Proceedings of ASME Turbo Expo 2011
GT2011
June 6-10, 2011, Vancouver, British Columbia, Canada

GT2011-46299

CONCEPTUAL DESIGN OF A MID-SIZED SEMI-CLOSED OXY-FUEL COMBUSTION COMBINED CYCLE

*** Majed Sammak, Klas Jonshagen, Marcus Thern, Magnus Genrup**
Lund University
SE-221 00 Lund, Sweden
**Corresponding author. Tel: +46 76 236 3637*
E-mail address: majed.sammak@energy.lth.se

Egill Thorbergsson, Tomas Grönstedt
Chalmers University of Technology
SE-412 96 Gothenburg, Sweden

Adrian Dahlquist
Siemens Industrial Turbomachinery AB
SE-612 83 Finspong, Sweden

ABSTRACT

This paper presents the study of a mid-sized semi-closed oxy-fuel combustion combined cycle (SCOC-CC) with net power output around 108 MW. The paper describes not only the power balance and the performance of the SCOC-CC, but also the conceptual design of the SCOC turbine and compressor. A model has been built in the commercial heat and mass balance code IPSEpro to estimate the efficiency of semi-closed dual-pressure oxy-fuel combustion combined cycle using natural gas as a fuel. In order to obtain the real physical properties of the working fluids in IPSEpro, the code was linked to the NIST Reference Fluid Thermodynamic and Transport Properties Database (REFPROP).

The oxy-fuel turbine was modeled with the in-house Lund University package LUAX-T. Important features such as stage loading, loss modeling, cooling and geometric features were included to generate more accurate results. The oxy-fuel compressor has been modeled using a Chalmers university in-house tool for conceptual design of axial compressors. The conceptual design of the SCOC-CC process has a net efficiency of 47 %. The air separation unit and CO_2 compression reduce the cycle efficiency by 10 and 2 percentage points, respectively.

A single-shaft configuration was selected for the gas turbine simplicity. The rotational speed chosen was 5200 rpm and the turbine was designed with four stages. All stage preliminary design parameters are within ranges of established industrial axial turbine design limits. The main issue is the turbine exit Mach number; the stage must be lightly loaded in terms of pressure ratio to maintain the exit Mach number below 0.6. The compressor is designed with 18 stages. The current value of the product of the annulus area and the blade rotational speed squared (AN^2) was calculated and found to be $40\cdot10^6$.

Keywords: SCOC-CC, Oxy fuel, gas turbine, mid-sized dual pressure combined cycle, CO_2.

NOMENCLATURE

A	Area [m^2]
C	Blade chord [m]
C_m	Meridional velocity [m/s]
CCS	Carbon capture and storage
cp	Specific heat capacity [kJ/kg.K]
Cp	Pressure recovery coefficient [-]
HP	High pressure
HRSG	Heat recovery steam generator
HTC	Heat transfer coefficient [W/m^2.K]
LP	Low pressure
$\dot{m}$	Mass flow [kg/s]
m^*	Dimensionless mass flow [-]
M	Molecular mass [kg/kmol]
M	Mach number [-]
N	Rotational speed [rpm]
P	pressure [bar]

PR	Pressure ratio [-]
r	Radius [m]
R	Gas constant [kJ/kg ^{o}C]
SCOC-CC	Semi-closed oxy-fuel combustion combined cycle
T	Temperature [°C or K]
U	Blade speed [m/s]

Greek symbols

γ	Gamma [-]
Δho	Stage work
ε	Cooling effectiveness [-]
η	Efficiency [%]
η_p	Polytropic efficiency [-]
Λ_p	Reaction degree based on pressure
ρ	Density [kg/m^3]
φ	Flow coefficient [-]
ψ	Blade loading coefficient [-]

Subscripts

ax	Axial
d	Discharge
h	hub
i	Inlet
m	Meridional
rel	Relative
rtr	Rotor
str	Stator
ti	Total inlet
td	Total discharge
t	Tip

INTRODUCTION

Secure, reliable and affordable energy supply is needed for economic growth. However, the increase in the carbon dioxide (CO_2) emissions associated with oil, coal and natural gas is a cause of major concern. Over the past decade, reducing carbon emissions and their impact on climate change has become the major focus of researchers and scientists. One option that has broad potential is CO_2 capture and storage; however, considerable development is needed to enable scale-up for industrial applications, and to make it more economical.

