temperature (°C)	25
inlet pressure (bara)	21.8
compression ratio	1.56
speed	6000 RPM
brake power shaft (KW)	2326
poly-efficiency (%)	85.77
turndown (%)	31.5 (estimated)
poly- efficiency at min flow (%)	83 (estimated)
capacity control type	VFD (variable frequency drive)
motor driver	1
total brake power shaft (sect. n° 1+2) (KW)	6498

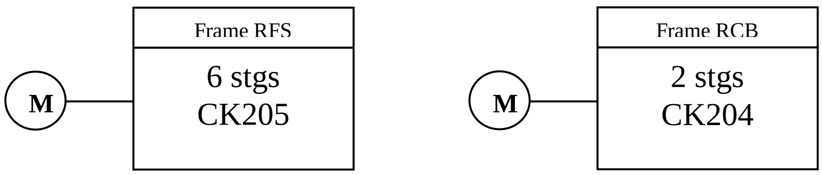
model	RBB
casing n° 1	
section n° 1	K105
stages	1
flowrate (Nm^3/h)	139079
inlet temperature (°C)	24
inlet pressure (bara)	32.9
compression ratio	1.55
speed	10000 RPM
brake power shaft (KW)	1930
poly-efficiency (%)	85.61
turndown (%)	31.3 (estimated)
poly- efficiency at min flow (%)	82.8 (estimated)
section n° 2	K106
stages	1
flowrate (Nm^3/h)	139079
inlet temperature (°C)	19
inlet pressure (bara)	50.9
compression ratio	1.38
speed	10000 RPM
brake power shaft (KW)	1075
poly-efficiency (%)	85.86
turndown (%)	30.7 (estimated)
poly- efficiency at min flow (%)	82.8 (estimated)
model	RAB
casing n° 2	
section n° 1	K107
stages	1
flowrate (Nm^3/h)	139079
inlet temperature (°C)	40
inlet pressure (bara)	69.9
compression ratio	1.3
speed	10000 RPM
brake power shaft (KW)	896
poly-efficiency (%)	86.04
turndown (%)	31.5 (estimated)
poly- efficiency at min flow (%)	79 (estimated)

section n° 2	K108
stages	1
flowrate (Nm^3/h)	139079
inlet temperature (°C)	40
inlet pressure (bara)	90.4
compression ratio	1.23
speed	10000 RPM
brake power shaft (KW)	440
poly-efficiency (%)	78.91
turndown (%)	34.5 (estimated)
poly- efficiency at min flow (%)	74 (estimated)
capacity control type	VFD (variable frequency drive)
motor driver	1
total brake power shaft (casing n° 1+2) (KW)	4341
Total brake power shaft (each train) (KW)	21747

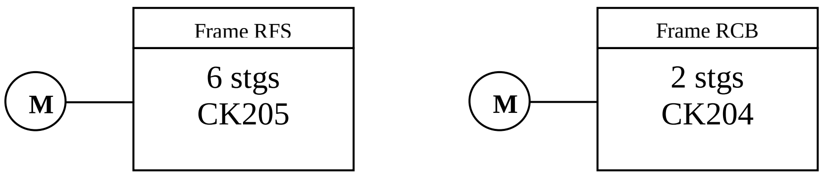
TOTAL BRAKE POWER (WHOLE PROCESS): **43494 KW**

OXY-FUEL - INTER-COOLING AS SPECIFIED

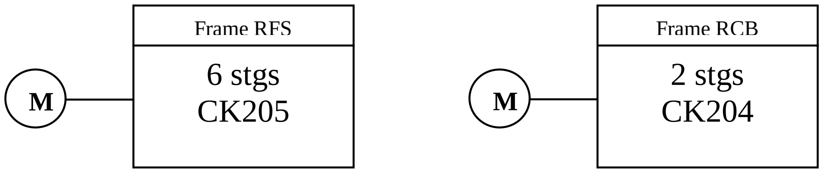
Train n° 1 (two compressor packages running in series)



Train n° 2 (two compressor packages running in series)

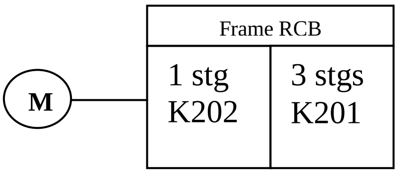


Train n° 3 (two compressor packages running in series)



Train n°1, n° 2 and train n° 3 run in parallel

Train n° 4 (running in series to the previous three ones)



The key process and mechanical characteristics of the machines selected by Rolls Royce for this case are shown in the following table.

OXY-FUEL PROCESS	
ROLLS-ROYCE	
Total n° of trains required	4
Composition train 1 / train 2 / train 3:	
model	RFS
casing n° 1	CK205
stages	6
flowrate (Nm^3/h)	111173
inlet temperature (°C)	12
inlet pressure (bara)	1
compression ratio	14.85
speed	4162 RPM
brake power shaft (KW)	14480
poly-efficiency (%)	87.71
turndown (%)	30.3 (estimated)
poly- efficiency at min flow (%)	85 (estimated)
capacity control type	VFD (variable frequency drive)
motor driver	1

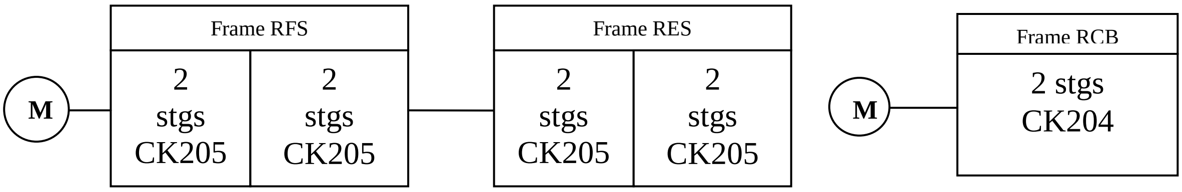
model	RCB 19.0
casing n° 1	CK204
stages	2
flowrate (Nm^3/h)	121857
inlet temperature (°C)	19
inlet pressure (bara)	14.4
compression ratio	2.16
speed	8565 RPM
brake power shaft (KW)	3559
poly-efficiency (%)	84.7
turndown (%)	33 (estimated)
poly- efficiency at min flow (%)	82 (estimated)
capacity control type	VFD (variable frequency drive)
motor driver	1
Total brake power shaft (each train) (KW)	18039
Composition train 4:	
model	RCB 19.0
casing n° 1	
section n°1	K202
stages	1
flowrate (Nm^3/h)	119780
inlet temperature (°C)	7.5
inlet pressure (bara)	9.3
compression ratio	2.01
speed	10427 RPM
brake power shaft (KW)	2998
poly-efficiency (%)	84.74
turndown (%)	25 (estimated)
poly- efficiency at min flow (%)	82 (estimated)
section n°2	K201
stages	3
flowrate (Nm^3/h)	236692
inlet temperature (°C)	13.3
inlet pressure (bara)	18.6
compression ratio	6
speed	10427 RPM
brake power shaft (KW)	16982
poly-efficiency	84.26

turndown (%)	32 (estimated)
poly- efficiency at min flow (%)	81 (estimated)
capacity control type	VFD (variable frequency drive)
motor driver	1
total brake power shaft (sect 1+2) (KW)	19980

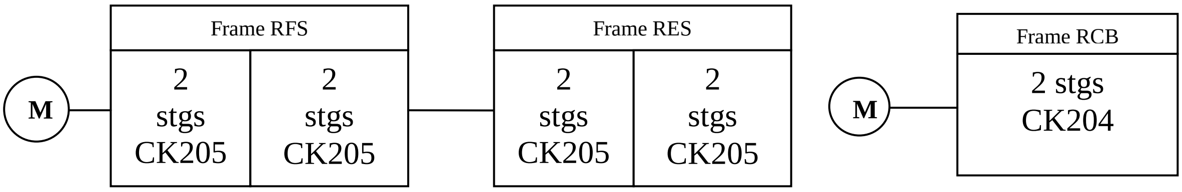
TOTAL BRAKE POWER (WHOLE PROCESS): **74097 KW**

OXY-FUEL – OPTIMISED INTER-COOLING AS SPECIFIED

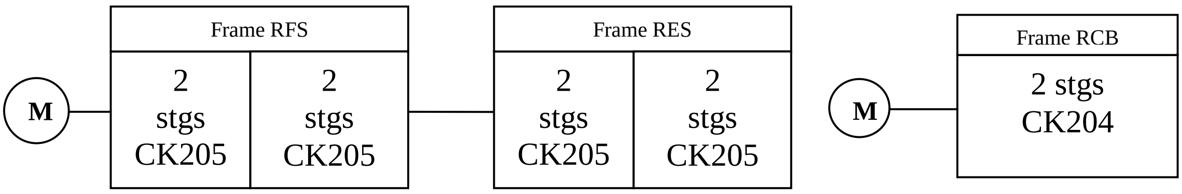
Train n° 1 (two compressor packages running in series)



Train n° 2 (two compressor packages running in series)

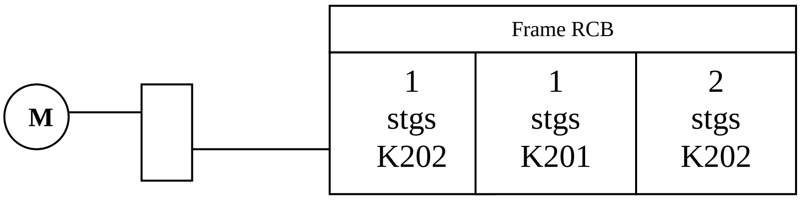


Train n° 3 (two compressor packages running in series)



Train n°1, n° 2 and train n° 3 run in parallel

Train n° 4 (running in series to the previous three ones)



The key process and mechanical characteristics of the machines selected by Rolls Royce for this case are shown in the following table.

OXY-FUEL PROCESS – OPTIMISED INTER-COOLING	
ROLLS-ROYCE	
Total n° of trains required	4
Composition train 1 / train 2 / train 3:	
model	RFS
casing n° 1	CK205
section n°1	
stages	2
flowrate (Nm^3/h)	117060
inlet temperature (°C)	12
inlet pressure (bara)	1
compression ratio	1.99
speed	4151 RPM
brake power shaft (KW)	2866
poly-efficiency (%)	86.93
section n° 2	
stages	2
flowrate (Nm^3/h)	116210
inlet temperature (°C)	19
inlet pressure (bara)	1.91
compression ratio	1.99
speed	4151 RPM
brake power shaft (KW)	2840

poly-efficiency (%)	87.94
casing n° 2	CK205
section n° 1	
stages	2
flowrate (Nm^3/h)	115370
inlet temperature (°C)	19
inlet pressure (bara)	3.7
compression ratio	2.01
speed	4151 RPM
brake power shaft (KW)	2888
poly-efficiency (%)	87.94
section n° 2	
stages	2
flowrate (Nm^3/h)	114930
inlet temperature (°C)	19
inlet pressure (bara)	7.37
compression ratio	2.03
speed	4151 RPM
brake power shaft (KW)	2921
poly-efficiency (%)	85.78
capacity control type	VFD (variable frequency drive)
motor driver	1
Brake power shaft (CK205 train) (KW)	11515
model	RCB 19.0
casing n° 1	CK204
stages	2
flowrate (Nm^3/h)	121857
inlet temperature (°C)	19
inlet pressure (bara)	14.4
compression ratio	2.16
speed	8565 RPM
brake power shaft (KW)	3559
poly-efficiency (%)	84.7
capacity control type	VFD (variable frequency drive)

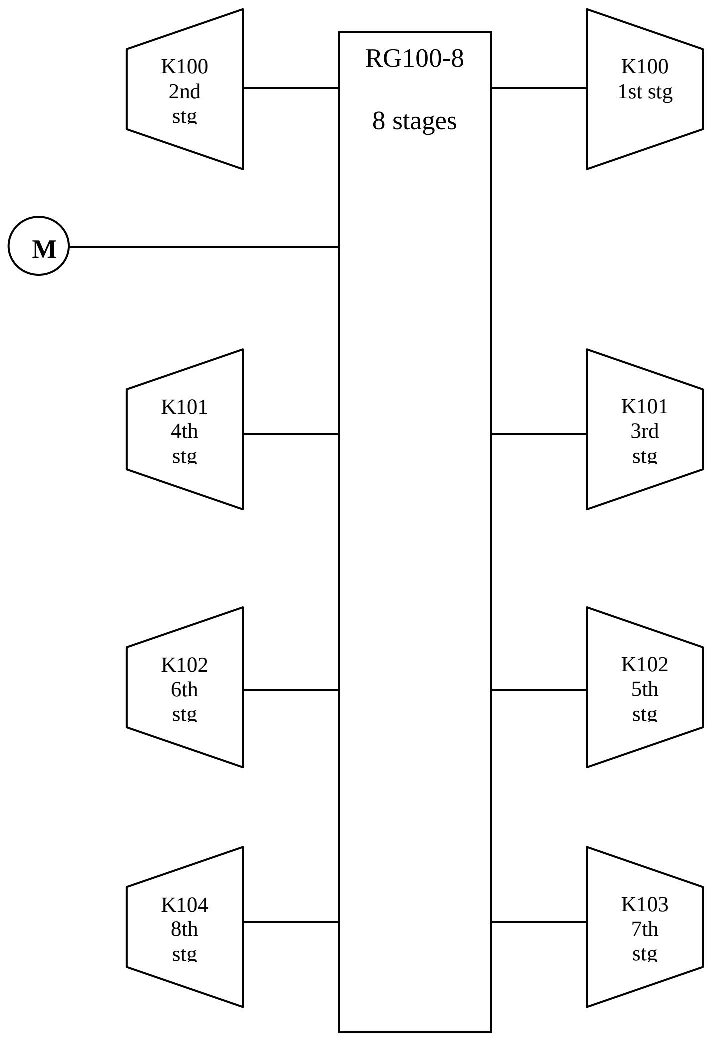
motor driver	1
Total brake power shaft (each train) (KW)	15074
Composition train 4:	
model	RCB
casing n° 1	
section n° 1	K202
stages	1
flowrate (Nm^3/h)	126300
inlet temperature (°C)	7.5
inlet pressure (bara)	9.3
compression ratio	2.01
speed	10181 RPM
brake power shaft (KW)	3040
poly-efficiency (%)	84.01
section n° 2	K201
stages	1
flowrate (Nm^3/h)	248460
inlet temperature (°C)	13.3
inlet pressure (bara)	18.55
compression ratio	1.75
speed	10181 RPM
brake power shaft (KW)	4453
poly-efficiency	84.61
section n° 3	K201
stages	1
flowrate (Nm^3/h)	248460
inlet temperature (°C)	19
inlet pressure (bara)	32.35
compression ratio	3.44
speed	10181 RPM
brake power shaft (KW)	10149
poly-efficiency	83.88

capacity control type	VFD (variable frequency drive)
motor driver	1
Brake power shaft (train 4) (KW)	17642

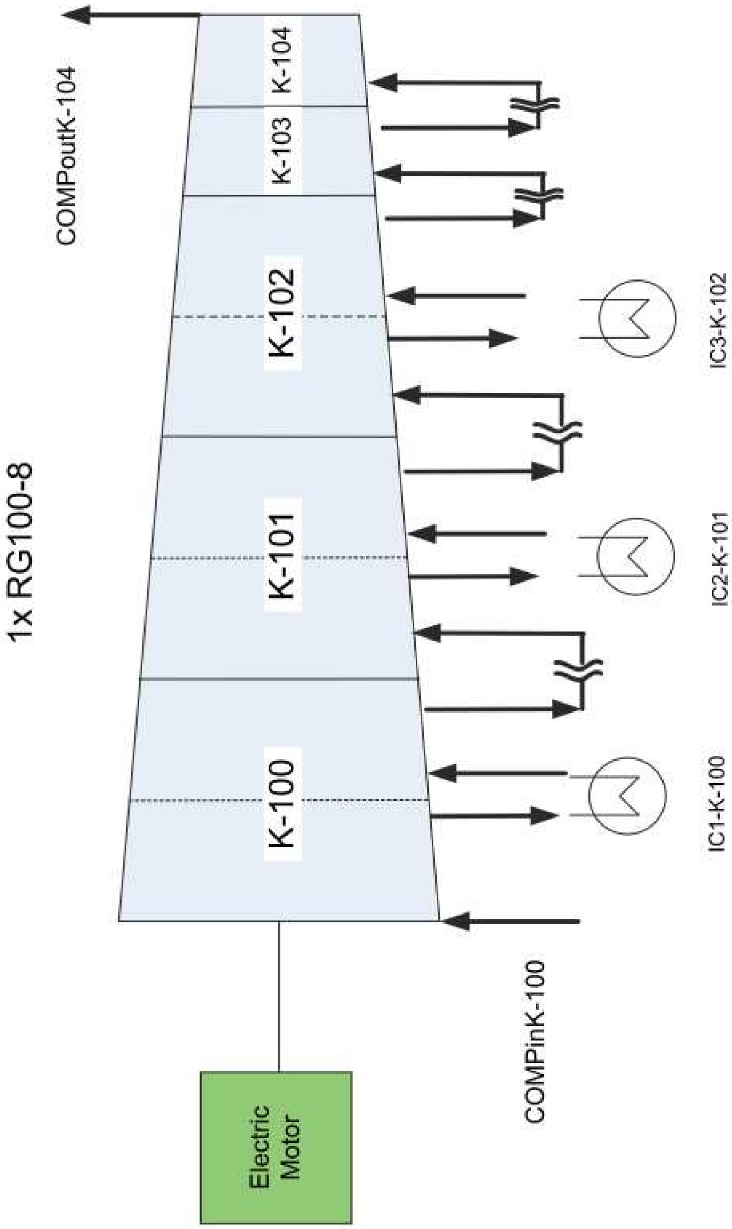
TOTAL BRAKE POWER (WHOLE PROCESS): **62864 KW**

2.2.2 MAN Diesel & Turbo

PRE-COMBUSTION



PRE-COMBUSTION – Gen. Overview



Note: only the intercoolers added by MANTURBO with respect to baseline configuration are indicated in the above overview.

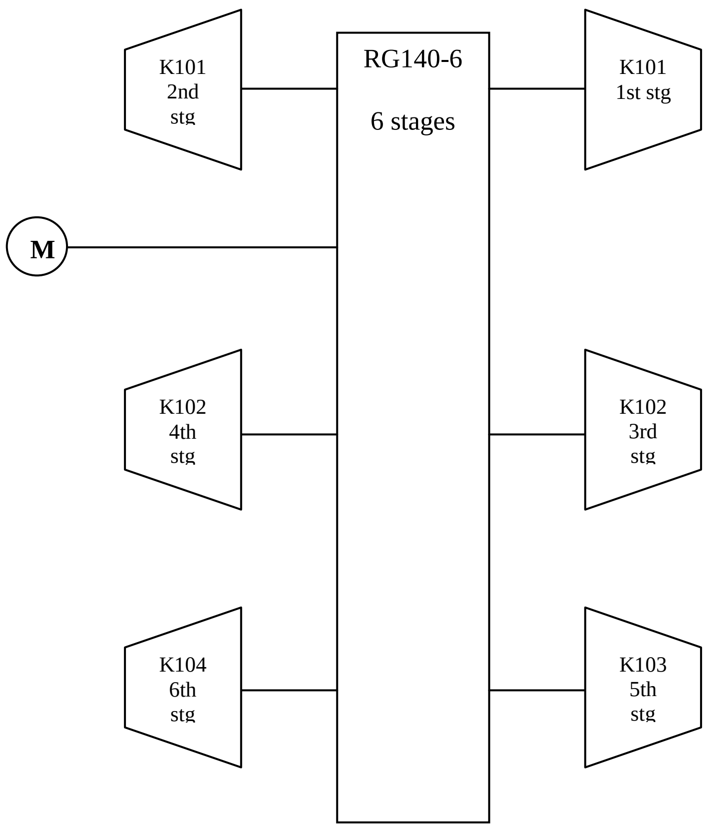
The key process and mechanical characteristics of the machines selected by MAN Diesel & Turbo are shown in the following tables.

PRE-COMBUSTION	
MAN TURBO	Integrally gear motor driven
Total n° of trains required	1
Ref. Train 1:	K-100 / K-101 / K-102 / K-103 / K-104
model	RG100-8
stages	5
flowrate K-100 (Nm^3/h)	105570
flowrate K-101 (Nm^3/h)	318604
flowrate K-102 (Nm^3/h)	354178
flowrate K-103 (Nm^3/h)	318643
flowrate K-104 (Nm^3/h)	318643
inlet temperature K-100 (°C)	-5
inlet temperature K-101 (°C)	6.1
inlet temperature K-102 (°C)	19.1
inlet temperature K-103 (°C)	24
inlet temperature K-104 (°C)	53
inlet pressure K-100 (bara)	1.2
inlet pressure K-101 (bara)	4.8
inlet pressure K-102 (bara)	11.8
inlet pressure K-103 (bara)	32.9
inlet pressure K-104 (bara)	65.3
compression ratio K-100	4.17
compression ratio K-101	2.5
compression ratio K-102	2.88
compression ratio K-103	1.99
compression ratio K-104	1.70
Speed shaft 1 K-100 (rpm)	5655
Speed shaft 2 K-101 (rpm)	4351
Speed shaft 3 K-102 (rpm)	7140
Speed shaft 4 K-103/K104 (rpm)	10563
brake power K-100 (KW)	6056
brake power K-101 (KW)	10809
brake power K-102 (KW)	13619
brake power K-103+K104 (KW)	13472
poly-efficiency 1st stage (%)	85.6
poly-efficiency 2nd stage (%)	87.8

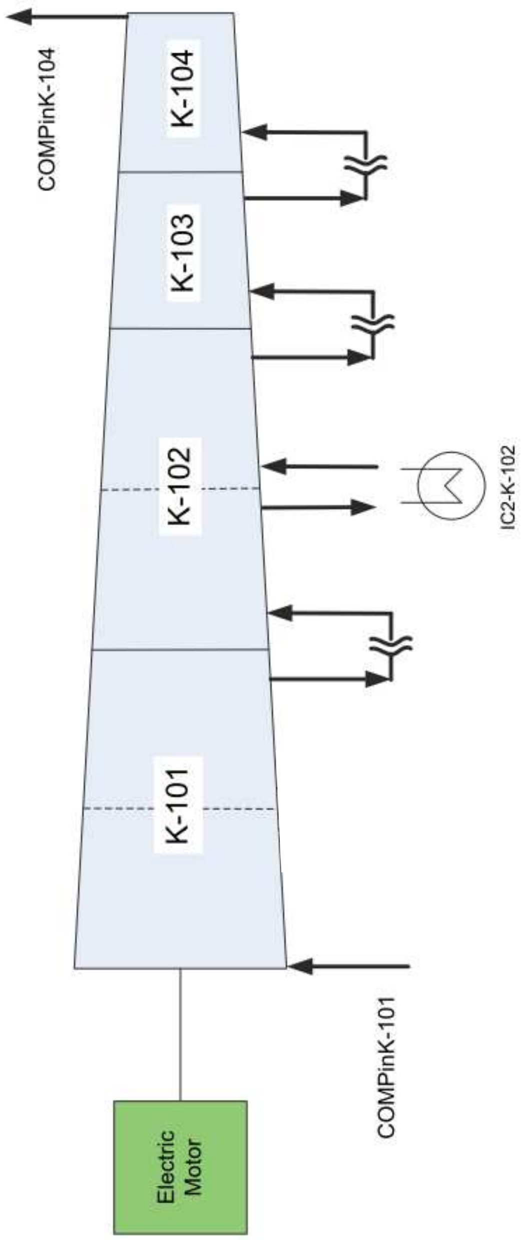
poly-efficiency 3rd stage (%)	86.9
poly-efficiency 4th - 5th stage (%)	83.4
turndown 1st stage (%)	33.6
turndown 2nd stage (%)	33.4
turndown 3rd stage (%)	33.6
turndown 4^{th}-5th stage (%)	35.7
capacity control type	IGV
poly-efficiency 1st stage at min flow (%)	82.5 (estimated)
poly-efficiency 2nd stage at min flow (%)	84 (estimated)
poly-efficiency 3rd stage at min flow (%)	84 (estimated)
poly-efficiency 4th/ 5^{th} stage at min flow (%)	80 (estimated)
Intercoolers included by vendor	3 (on 1st-2nd-3rd stages)

TOTAL BRAKE POWER (WHOLE PROCESS): **43956 KW**

POST-COMBUSTION – INTERCOOLING AS SPECIFIED



POST-COMBUSTION – INTERCOOLING AS SPECIFIED, Gen. Overview



Note: only the intercoolers added by MANTURBO with respect to baseline configuration are indicated in the above overview. In addition the intercooler between process K-103 and K-104 is removed.

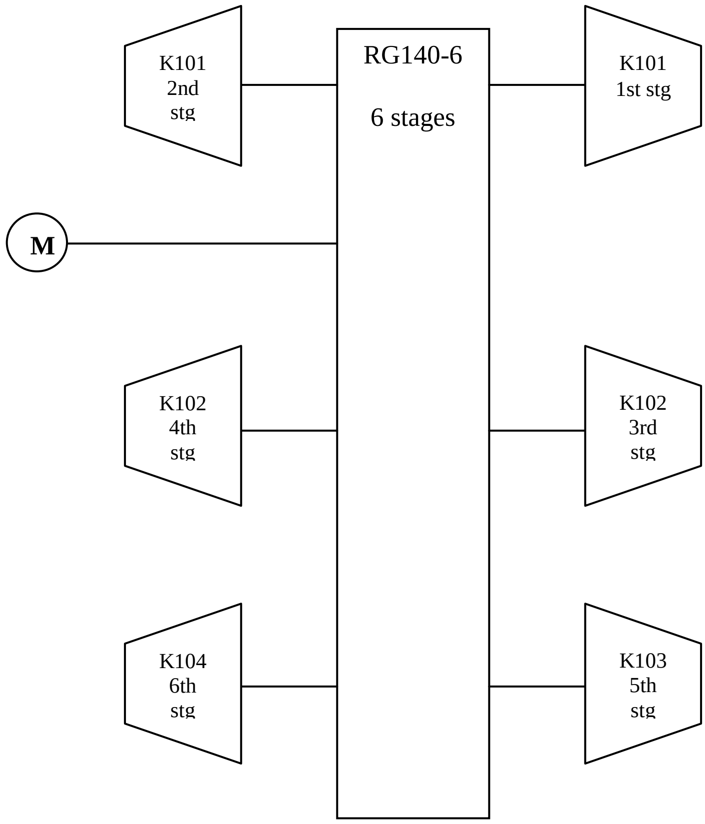
The key process and mechanical characteristics of the machines selected by MAN Diesel & Turbo are shown in the following tables.

POST-COMBUSTION PROCESS – INTERCOOLING AS SPECIFIED	
MAN TURBO	Integrally gear motor driven
Total n° of trains required	1
Ref. Train 1:	K-101 / K-102 / K-103 / K-104
model	RG140-6
stages	6
flowrate K-101 (Nm^3/h)	276677
flowrate K-102 (Nm^3/h)	307506
flowrate K-103 (Nm^3/h)	276321
flowrate K-104 (Nm^3/h)	276321
inlet temperature K-101 (°C)	37.8
inlet temperature K-102 (°C)	19
inlet temperature K-103 (°C)	24
inlet temperature K-104 (°C)	89.8 (note 1)
inlet pressure K-101 (bara)	1.55
inlet pressure K-102 (bara)	6.76
inlet pressure K-103 (bara)	32.85
inlet pressure K-104 (bara)	68.89
compression ratio K-101	4.50
compression ratio K-102	5.0
compression ratio K-103	2.10
compression ratio K-104	1.61
brake power K101 (KW)	20706
brake power K102 (KW)	19655
brake power K103+ K104 (KW)	12796
Speed shaft 1 K101 (rpm)	4080
Speed shaft 2 K102 (rpm)	6969
Speed shaft 3 K103/K104 (rpm)	10496
poly-efficiency 1^{st}- 2nd stage (%)	86.1
poly-efficiency 3^{rd} - 4th stage (%)	85
poly-efficiency 5^{th} - 6th stage (%)	81.9
capacity control type	IGV
Intercoolers included by vendor	1 (3rd/4th stages)

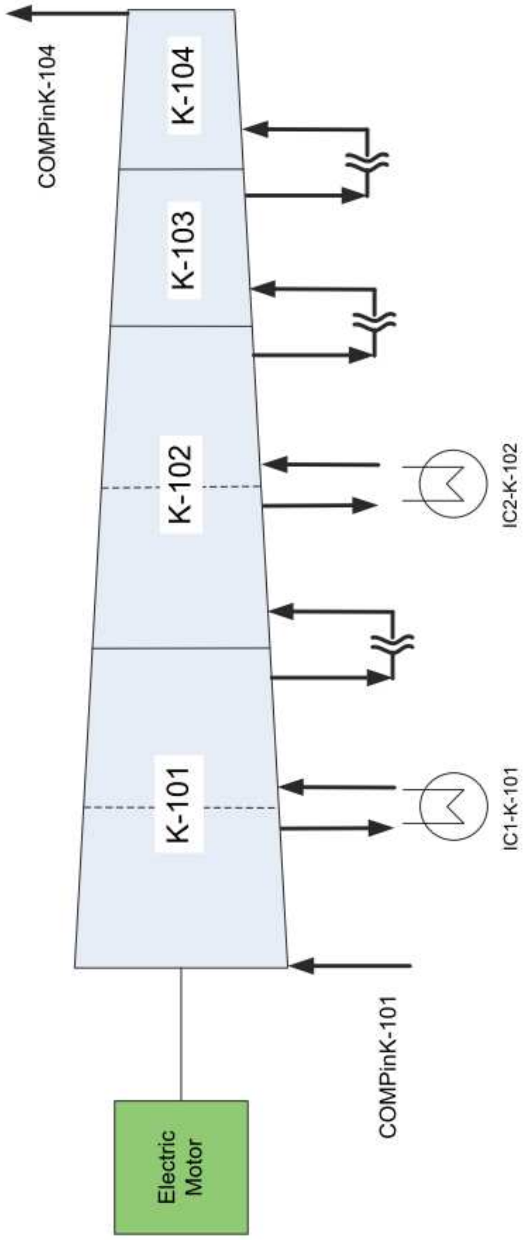
TOTAL BRAKE POWER (WHOLE PROCESS): **53157 KW**

Note 1: the intercooler between process K103 and K104 is deleted so that the discharge temperature of K103 coincides with suction temperature of K104. The process K-104 will deliver the gas at 138°C.

POST-COMBUSTION – OPTIMISED INTER-COOLING



POSTCOMBUSTION – OPTIMISED INTERCOOLING, Gen. Overview



Note: only the intercoolers added by MANTURBO with respect to baseline configuration are indicated in the above overview.

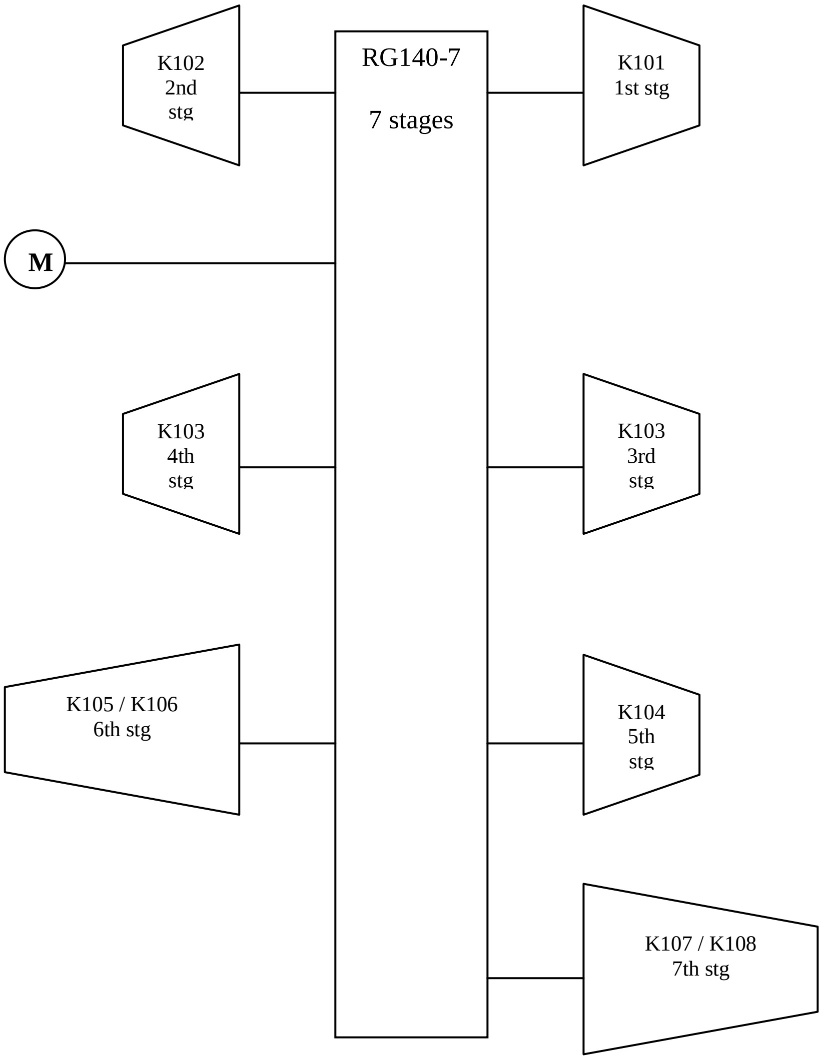
The key process and mechanical characteristics of the machines selected by MAN Diesel & Turbo are shown in the following tables.

POST-COMBUSTION - OPTIMISED INTERCOOLING	
MAN TURBO	Integrally gear motor driven
Total n° of trains required	1
Ref. Train 1:	K-101 / K-102 / K-103 / K-104
Model	RG140-6
Stages	6
flowrate K-101 (Nm^3/h)	276677
flowrate K-102 (Nm^3/h)	307506
flowrate K-103 (Nm^3/h)	276321
flowrate K-104 (Nm^3/h)	276321
inlet temperature K-101 (°C)	37.8
inlet temperature K-102 (°C)	19
inlet temperature K-103 (°C)	24
inlet temperature K-104 (°C)	48
inlet pressure K-101 (bara)	1.55
inlet pressure K-102 (bara)	6.7
inlet pressure K-103 (bara)	32.8
inlet pressure K-104 (bara)	64.2
compression ratio K-101	4.50
compression ratio K-102	5.0
compression ratio K-103	1.96
compression ratio K-104	1.73
brake power K101 (KW)	18052
brake power K102 (KW)	19655
brake power K103+ K104 (KW)	11102
Speed shaft 1 K101 (rpm)	4058
Speed shaft 2 K102 (rpm)	6969
Speed shaft 3 K103/K104 (rpm)	12026
poly-efficiency 1st- 2nd stage (%)	86.2
poly-efficiency 3rd - 4th stage (%)	85
poly-efficiency 5th - 6th stage (%)	84.2
turndown 1st - 2nd stage (%)	33.8
turndown 3rd - 4th stage (%)	33.6
turndown 5th - 6th stage (%)	35.6
poly-efficiency 1st - 2nd stage at min flow (%)	82.5 (estimated)

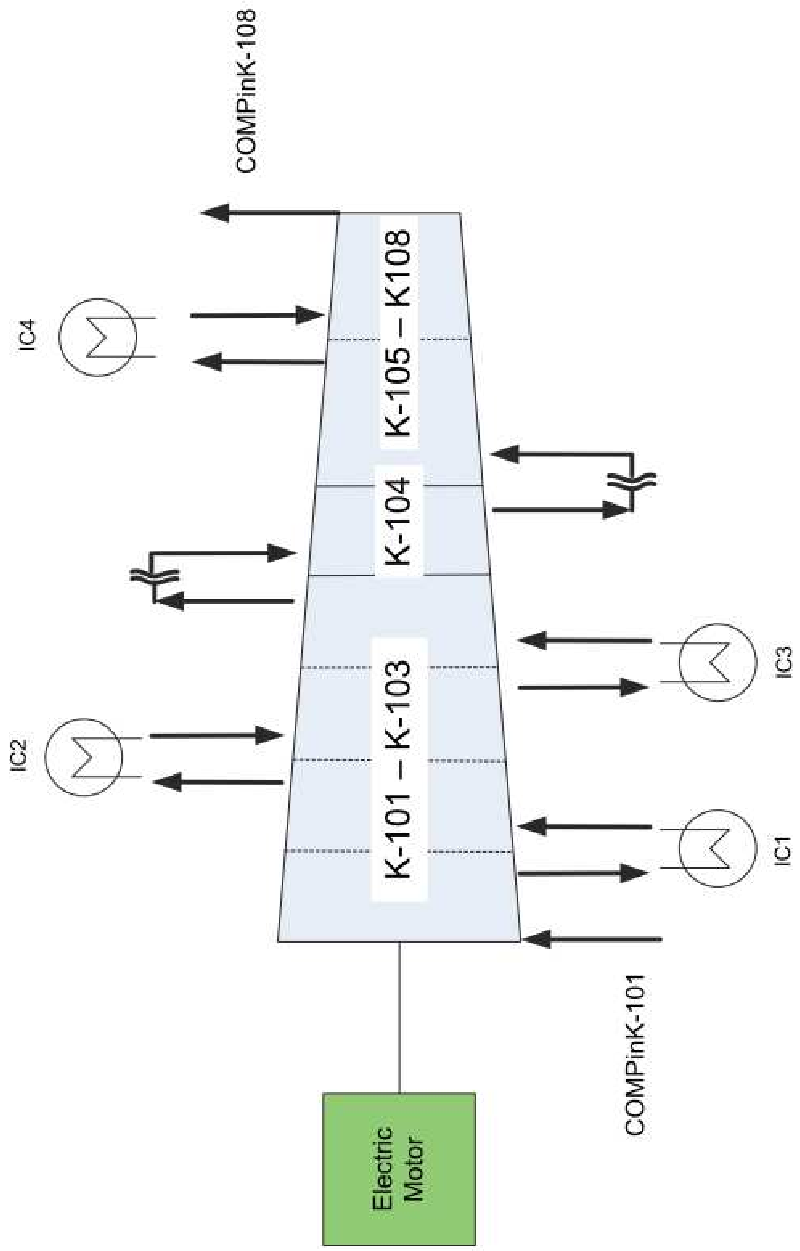
poly-efficiency 3rd - 4th stage at min flow (%)	81 (estimated)
poly-efficiency 5th - 6th stage at min flow (%)	79 (estimated)
capacity control type	IGV
Intercoolers included by vendor	2 (1st/2nd, 3rd/4th stages)

TOTAL BRAKE POWER (WHOLE PROCESS): **48809 KW**

POST-COMBUSTION – INCREASED STAGES



POSTCOMBUSTION – INCREASED STAGES, Gen. Overview



Note: MAN Diesel & Turbo have re-arranged stages split. Only the inter-coolers placed at different inter-stage pressure with respect to baseline configuration are shown in the overview. Total Number of inter-coolers is 6.

The key process and mechanical characteristics of the machines selected by MAN Diesel & Turbo are shown in the following tables.

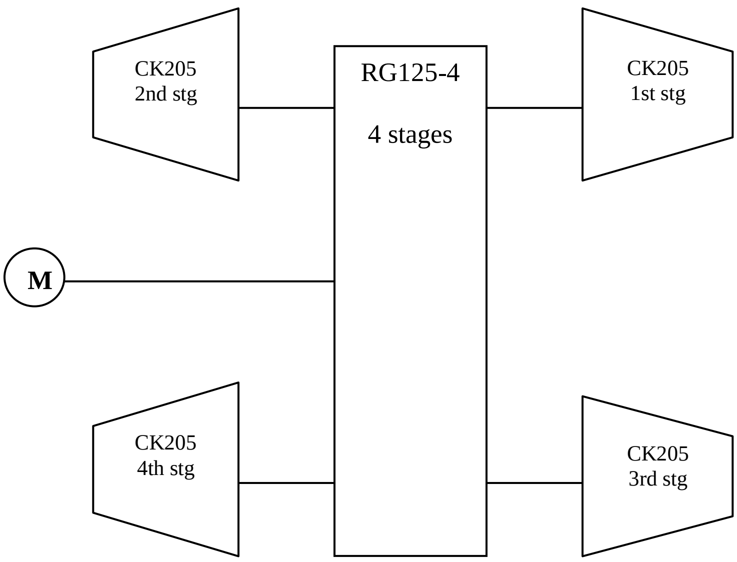
POST-COMBUSTION PROCESS– INCREASED STAGES	
MAN TURBO	Integrally gear motor driven
Total n° of trains required	1
Ref. Train 1:	K-101/K-102/K-103/K-104/K105/K106/K107/K108
Model	RG140-7
Stages	7
brake power K101+K102+K103 (KW)	29837
brake power K104 (KW)	4831
brake power shaft K105+K106+K107+K108 (KW)	10979
flowrate K-101 (Nm^3/h)	276662
flowrate K-102 (Nm^3/h)	276662
flowrate K-103 (Nm^3/h)	276662
flowrate K-104 (Nm^3/h)	307465
flowrate K-105/K-106 (Nm^3/h)	276512
flowrate K-107/K-108 (Nm^3/h)	276512
inlet temperature K-101 (°C)	37.8
inlet temperature K-102 (°C)	19
inlet temperature K-103 (°C)	19
inlet temperature K-104 (°C)	25
inlet temperature K-105/K-106 (°C)	24
inlet temperature K-107/K-108 (°C)	40
inlet pressure K-101 (bara)	1.6
inlet pressure K-102 (bara)	5.4
inlet pressure K-103 (bara)	9.8
inlet pressure K-104 (bara)	21.8
inlet pressure K-105/K-106 (bara)	32.9
inlet pressure K-107/K-108 (bara)	56.3
compression ratio K-101	3.46
compression ratio K-102	1.85
compression ratio K-103	2.24
compression ratio K-104	1.56
compression ratio K-105/K-106	1.71
compression ratio K-107/K-108	1.97

Speed shaft 1 K101/K102 (rpm)	3940
Speed shaft 2 K103 (rpm)	6896
Speed shaft 3 K104/K105/K106 (rpm)	8545
Speed shaft 4 K107/K108 (rpm)	18438
poly-efficiency 1st- 2nd -3rd - 4th stage (%)	86.3
poly-efficiency 5th (%)	86.4
poly-efficiency 6th - 7th stage (%)	84.2
turndown 1st - 2nd -3rd - 4th stage (%)	33
turndown 5th (%)	35
turndown 6th - 7th stage (%)	33
poly-efficiency 1st - 2nd -3rd - 4th stage at min flow (%)	82 (estimated)
poly-efficiency 5th at min flow (%)	82 (estimated)
poly-efficiency 6th - 7th stage at min flow (%)	79 (estimated)
capacity control type	IGV
Intercoolers included by vendor	4 (1st/2nd, 2nd/3rd, 3rd/4th, 6th/7th stages)

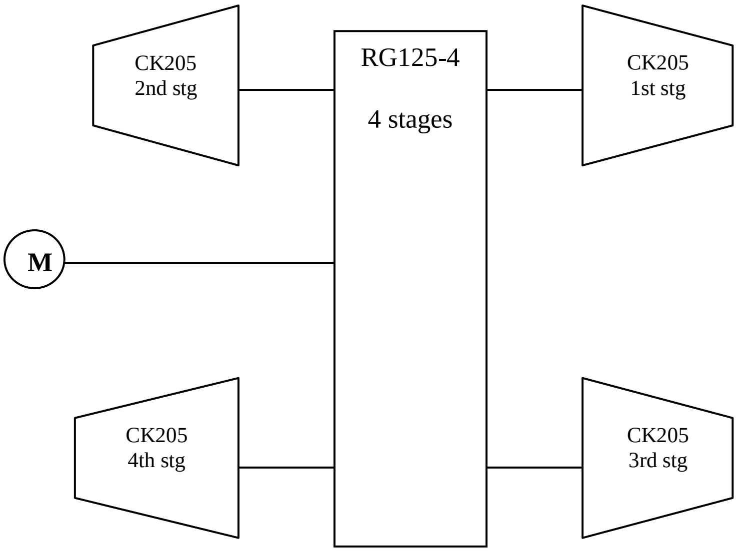
TOTAL BRAKE POWER (WHOLE PROCESS): **45647 KW**

OXY-FUEL – INTER-COOLING AS SPECIFIED

Train n° 1

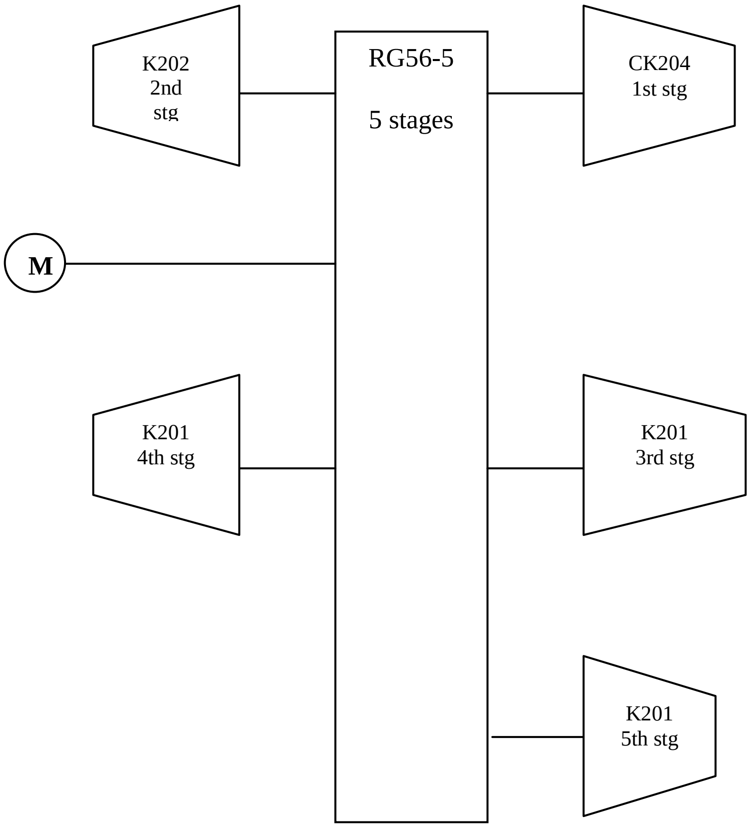


Train n° 2

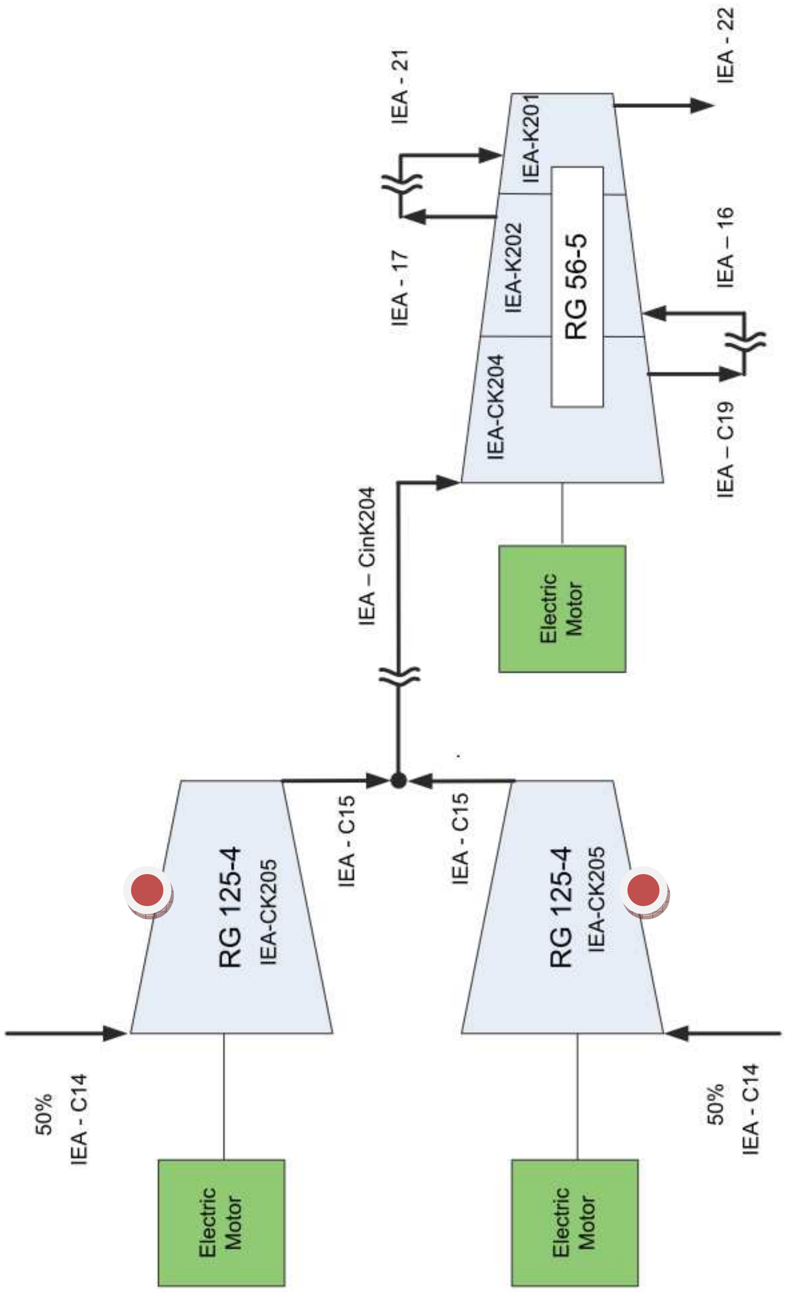


Train n°1 and train n° 2 run in parallel

Train n° 3 (running in series to trains 1-2).



OXYFUEL – INTER-COOLING AS SPECIFIED, General Overview



Note: only the presence of additional intercoolers with respect to baseline configuration is indicated in the above overview with red bullets.

The key process and mechanical characteristics of the machines selected by MAN Diesel & Turbo are shown in the following tables.

OXY-FUEL PROCESS – INTERCOOLING AS SPECIFIED	
MAN TURBO	Integrally gear motor driven
Total n° of trains required	3
Ref. Train 1:	CK205 (50% capacity)
model	RG125-4
stages	4
Flowrate CK205 (Nm^3/h)	163758
inlet temperature (°C)	12
inlet pressure (bara)	1
compression ratio	14.85
Speed shaft 1 (CK-205) rpm	4464
Speed shaft 2 (CK-205) rpm	8058
brake power shaft 1 +2 CK-205 (KW)	19978
total brake power (KW)	19978
poly- efficiency 1^{st}-2nd -3^{rd}-4th stage (%)	86.05
capacity control type	IGV
Intercoolers included by vendor	1 (; 2nd/3rd;) note 1
Ref. Train 2:	CK205 (50% capacity)
model	RG125-4
stages	4
Flowrate CK205 (Nm^3/h)	163758
inlet temperature (°C)	12
inlet pressure (bara)	1
compression ratio	14.85
Speed shaft 1 (CK-205) rpm	4464
Speed shaft 2 (CK-205) rpm	8058
brake power shaft 1 +2 CK-205 (KW)	19978
total brake power (KW)	19978
poly- efficiency 1^{st}-2nd -3^{rd}-4th stage (%)	86.05
capacity control type	IGV

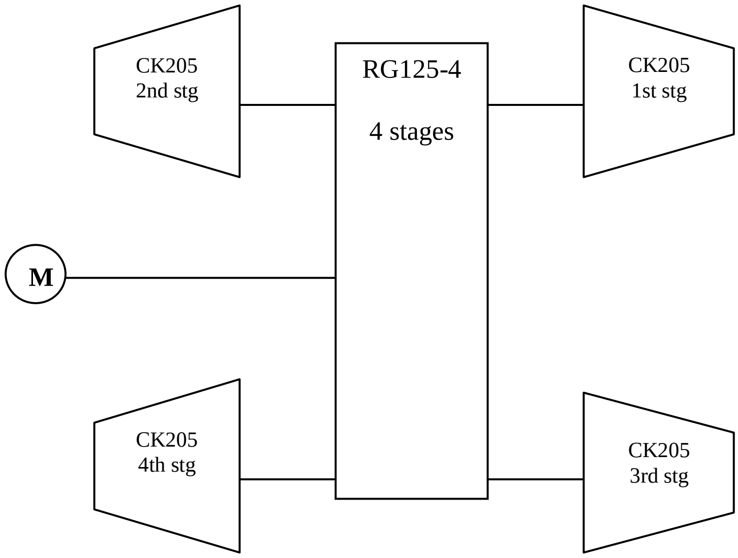
Intercoolers included by vendor	1 (; 2nd/3rd;) note 1
Ref. Train 3:	CK204 / K202 / K201
model	RG56-5
stages	5
flowrate CK204 (Nm^3/h)	363454
flowrate K202 (Nm^3/h)	119033
flowrate K201 (Nm^3/h)	235200
inlet temperature CK204 (°C)	19
inlet temperature K202 (°C)	7.5
inlet temperature K201 (°C)	13.3
inlet pressure CK204 (bara)	14.4
inlet pressure K202 (bara)	9.3
inlet pressure K201 (bara)	18.6
compression ratio CK204	2.16
compression ratio K202	2.01
compression ratio K201	6
Speed shaft 1 (CK204+K202) rpm	9844
Speed shaft 2 (K201) rpm	10881
Speed shaft 3 (K201) rpm	16036
brake power shaft CK204+K202+K201 (KW)	31586
total brake power (KW)	31586
poly-efficiency 1st stage	86.4
poly-efficiency 2nd stage	85.2
poly-efficiency 3rd -4th -5th stage	84.6
capacity control type	IGV
Intercoolers included by vendor	0 note 2

TOTAL BRAKE POWER (WHOLE PROCESS): **71542 KW**

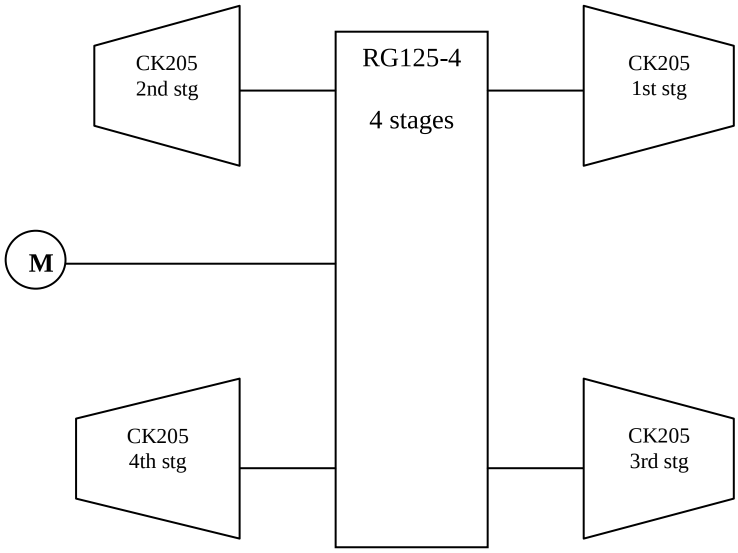
One additional intercooler between 2nd/3rd stage of process CK205 is included. The discharge temperature of CK205 is 142.1°C.