A number of power plant concepts for CO_2-neutral power production have been developed. All have advantages and disadvantages, and no single technique has shown to be superior in terms of performance and cost. The choice of capture technology is determined largely by the process conditions under which it the process must operate. Current carbon dioxide capture systems for power plants have the potential to capture some about 85-95 % of the CO_2 produced [1]. However, carbon capture and compression come at a price, namely a decrease in performance of the plant.

One promising candidate for the separation of carbon dioxide emitted by power plants is the semi-closed oxy-fuel combustion combined cycle (SCOC-CC). SCOC-CC concept is based on replacing air with oxygen in the combustion of the fuel so that the main products of combustion will be carbon dioxide and water. The steam is condensed in the flue gas condenser, while the CO_2 rich flue gas is separated into two streams. The main stream is re-circulated to the gas turbine, while the much smaller bleed stream is compressed for transport and storage [1].

Previous published work on SCOC-CC has been focused on large single-shaft gas turbines. There are several publications from ENCAP, Bolland and Sanz [2-9], among others. This paper differs from the previous work in that the studied SCOC-CC cycle is a mid-sized cycle with net power output around 108 MW and a gas turbine rotational speed of 5200 rpm leading to fewer stages in the compressor. The issues of air separation and CO2 transport and storage are not discussed although their impact is taken into consideration in the cycle assessments.

The change in the compressor working fluid, from air to CO_2, requires a number of important changes in the properties for the design of the gas turbine compressor, combustion chamber, and turbine.

Table 1 presents the main thermodynamic properties of air and carbon dioxide at 15 °C and 1 bar.

Table 1: Thermodynamic properties of air and carbon dioxide at 15 °C and 1 bar [10].

Thermodynamic properties	*Units*	*Air*	*Carbon dioxide*
C_p	kJ/kg °C	1.005	0.846
C_v	kJ/kg °C	0.718	0.657
R	kJ/kg °C	0.287	0.189
γ	-	1.4	1.289
ρ	kg/m^3	1.2	1.84
M	kg/kmol	28.9	44.01

From Table 1 it can be seen that the carbon dioxide has a lower value of gamma γ and a higher density than air. The speed of sound in the denser gas will be slower, and this will have a great impact on the design of the gas turbine. Issues such as Mach numbers in the turbomachinery components and maintaining a high performance in the bottoming cycle will have to be addressed.

Land based gas turbines can be categorized as single-, twin- or three-shaft units. The single-shaft configuration is less complex, but has poor torque characteristics for mechanical drive. The gas turbine mass flow can be controlled using a fixed-speed, single-shaft turbine by means of the variable guide vanes. This means that part-load operation can be achieved with a, more-or-less, nominal firing level, maintaining a high exhausts temperature over a wide load range. This feature results in high combined-cycle performance. A multi-shaft unit typically has about twice the full-load torque at starting speed, and is capable of satisfying both electric power and mechanical drive applications. Thus, a SCOC-CC plant based on a single-shaft turbine offers several advantages.

THE SEMI-CLOSED OXY-FUEL COMBUSTION COMBINED CYCLE PROCESS

Figure 1 show a schematic of the SCOC-CC cycle. The SCOC-CC consists of five main parts: the topping cycle, the bottoming cycle, the air separation unit, the CO_2 compression and the flue gas condenser. A brief description of the proposed SCOC-CC is given below.

The topping cycle consists of a compressor, a combustion chamber and a turbine.
Compressor: The compressor working fluid consists mainly of CO_2 re-circulated from the flue gas condenser. (The flue gas is divided into two streams; about 93 % being returned to the compressor). As a direct consequence of the high CO_2 content, the pressure ratio in the SCOC-CC is calculated to 39, which is higher than the operating pressure ratio for a conventional industrial gas turbine.
Combustion Chamber: In the SCOC-CC, combustion of hydrocarbons is performed in oxygen with a purity of 95 % at close to the stoichiometric conditions. The oxygen is obtained from the air separation unit. The SCOC-CC combustion chamber products are thus quite different from conventional combustion chamber products. The outlet temperature from the combustion chamber is set to 1400 °C. This gives a good turbine effect taking into consideration material limitation in the turbine as well as cooling effectiveness that is a function of degree of technology. The theoretical flame temperature in the combustion chamber may reach 3500 °C [1] while the actual flame temperature is lower. This is considerably higher than any state-of-the-art gas turbine can handle, and must be addressed in subsequent design phases.
Turbine: Due to the high temperature at the turbine inlet the turbine is cooled with CO_2 from the compressor. The mass flow of the cooling CO_2 is about 30 % of the turbine inlet mass flow. The flue gas is expanded in the turbine to a pressure of 1.06 bar (including the diffuser loss), resulting in a temperature of 620 °C.