OXY-FUEL – OPTIMISED INTER-COOLING

Train n° 1

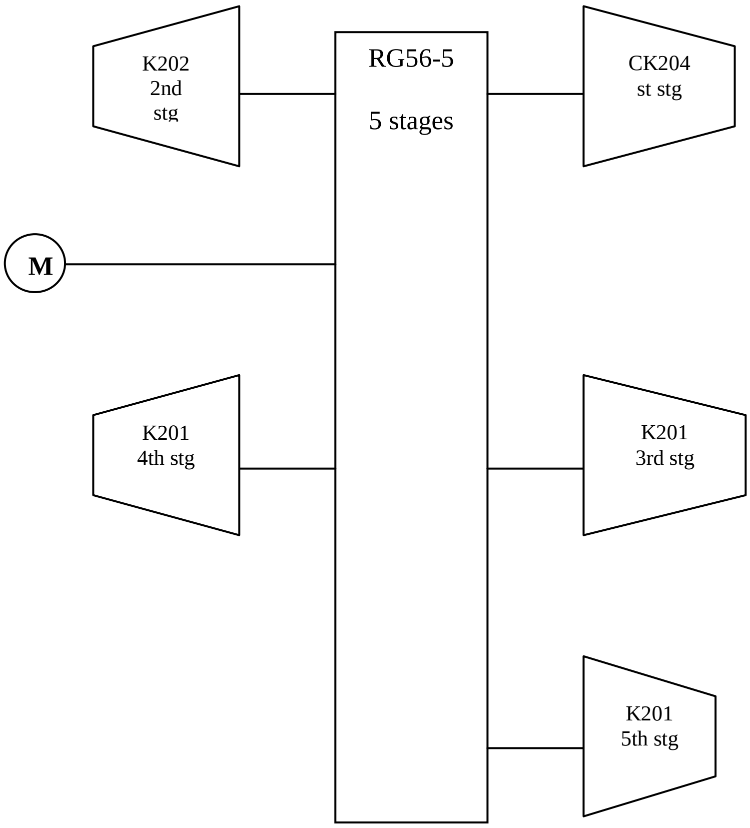


Train n° 2

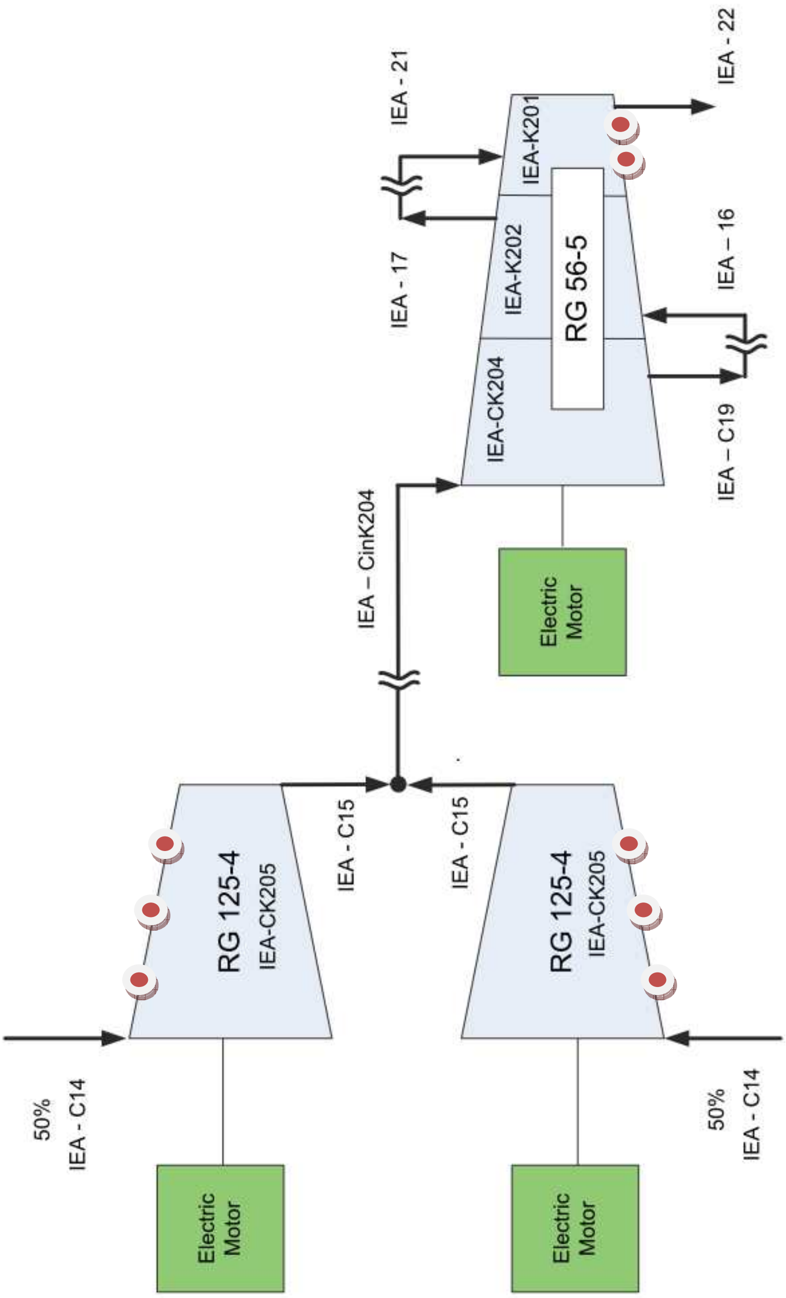


Train n°1 and train n° 2 run in parallel

Train n° 3 (running in series to train 1-2)



OXYFUEL – OPTIMISED INTER-COOLING, General Overview



Note: only the presence of additional intercoolers with respect to baseline configuration is indicated in the above overview with red bullets.

The key process and mechanical characteristics of the machines selected by MAN Diesel & Turbo are shown in the following tables.

OXY-FUEL PROCESS – OPTIMISED INTERCOOLING	
MAN TURBO	Integrally gear motor driven
Total n° of trains required	3
Ref. Train 1:	CK205 (50% capacity)
Model	RG125-4
Stages	4
Flowrate CK205 (Nm^3/h)	163758
inlet temperature (°C)	12
inlet pressure (bara)	1
compression ratio	14.85
Speed shaft 1 (CK-205) rpm	4409
Speed shaft 2 (CK-205) rpm	7246
brake power shaft 1 +2 CK-205 (KW)	18185
total brake power (KW)	18185
poly- efficiency 1st-2nd -3rd-4th stage (%)	86.05
turndown each stage (%)	33.4
capacity control type	IGV
poly- efficiency at min flow (%)	82.5
Intercoolers included by vendor	3 (1st/2nd; 2nd/3rd; 3rd/4th stages)
Ref. Train 2:	CK205 (50% capacity)
Model	RG125-4
Stages	4
Flowrate CK205 (Nm^3/h)	163758
inlet temperature (°C)	12
inlet pressure (bara)	1
compression ratio	14.85
Speed shaft 1 (CK-205) rpm	4409
Speed shaft 2 (CK-205) rpm	7246
brake power shaft 1 +2 CK-205 (KW)	18185
total brake power (KW)	18185
poly- efficiency 1st-2nd -3rd-4th stage (%)	86.05
turndown each stage (%)	33.4
capacity control type	IGV
poly- efficiency at min flow (%)	82.5

Intercoolers included by vendor	3 (1st/2nd; 2nd/3rd; 3rd/4th stages)
Ref. Train 3:	CK204 / K202 / K201
Model	RG56-5
Stages	5
flowrate CK204 (Nm^3/h)	363454
flowrate K202 (Nm^3/h)	119033
flowrate K201 (Nm^3/h)	235200
inlet temperature CK204 (°C)	19
inlet temperature K202 (°C)	7.5
inlet temperature K201 (°C)	13.3
inlet pressure CK204 (bara)	14.4
inlet pressure K202 (bara)	9.3
inlet pressure K201 (bara)	18.6
compression ratio CK204	2.16
compression ratio K202	2.01
compression ratio K201	6
Speed shaft 1 (CK204+K202) rpm	9844
Speed shaft 2 (K201) rpm	10832
Speed shaft 3 (K201) rpm	19545
brake power shaft CK204+K202+K201 (KW)	27965
total brake power (KW)	27965
poly-efficiency 1^{st} stage	86.4
poly-efficiency 2nd stage	85.2
poly-efficiency 3^{rd} -4^{th} -5^{th} stage	85.2
turndown 1^{st} stage (%)	32.6
turndown 2nd stage (%)	34.4
turndown 3rd - 4^{th}-5^{th} stage (%)	33.2
capacity control type	IGV
poly- efficiency 1^{st} stage at min flow (%)	83.8 (estimated)
poly- efficiency 2nd stage at min flow (%)	82 (estimated)
poly- efficiency 3^{rd}-4^{th}-5^{th} stage at min flow (%)	81 (estimated)
Intercoolers included by vendor	2 (3rd/4th; 4th/5th stages)

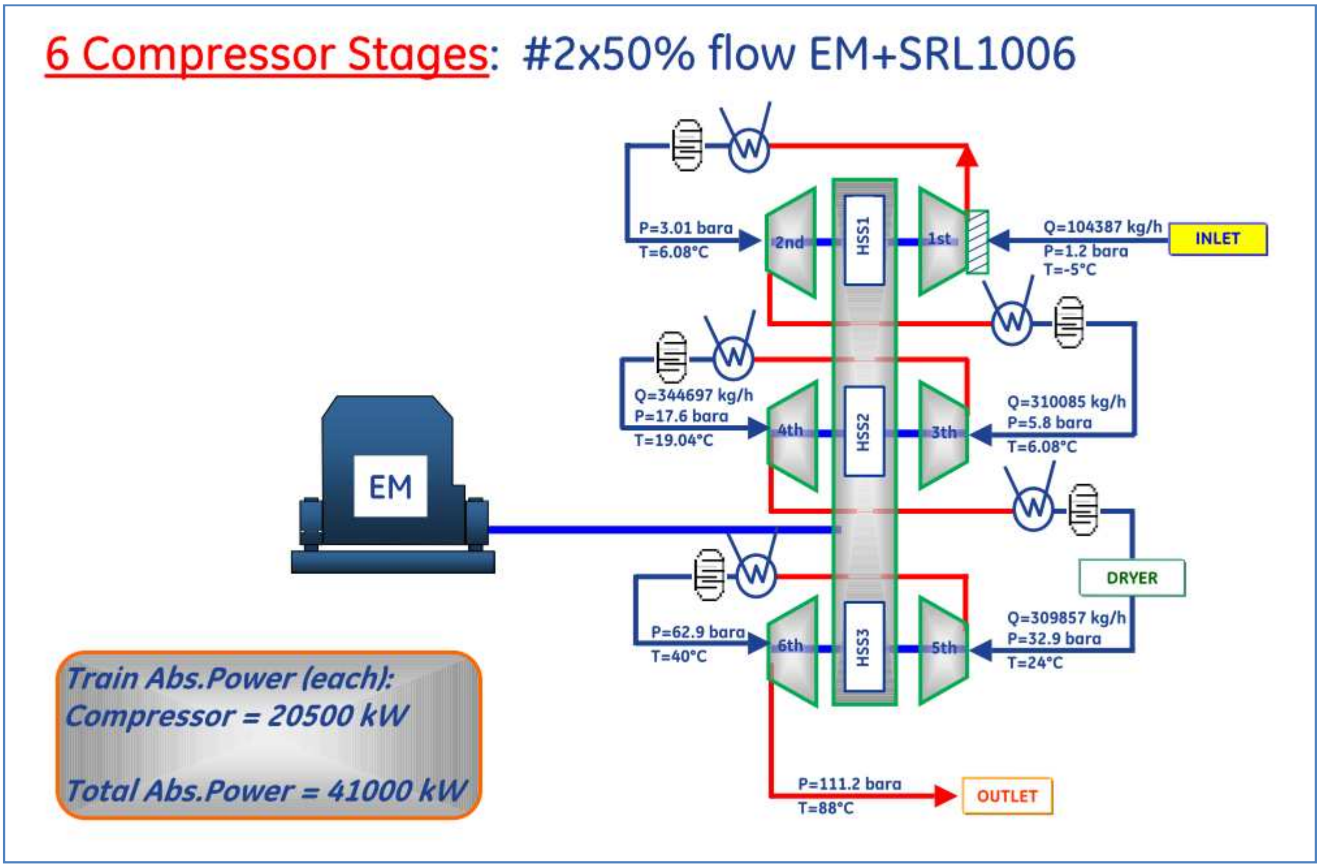
TOTAL BRAKE POWER (WHOLE PROCESS): **64335 KW**

2.2.3 GE

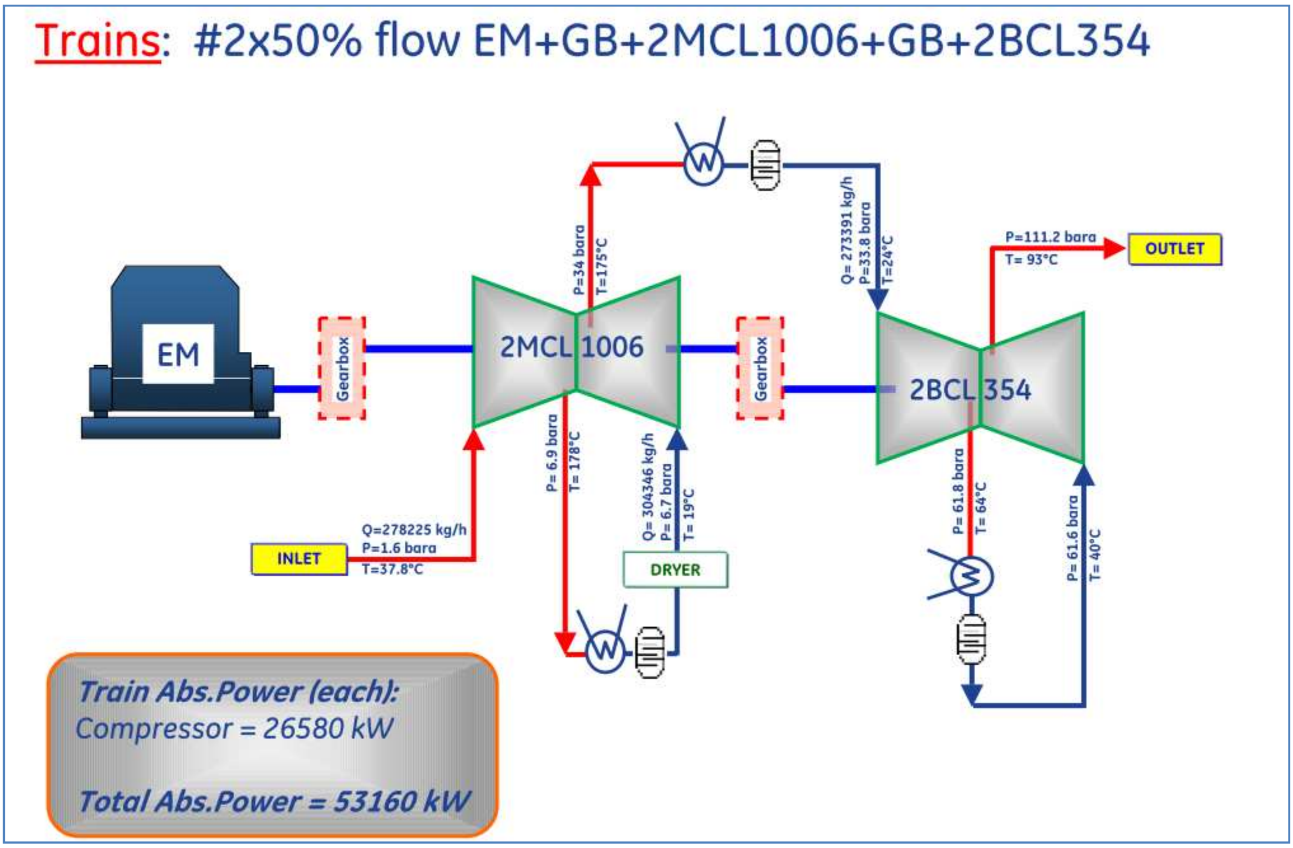
The following pages outline the selection of the compressors for the three different process technologies.
Briefly they can be summarized as follows:

Process technology	Type of compressor
Pre-combustion – 6 stages	Integral gear
Post-combustion – 4 stages	Conventional in-line
Post-combustion – 8 stages	Integral gear
Post-combustion – alternative 6 stages	Integral gear
Oxy-fuel (duty K205, K204)	Conventional in-line
Oxy-fuel (duty K202, K201)	Conventional in-line