The bottoming cycle main components are the heat recovery steam generator (HRSG) and steam turbine and condenser. The heat recovery system employs two pressure levels. A third level would probably give better performance, but is not considered here due to the cost. The heat recovery steam generator has a standard configuration including a low-pressure (LP) economizer, a low-pressure evaporator, a low-pressure superheater, a high-pressure (HP) economizer, a high-pressure evaporator and finally a high-pressure superheater. The HP inlet steam temperature is set at 565 °C and a pressure of 127 bar at steam quality around 85 %. The static pressure of the condenser is set to 0.034 bar, assuming a cooling water temperature of 15 °C. The performance of the steam turbine has been assessed in detail with the Siemens in-house design code for steam turbines, however, due to the proprietary nature of this information; no details can be given here. The deaerator is operating at 105 °C and the energy is taken from the LP-economizer. The deaerator outgoing water is cooled by the incoming condensate, to approximately 50 °C. This is limited by the requirement of a temperature difference of 13 °C for efficient deaeration. This configuration increases the temperature of the inlet water in the economizer which prevents flue-gas condensation. The flue-gas exit temperature from the HRSG is of 64 °C is lower than for a regular dual-pressure combined cycle and this is due to the fact that the specific heat of the flue gas is lower than in a conventional cycle.

The flue gas condenser is assumed to be a packed column. The purpose of this is to condense water from the exhaust gases and remove pollutants and non-condensed gases from the flue gas. The flue gas leaving the flue gas condenser consists mainly of CO_2 with a temperature around 20 °C. Most of the CO_2, around 93 %, is recycled to the compressor, while the remainder is bled off and compressed for storage.

CO_2 is compressed and liquefied at a pressure of 200 bar to enable transport and storage in the dense phase.

There are three commercially available methods of separating air, namely; cryogenic separation, pressure swing absorption, and membranes. The very large quantities of oxygen required in oxy-fuel combustion can only be produced economically using cryogenic separation [9,11].

MODELING AND PROCESS SIMULATION

All thermodynamic and process simulations were performed using the commercial software IPSEpro, developed by SimTech Simulation Technology [12]. This software allows the implementation of user-defined fluid properties to simulate the real gas properties of the working fluids. The physical properties of water and steam were calculated using the standard IAPWS-IF97 formulations in IPSEpro [12]. IPSEpro was linked to the NIST Reference Fluid Thermodynamic and Transport Properties Database (REFPROP) to obtain more realistic results [13].

Modeling of e.g. turbomachinery components becomes more intricate with rigorous-state models. The assumption of the reversible polytropic path for the compression works should not be used, since they are typically derived from semi-perfect gas models. Instead, one should use the basic state equations for such process.
The Mallen-Saville model [14, 15] was implemented in this study.

$$T\,\mathrm{ds/dT} = constant \quad (1)$$

$$\Delta H_p = \Delta H - (S_d - S_i)(T_{td} - T_{ti})/\ln(T_{td} - T_{ti}) \quad (2)$$

$$\eta_p = \Delta H_p/\Delta H \quad (3)$$

The polytropic efficiency calculated by the model was 92 %.

The SCOC-CC was designed as a mid-sized plant. Such a plant would typically have two pressure levels in the bottoming cycle, in contrast to larger plants with three pressure levels and reheat. Choosing a smaller plant offers the possibility of using geared gas- and steam turbines, hence allowing compact parts to be used. Bolland and Mathieu found that the difference between dual- and triple-pressure steam cycles was very small [9].
The work necessary to obtain oxygen with 95% purity from air has been estimated to approximately 900 kJ/kg O_2 and 325 kJ/kg O_2 for compression [2]. The energy required for CO_2 compression from 1 to 200 bar for transport and storage has been estimated to be 350 kJ/kg CO_2 [2].