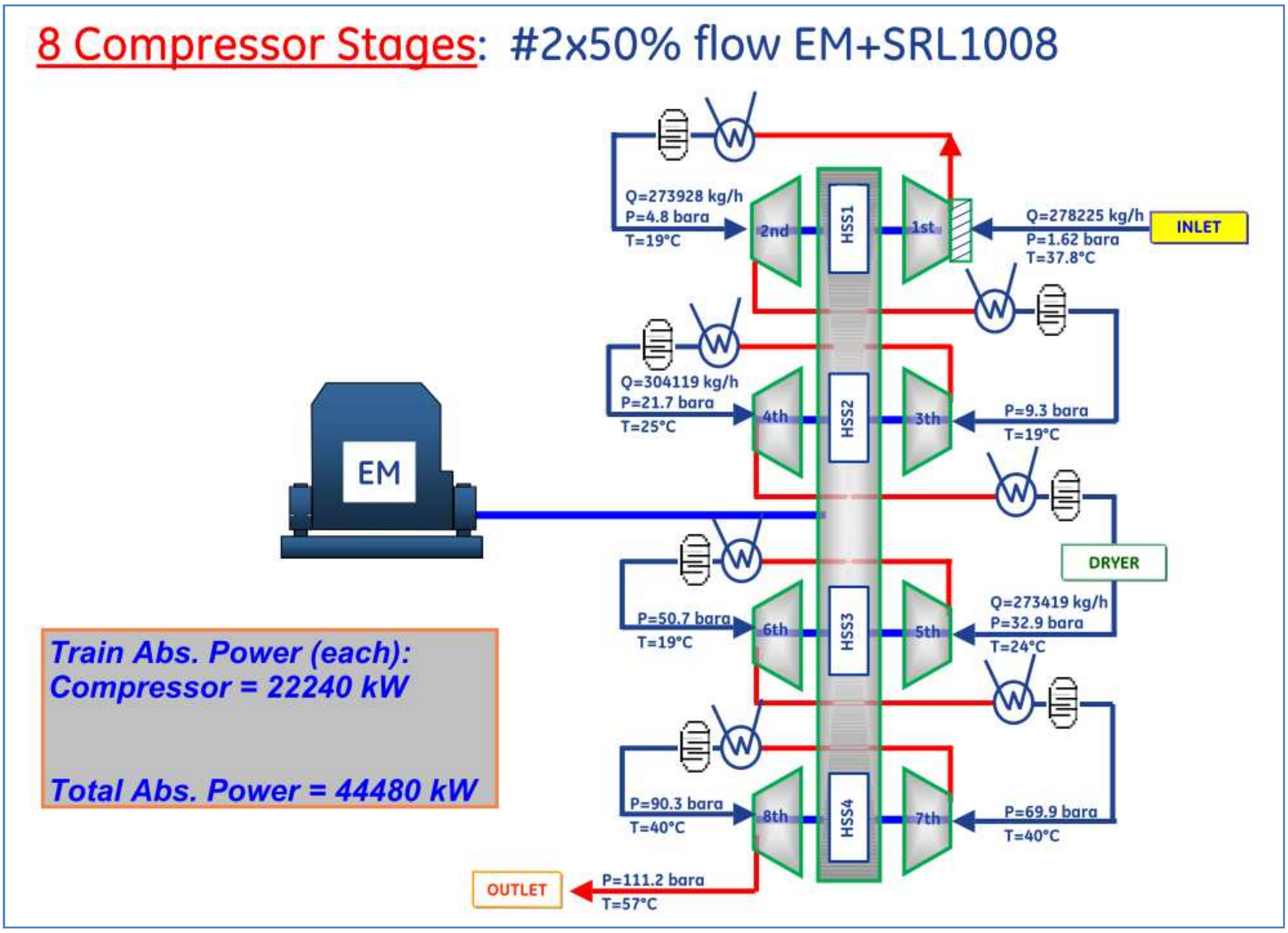
PRE-COMBUSTION



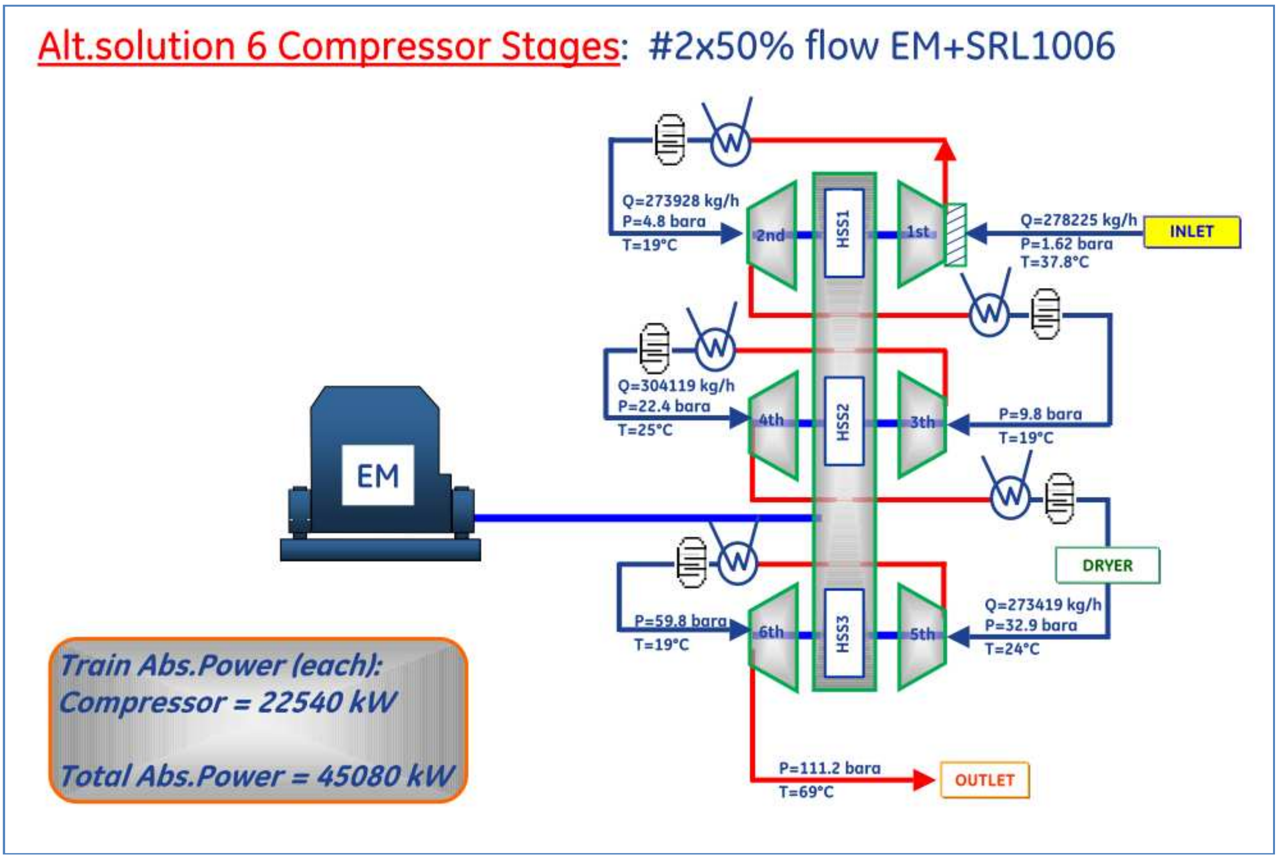
POST-COMBUSTION – 4 COMPRESSOR STAGES



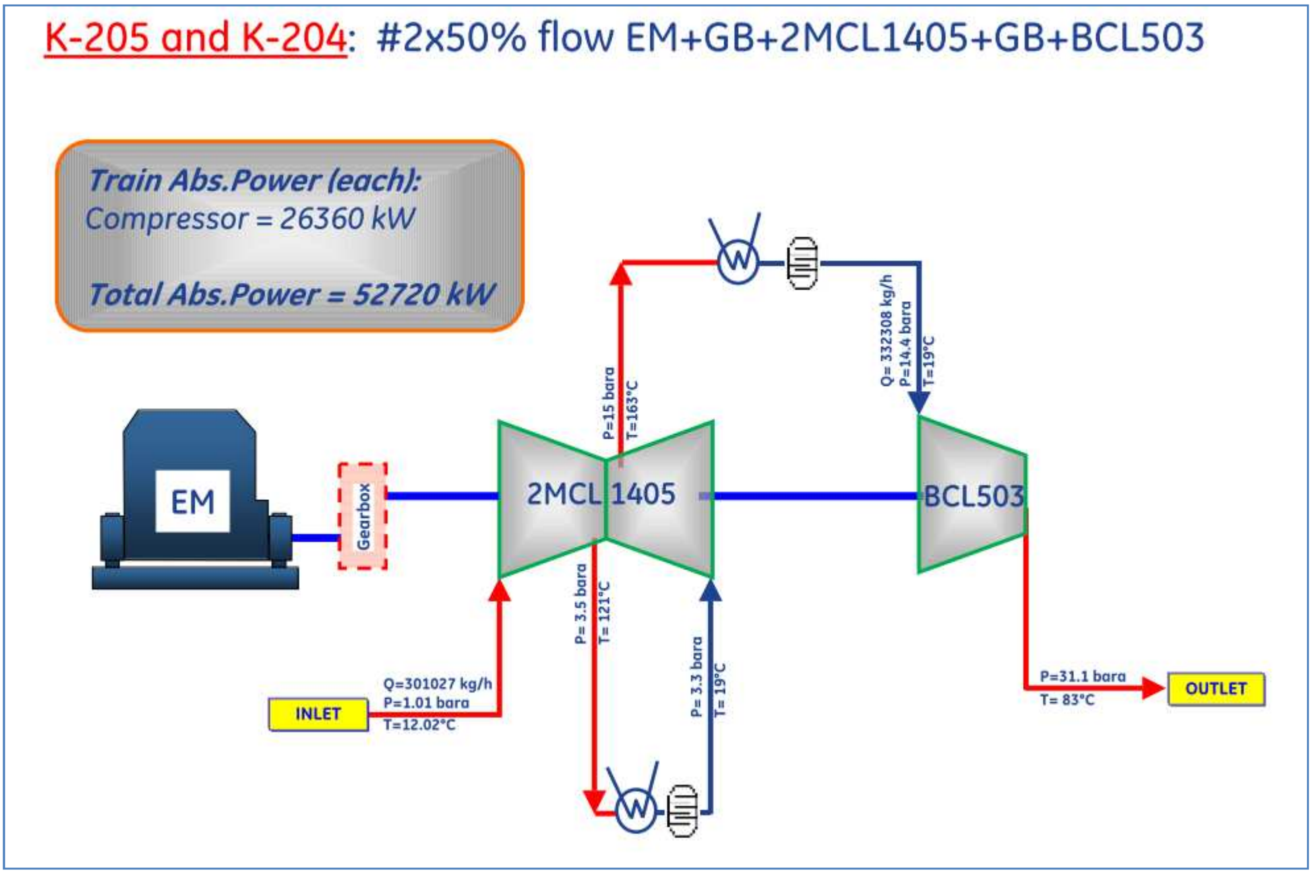
POST-COMBUSTION – 8 COMPRESSOR STAGES

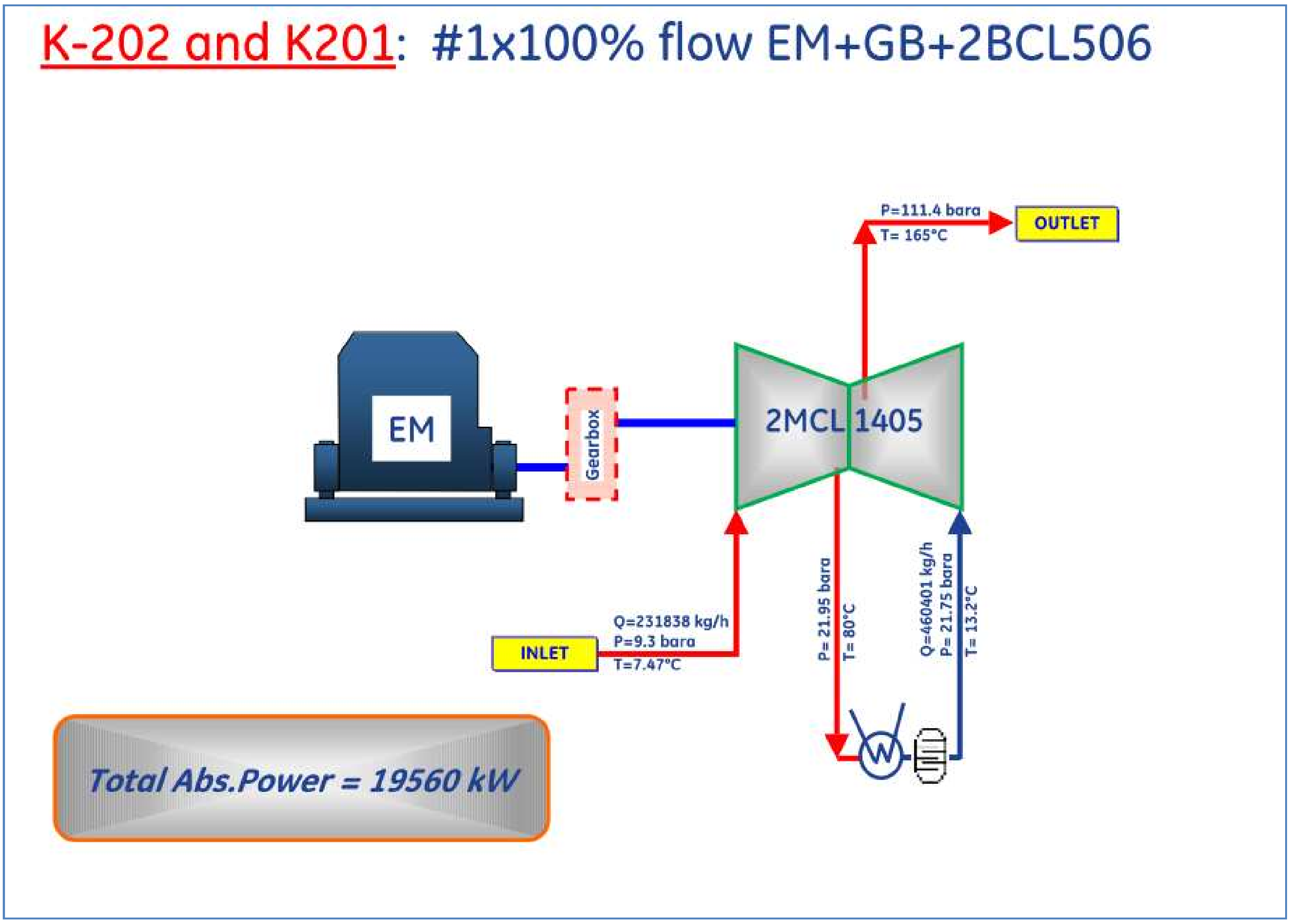


POST-COMBUSTION – ALTERNATIVE 6 COMPRESSOR STAGES



OXY-FUEL


K-205 and K-204: #2x50% flow EM+GB+2MCL1405+GB+BCL503
Train Abs.Power (each):
Compressor = 26360 kW
Total Abs.Power = 52720 kW
EM
Gearbox
2MCL 1405
BCL503
P=15 bara
T=163°C
Q= 332308 kg/h
P=14.4 bara
T=19°C
P= 3.5 bara
T= 121°C
P= 3.3 bara
T= 19°C
INLET
Q=301027 kg/h
P=1.01 bara
T=12.02°C
P=31.1 bara
T= 83°C
OUTLET


K-202 and K201: #1x100% flow EM+GB+2BCL506
P=111.4 bara
T= 165°C
OUTLET
EM
Gearbox
2MCL 1405
Q=231838 kg/h
P=9.3 bara
T=7.47°C
INLET
P= 21.95 bara
T= 80°C
Q=460401 kg/h
P= 21.75 bara
T= 13.2°C
W
Total Abs.Power = 19560 kW

2.2.4 Overview of absorbed powers

Table 2-2 summarises the main information on train arrangement and machine selection undertaken by the different manufacturers involved in the study, indicating the absorbed powers for each case. It is noted that the performance figures provided by the compressor manufacturers are for the purposes of the study only and do not represent performance guarantees. All manufacturers have included flange to flange losses, indicating the shaft power at the driver.

Table 2-2 Configuration proposed and shaft power (kW)

Case	Rolls-Royce	MAN Diesel & Turbo	GE
Pre-combustion	40,540 kW Trains: 1x100% 4 in-line machines 13 compression stages, 6 inter-cooling steps	43,960 kW Trains: 1x100% 1 integral gear machine 8 compression stages, 7 inter-cooling steps	41,000 kW Trains: 2x50% 2 integral gear machine #.stages not avail. 5 inter-cooling steps
Post-combustion inter-cooling as specified	50,460 kW Trains: 2x50% 4 in-line machines/train 13 compression stages, 3 inter-cooling steps T range: 80÷170 °C (1)	53,160 kW Trains: 1x100% 1 integral gear machine 6 compression stages, 3 inter-cooling steps T range: 85÷180 °C (1)	53,160 kW Trains: 2x50% 4 in-line machines #.stages not avail. 3 inter-cooling steps T range: 60÷180 °C (1)
Post-combustion optimised inter-cooling	44,410 kW Trains: 2x50% 4 in-line machines/train 12 compression stages, 5 inter-cooling steps T range: 60÷115 °C (1)	48,810 kW Trains: 1x100% 1 integral gear machine 6 compression stages, 5 inter-cooling steps T range: 80÷100 °C (1)	–
Post-combustion Increased stages	43,490 kW Trains: 2x50% 4 in-line machines/train 13 compression stages, 6 inter-cooling steps T range: 50÷150 °C (1)	45,650 kW Trains: 1x100% 1 integral gear machine 7 compression stages, 6 inter-cooling steps T range: 65÷100 °C (1)	44,480 kW Trains: 2x50% 2 integral gear machine #.stages not avail., 7 inter-cooling steps T range: not avail. (1)

Case	Rolls-Royce	MAN Diesel & Turbo	GE
Oxy-fuel inter-cooling as specified	74,100 kW Trains: 3x33%+1x100% 7 in-line machines 12 compression stages, 3 inter-cooling steps T range: 65÷280 °C (1)	71,540 kW Trains: 2x50%+1x100% 3 integral gear machines 9 compression stages, 4 inter-cooling steps T range: 65÷185 °C (1)	72,280 kW Trains: 2x50%+1x100% 5 in-line machines #.stages not avail., 4 inter-cooling steps T range: not avail. (1)
Oxy-fuel optimised inter-cooling	62860 kW Trains: 3x33%+1x100% 7 in-line machines 14 compression stages, 7 inter-cooling steps T range: 70÷135 °C (1)	64340 kW Trains: 2x50%+1x100% 3 integral gear machines 9 compression stages, 8 inter-cooling steps T range: 65÷95 °C (1)	_

Notes:
1) The stage discharge temperatures range is indicated only for oxy-fuel and post-combustion processes, in which the compression heat is recovered through thermal integration with the Power Island.

Generally, for the Base Cases the table shows absorbed powers lower than the figures estimated in the study (ref. Section B), as summarized in the following:

- Oxy-fuel: (from 7 to 21% lower)
- Post-combustion: (from 8 to 23% lower)
- Pre-combustion: (from 7 to 14% lower).

This is mainly due to both the higher stage efficiencies proposed by the Vendors with respect to the assumptions made in the reference studies and, in most cases, the use of additional inter-cooling steps.

For the pre-combustion case, the reduction in terms of power consumption would be entirely reflected into a net power output improvement, since the CO_2 compression waste heat is disposed to the cooling water only, i.e. there is no recovery of waste heat from the CO_2 compression.

As far as Oxy-fuel combustion and Post Combustion cases are concerned, the reduction of power consumption due to the higher efficiency or increased inter-cooling would be partially off-set by the consequent decrease of compression heat available at the CO_2 compression stage outlet, a fraction of which is recovered into the Steam Condensate / Boiler Feed Water systems of the Boiler / Steam Turbine Island. This is particularly true for the machinery selections in which the Suppliers have included additional inter-cooling steps with respect to the original design, in order to optimise the selection and minimise compressor electrical consumption.

The feedback from MAN Diesel & Turbo for the oxy-fuel combustion base case indicates that integral gear machines may have some technical limitations in handling the first un-cooled section of the CO_2 compression process, due to the high discharge temperature. Hence, with the integral gear compressors proposed by MAN Diesel & Turbo, the deep thermal integration with the Power Island can not be implemented as foreseen in the reference case. The compressor itself would have lower parasitic load but elsewhere in the plant (e.g. Steam Turbine Island) there will be an increased thermal energy demand.

Generally, in terms of performance evaluation, it would be a mistake to draw conclusions based just on the compressor electrical consumption. The performance of the machines has to be considered in the context of overall plant performance, which is influenced by the different integration into the overall process depending on the proposed configuration.

For the post combustion capture, the comparison between 4 stages and 8 stages from Vendors data confirm the beneficial effects in terms of electrical consumption reduction, as expected in case D1 (ref. Section C).

2.2.5 Budget cost of machine packages

An indication of the expected specific cost range for each machinery class is included in this report, based on Foster Wheeler judgements form a range of sources. The cost ranges, reported in the following table, cover the selections for the operating envelope given in the different applications investigated (pre-combustion, post-combustion and oxy-fuel combustion).

Table 2-3 Specific investment cost range for each machine type.

Type	Specific cost range
In-line centrifugal (Rolls Royce)	600÷900 €/kW
Integral-gear (MAN Diesel & Turbo)	300÷600 €/kW

Table 5.2 generally shows a higher investment cost for in-line centrifugal machines than integral-gear, the delta cost being mainly justified by the following reasons:

- In-line centrifugal compressors need multiple train solutions;
- More compact design of the integrally geared centrifugal compressors, which results in a lower machine investment cost;

- Variable Speed Drivers are required for capacity control at partial load in the in-line machines, whereas the integral-gear type is supplied with Inlet Guide Vanes.

Despite the generally higher cost, the “in-line” machines offer the following technical advantages over the integral-gear type:

- Better maintainability, due to easier access, as explained in paragraph 1.
- Higher operating flexibility, due to the multiple parallel trains configuration, with the turndown capability of the single train being very similar for both types. Also, the VFD provided with the in-line machines ensures better efficiency (i.e. lower parasitic consumption) when the plant operates at partial load. This is a very important feature since it is expected that CCS power plants will be required to operate in the actual electricity market, responding to the normal daily and seasonal variability of electricity demand.
- Higher reliability, typically by 2% with respect to the integral-gear compressors.
- Reduced impact on the electrical system design. The impact of using large motors is mainly represented by the necessity for a significant over design of the electrical systems equipment (transformers, cables, etc.) to support the peak current demand at motor starting. For the in-line compressor, smaller size (roughly half) for the largest motors has been proposed with respect to the integral–gear compressors. Also, VFD’s have been included for in-line machines capacity control, which are expected to perform better in smoothing the peak demand at motor starting than the soft starters proposed with the integral gear type.

Hence, from the indications reflected in this report, it is not possible to draw any definitive conclusion on the economics of the different machine types. The selection of the machine, as usual, is case-specific and not driven by machine investment cost only. Other features like reliability, flexible operation, easy maintainability and associated impacts on other systems in the plant shall be also accurately assessed. Also, the cost of the machines has to be considered in the context of overall plant cost and performance, which may differ as each requires slightly different integration into the overall process.

3 Drive and capacity control options

Usually the drives that conventionally might be used for compressors in Power Stations with CCS can be selected between electric motors and steam turbines, considering both the typical power range of the prime movers and the experience gained by compressor manufacturers.
Here below the main pros and cons are listed for each solution.

3.1 Steam turbine option

Technical Pros and cons associated to the application of steam turbines for machinery driving are reported below.

Pros

- Rating covers wide range of power (typically 130MW is not an issue)
- It is particularly adaptable for direct connection to equipment rotating at high speed
- Operate over a broad speed range
- Steam is often used elsewhere in process
- Better efficiency at part load conditions.

Cons

- Physically very large, layout requires more space and more extensive civil works
- Overhaul is more complex and maintenance is more costly
- Operating procedures are more complex and take more time especially during start-up and stop (running the plant without CCS capture is easier if this involves simply switching off an electric motor)

3.2 Electric motor option

Technical Pros and cons associated to the use of electric motors are reported below.

Pros

- Higher availability than steam turbines
- Reduced manning level
- Simple layout, reduced civil works
- Easy operation, start-up procedure and switch-off

Electric motors –Cons

- Limited experience with high power VSD (typically up to 65 MW); however, this is not a major issue as in VSD solutions multiple drivers are likely to be used.
- Electrical issues at compressor start-up when VSD is used (peak voltage, harmonic distortion, etc)
- Speed control range: electric motors can potentially go down to 55% speed, as per Rolls Royce feedback, however, the range is restricted by VFD efficiency drop due to the additional cooling required at minimum speed.

3.3 Economics qualitative comparison of the options

In terms of economics the comparative analysis can only be case specific, depending on unit scale and type of solutions proposed by Vendors.

In a Power Station with CCS, the OPEX of the Steam Turbine drive option are expected to be worse than the electric motor option, since the steam to the drives is taken from the main steam system in the Power Island and, generally, the adiabatic efficiency of the main Steam Turbine is expected to be noticeably higher than that of steam turbine used for driving the compressors.

As far as CAPEX is concerned, for large scale application, the steam turbines may turn out to be less costly than the electrical motors. However, this depends on many factors, the most important ones being:

- need for a variable speed driver with the electric motor, if this is proposed by Vendors to provide a load flexibility comparable to steam turbine drives.
- Use of condensing turbine vs. backpressure turbine, as reflected in the quotation provided by MAN Diesel & Turbo.
- Cost of steam piping (high/low pressure) and valves for the steam turbine option, which may not be negligible if the compression plant is located some distance from the steam plant (steam pipe would incur significant losses)

3.4 Capacity control options proposed by Vendors

In the power generation business turndown requirements may become increasingly important as dependence on renewables increases. Turndowns of up to 50% are not unusual today and this is likely to be the requirement for future power stations with

CCS, being reflected in a 50% turndown requirement for the CO_2 compression train as well.

In applications that require flow control the most energy effective technique is often variable speed control; alternatively inlet guide vanes can be used, depending on the type of machine.

Generally the variable speed control is used when conventional machine trains are involved and for instance this applies to the Rolls-Royce proposal.
It is worth to mention that, to complement its current portfolio of products, Rolls-Royce is also developing an advanced CO_2 compressor for future carbon capture and storage applications. Through its two-year collaborative CO_2 Optimised Compressor project, COZOC, with partners E.ON Engineering and the University of Nottingham, Rolls-Royce is developing compressor concepts specifically designed to minimise power consumption across a range of operating conditions. These include novel approaches to power optimised base load and part load operation. Whereas existing compressors meet turndown requirements between 75-80% flowrate via re-circulating bleed and/or multiple parallel trains, Rolls-Royce advanced concepts utilise almost no bleed. This results in significant power savings.

On the other hand, the inlet guide vanes are employed with integral-gear compressors likewise has been proposed by MAN Diesel & Turbo.
The possibility to install the IGV's is facilitated by the direction of the gas flow at inlet of impeller (axial), the shape of the casing and the space available that allows the application of this solution on integral gear machine rather the conventional machine.

In conjunction with the description of the machinery selection for each process technology and for each manufacturer at paragraph **Error! Reference source not found.**, the outcomes of the market survey regarding capacity control are reported in the following sections, where the turndown rates are indicated in accordance to the following definition:

$$\text{Turn Down} = \frac{Rated\ Capacity - Min\ Capacity\ (at\ rated\ head)}{Rated\ Capacity}$$

<u>Rolls-Royce selection</u>

<u>Pre-combustion</u>
For this case a single compressor train has been selected by vendor to comply with the process duty, consisting of two compressor machines.

The expected performance curve of the first compressor machine (duty K-100, K101, K102) exhibits a minimum flow of about 60000 am3/h at inlet of first stage (duty K100), while the rated capacity is 87493 am3/h. This corresponds to the theoretical compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 31.4 %.
The regulation of capacity is achieved by means of VSD (variable speed drive) which reduces the speed by about 5% to about 4560 rpm, while the speed corresponding to rated capacity is 4800 rpm.
The expected performance curve of the second compressor machine (duty K-103, K104) exhibits a minimum flow of about 6100 am3/h at inlet of first stage (duty K103) while the rated capacity is 8793 am3/h.
This corresponds to the theoretically compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 30.6 %.
The regulation of capacity is achieved by means of VSD (variable speed drive) which reduces the speed at about 9600 rpm, while the speed corresponding to rated capacity is 10000 rpm.

However it has to be highlighted that some margin on the surge line is taken into account by the compressor vendor to allow the anti-surge protection system to react promptly, hence the practical turndown achievable would be slightly higher and depends on the algorithms pertaining to the anti-surge protection system. These consideration is valid for all operating cases, thus not only for pre-combustion, but also for post-combustion and oxy-fuel.

In conclusion, for the pre-combustion single train, being the turndown of the overall unit equal to 30.6%, it is not possible to achieve 50% of turndown capacity relying upon the design of machine itself and therefore the solutions to accomplish this requirement are the use of spillback lines or multiple parallel trains.

<u>Post-combustion - inter-cooling as specified</u>
For this case two parallel compressor trains have been selected by vendor to comply with the process duty, consisting of total four compressor machines, two machines group running in parallel and consisting of first compressor unit (K101, K102) and second compressor unit (K103, K104) per each train.
Due to this duplication, it is possible to achieve 50% of total compressor capacity of post-combustion process simply switching – off the two compressor units pertaining to one of the two trains. However the range from around 30%-50% will not be covered.

Moreover, even with one train running it is possible to further reduce the capacity if necessary.
In fact the expected performance curve of the first compressor machine (duty K-101, K102) exhibits a minimum flow of about 70000 am3/h at inlet of first stage (duty

K101), while the rated capacity is 102620 am3/h. This corresponds to the theoretically compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 31.7 % for each of the two parallel trains.
The regulation of capacity is achieved by means of VSD (variable speed drive) which reduces the speed at about 3900 rpm, while the speed corresponding to rated capacity is 4200 rpm.
The expected performance curve of the second compressor machine (duty K-103, K104) exhibits a minimum flow of about 2600 am3/h at inlet of first stage (duty K103) while the rated capacity is 3757.7 am3/h.
This corresponds to the theoretically compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 30.8 % for each of the two parallel trains.
The regulation of capacity is achieved by means of VSD (variable speed drive) which reduces the speed at about 9500 rpm, while the speed corresponding to rated capacity is 10000 rpm.

The overall unit turndown is therefore equal to 65.6%, considering the double parallel train configuration. but this is not a continuously variable turndown, there is a gap between around 30% and 50%.

<u>Post-combustion – increased stages</u>
For this case two parallel compressor trains have been selected by vendor to comply with the process duty, consisting of total six compressor machines, three machines group running in parallel and consisting of of first compressor unit (K101, K102), second compressor unit (K103, K104) and third compressor unit (K105, K106, K107, K108) per each train.
Due to this duplication, it is possible to achieve 50% of total compressor capacity of post-combustion process simply switching –off the three compressor units pertaining to one of the two trains.
Moreover, even with one train running it is possible to further reduce the capacity if necessary.
In fact the expected performance curve of the first compressor machine (duty K-101, K102) exhibits a minimum flow of about 70000 am3/h at inlet of first stage (duty K101), while the rated capacity is 102620 am3/h. This corresponds to the theoretically compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 31.7 % for each of the two parallel trains.
The regulation of capacity is achieved by means of VSD (variable speed drive) which reduces the speed at about 3900 rpm, while the speed corresponding to rated capacity is 4200 rpm.
The expected performance curve of the second compressor machine (duty K-103, K104) exhibits a minimum flow of about 10000 am3/h at inlet of first stage (duty K103) while the rated capacity is 14586 am3/h.

This corresponds to the theoretically compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 31.5 % for each of the two parallel trains..
The regulation of capacity is achieved by means of VSD (variable speed drive) which reduces the speed at about 5700 rpm, while the speed corresponding to rated capacity is 6000 rpm.
The expected performance curve of the third compressor machine (duty K105, K106, K107, K108) exhibits a minimum flow of about 2580 am3/h at inlet of first stage (duty K105) while the rated capacity is 3758.2 am3/h.
This corresponds to the theoretically compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 31.3 % for each of the two parallel trains.
The regulation of capacity is achieved by means of VSD (variable speed drive) which reduces the speed at about 9500 rpm, while the speed corresponding to rated capacity is 10000 rpm.
The overall unit turndown is therefore equal to 56.7 %, considering the double parallel train configuration.
Considering two trains running the capacity ranges between about 70 and 100% while one train running ranges between about 35% and 50%; hence the capacity band 50% - 70% is not possible.

<u>Oxy-fuel – inter-cooling as specified</u>
For this case seven compressor machines have been selected by vendor to comply with the process duty. For first and second compressor units (CK205 and CK204) three parallel trains, each one is composed by two compressors, are selected while the fourth compressor unit (K202, k201) is a single train unit.
As far as first compression stages are concerned, due to the train triplication, switching –off one of the three trains, a turndown of 33% is achievable and regulating further the capacity of each compressor CK205 it is possible to obtain the required minimum turndown of 50%.
In fact the expected performance curve of the first compressor machine (duty CK-205) exhibits a minimum flow of about 80770 am3/h at inlet of first stage (duty CK205), while the rated capacity is 115980 am3/h. This corresponds to the theoretically compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 30.3 % for each parallel train. The regulation of capacity is achieved by means of VSD (variable speed drive) which reduces the speed at about 3980 rpm, while the speed corresponding to rated capacity is 4162 rpm.
The expected performance curve of the second compressor machine (duty CK-204) exhibits a minimum flow of about 5800 am3/h at inlet of first stage (duty CK204) while the rated capacity is 8690.2 am3/h. This corresponds to the theoretically compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 33 % for each parallel train. The regulation of capacity is achieved

by means of VSD (variable speed drive) which reduces the speed at about 8140 rpm, while the speed corresponding to rated capacity is 8565 rpm.

The expected performance curve of the forth train consisting of one single compressor machine (duty K202, K201) exhibits a minimum flow of about 9500 am3/h at inlet of first stage (duty K202) while the rated capacity is 12660 am3/h.
This corresponds to the theoretically compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 25 %.
The regulation of capacity is achieved by means of VSD (variable speed drive) which reduces the speed at about 10200 rpm, while the speed corresponding to rated capacity is 10414 rpm.

In conclusion, for the oxy-fuel combustion case, being the turndown of the overall compression unit equal to 25%, it is not possible to achieve 50% of turndown capacity relying upon the design of machine itself and therefore the solutions to accomplish this requirement are the spillback lines or duplication of units K-201 and K-202.

For all processes it has to be noticed that the speed variation ranges within 7% max (i.e. from 100% down up to 93% max).
Usually the standard construction of variable speed drives (VSD) is based on speed variation 70-100 % and the technology in use for such range applies as such even when small speed variations are necessary as in this specific case.
Therefore the VSD will result underused and could be compensated by energy saved as a function of the time running at part load conditions.

MAN Diesel & Turbo selection

Pre-combustion – solutions 1 and 2
One single compressor train has been selected by vendor to comply with the process duty, consisting of one compressor machine.
The expected performance curve of the compressor machine for duty K-100 exhibits a minimum flow of about 70000 Nm3/h at inlet of first stage (duty K100), while the rated capacity is 105568 Nm3/h. This corresponds to the theoretical compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 33.6 %.
The regulation of capacity is achieved by means of IGV (inlet guide vanes) which are rotated to about 42° (at rated capacity is 0°).
The expected performance curve of the machine for duty K-101 exhibits a minimum flow of about 210000 Nm3/h at inlet of second stage (duty K101) while the rated capacity is 315734 Nm3/h.

This corresponds to the theoreticall compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 33.4 %.
The regulation of capacity is achieved by means of IGV (inlet guide vanes) which are rotated to about 45° (at rated capacity is 0°).
The expected performance curve of the machine for duty K-102, exhibits a minimum flow of about 235000 Nm3/h at inlet of third stage (duty K102), while the rated capacity is 354177 Nm3/h. This corresponds to the theoretical compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 33.6 %.
The regulation of capacity is achieved by means of IGV (inlet guide vanes) which are rotated to about 50° (at rated capacity is 0°).
The expected performance curve of 4th and 5th stages (duty K103, K104) exhibits a minimum flow of about 205000 Nm3/h, while the rated capacity is 318643 Nm3/h. This corresponds to the theoretically compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 35.7 %.
The regulation of capacity is achieved by means of IGV (inlet guide vanes) which are rotated to about 42° (at rated capacity is 0°).

In conclusion, for the pre-combustion single train, being the turndown of the overall unit equal to 33.4%, it is not possible to achieve 50% of turndown capacity relying upon the design of machine itself and therefore the solutions to accomplish this requirement are the spillback lines or train duplication.

<u>Post-combustion – optimised inter-cooling</u>
For this case only one single compressor train has been selected by vendor to comply with the process duty, consisting of one compressor machine.
The expected performance curve (duty K-101) exhibits a minimum flow of about 183000 Nm3/h at inlet of first stage (duty K101) while the rated capacity is 276677 Nm3/h.
This corresponds to the theoretical compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 33.8 %.
The regulation of capacity is achieved by means of IGV (inlet guide vanes) which are rotated to about 45° (at rated capacity is 0°).
The expected performance curve (duty K-102) exhibits a minimum flow of about 204000 Nm3/h at inlet of third stage (duty K102), while the rated capacity is 307506 Nm3/h. This corresponds to the theoretical compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 33.6 %.
The regulation of capacity is achieved by means of IGV (inlet guide vanes) which are rotated to about 45° (at rated capacity is 0°).
The expected performance curve of 4th and 5th stages (duty K103, K104) exhibits a minimum flow of about 178000 Nm3/h, while the rated capacity is 276321 Nm3/h. This corresponds to the theoretical compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 35.6 %.

The regulation of capacity is achieved by means of IGV (inlet guide vanes) which are rotated to about 40° (at rated capacity is 0°).

In conclusion, for the post-combustion 4-stages single train, being the turndown of the overall unit equal to 33.6%, it is not possible to achieve 50% of turndown capacity relying upon the design of machine itself and therefore the solutions to accomplish this requirement are the spillback lines or train duplication.

Post-combustion – increased stages

For this case one single compressor train has been selected by vendor to comply with the process duty.

The expected performance curve (duty K-101, K102, K103) exhibits a minimum flow of about 185000 Nm3/h at inlet of first stage (duty K101) while the rated capacity is 276662 Nm3/h.

This corresponds to the theoretical compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 33 %.

The regulation of capacity is achieved by means of IGV (inlet guide vanes) which are rotated to about 52° (at rated capacity is 0°).

The expected performance curve (duty K-104) exhibits a minimum flow of about 197000 Nm3/h at inlet of fifth stage (duty K104), while the rated capacity is 307465 Nm3/h. This corresponds to the theoretical compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 35 %.

The regulation of capacity is achieved by means of IGV (inlet guide vanes) which are rotated to about 30° (at rated capacity is 0°).

The expected performance curve (duty K105, K106, K107, K108) exhibits a minimum flow of about 184000 Nm3/h, while the rated capacity is 276512 Nm3/h. This corresponds to the theoretical compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 33.4 %.

The regulation of capacity is achieved by means of IGV (inlet guide vanes) which are rotated to about 52° (at rated capacity is 0°).

In conclusion, for the post-combustion 8-stages, being the turndown of the overall unit equal to 33.0%, it is not possible to achieve 50% of turndown capacity relying upon the design of machine itself and therefore the solutions to accomplish this requirement are the spillback lines or train duplication.

Oxy-fuel – optimised inter-cooling

For this case three compressor machines have been selected by vendor to comply with the process duty. For first compressor unit (CK205), two parallel trains are selected, while for the other downstream units (CK204, K202, K201) consist of a single train.

For compressor CK205, due to the train duplication, it is possible to achieve 50% of total compressor capacity simply switching – off the two parallel trains. Moreover,

even with one train running it is possible to further reduce the capacity if necessary. In fact the expected performance curve of the first compressor machine (duty CK-205) exhibits a minimum flow of about 109000 Nm3/h at inlet of first stage, while the rated capacity is 163758 Nm3/h. This corresponds to the theoretical compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 33.4 % per each parallel train. The regulation of capacity is achieved by means of IGV (inlet guide vanes) which are rotated to about 55° (at rated capacity is 0°).
The expected performance curve of the third compressor (duty CK204, K202, K201) exhibits a minimum flow of about 245000 Nm3/h at inlet of CK204, while the rated capacity is 363454 Nm3/h. This corresponds to the theoretical compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 32.6 %. The regulation of capacity is achieved by means of IGV (inlet guide vanes) which are rotated to about 30° (at rated capacity is 0°).
The performance curves at inlet of second stage (K202) exhibits a minimum flow of about 78000 Nm3/h, while the rated capacity is 119033 Nm3/h. This corresponds to the theoretical compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 34.4 %. The regulation of capacity is achieved by means of IGV (inlet guide vanes) which are rotated to about 40° (at rated capacity is 0°).
The performance curves at inlet of second stage (K201) exhibits a minimum flow of about 157000 Nm3/h, while the rated capacity is 235200 Nm3/h. This corresponds to the theoretical compressor surge point leading to a capacity turndown (at constant discharge pressure) of about 33.2 %. The regulation of capacity is achieved by means of IGV (inlet guide vanes) which are rotated to about 50° (at rated capacity is 0°).

In conclusion, for the oxy-fuel combustion case, being the turndown of the overall compression unit equal to 25%, it is not possible to achieve 50% of turndown capacity relying upon the design of machine itself and therefore the solutions to accomplish this requirement are the spillback lines or duplication of units CK204, K-201 and K-202.

4 Reliability and operability

4.1 Reliability and Availability of Rolls-Royce compressors

As a general rule, compressors in CO_2 service are not expected to be worse than for other gas applications with respect to reliability and availability.
Reliability data for compressors in any gas service is difficult to obtain. Many customers don't track the issues and those that do keep the data to themselves. R-R do have some reliability/availability data for their compressors (in a variety of services) however unfortunately they would only be able to share this under a Non Disclosure Agreement which R-R imagine is not particularly useful as the present study will be published.
R-R however comments that their gas compressors are extremely reliable and often have very long lives (they state to have known of examples of R-R compressors in continuous operation for 50 years+). This is also why there is limited customer feedback/reliability data available because customers generally only come back to manufacturer if there are problems.
R-R also stated that including a long term service agreement for a compressor adds only a minimal amount more (a few 10s of thousands of $ vs. the 1+million they charge for the gas turbine.) to the price quotation. This provides a good indication of the reliability of this hardware.

4.2 Operability of Rolls-Royce compressors

As stated in the above paragraph, RR compressors have extremely high reliability and, as a result, customer feedback regarding operating issues is limited. Field service engineers have however commented that the best way to avoid operating/maintenance issues is to do a full set of commissioning tests and also try to ensure that the customer is really going to operate at the points the compressor is designed for (noting however that more flexible operation can be designed for, such as part load conditions for CCS applications). Keeping process flows clean also helps to prolong the life of components which are more prone to issues (bearings, seals, etc.)

RR centrifugal inline compressors have an advantage over integrally geared compressors with respect to maintenance access, as compressor bundles can be removed for easy inspection/maintenance from the end (barrels) or from the top (horizontally split casings) generally without disturbing the process piping.

4.3 Reliability and Availability of MAN Diesel & Turbo compressors

Similarly to Rolls-Royce, information related to availability of the compressors is not readily available, because malfunctions or problems are track but not always disclosed to compressor manufacturer.
However as a rough idea, for integrally geared centrifugal CO2 compressor, it is expected an availability figure of around 97%.

4.4 Operability of MAN Diesel & Turbo compressors

As stated in the above paragraph, MAN Diesel & Turbo compressors have extremely high reliability and as a result customer feedback regarding operating issues is limited.
For instance the Customer Plant reliability superintendent of Dakota Gasification Company has commented that they initially did experience problems with high pressure stages seals and the Turbolog control system. After the correction of such deficiencies by MAN Diesel & Turbo the current operation of the machines has been very reliable and consistent.

4.5 Ramp up / ramp down of MAN Diesel & Turbo compressors

Regardless the type of driver (electric motor or steam turbine) the compressor will run at constant speed and the capacity control is achieved by means of inlet guide vanes.
For plant stability the inlet guide vanes are stroke from minimum to maximum setting and vice-versa in about 30 seconds.
The start up for a motor can be as short as 15-20 seconds to full speed in the unloaded condition, it will than take 20 seconds to close the bypass valves to start delivering at minimum IGV setting and than a further 30 seconds to reach full load.
The ramp down is expected to be within one minute, depending on size of machine, inertia forces and load.
Start-up time of a steam turbine depends upon whether it is cold, warm, or hot and it will have to go through its start-up ramp procedures which are machine and live steam conditions dependant.

4.6 Ramp up / ramp down of Rolls-Royce compressors

In single speed motor applications ramp up to full speed (0 to 10000 rpm) can be completed in just 8 seconds, and ramp up to full load (26.000 HP) has been obtained

in 17 seconds. As long as the thrust and journal bearing lube oil supply is at the specified pressure and temperature, there are no adverse effects to the compressor.
Ramp down times are typically less than one minute (30 to 45 seconds), but can take up to 2 minutes for a complete stop. Rolls-Royce prefers coast-down decelerations where frictional and inertial forces along with the gas load on the compressor are controlling the ramp down.

5 Manufacturing capacity of industry with respect to CCS capacity projection

The contacted compressors' Vendors were asked to provide general information regarding their manufacturing capacity for large compressor units like the ones proposed for the present study.

Some feedback was received from MAN Diesel & Turbo and Rolls Royce.

MAN Diesel & Turbo stated they are currently building up to 12 large (RG160) compressor units per year and numerous smaller sized RG machines. With 5 European manufacturing facilities and a full turbo-machinery programme, MDT believe they are well positioned to adjust their production slate to accommodate higher demands for particular machine types, should the business case demand it.

Rolls Royce stated their current production capacity is approximately 30 units per year. If the market for CCS is attractive they may seek to increase the production capacity to help accommodate this demand.

Based on the IEA Blue Map Scenario, the projection of the number of units needed over time can be summarised as follows:

- Approx 40 compressor units per year up to 2030.
- Approx 100 compressor units per year from 2030 to 2050.

Considering the feedback from two of the main Suppliers in the market and the potential contribution of other Suppliers to the overall manufacturing capacity, it can be concluded that industry has the capability to accommodate future potential demand in case CCS in Power Generation will effectively be a leading strategy in reducing the CO_2 emissions.

6 Contrast with operating envelopes for the most promising compression strategies

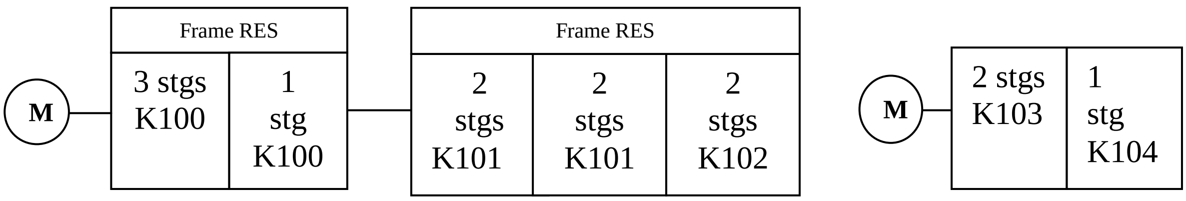
This section defines the impacts and the possible modifications of the machinery selection, due to changes of some process parameters, as discussed in section C, for the following process technologies:

a- Pre combustion – Case A2
b- Pre-combustion – Case D2b
c- Post-combustion – Case B2A
d- Post-combustion – Case B2B
e- Oxy-fuel – Case C3

6.1 Machinery selection by Rolls-Royce

PRE-COMBUSTION – Case A2

Train n° 1 (two compressor packages running in series)



TOTAL BRAKE POWER (WHOLE PROCESS): **37510 KW**

The main changes in respect to the base machine selection for the case PRE-COMBUSTION Case A2 are the following:

Process K100

- The rotating speed remains the same, while the absorbed shaft power slightly increases from 5630 KW to 5760 KW.

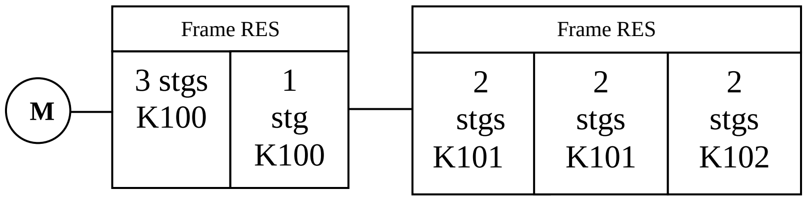
Process K101

- The absorbed shaft power reduces from 15880 KW to 12720 KW.

No impact is foreseen on processes K102, K103 and K104 that remain as per original base case selection.

PRE-COMBUSTION – Case D2B

Train n° 1 (one single compressor package)



TOTAL BRAKE POWER (WHOLE PROCESS): **31130 KW**

The main changes in respects to the base machine selection for the case PRE-COMBUSTION Case D2B are the following:

<u>Process K100</u>

- The rotating speed slightly reduces from 4800 RPM to 4700 RPM, while the absorbed power slightly increases from 5630 KW to 5700 KW .

<u>Process K101</u>

- The absorbed power increases from 15880 KW to 17530 KW.

<u>Process K102</u>

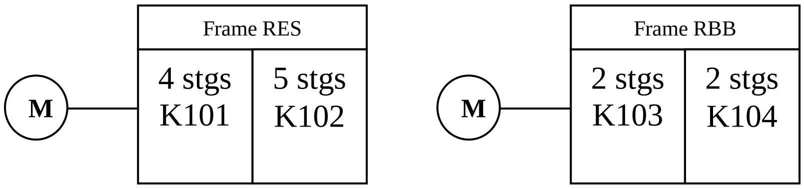
- The absorbed power increases from 7230 KW to 7910 KW.

The processes K103 and K104 are eliminated.

POST-COMBUSTION PROCESS (INTER-COOLING AS SPECIFIED) – Case B2A

Train n° 1 (two compressor packages running in series)

Train n° 2 (two compressor packages running in series)



Train n°1 and train n° 2 run in parallel

TOTAL BRAKE POWER (WHOLE PROCESS): **46044 KW**

The main changes in respects to the base machine selection for the case POST-COMBUSTION (INTER-COOLING AS SPECIFIED) Case B2A are the following:

<u>Process K101</u>
- The number of stages decreases from 5 to 4 as well as the frame size changes from RFS to RES.
- The rotating speed increases from 4200 RPM to 4900 RPM, while the absorbed power slightly decreases from 9366 Kw to 8416 Kw.

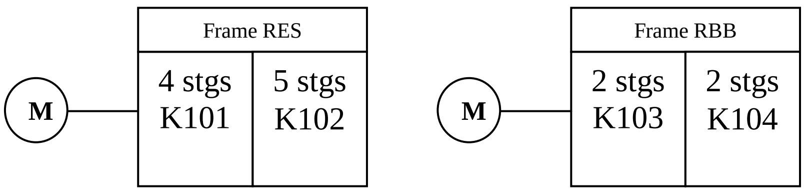
<u>Process K102</u>
- Within the frame RES the rotating speed increases from 4200 RPM to 4900 RPM, while the absorbed power decreases from 10586 Kw to 9329 Kw.

There is no impact on processes K103 and K104 that remains unchanged.

POST-COMBUSTION PROCESS (INTER-COOLING AS SPECIFIED) – Case B2B

Train n° 1 (two compressor packages running in series)

Train n° 2 (two compressor packages running in series)



Train n°1 and train n° 2 run in parallel

TOTAL BRAKE POWER (WHOLE PROCESS): **45930 KW**

The main changes in respects to the base machine selection for the case POST-COMBUSTION (INTER-COOLING AS SPECIFIED) Case B2B are the following:

Process K101

- The number of stages decreases from 5 to 4 as well as the frame size changes from RFS to RES.
- The rotating speed increases from 4200 RPM to 4900 RPM, while the absorbed power slightly decreases from 9366 KW to 8368 KW.

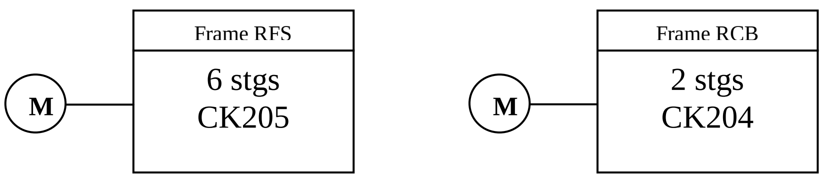
Process K102

- Within the frame RES the rotating speed increases from 4200 RPM to 4900 RPM, while the absorbed power decreases from 10586 KW to 9320 KW.

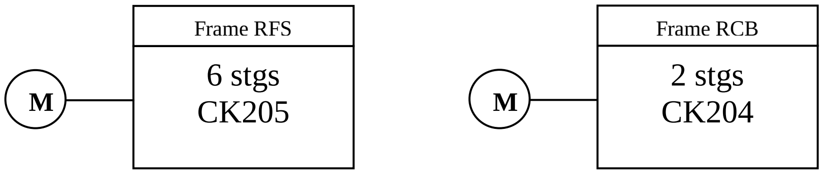
There is no impact on processes K103 and K104 that remains unchanged.

OXY-FUEL PROCESS (INTER-COOLING AS SPECIFIED) – Case C3

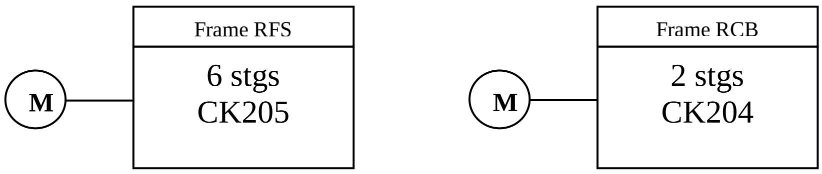
Train n° 1 (two compressor packages running in series)



Train n° 2 (two compressor packages running in series)

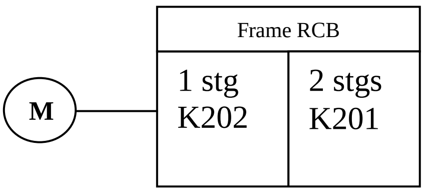


Train n° 3 (two compressor packages running in series)



Train n°1, n° 2 and train n° 3 run in parallel

Train n° 4



TOTAL BRAKE POWER (WHOLE PROCESS): **68631 KW**

The main changes in respects to the base machine selection for the case OXY-FUEL (INTER-COOLING AS SPECIFIED) Case C3 are the following:

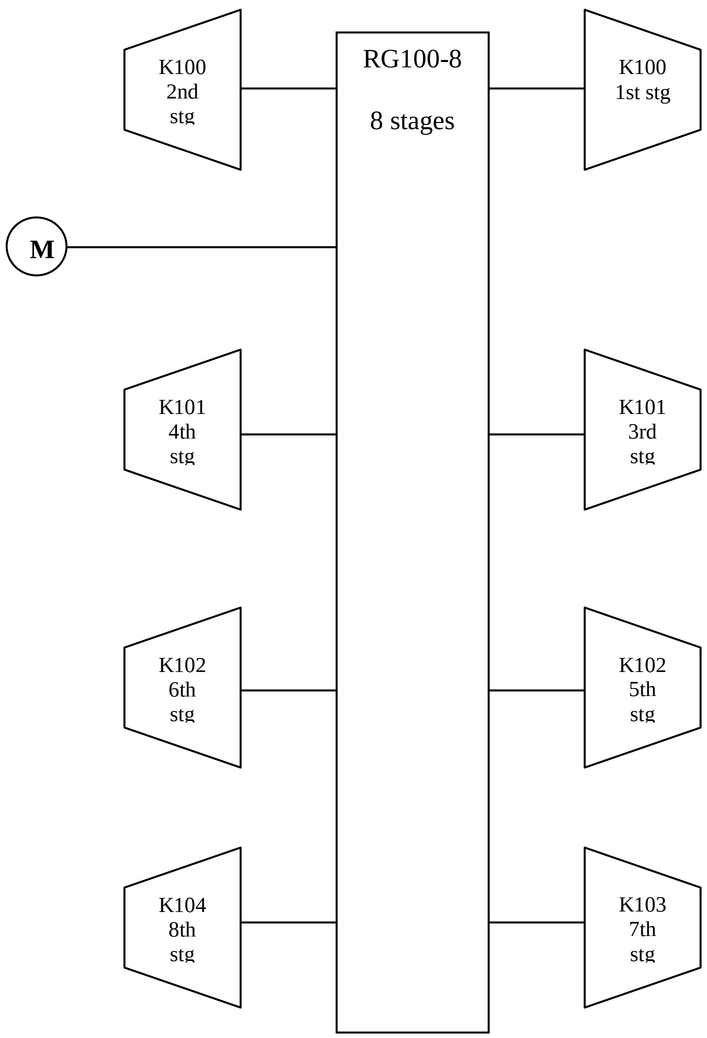
<u>Process K202</u>
- Within the frame RCB the rotating speed slightly decreases from 10427 RPM to 10393 RPM, while the absorbed power increases from 2998 KW to 3447 KW.

<u>Process K201</u>
- The number of stages reduces from 3 to 2 while the rotating speed slightly decreases from 10427 RPM to 10393 RPM.
- The absorbed power decreases from 16982 KW to 11067 KW.

6.2 Machinery selection by MAN Diesel & Turbo

PRE-COMBUSTION – Case A2

Train n° 1



TOTAL BRAKE POWER (WHOLE PROCESS): **41926 KW**

The main changes in respect to the base machine selection for the case PRE-COMBUSTION Case A2 are the following:

Process K100
- The rotating speed slightly decreases from 5655 rpm to 5462 rpm, while the absorbed power slightly increases from 6056 KW to 6161 KW.

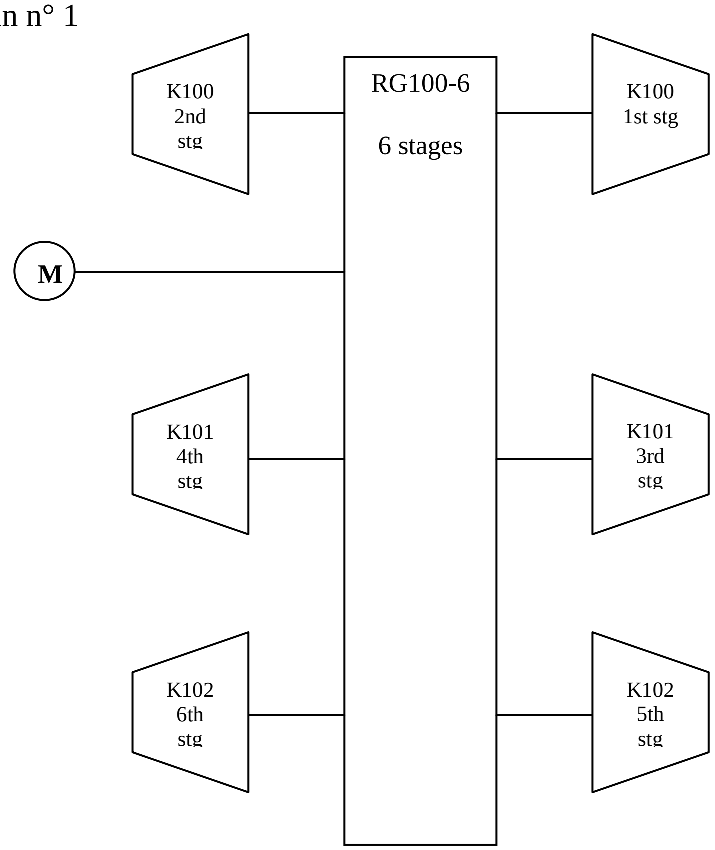
Process K101
- The rotating speed slightly increases from 4351 rpm to 4556 rpm, while the absorbed power reduces from 10809 KW to 8286 KW.

Process K102
- The rotating speed slightly decreases from 7140 rpm to 6859 rpm, while the absorbed power increases from 13619 KW to 14007 KW.

Process K103 / K104
- No changes.

PRE-COMBUSTION – Case D2B

Train n° 1



TOTAL BRAKE POWER (WHOLE PROCESS): **31431 KW**

The main changes in respect to the base machine selection for the case PRE-COMBUSTION Case D2B are the following:

The frame size incorporates 6 stages instead of 8 stages.

Process K100

- The rotating speed slightly decreases from 5655 rpm to 5462 rpm, while the absorbed power slightly increases from 6056 KW to 6161 KW.

Process K101

- The rotating speed slightly increases from 4351 rpm to 4556 rpm, while the absorbed power reduces from 10809 KW to 8286 KW.

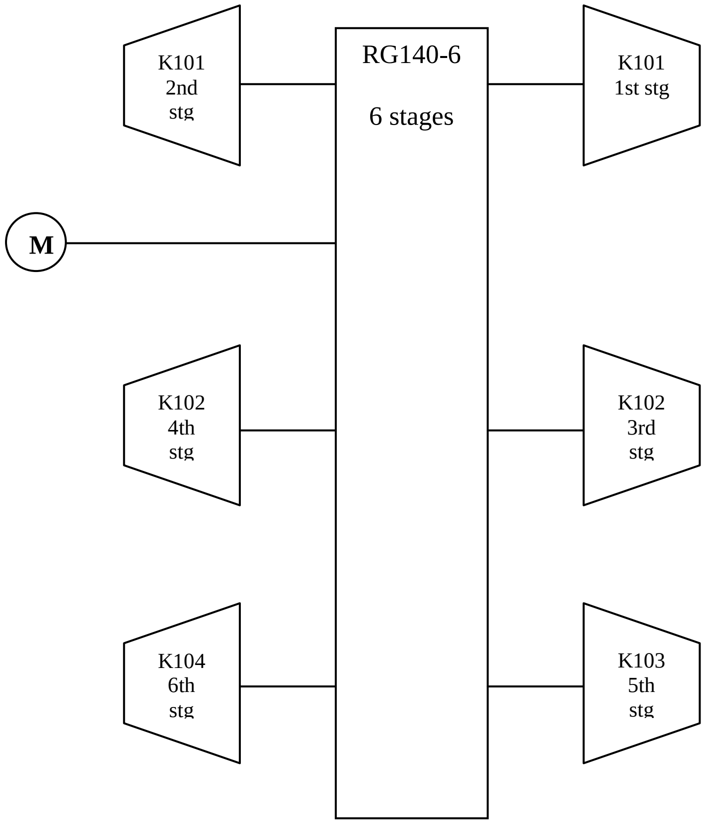
Process K102

- The rotating speed slightly decreases from 7140 rpm to 7095 rpm, while the absorbed power increases from 13619 KW to 16984 KW.