The thermodynamic states and compositions of the streams are given in Tables 2 and 3 respectively. Table 4 lists the main cycle parameters.

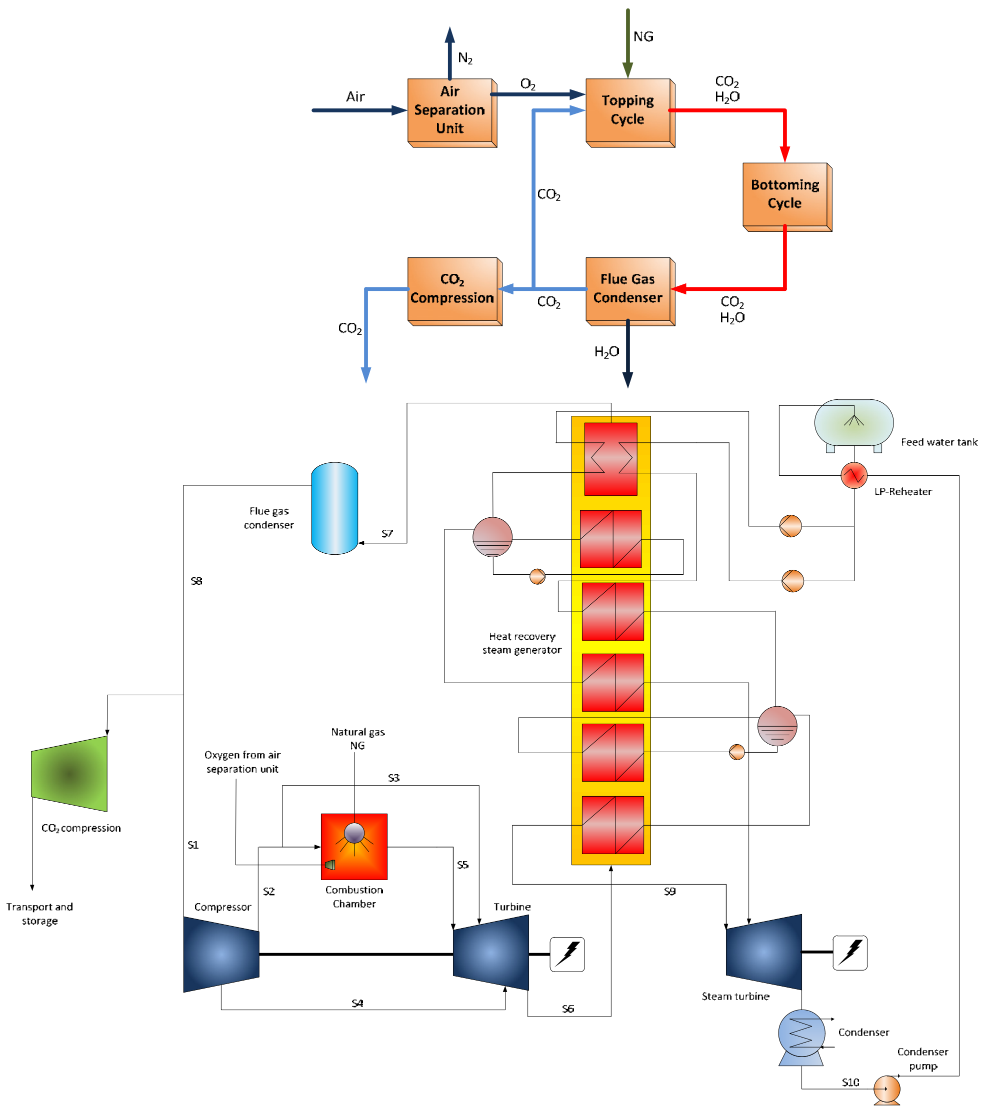

Figure 1: Principle flow scheme for SCOC-CC.

Table 2: Composition of streams S1-S5.

Stream	*Unit*	*S1*	*S2*	*S3*	*S4*	*S5*
T	°C	20	394	394	265	1400
P	bar	1.013	39	39	14	39
m	kg/s	190	171	31	19	164
CO_2	%wt	92	92	92	92	86.5
N_2	%wt	2.85	2.85	2.85	2.85	2.7
Ar	%wt	4.06	4.06	4.06	4.06	3.8
H_2O	%wt	0.97	0.97	0.97	0.97	6.8
O_2	%wt	0.11	0.11	0.11	0.11	0.11

Table 3: Composition of streams S6-S10.

Stream	*Unit*	*S6*	*S7*	*S8*	*S9*	*S10*
T	°C	620	64	20	565	26
P	bar	1.013	1.013	1.013	127	0.034
m	kg/s	214	214	204	35.6	39.5
CO_2	%wt	87.7	87.7	92	0	0
N_2	%wt	2.7	2.7	2.85	0	0
Ar	%wt	3.9	3.9	4.06	0	0
H_2O	%wt	5.5	5.5	0.97	100	100
O_2	%wt	0.11	0.11	0.11	0	0

Table 4: SCOC-CC cycle power balance.

	Unit	
Compressor mass flow	kg/s	190
Compressor pressure ratio	-	39
Combustor outlet temp.	°C	1400
Gas turbine power, Turbine	MW	156
Gas turbine power, Compressor	MW	67
Gas turbine power	MW	87
Total heat input	MW	230
Steam turbine power	MW	49
Gross power output	MW	136
Gross efficiency	%	59
O_2 generation+compression	MW	23
CO_2 compression to 200 bar	MW	5
Net power output	MW	108
Net efficiency	%	47

The designed process has a net effect of 108 MW and a net efficiency of 47 %. The penalties for air separation and compression are 10 % while CO_2 compression is 2 %.

CONCEPTUAL DESIGN OF THE SCOC-CC GAS TURBINE

As with the compressor, the working fluid in the turbine is mainly CO_2, with a purity of almost 92 %. Changing the fundamental physical properties of the working fluid will have a significant impact on the design of the turbine. The standard set of design rules for a normal gas turbine can therefore not be directly applied to the design of a SCOC cycle. The SCOC-CC turbine has been modeled with the Lund University in-house turbine design tool [1]LUAX-T. The code is a reduced-order throughflow tool which is capable of designing highly cooled turbines. The code uses the [2]AMDC-KO-MK loss model and fundamental equations for momentum, energy and continuity for assessing losses due to cooling, purging and packing flows.

The chosen layout is a straight forward CO_2-cooled single-shaft unit. As mentioned the principal reasons for a single-shaft are the simplicity and related low cost. Furthermore, a single-shaft unit offers significantly less complexity in terms of shaft and bearing systems. Since there is no need for a collector with cold-end drive a more efficient exhaust system could be designed [16]. A conical diffuser can reach a recovery level of 0.8 in comparison to 0.6-0.7 for a system with a 90° bend. Figure 2 presents turbine exhaust diffuser performance which presents the relation between the pressure recovery coefficient and the static/total outlet pressure. A single-shaft unit has typically a beam type rotor whereas the twin-shaft needs bearings within the high pressure section of the gas turbine. Before describing the features of a SCOC turbine, it is instructive to discuss the design of a conventional (air and flue gas) single-shaft unit. One of the most critical design choices is the speed level since it has a profound impact on the stage count and cost.

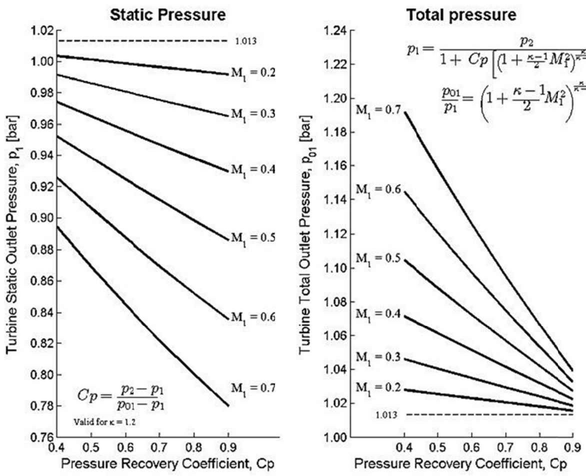

Figure 2: Turbine exhaust diffuser performance.

[1] LUAX-T is freely available for academic work outside Lund University.