Process K103 / K104 are eliminated.

POST-COMBUSTION (OPTIMISED INTER-COOLING) – Case B2A

Train n° 1



TOTAL BRAKE POWER (WHOLE PROCESS): **44816 KW**

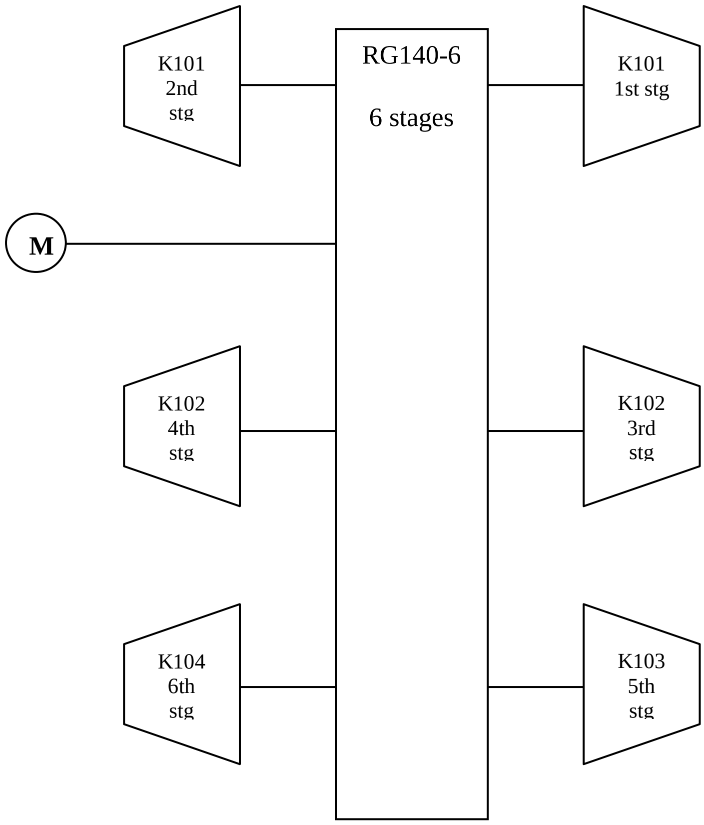
The main changes with respect to the base machine selection for the case POST-COMBUSTION (OPTIMISED INTER-COOLING) Case B2A are the following:

Process K101
- The rotating speed slightly decreases from 4058 rpm to 3818 rpm, while the absorbed power reduces from 18052 KW to 14059 KW.

Process K102/K103/K104
- No changes.

POST-COMBUSTION (OPTIMISED INTER-COOLING) – Case B2B

Train n° 1



TOTAL BRAKE POWER (WHOLE PROCESS): **44758 KW**

The main changes in respect to the base machine selection for the case POST-COMBUSTION (OPTIMISED INTER-COOLING) Case B2B are the following:

<u>Process K101</u>

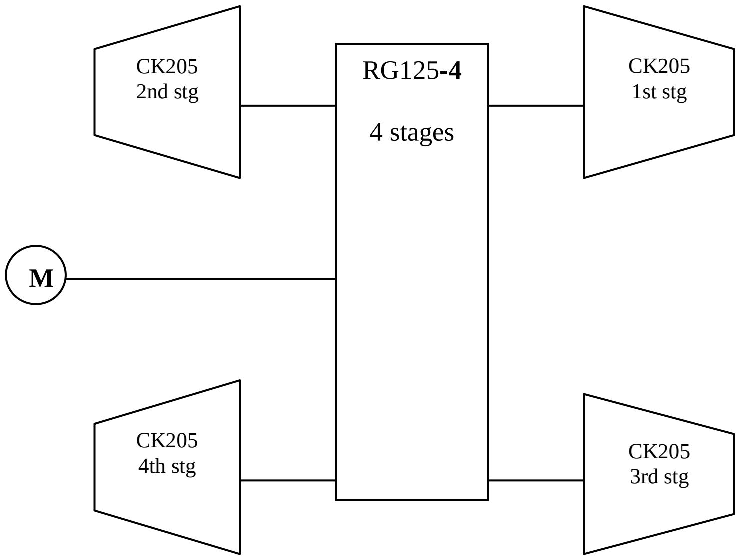
- The rotating speed slightly decreases from 4058 rpm to 3823 rpm, while the absorbed power reduces from 18052 KW to 14001 KW.

<u>Process K102/K103/K104</u>

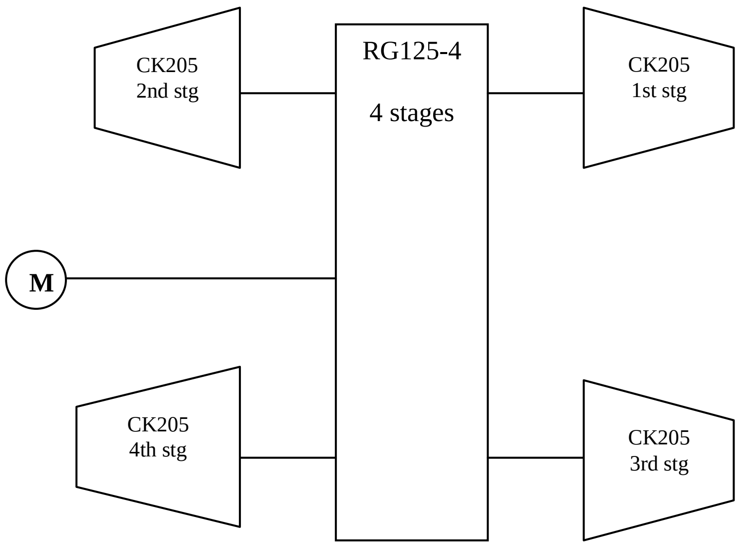
- No changes.

OXY-FUEL (OPTIMISED INTER-COOLING) – Case C3

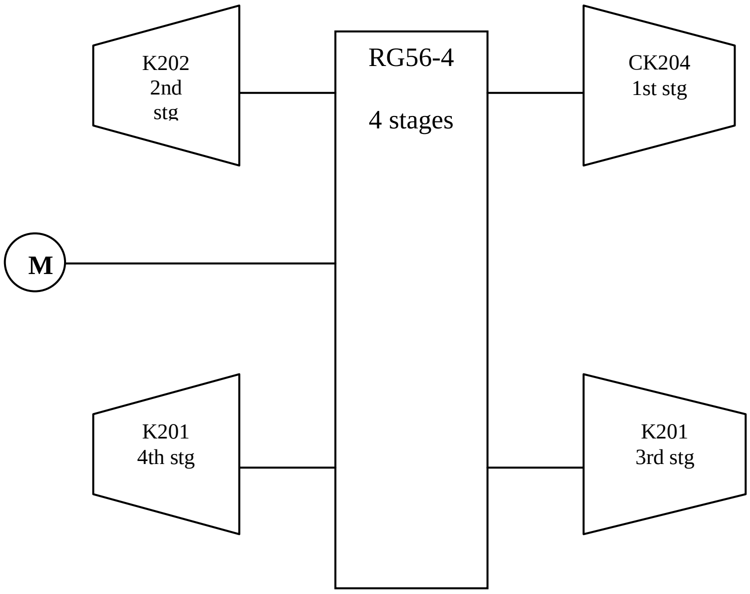
Train n° 1



Train n° 2



Train n° 3



Train n°1 and train n° 2 run in parallel, train 3 runs in series to train 1-2.

TOTAL BRAKE POWER (WHOLE PROCESS): **61745 KW**

The main changes in respect to the base machine selection for the case OXY-FUEL (OPTIMISED INTER-COOLING) Case C3 are the following:

<u>Process CK205</u>
- No changes.

<u>Process CK204/K202</u>
- No changes.

<u>Process K201</u>
- Number of stages reduces of one stage. The rotating speed slightly decreases from 10832 rpm to 10633 rpm, while the absorbed power reduces from 27965 KW to 25375 KW.

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SECTION E

NOVEL CONCEPTS FOR CO_2 COMPRESSION

I N D E X

1 Introduction

The main purpose of this section is to assess novel CO_2 compression concepts, which may find application in Carbon Capture and Storage plants in next few years.

Firstly, the Ramgen compression concept is investigated, mainly considering possible advantages and disadvantages of its potential application to CO_2 compression in CCS plants. A detailed description of the device, its state of development and its key characteristics are described in this section. A “strategy” for incorporating it in a typical captured CO_2 compression system, the post combustion alternative, is also shown. Furthermore, the range of capacities to which such a device might be applied is investigated.

In addition, an alternative novel compression is assessed, based on a first stage with a single train axial compressor that can handle a much higher flow rate than the centrifugal compressors, having also higher efficiencies.

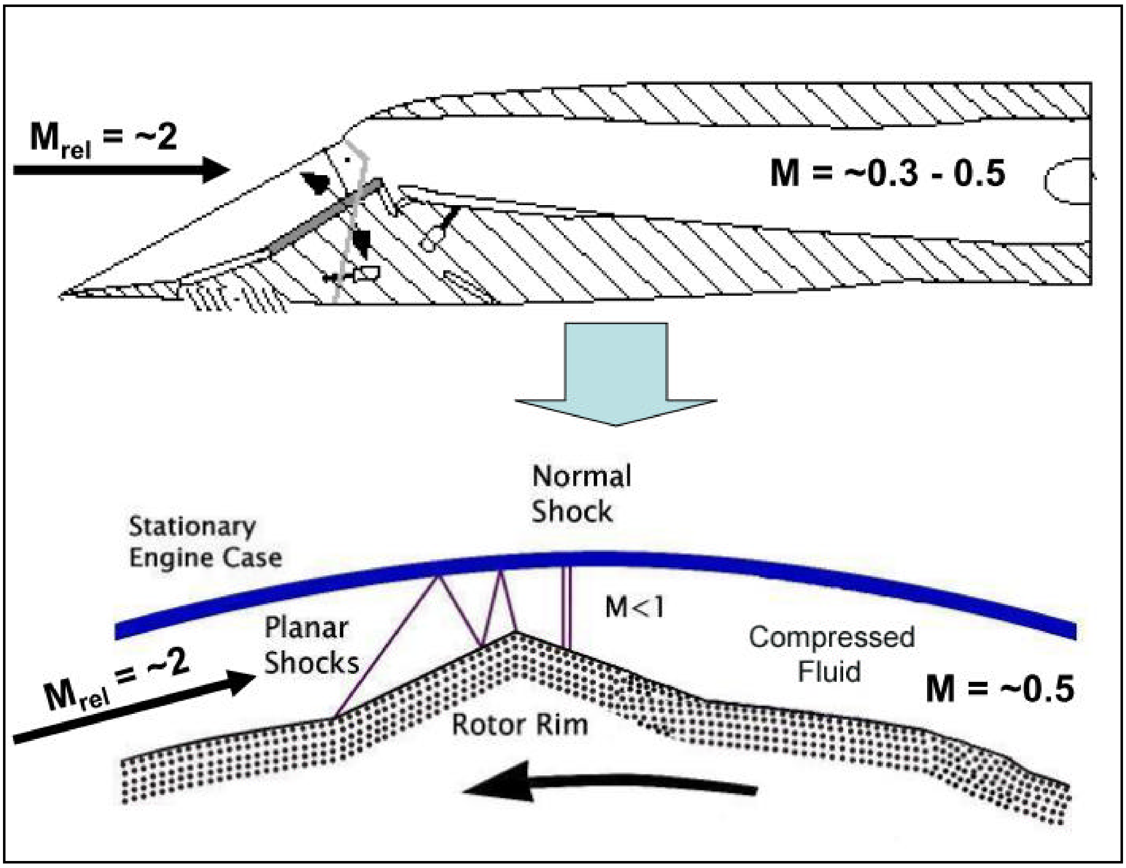
2 Ramgen technology

2.1 Overview of the compression concept

Ramgen is a novel technology based on the supersonic shock wave compression. It uses the same principle as a supersonic aircraft, where the engine forward motion is used to compress the air.

In fact, at supersonic speeds, air incomes the engine, then flows around an obstructing centre-body that creates a ramming effect when the air is forced through the area between the body and the engine sidewall (reference shall be made to Figure 2-1).

Figure 2-1 Supersonic aircraft engine inlet cross section compared to Ramgen rotor

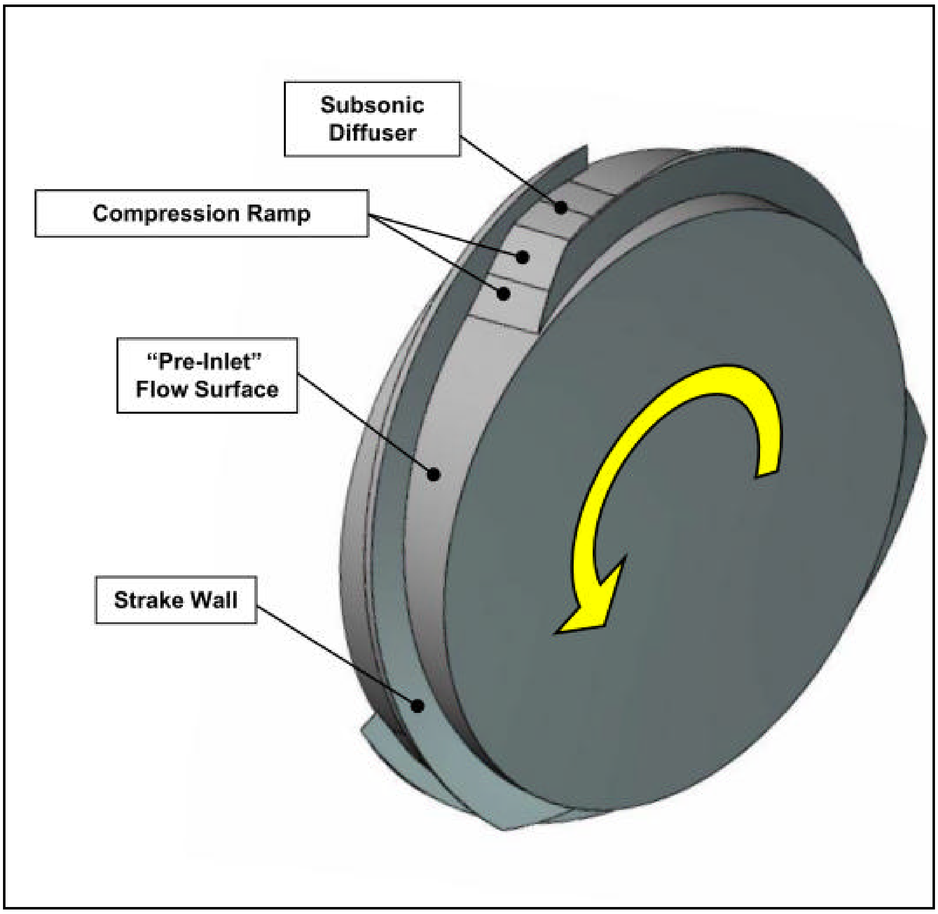


The shock waves create a sudden compression (the pressure increases immediately), after which the airflow is slowed down to subsonic speed.

In the Ramgen compressor, a rotating disc (see Figure 2-2) simulates forward motion of the aircraft. It spins at high speed to create a supersonic effect like the centre-body in a supersonic aircraft.

The fluid enters through a common inlet, flows into the annular space between the disc and the casing, where the three raised sections create shock waves. The shock waves generate a pressure increase, compressing the fluid at the required pressure level.

Figure 2-2: Rotating disc



The compression ratio is highly dependent on:

- the intensity of the shock wave, which increases with the Mach number;
- where the oblique shock wave takes place.

The Mach number increases with the increase of the speed of the rotating disc.

The compression is developed into two stages, with a pressure ratio of approximately 10: 1 and can be fitted with both an inter-cooler and after-cooler, depending on the application. Typically, the temperature rise is approximately 200 °C over the compressor stage inlet temperature, thus the relevant heat may be recovered in other units of the plant.

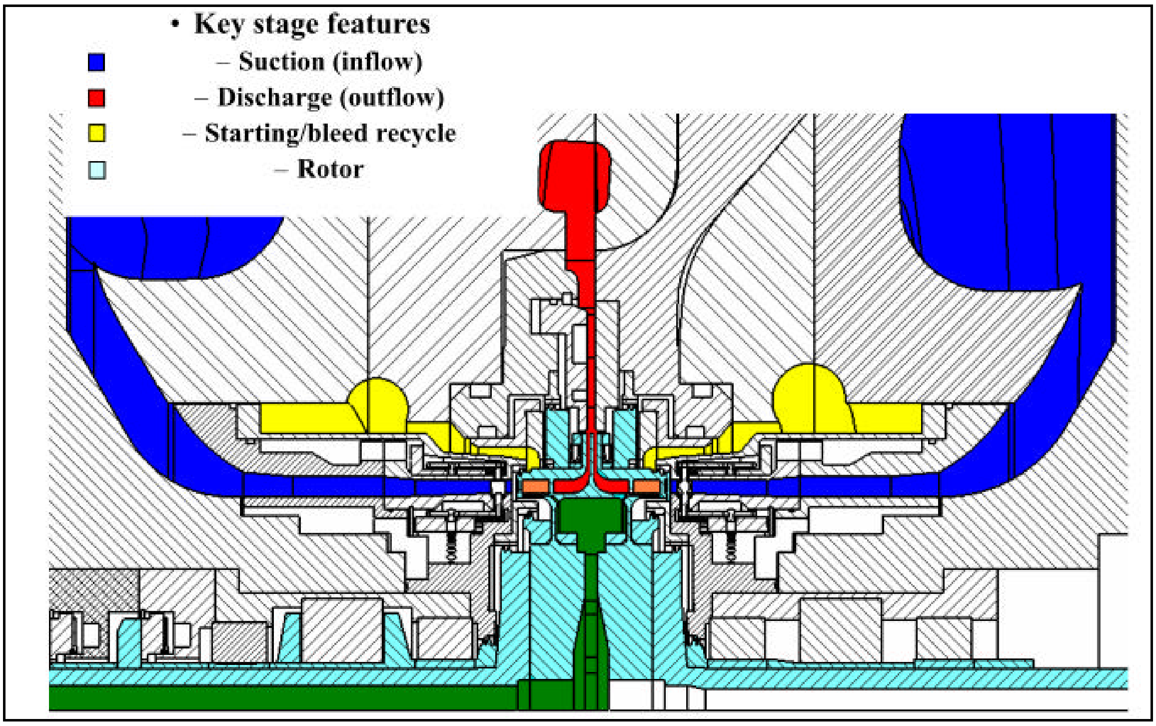
Ramgen claims high efficiency for their compressors, because of the relatively simple design, having low number of leading edges that reduce the drag and therefore minimize the losses.

Another advantage over other compressor types is the high pressure ratio per stage, which reduces footprint requirement and cost, and the possibility to use the high-grade waste heat due to the high discharge temperature.

2.2 Design features

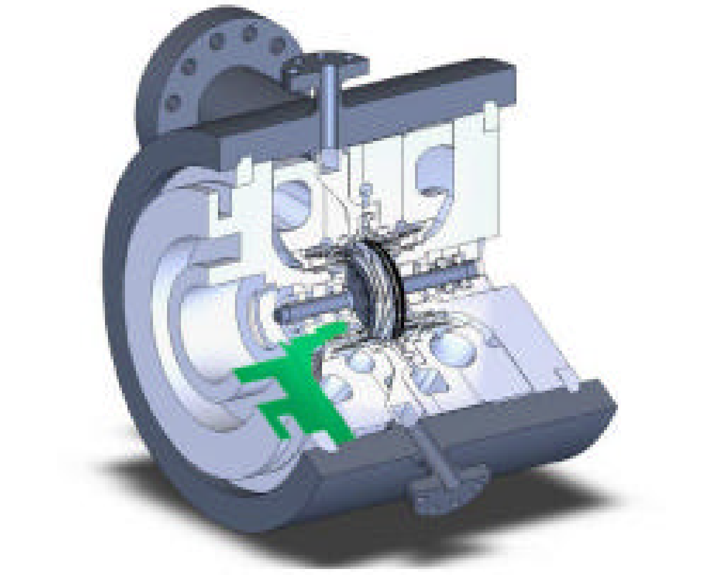
The rotor configuration is a double-suction design, with a common radial discharge (see Figure 2-3).

Figure 2-3 Typical cross section of HP stage back-to-back configuration



A general view of the Ramgen HP stage is outlined in Figure 2-4.

Figure 2-4 Ramgen HP stage



Ramgen Discrete Drive HP-Stage

Each stage of compression is coupled to its drive, so to optimise the speed, using the step-up external gearbox. Doing so, there is the possibility to regulate the LP stage from the HP stage and use as an option the VFD (variable frequency drive) for those cases where the plant resistance load curve is variable.

The design of compressor incorporates IGV's (inlet guide vanes) in both suction flow path that are used to provide pre-swirl and capacity turn-down, that Ramgen expects to target at 30% (at constant discharge pressure).

The size selection is done using the compressor charts shown for each stage (LP and HP). Figure 2-5 illustrates the various parameters of the chart, in which the inlet capacity and differential head allow to select the best frame. Once the frame (rotor diameter) is selected, the operating speed of that frame is calculated to achieve the requested differential head.
An external gearbox is purchased to match the calculated speed and power required.

Figure 2-5 Compressor selection

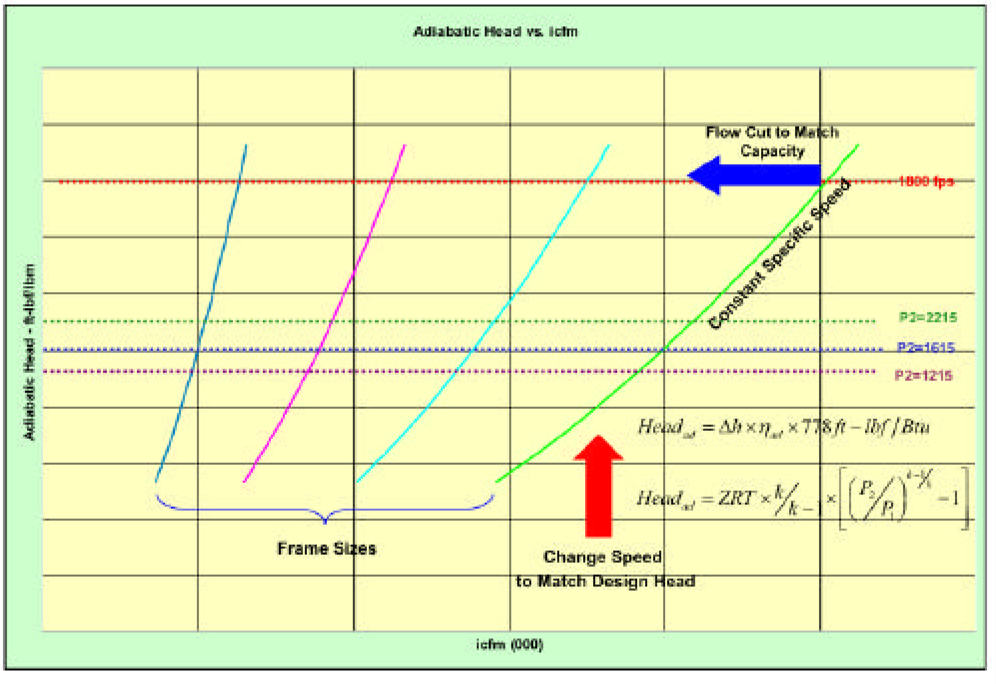


Figure 2-6 Compressor chart for LP stage

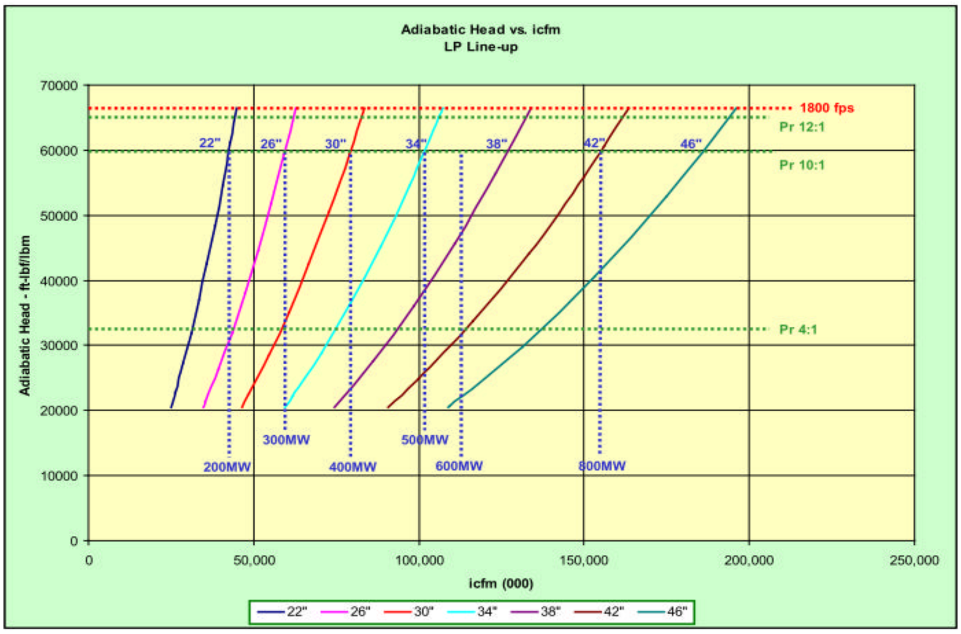
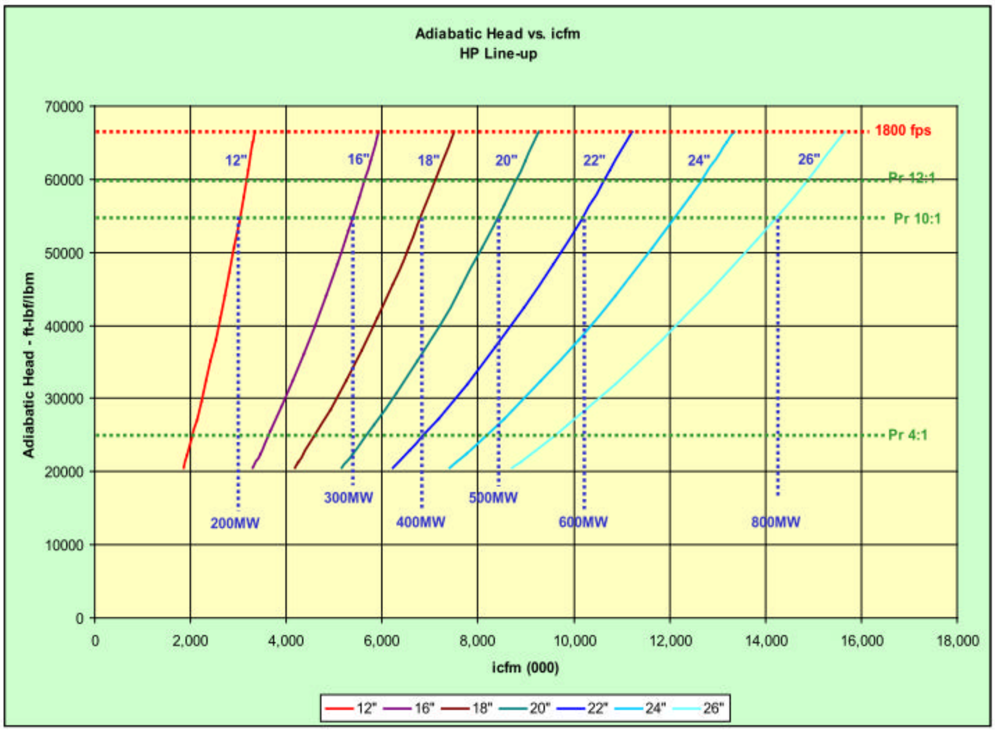


Figure 2-7 Compressor chart for HP stage



2.3 Expected performance

Ramgen calculation and machine selection is based on the specification of the study, i.e. 12°C cooling water temperature, CO_2 dehydration at 50 ppmv and the possibility to adopt a 1 x 100% configuration, if available. CO_2 dehydration is not intended to be in the Ramgen scope, however it is noted that the water content spec and consequent design of the Dryer may affect compressor design as well (e.g. dried CO_2 recycle for regeneration).