[2] AMDC-KO-MK: Ainley+Mathieson+Dunham+Came-Kacker+Okapuu- Moustapha+Kacker [17]

The SCOC-CC Turbine design

The rotational speed is typically set by either the turbine exhaust Mach number at a maximum available AN^2 (~root stress level) or the relative tip Mach number at the first compressor rotor. The importance of the AN^2 parameter cannot be overemphasized since a reduction from e.g. $50{\cdot}10^6$ to $40{\cdot}10^6$ may translate into two additional compressor stages. The exit Mach number is preferable within the range of 0.5-0.6 while the maximum tip Mach number is in the order of 1.3 [16]. The meridional Mach number should be less than 0.7 at the compressor inlet.
The heavy gas introduces issues with the last stage loading due to the low speed of sound. For reasonable exit losses, the last turbine stage should be designed with an exit Mach number below 0.6 (see figure 2).

The pressure ratio of the last stage needs to be low, but still give sufficient rotor turning to keep blade frequency high enough in order to avoid rotor dynamic difficulties. One can argue whether a twin-shaft unit should provide a better platform. The additional freedom of having another speed level would be advantageous in terms of optimizing the loading distribution. The speed level was calculated to 5200 rpm and the first turbine stage rim speed to 380 m/s. The calculated rotation speed takes into account the aerodynamics requirements that are leveling between exit Mach number and the last stage loading coefficient. The relative tip Mach number of the first compressor stage does not pose any limitations at the chosen speed level. Efficient gears are available exceeding 100 MW. The associated losses are approximately 1.2 % and 0.6 % for normal and vacuum respectively.

The first stage has almost no inner wall hade, while the later next one has about eight degrees. The first two rotors are cylindrical to satisfy minimum running clearance requirements. The outer wall hade was set to a maximum of 25° to maintain a maximum opening of approximately 35°. These values are somewhat arbitrary, but well within current practice. The actual limitations must be assessed in a later, more detailed design phase. With these design constraints, the exit Mach number is 0.60 and the value of AN^2 is close to $40{\cdot}10^6$. The last rotor has a mid-span turning exceeding 65°. A three-stage design would not be realistic, even at higher speed, due to the inherent limitation on the exit Mach number. Figure 3 presents the annulus of a four stages SCOC designed turbine.

The designed four stages SCOC turbine flow coefficients, stage loading coefficients and reactions degree of the four-stage are presented in Table 5.

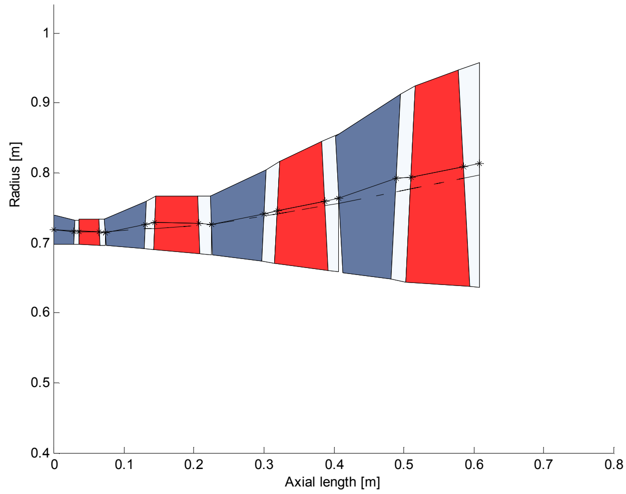

Figure 3: Annulus of the conceptual design of the SCOC turbine.

Table 5: Turbine design data.

	Stage 1	*Stage 2*	*Stage 3*	*Stage 4*
$\Delta h_0/U^2$	1.7	1.3	1.1	1.0
ψ		1.30		
C_m/U	0.39	0.39	0.42	0.44
PR	2.57	2.32	2.34	2.74
Λ_p	0.30	0.38	0.38	0.40
$\varepsilon_{str}/\varepsilon_{rtr}$	0.47/0.4	0.28/0.16	0.05/0	-

The combustor outlet temperature was set to 1400 °C to obtain high power density and good cycle performance. This level of firing requires an efficient cooling system. The firing level alone is not sufficient to describe the required cooling, and the cooling effectiveness must be used to describe the cooling duty (ε). At first glance, the values of the cooling effectiveness are low – and well within common practice. The cooling flows can be assessed with the standard m-star model [17].

$$\varepsilon = \frac{T_{gas.rel} - T_{metal}}{T_{gas.rel} - T_{cool.rel}} \qquad \varepsilon = \frac{m^* \eta_{cool}}{1 + m^* \eta_{cool}} \qquad m^* = \frac{\dot{m}_{cool} \cdot c_{p.cool}}{HTC \cdot 2.2 \cdot C \cdot h} \qquad (4)$$

Equations (4) provide a link between the thermodynamic properties of the cycle, the maximum metal temperature, the level of technology and the velocity triangles. All temperatures should be relative, since both the inlet temperature and coolant temperature are dependent on the velocity triangles. The relative inlet temperature can be controlled effectively by lowering the reaction level. Another possibility is to reduce the relative inlet velocity by using a lower value of C_m/U, resulting in lower local heat transfer at the first part of the blade. However, it is not possible to assess this heat transfer with a mid-span code. The present version of LUAX-T has three different ways of mixing coolant into the main stream, namely: (i) film, (ii) trailing-edge ejection, and (iii) packing and cavity purge flows. Losses due to coolant- and packing flows are assessed by conservation of momentum and energy. The mixing routines result in a non-linear system that must be solved iteratively for conservation of mass. Film losses are assessed through a generic free stream velocity distribution and the Hartsel mixing method [18]. The generic velocity distribution assumes a certain loading distribution and suction-side diffusion level. Injection positions, angles and momentum ratios are typical in terms of normal showerhead and profile configurations. Common practice is not to have injection cooling after the throat point – driven by that the loss scales with Mach number squared.

The total cooling flow, including 1.5 % per disc cooling and packing, is 49.8 kg/s or 26.2 % of the compressor inlet flow. This figure is rather high, and must be further investigated. The specific heat ratio between air and carbon dioxide is about 1.2, and a corresponding influence on cooling consumption is to be expected by virtue of the medium alone. It should be borne in mind that the m-star model typically underestimates the total cooling flow for highly fired engines. It is common to introduce an adjustment factor which is dependent on the firing level. Early profiling and a heat transfer code should be able to verify the coolant consumption in a later design stage. The drop in combustion pressure is a limiting factor in terms of the pressure difference for film cooling of the first vane. For a normal gas turbine, the pressure drop should not fall below 3 % to ensure sufficient film pressure drop. The entire cooling and secondary air system must be analyzed before any firm conclusions can be drawn.

The SCOC-CC compressor design

As mentioned previously the design of the SCOC-CC compressor is constrained by the relative tip Mach number of the first rotor. To allow a high blade speed and keep first stage transonic losses down an inlet guide vane is employed. Here, a 15° swirl was selected. This was a sufficient level for the relatively moderate Mach number that resulted from the design process described below.

Reducing the flow coefficient at the first stage will help to keep the first stage specific work up. For a given hub tip ratio reducing flow speed will increase speed at the blade mid allowing a higher first stage temperature rise to be achieved for a given stage loading parameter ψ. However, it is usually desirable to reduce axial flow speed and the flow coefficient through the compressor to allow a relatively low flow coefficient at the compressor exit. This makes the process of establishing a desirable surge margin easier. In fact, it is possible to select a somewhat higher first stage flow factor and simultaneously obtain an operation point close to the minimum tip Mach number, i.e. close to the minima of the family of curves presented in Figure 4. This will reduce the transonic losses incurred in the first stage rotor. Here, a first stage hub tip ratio of 0.483 is selected together with a flow factor of 0.65.

To keep blade lengths at a reasonable level and thereby limit tip leakage losses in the last stages of the compressor the hub-tip ratio must be kept at a reasonable number. Here a constraint of 0.92 in exit hub tip ratio has been imposed together with an axial exit Mach number of 0.25. Together with the mass flow, the compressor efficiency and the pressure ratio the hub tip ratio limit will define the compressor exit tip and hub radius; here these numbers turn out to be 0.374 and 0.345 meters respectively.

To ensure that accurate thermodynamic gas properties are obtained the Chalmers university in-house tool for conceptual compressor design couples the REFPROP tool to the compressor design process.