The specified operating envelope is the same as the post combustion base case (case B0, ref. Section B). However, as far as dehydration level specification is concerned, Ramgen challenged the basic assumption made for this study, particularly the application of desiccants to achieve 50 ppmv in the post combustion case. Making reference to Appendix 1 of the present report, where the uncertainties on the required moisture levels are addressed and the available technologies for CO2 dehydration are introduced, Foster Wheeler, Ramgen and IEA GHG agreed that the performance of the compressor are estimated for the following cases:

- Dehydration through a TEG absorption unit (no CO_2 recycle needed), with a minimum achievable moisture spec of 30 ppmv.
- Dehydration through desiccants (CO_2 recycle needed for beds regeneration) to achieve moisture specification below 10 ppmv.

Two stages compressor have been selected and relevant performances are reported in Table 2-1 and Table 2-2.

Table 2-1: Ramgen performance and cost, Dehydration by means of TEG.

Platform	**LP**	**HP**	**Total**
Model	38	26	
Stage	1st	2nd	
Quantity	1	1	
Barometric – bara	1.0135	1.0135	
Inlet pressure – bara	1.62	12.90	
Inlet temperature - °C	37.8	24	
Humidity –RH %	100	2.2	
Outlet pressure – bara	14	111.5	111
Pressure ratio	8.6	8.6	68.5
Stage efficiency – isentropic%	86.5	86.5	
Coolant temperature - °C	12	12	
Approach temperature - °C	12	28	
Mole weight	42.955	44.007	
K - ratio of specific heats	1.291	1.376	
Z – compressibility	0.993	0.932	
Mass flow (wet) – kg/hr	556451	547007	
CO2 – kg/hr	546960	546960	
Volume flow – m3/hr	71161	7690	
Discharge temperature - °C	236.5	225.3	
Total power – Kw	29988	25892	**55880**
Motor power – Kw	33000	28500	
Polytropic efficiency - %	89.3	89.8	
Heat recovery			
- Potential KJ/Kg CO2	212.6	357.3	569.9

Table 2-2: Ramgen performance and cost, Dehydration by means of desiccants.

Platform	**LP**	**HP**	**Total**
Model	40	26	
Stage	1st	2nd	
Quantity	1	1	
Barometric – bara	1.0135	1.0135	
Inlet pressure – bara	1.62	12.65	
Inlet temperature - °C	35.4	24	
Humidity –RH %	100	0.4	
Outlet pressure – bara	14.3	111.5	111
Pressure ratio	8.8	8.8	68.5
Stage efficiency – isentropic%	86.3	86.3	
Coolant temperature - °C	12	12	
Approach temperature - °C	12	28	
Mole weight	43.085	44.008	
K - ratio of specific heats	1.292	1.374	
Z – compressibility	0.993	0.933	
Mass flow (wet) – kg/hr	617410	546836	
CO2 – kg/hr	608198	546798	
Volume flow – m3/hr	78094	7852	
Discharge temperature - °C	235.6	227.6	
Total power – Kw	33395	26267	**59662**
Motor power – Kw	36800	28900	
Polytropic efficiency - %	89.2	89.7	
Heat recovery			
- Potential KJ/Kg CO2	211.9	323.7	535.6

Ramgen have also provided budget cost of the proposed selection, which shall be deemed as preliminary only. As per the previous cost information, only an indication of the expected specific cost range is included in this report. Based on the budgetary information received, the specific cost for the Ramgen compressor is expected to be in the range 170 ÷ 280 €/kW.

2.4 Strategy for integration in the post combustion capture plant

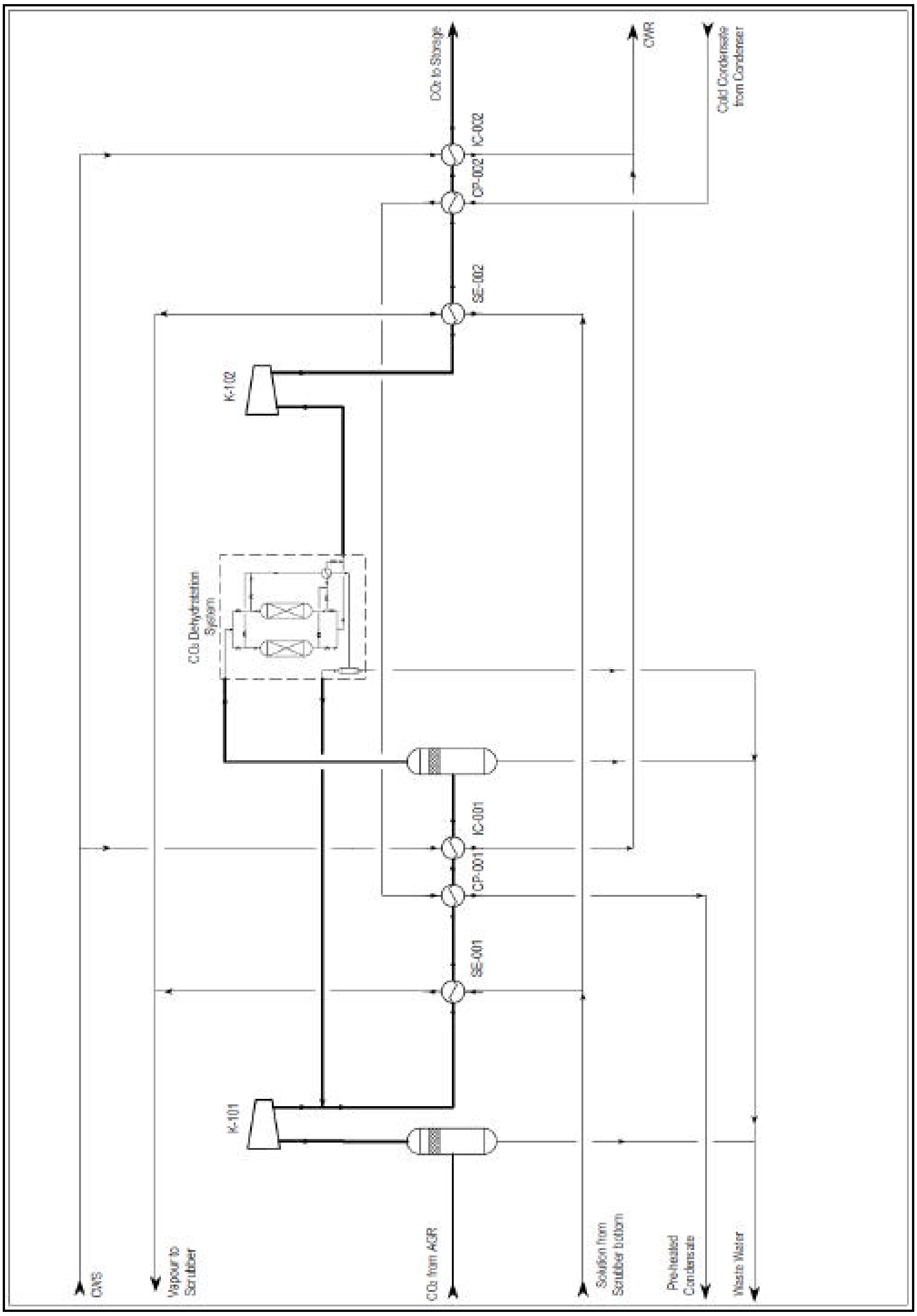
Ramgen proposed a two-stage compressor configuration, with a relatively high pressure ratio (approximately 9:1), which leads to a large amount of recoverable compression heat available at both the inter-cooler and the after-cooler.

FW and IEA GHG agreed that the best case for incorporating the Ramgen compressor into the plant is the post combustion CCS alternative; the objective is to demonstrate how to use the heat as best as possible.

The option considered for the present study is to use the recoverable compression heat, which is available at relatively high temperatures (around 230 °C), in the stripper reboiler to the maximum extent and then for the ST condensate preheating. The latter thermal integration is already incorporated in the post combustion capture reference case, as described in section B. From the energetic point of view this is the optimum configuration, as it allows recovering most of the low-grade heat generally available from the CO_2 compression.

The process flow scheme of the proposed solution is shown in Figure 2-8.

Figure 2-8 Ramgen compressor integration into the post combustion capture process



An evaluation of the overall performance impact of the Ramgen compressor in the plant (see Table 2-3) has been performed through a comparison in the post-combustion case with the integrally geared centrifugal compressor, proposed by MAN Diesel & Turbo, and with the in-line machines configuration with optimised inter-cooling, proposed by Rolls Royce, the latter being the minimum power demanding scheme among the centrifugal options (ref. 5.2.1).

A process simulation of the three cases allows comparing the relevant equivalent consumptions (ref. Section B), which take into account the effects on the plant electrical power output of the factors listed below, according to the methodology for the assessment of the compression strategies shown in section C:

- Thermal integration of the CO_2 compression with the amine regeneration system and consequent changes in steam demand from the Power Island;
- Thermal integration of the CO_2 compression with the ST condensate preheating system and consequent changes in the steam consumption within the Power Island for this service;
- Cooling Water demand.

The comparative analysis has been undertaken on the basis of a moisture spec of 50 ppmv, achieved through the absorption process of a TEG unit. In fact, it is recognised that with, the 10% recycle assumed for the basic Dryer configuration (i.e. adsorption in desiccant beds, ref. Appendix 1), Ramgen power penalty would be aggravated by the added mass flow compressed from lower pressures, the result of fewer discrete stages to work with. Having assessed that, for the given basic moisture specification, a TEG system can be successfully applied to both Ramgen compressor and conventional machines, requiring no CO_2 recycle (ref. ref. Appendix 1), it is acceptable to deviate from the basic configuration selected for the study, as far as Ramgen performance evaluation is regarded.
Furthermore, Ramgen compressor shows unique potential for combination with the HOC (Heat Of Compression) Drying system, developed by SPX, to achieve moisture content even lower than 10 ppmv in the dried stream (ref. Appendix 1), without any recycle for adsorption bed regeneration. Hence, Ramgen compression concept can potentially work with low moisture spec (e.g. as required in the CO_2 purification process of the oxy-fuel combustion technology) without major energetic penalties.

The results of the comparative analysis are presented in Table 2-3, where the Ramgen consumption deltas with respect to the centrifugal compressors are reported, i.e. a negative figure indicates a lower demand of the Ramgen compressor.

Table 2-3: Ramgen performance delta with respect to centrifugal compressors

Ramgen Thermal Integration with the Power Plant / CO_2 capture unit		
	Comparison with integra-gear machine (inter-cooling as specified)	Comparison with in-line machine (optimised inter-cooling)
Steam cons. for Condensate Pre-heating	+ 9.4 MW_{th} ? + 2.5 MW_e	-7.1 MW_{th} ? - 1.9 MW_e
Steam cons. for MEA Reboiling	- 35.3 MW_{th} ? - 9.3 MW_e	- 35.3 MW_{th} ? - 9.3 MW_e
Cooling water		
CW consumption	~ 0.0 t/h ? ~ 0.0 MW_e	- 1905 t/h ? - 0.2 MW_e
Compressor Electrical Consumption		
Overall electrical consumption difference	+ 5.4 MW_e	+11.1 MW_e
Overall Plant Electrical Power Gap		
TOTAL	**- 1.4 MW_e**	**- 0.3 MW_e**

Table 2-3 shows that the Ramgen compression strategy has potential to offer not only low cost and simplicity, but also a lower equivalent power demand of the whole system. It has to be noted that the net reduction of the equivalent compression parasitic load is diminished but not overtaken even when compared to the in-line machines configuration with optimised inter-cooling, the minimum power demanding scheme among the centrifugal options.

As far as this comparison exercise is concerned, the two following factors should be taken into consideration:

- The low cooling water temperature assumed for this study encourages compression staging rather than the de-staging approach proposed by Ramgen. As a matter of fact, Ramgen provided performance figures also with cooling water at 30°C, showing a parasitic load increase of 3.5% only with respect to the 12°C case, whereas for centrifugal compressor the expected penalty would be approx 5÷6%.
- The significant potential for high grade heat recovery offered by Ramgen is not fully exploited in the present analysis for the post combustion process, as the MEA stripper reboiler operates at about 120 °C, i.e. over 100°C less than compression discharge temperature

Regarding other possible integration strategies, the option of using the CO_2 compression heat through either an Organic Rankine cycle or a Kalina cycle in lieu of the ST condensate preheating has been assessed as well. However, the CO_2 compression is basically the only source of waste heat within the post combustion scheme. If the thermal integration between the CO_2 compression and Power Island were removed to pursue the addition of such low temperature cycles, then the steam consumption within the Power Island would increase due to the lower ST condensate temperature at the preheating train inlet. Having assessed the high energetic value of the steam turbine extractions, even at low pressure (ref. Section B), it is believed that the low temperature cycle option should not be considered for heat recovery from the CO_2 compression. Also, it is noted that this solution would tend to offset cost and simplicity advantages which the Ramgen compressor typically offers.

2.5 State of development of the technology

Despite the promising characteristics, the Ramgen compressor concept is not yet a proven technology. Further development and testing are required to demonstrate its capability at a commercial scale.

Ramgen has recently developed a test program on the frame HP-16, which can support a CCS power plant in the capacity range of 200 - 250MW$_e$. These tests are scheduled to start on 2nd quarter 2011.
On that machine Ramgen expect to be able to offer commercial performance guarantees and terms by 1st quarter 2012.

Further development activities will then be carried out on HP-32 and LP-48 as required. These are the largest anticipated sizes for approximately 800MW$_e$ CCS applications, for HP and LP stages respectively.

2.6 Effects on compression strategies

General understanding of Ramgen compression concept is that the proposed 10:1 stage ratio leads to a techno-economic optimum for the machine.

Even in the post combustion base case, the overall pressure ratio for the CO_2 in the present study (approx 70) does not suit perfectly this ideal stage pressure ratio.

Compression Strategies aiming at an increase of the inlet pressure to the main CO_2 compressor would lead to a further reduction of the pressure ratio to Ramgen and therefore to a further gap from the optimum. For instance, if the final delivery pressure to the compression unit were 3 bara, the overall compression ratio would be approx 37. At such a pressure ratio, Ramgen could still offer a competitive two-stage solution with each stage rated at approximately 6:1, as in the design it is possible to vary speed or wheel diameter to accommodate alternative pressure ratios; however, from a techno-economic point of view, these would not be the optimum conditions for the considered compression concept.

For the above reason, the strategy of the staged regeneration through a multiple pressure stripper does not seem to be compatible with the Ramgen concept. Also, the complication associated to the addition of low head compressor within the CO_2 capture unit (whose duty cannot be taken by the Ramgen compressor) would further discourage the implementation of this strategy in case the relatively simple Ramgen device were used.

In this sense, a good technical compromise solution would be represented by a slight increase of the stripper/reboiler operating pressure (for instance from 1.6 bar to 2.1 bara, as per case B2a shown in section C). In fact, this strategy would allow:

- recovering waste heat from CO_2 at slightly higher temperature (approx 130 °C instead of 120 °C), which is recommended form an energetic point of view;
- improving CO_2 capture performance (less solvent regeneration heat) without penalizing too much Ramgen compressor.
- avoiding excessive complication of the process scheme.

However, as stated in section C, the real possibility of different regenerating conditions has to be confirmed by the solvent technology Licensors, especially in relation to the issue of higher amine degradation and consequent higher operating costs.

Regarding general strategies, the options associated to liquefaction of the CO_2 and increased number of stages would not be applicable to the Ramgen compressor, as they would lead again to an excessive reduction of the single stage pressure ratio to the machine.

3 Axial machine at the front end of CO_2 compression

Generally, axial compressor can handle a much higher flowrate than the centrifugal compressors, with a higher efficiency as well. In addition, the limitation on the maximum capacity of the integrally geared compressors leads, for large power plants, to the need of splitting the flowrate in multi-parallel trains, with a significant investment cost increase.

An alternative novel compression concept is therefore represented by the use of a single train axial machine for the first compression stage, which leads to a significant reduction of the volumetric flow rate, thus allowing the installation of a single train integrally geared compressor for the downstream compression section.

The Vendors that have supported the study have been asked to provide some feedback on the feasibility of this concept for the post combustion capture case.

Rolls Royce stated that, though they used axial compressors extensively in Aero Engines, these machines are not available at the moment and there are no plans for their development. Rolls Royce believes the design flexibility given by their current, pre-customised centrifugal approach is preferable. If the volume flow rates are higher than those that this system can cover, then parallel trains would be appropriate and also offer improved turndown capability/redundancy.

MAN Diesel & Turbo already selected a single train machine for their conventional approach (integrally geared compressor) for the post combustion application. The necessity of multiple trains is thus not an issue of their proposal. However, MAN Diesel & Turbo have provided further information regarding the possible use of an axial compressor at the front end of the CO_2 compression train. They are offering axial flow compressor solutions for CO_2 applications where the duty is too high for centrifugal designs or the adiabatic heat of compression is required by the process. Though the particular application investigated in the present work clearly fits the front stages of an integrally geared centrifugal compressor, which benefits from a stage of inter-cooling to achieve the first stage discharge pressure, they can potentially replace these two centrifugal stages by an un-cooled axial compressor with a pressure ratio of ~3.5 and polytrophic efficiency ~90%.

The impacts on the overall performance of the unit with respect to their integrally geared centrifugal solution have been estimated, as shown in Table 3-1.

Table 3-1: Performance of MAN Diesel and Turbo axial front-end machine with respect to their integrally geared centrifugal compressor.

MAN DIESEL AND TURBO AXIAL FORNT-END MACHINE Performance delta with respect to the conventional integrally geared compressor					
Thermal Integration with the Power Plant / CO_2 capture unit					
Steam cons. for Condensate Pre-heating	-8.3	MW_{th}	=	-2.2	MW_e
Cooling water					
CW consumption	- 972	t/h	=	- 0.1	MW_e
Compressor Electrical Consumption					
Overall electrical consumption difference				- 0.2	MW_e
Overall Plant Electrical Power Gap					
TOTAL				**- 2.5**	**MW_e**

Table 3-1 shows an overall consumption reduction of 2.5 MWe, whose major contribution is not the compressor shaft power itself, but the equivalent gain on the Steam Turbine output in the Power Island. In fact, the use of an un-cooled compressor at the front end makes more compression heat available, thus reducing the steam requirement for the ST condensate preheating in the Power Island.

On the other hand, in terms of investment cost, MAN Diesel & Turbo stated this is a much more expensive solution as the complete axial compressor must be manufactured in acid-resistant materials (i.e. similar to the MDT axial machines for nitric acid service), whereas for the integrally geared compressor only the impellers and volutes are in acid resistant materials.

In conclusion, the only Vendor that confirmed this option is feasible already provided single train centrifugal compressors for the applications considered in the study, whereas the necessity for multiple trains was the driving factor to seek the axial front end solution. Also, from the economic point of view, qualitative feedback on much higher CAPEX leads to the expectation that benefits in terms of lower consumption will be off-set by the additional investment cost of this syetm.

CLIENT : IEA GREENHOUSE GAS R&D PROGRAMME
PROJECT NAME : ROTATING MACHINERY FOR CO_2 COMPRESSION IN CCS SYSTEMS
DOCUMENT NAME : APPENDIX 1

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CHECKED BY : V. TOTA
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May 2011	First issue	F.Fazi	V.Tota	L.Mancuso

APPENDIX 1

EXECUTIVE SUMMARY

INDEX

1 Dehydration Options

The carbon Dioxide flowing to the CO_2 compression unit usually presents a moderate level of moisture, so dehydration is needed before delivering CO_2 to the export pipeline, in order to prevent potential hydrate formation, two-phase flow and corrosion in the export line.
To avoid these problems, the CO_2 stream has to be dehydrated before final delivery to the transport pipeline.

Two main basic processes are generally considered for the dehydration of the CO_2 stream:

- TEG (triethylene glycol) process;
- Solid bed dessicant.

1.1 TEG process

TEG is a proven technology for gas dehydration. A wet stream enters from the bottom of the absorber column and is contacted with descending glycol, which absorbs the water from the gas. Dry gas leaves the top of the absorber. The glycol then passes to a regeneration section where a reboiler operating around 180°C vaporizes the water and water-free glycol returns to the absorber. This is a continuous circulating process.

TEG systems have been widely and successfully used for Natural gas dehydration. When used for CO_2 dehydration in a CCS application, there are potential issues related to the higher affinity of glycol with CO_2 rather than other gases like methane.
In fact, part of the CO_2 in the wet gas is absorbed in the glycol stream and released from the glycol regeneration section. If no recycle of this vent were provided, the CO_2 stream would be released to atmosphere, thus affecting the overall carbon capture rate of the CCS system.
No significant reference was found in the literature to quantify the faction of CO_2 that can be potentially absorbed by the glycol. Should it be a significant amount, a recycle to the CO_2 compression system would effectively be needed, with impacts on machines selection and performance. For this reason it was believed that the issue had to be further addressed during the execution of the study.

In terms of dew point depression, approximately 30 ppm in the dry gas is considered to be the limit achievable with a standard TEG process.

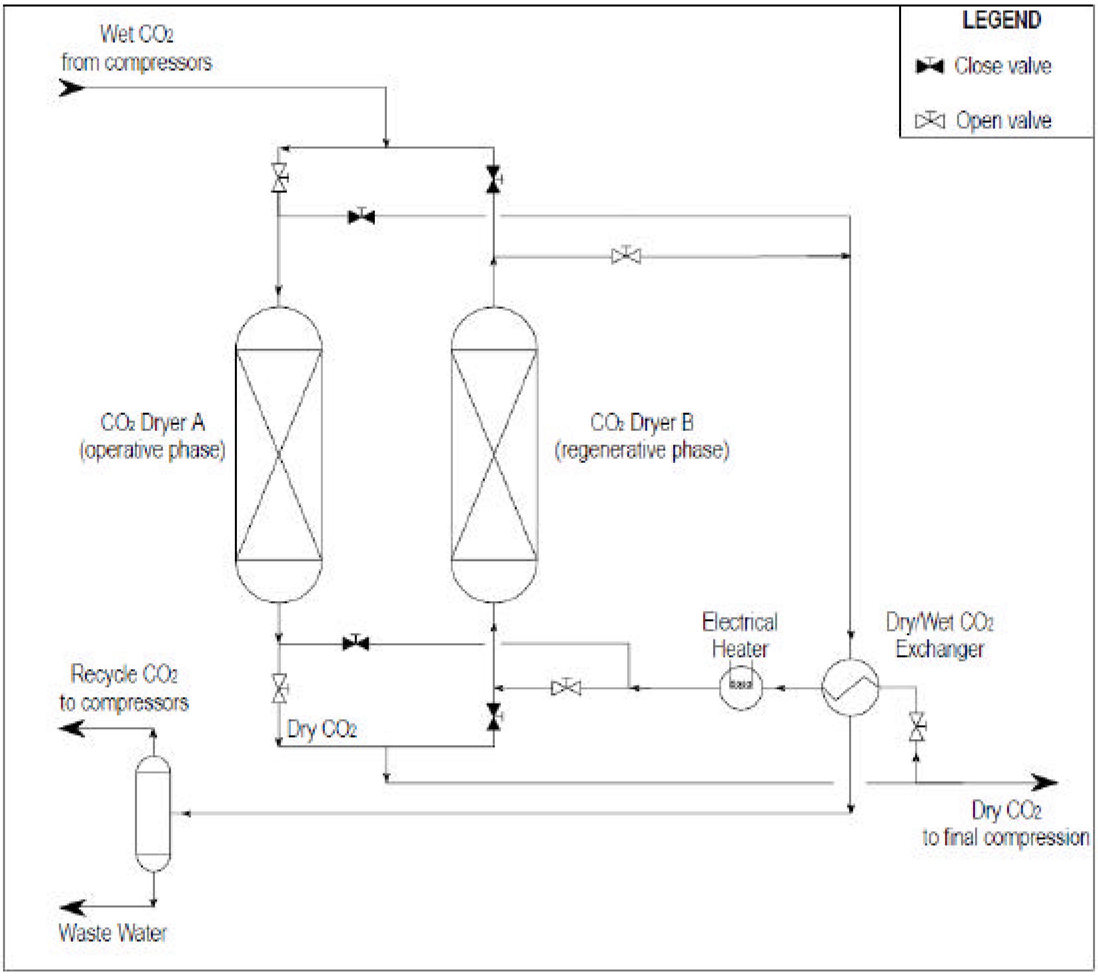
1.2 Solid Bed Dessicant

The drying process is made with specific solid materials that have the peculiarity to remove the humidity from gas streams. In the solid bed desiccant dehydration process the wet gas passes through a bed of solid dessicant contained in a vessel. The solid bed adsorbs the humidity in the gas phase via physisorption (Van der Waals adsorption), chemisorption, capillary condensation or molecular filtration. In general the absorption process is favoured by high partial pressure and low temperature.

Once the solid phase is saturated with water, it is necessary regenerate the dryer and to redirect the gas to be dried to another clean dessicant. Therefore the dehydration system is composed by multiple dryers (at least two) that are alternatively working in adsorption or regenerative manner: while one bed is processing, the other bed is under regeneration. A sequence of valve system switches the duties of the two vessels as one bed is exhausted. Process scheme is shown in Figure 1-1.

The solid dessicant bed can be regenerated by heating with hot gas (temperature swing adsorption), by changing the partial pressure (pressure swing adsorption) or a combination of both. In case of thermal swing, the regeneration is carried out using a small portion of the dried gas product (typically around 10%), which is heated in the range of 200-285°C. The quantity and temperature of regeneration gas also vary from one process to the other. The heat source for regeneration may be via electrical heater, steam heater or direct fire heater.
In both temperature and pressure swing cases the regeneration gas at dryer outlet is cooled to separate the entrained moisture by condensation and then recycled back to the CO_2 compressors at the most adequate inter-stage pressure.

Figure 1-1 Dehydration System: Solid Bed Dessicant



Three main dehydration technologies are typically adopted to remove water from gaseous streams:

- **Activated alumina** It is a porous dessicant where the humidity in the gas to be dried is adsorbed via physisorption, chemisorption or capillary condensation. The pores in the solid dessicant provide high surface area to create adsorption sites (>30Å are selective for H_2O molecules).

- **Molecular sieve** Molecular sieves are zeolite based adsorbent consisting of crystalline aluminosilicate (zeolites) and clay. The zeolite represents the active phase while a small amount of clay acts as a binder enhancing mechanical properties. Network of cavity and narrow pores provide high internal surface that enable the physical adsorption process. Liquid condensed water destroys the binder between clay and zeolite.

- **Silica gel** Silica gel is an amorphous form of silicon dioxide, which are synthetically produced. A microporous structure of interlocking cavities gives a very high surface area (800 m^2/gr) enabling high physical dessicant capacity.

Adsorption capacity together with adsorption kinetics defines the volume of desiccant to be used. The more amount desiccant is used, the longer would be the time needed to saturate the bed with water. Therefore, this time defines the adsorption cycle time. In addition to that, the depth of the adsorption front (which represents the zone where the adsorption takes place effectively) is related to the kinetic needed to reach the equilibrium in the adsorption process. The slower the kinetic of this process, the deeper this zone would be and such that more desiccant would be required. Among the considered solid desiccant, the silica gel has the slowest kinetic.

The efficiency measured with the water content of the dry gas. Among the three desiccant, the most efficient are the molecular sieve that can achieve less then 1 ppm of water in the dried gas.

On the other hand, the activated alumina is the strongest, i.e. it is less susceptible to deterioration and has the longest overall life.

2 Dehydration system design basis

2.1 CO2 moisture specification

For the purpose of the study, the moisture specification of the final CO_2 product was set to 50 ppmv. This figure has been considered as typical for the relatively low ambient conditions used in the study. However, it has to be noted that there is yet no consensus on a widely recognised pipeline specification. For further considerations on this topic, reference is made to para 3.1.

2.2 Selection of the technology for the base cases

A dehydration unit based on solid bed adsorption technology has been selected for the analysis of the CO_2 compression systems.
As described in para. 1.2, in the adsorption based technology, the regeneration of the saturated bed is carried out using a portion of the dried gas product, which is then recycled back to the CO_2 compressor. A fraction equal to 10% of the total CO_2 flow has been considered for the study, this being a typical average value from in-house set of data. It is recognised that the recycle represents an energy penalty, as one of the compressor stages has to handle an additional 10% flow rate.

On the other hand, there are still some uncertainties related to the impacts of the application of a TEG system for CO_2 drying (ref. 1.1), in terms of either carbon capture rate or energy penalties on the compression (ref. 1.1).

Solid bed adsorption units generally have a lower whole life cost than TEG units and provide higher flexibility in terms of dew point depression, being capable to achieve lower moisture in the export gas. As the debate on the moisture spec to be used in the different applications associated to CO_2 capture is still ongoing (ref. chapter 3), these advantages lead to select the solid bed adsorption as the reference configuration for the present study.
Also, even though it is recognised that the need of recycling a portion of the CO_2 for bed regeneration causes an energy penalty, it is appreciable that, including this configuration as reference for the study, it is possible to investigate the impacts which the recycle may have on machinery performance and selection.

3 Further considerations and investigations on CO_2 dehydration

3.1 Required water dew point specification

During the investigation of the novel compression concepts (ref. Section E of the report), Ramgen expressed concern over the specified water content in the discharge as not being representative of power plant design and CCS requirements worldwide. Ramgen believe that low moisture level drives the need for recirculation loops and additional equipment into the system, thus leading to unnecessary cost and complexity increase for no apparent benefit, especially for pre-combustion and post-combustion carbon capture. Also, Ramgen stated that acceptable levels of water content can be achieved with straight forward TEG systems without incurring such penalties. They reported the feedback from companies in the business of providing drying equipment and systems, according to whose opinion, for CO_2 pipeline, depending on location, the typical dehydration levels correspond to a dew point in the order of 0 to -10°C, which equates to approximately 20-30 lb/MMSCF (400-600 ppmv) under typical transport conditions.

In general, as far as required water dew point specification is concerned there is yet no consensus on a common pipeline specification and the requirements are to some extent more driven by the process. The oxy-combustion process requires very low dew points and hence the figure of 10 ppm is often taken as a suitable specification. For other processes the main limit would appear to be the avoidance of hydrate formation, which from measurements of equilibria seems to impose a greater restriction than the need to avoid free water to prevent corrosion of carbon steels. Also, water solubility in dense phase CO_2 is higher than equilibrium concentrations in the gas phase (ref. Figure 3-1).

Figure 3-1 Solubility of water in pure CO_2 as a function of pressure and temperature [1].

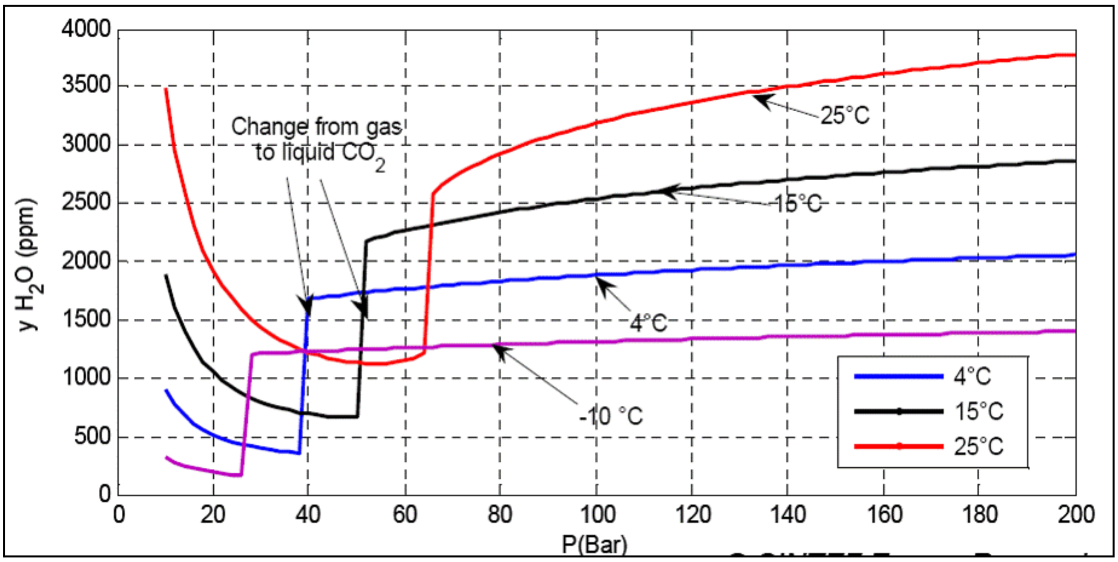
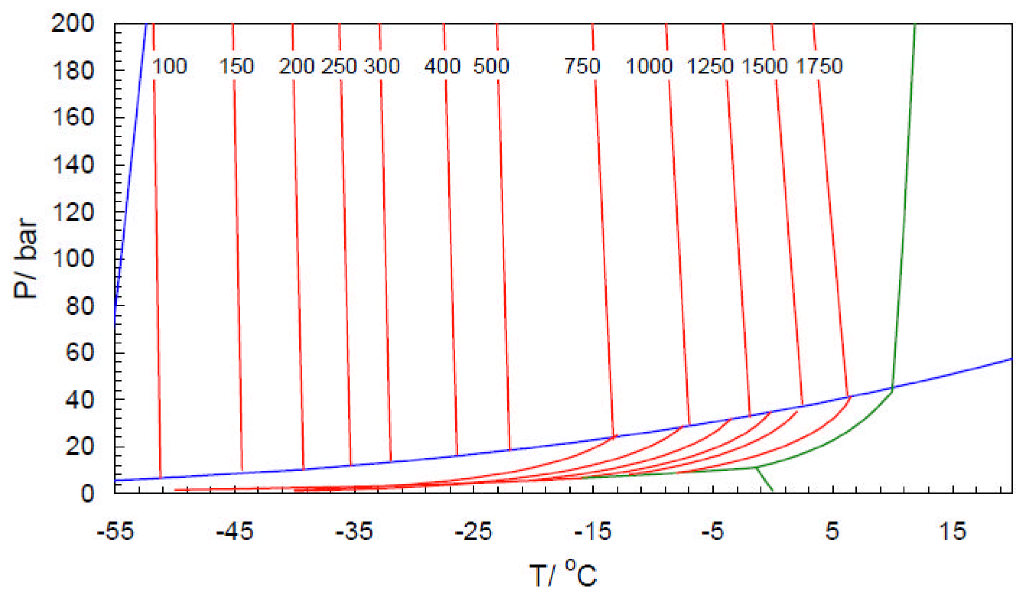


Figure 3-2 - Hydrates equilibria in pure CO_2.



Green lines: Hydrate stability of CO_2 in the presence distilled water or ice; Blue lines: VLE and SLE lines of pure CO_2; Red lines: Hydrate stability of CO_2 at fixed water content (Note in the vapour region, the water contents are (from left to right): 100, 150, 200, 250, 300 and 400 ppm)

In the dense phase much higher levels of water are needed to form hydrates at lower temperatures but this does remain the dominant factor as illustrated by the comparison between Figure 3-1 and Figure 3-2. For example at -10 °C water solubility is shown below as being around 1200ppm whereas hydrates are forming between the 750 and 1000ppm contours.

In practice the allowable concentration of CO_2 might be set at a lower level because of some additional occurrences such as the possible need for safe depressurisation and cross-effects with other impurities.

In many applications it is likely that values as high as 500 ppmv [1] will be acceptable in the pipeline and even in colder climates 200-300 ppmv should be sufficient to prevent hydrates formation even in the gas phase.

It is noted that such specifications are not achievable through a simple cooling, even with reasonably low cooling water temperature. For instance, in the post combustion capture the residual moisture content after cooling to 20°C below CO2 liquefaction pressure is approx. 1000 ppmv, which means that it is not possible to avoid further drying.

3.2 Further investigation on application of TEG Units

Regarding the issue raised on the potential application of the TEG technology to CO_2 dehydration (ref. 1.1), i.e. the uncertainties related to the impacts in terms of either carbon capture rate or energy penalties on the compression, Foster Wheeler has carried out further investigation both with drying systems Vendors and through simplified process simulations.

The generalised feedback is that only a minor portion of the incoming CO_2 (approx. 0.3%) is expected to be absorbed by the glycol, therefore the overall carbon capture rate is not significantly affected if the vent from the glycol regeneration is routed to atmosphere. Hence there is no need to recycle CO_2 from solvent regeneration to the CO_2 main path.

In conclusion, TEG process can be regarded as a technically viable option of CO_2 dehydration for CCS applications when the moisture specification for the dried carbon dioxide is above the limit of 30 ppmv.

4 Alternative options – Heat Of Compression Drier

During the execution of the study, in the context of debating on the moisture spec and the drying processes, Ramgen introduced a novel dehydration concept which seems to be capable of supporting down to 10 ppmv level requirement without the need for recycle flow.

This process, know as Heat Of Compressor (HOC) Dryer, has been developed by SPX, who supported the study providing a general description of the system and an indication of its potential.

4.1 General Description

The concept behind the "HOC Dryer" is to use the heat of compression for regeneration of the adsorbent. The system is generally composed by two dryers that are alternatively working in absorption (i.e. drying) or regeneration mode. The operation of the CO_2 HOC Dryer is described as follows.

4.1.1 Drying

Making reference to the flow diagram shown in Figure 4-1, before entering the drying bed, the inlet gas first passes through a gas cooler to lower the gas temperature then to a separator to remove condensed liquids. As the gas passes through the drying bed, water vapour is adsorbed. The dried gas is routed to the final stage of compression.

4.1.2 Adsorbent Bed Regeneration

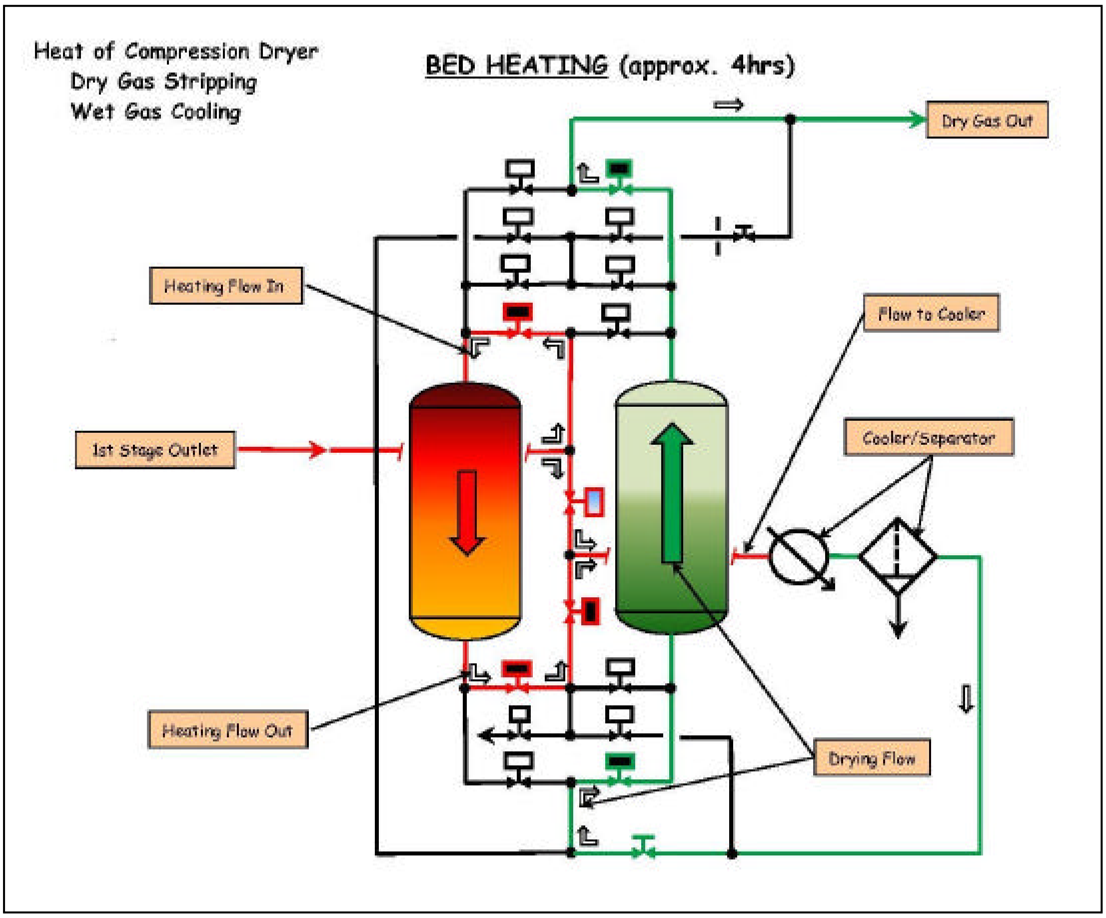
The regeneration of the absorbent bed is carried out in three main phases:

Bed Heating (Figure 4-1)

A fraction of the hot CO_2 from compressor inter-stage discharge passes through the bed being heated. This gas provides enough energy to release moisture from the adsorbent media and the relative humidity is low enough to create the needed differential for desorption and to carry off the liberated moisture. This first stage of the regeneration process removes the bulk of the previously adsorbed water vapour.

Warm, wet gas exiting the bed under regeneration recombines with the remaining hot gas from the compressor and is routed to the working bed.

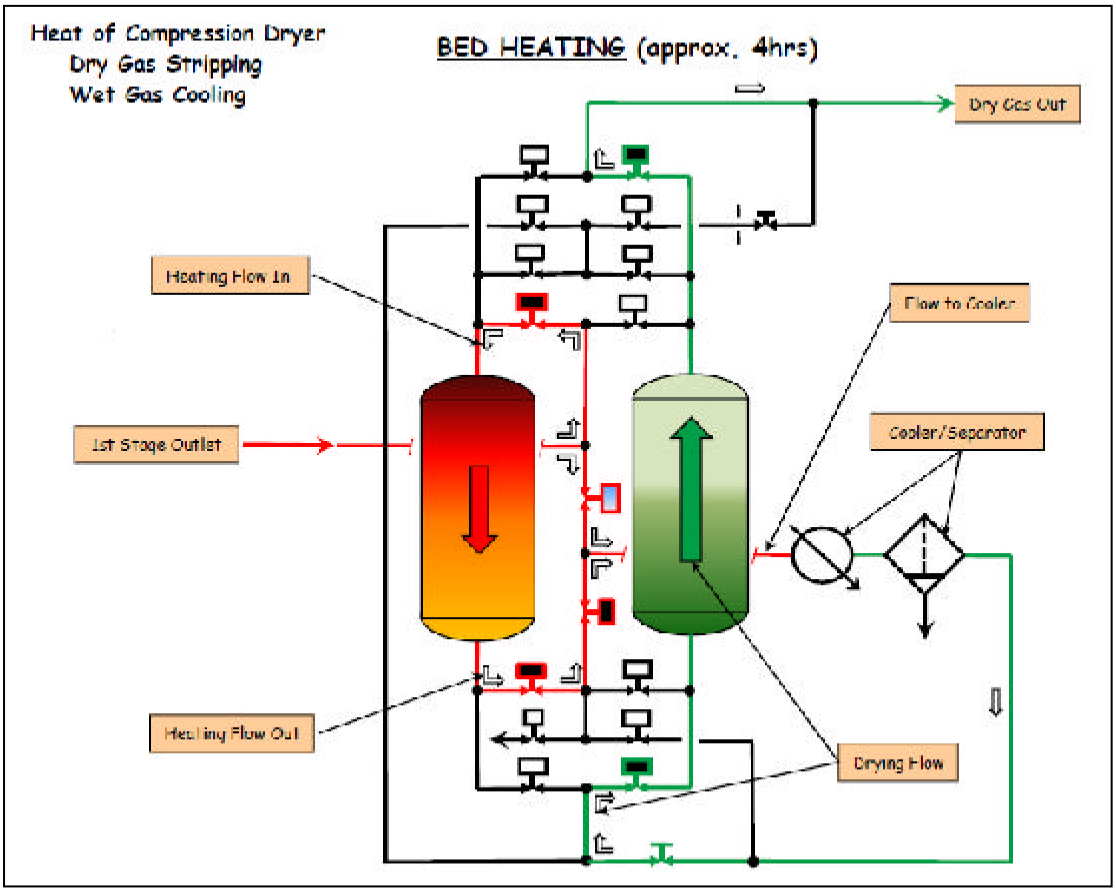
Figure 4-1 – Bed heating phase.



Dry Gas Stripping (ref Figure 4-2)

Once the bed is heated through, the total dryer inlet flow is directed to the on-line drying bed. The off-line tower is isolated, depressurised and a small amount of dried gas is used to strip additional moisture from the adsorbent. Dry Gas Stripping improves dew point suppression when the tower goes on-line and provides partial bed cooling.

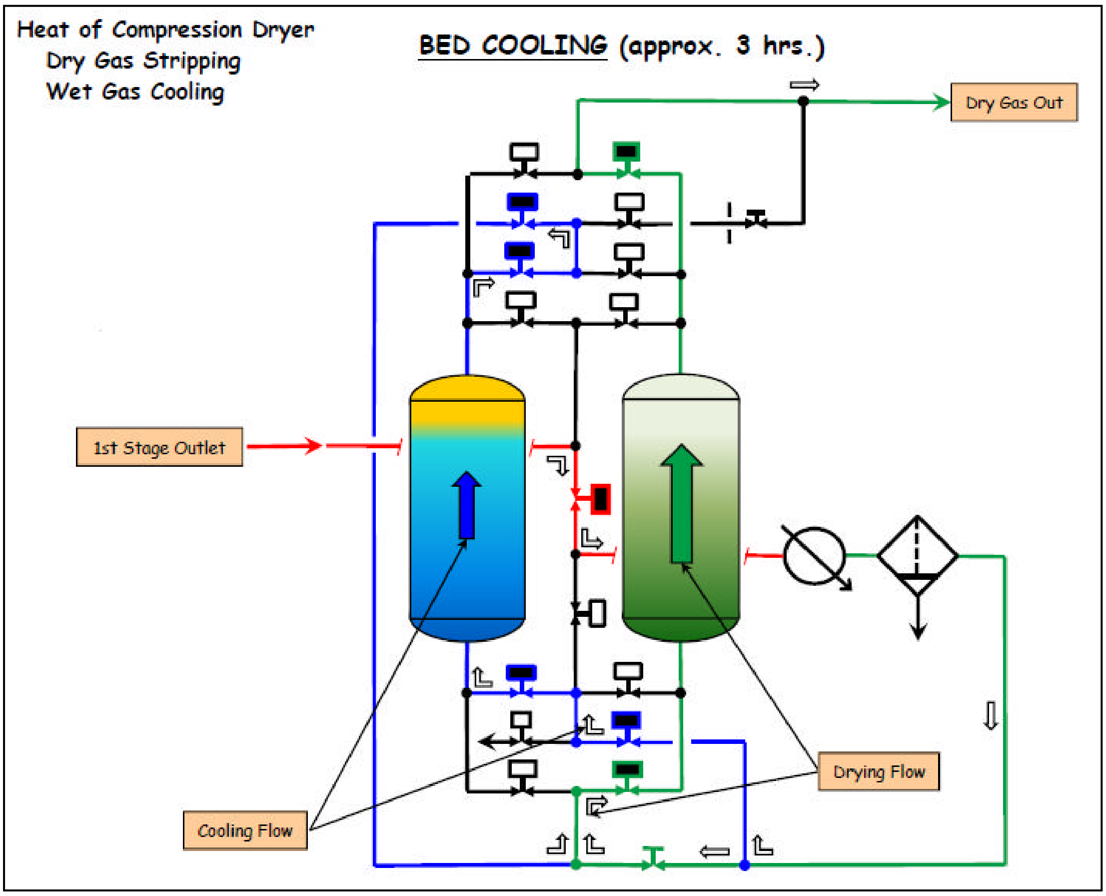
Figure 4-2 – Dry Gas Stripping phase.



Bed Cooling (ref Figure 4-3)

Once the stripping period is complete, the regenerating tower is re-pressurised and a portion of the "wet", cooled inlet gas is split off to cool the off-line tower as the final step to prepare the tower for another drying period. The cooling gas passes through the off-line tower, rejoins with the main flow and the combined flow is then dried.

Figure 4-3 – Bed Cooling phase.



4.2 Final considerations

SPX stated that, when the compressor inter stage discharge temperature is approx 240°C (like Ramgen case), the heat of compression is adequate to provide the desired dehydration (outlet moisture below 10 ppmv) with no need for additional external regeneration heaters.

Dry Gas Sweep is required because of the very high entering moisture content. Even with the relatively high compressor discharge temperature, the high moisture content of the entering gas leaves significant residual moisture on the bed at the end of the heating phase. Without dry gas sweep, there would be a significant moisture "bump" at following switchover.

Total dry gas "consumed" for Dry Gas Stripping and eventually vented to atmosphere is a small fraction of the inlet flow and lasts for a relatively short period with respect to the drying period, thus average CO_2 consumption will be less than 0.5% of the process flow.

It is noted that throttling is required on the main CO_2 compression path, in order to divert a portion of the inlet flow to heat/cool the off-line tower during heating and cooling phases, thus introducing additional pressure drop on the overall compression system. However, the required throttling, which is equal to the pressure drop through the tower being heated/cooled, plus piping losses, is expected to be minor, i.e. typically in the order of 0.1 to 0.2 bar. The associated energy penalty is significantly lower than with a conventional solid adsorption system using dried gas recycle for thermal swing regeneration.

It is also noted that the minimum required heating temperature falls as the outlet moisture specification rises.
For instance, if the temperature from the compressor were as low as 100°C, the adsorption bed would be left with relatively high residual moisture content at the end of the heating period. Dry gas stripping tends to be ineffective in lowering the exit gas moisture content in these conditions, so the resultant moisture content when the regenerated tower is put online could exceed 300 ppmw, as indicated by SPX.
At 150°C, depending also on the moisture content of the inlet gas, SPX expect it would be reasonable to achieve 30 ppmw or better.
Various design options for an HOC dryer are available to lower the exit gas dew point, if required, but these options generally increase complexity, investment cost and operating cost.

In conclusion, the combination of the HOC Dryer with the Ramgen compressor (which offers high temperature gas discharge) looks promising in achieving very low moisture content of the final CO_2 product with minimum impact on CO_2 capture rate, avoiding the major energy penalty of recycling back a portion of the dried CO_2 during adsorbent bed regeneration.
If the dew point depression requirement is stringent, this option is unique to Ramgen high inter stage temperature. However, if the outlet moisture specification is relaxed (ref. 3.1) it is possible to take advantage of the HOC dryer benefits even with more conventional compression applications, in which the heat of compression is available at lower temperature, i.e. in the range of 100 to 150 °C.

5 Conclusions

The main conclusions of the analysis presented in this appendix are summarised as follows:

- For the purpose of the study, the moisture specification of the final CO_2 product was set to 50 ppmv. This figure has been considered as typical for the relatively low ambient conditions used in the study. However, it has to be noted that there is yet no consensus on a widely recognised pipeline specification.

- TEG absorption and solid desiccant are well-known dehydration technologies applicable to carbon dioxide drying. For this study the reference configuration of Dehydration unit is based on solid bed adsorption for the following main reasons:
  - Solid bed adsorption units generally have a lower whole life cost than TEG units and provide higher flexibility in terms of dew point depression, being capable to achieve lower moisture in the export gas; this is a key factor as the debate on the moisture spec to be used in the different applications associated to CO_2 capture is still ongoing.
  - Even though it is recognised that the need of recycling a portion of the CO_2 for bed regeneration causes an energy penalty, it is appreciable that, including this configuration as reference for the study, it is possible to investigate the impacts which the recycle may have on machinery performance and selection.

- There is yet no consensus on a common pipeline specification as far as required water dew point specification is concerned. However, recent works report that, in many applications, it is likely that the specification can be relaxed with respect to the 50 ppmv used for the study. Values as high as 500 ppmv may be acceptable in the pipeline and even in colder climates 200-300 ppmv should be sufficient to prevent hydrates formation even in the gas phase.

- The Heat Of Compression (HOC) Dryer technology, developed by SPX, represents an alternative option for CO_2 dehydration. In particular, the combination of the HOC Dryer with the Ramgen compressor (which offers high temperature gas discharge) looks promising in achieving very low moisture content of the final CO_2 product with minimum impact on CO_2 capture rate, avoiding the major energy penalty of recycling back a portion of the dried CO_2 during adsorbent bed regeneration. It is noted that, if the moisture spec for the Carbon Dioxide is relaxed (for instance to values as high as 300 ppmv), the HOC can be profitably used even with more conventional compression unit, in which the waste heat is available at lower temperatures.

6 Bibliography

1. De Visser E., Hendriks C., Towards Hydrogen and Electricity Production with carbon Dioxide Capture and Storage, D.3.1.3 DYNAMIS CO_2 Quality recommendations, 2007.