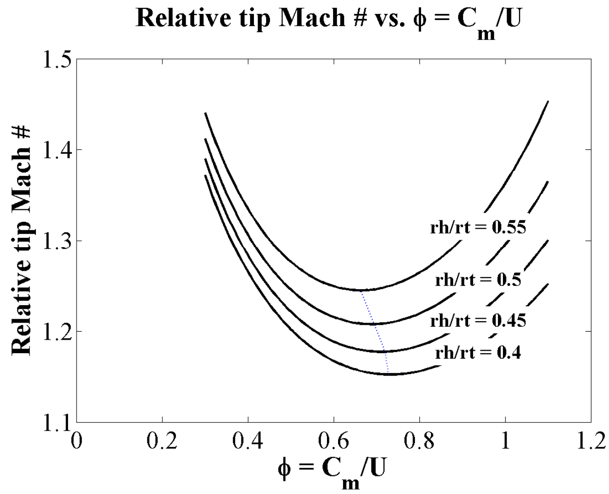

Figure 4: First stage rotor relative tip Mach number as a function of flow coefficient for a range of hub-tip-ratios (0.4, 0.45, 0.50, 0.55).

The resulting compressor is an 18-stage design. It should be noted that the initial assumption of a first stage rotor hub-tip ratio of 0.50 and the limiting hub tip ratio of 0.92 at the compressor exit produced an annulus very close to a constant hub design after the first stage. Data on axial flow velocities, blade speeds at blade mid, flow coefficients, stage loading coefficients, stage hub tip ratio, relative tip Mach numbers and stage pressure ratios have been collected in Table 6. A cross sectional drawing of the compressor annulus is reproduced in Figure 5.

The design studies carried out strongly suggest that the number of stages and the feasibility of a single-shaft compressor are quite sensitive to the exit hub tip ratio. Reducing the allowable exit hub tip ratio to 0.90 typically adds 2-4 stages on the compressor due to the reduced blade speed. This gives some insight into how the selection of single-shaft compressor architecture imposes restrictions on the pressure ratio of the cycle. Increasing the pressure ratio further will necessitate that the annulus is allowed to move to lower radii, a decreasing hub radius design, to allow the shorter blades arising from the increased pressure and density to stay within the hub tip ratio bound. This will add stages rapidly making the single-shaft design less attractive for very high cycle pressure ratios.

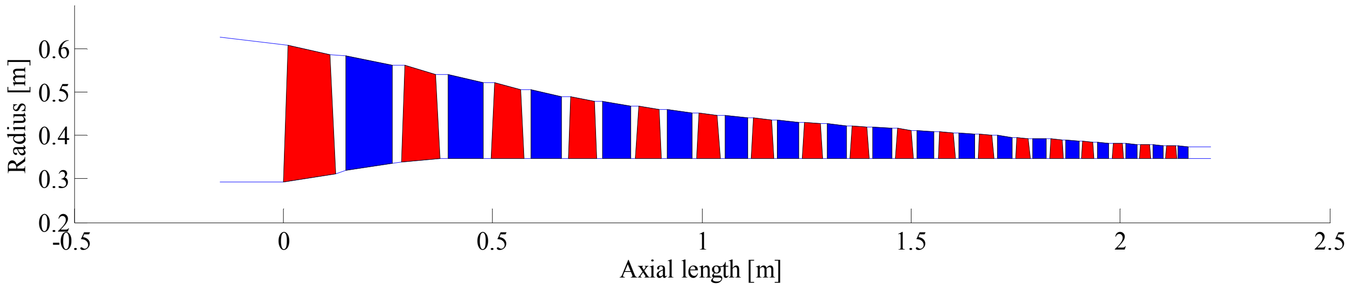

Figure 5: Annulus of the conceptual design of the SCOC compressor.

Table 6: SCOC-CC compressor data.

Stage		*1*	*3*	*5*	*7*	*9*	*11*	*13*	*15*	*17*	*18*
C_{ax}	[m/s]	155	138	127	117	106	100	100	100	100	100
U_m	[m/s]	246	232	219	212	208	204	201	199	197	196
φ_m	-	0.63	0.60	0.58	0.55	0.51	0.49	0.50	0.50	0.51	0.51
ψ_m	-	0.45	0.45	0.45	0.45	0.45	0.45	0.45	0.45	0.45	0.45
rh/rt	-	0.55	0.68	0.75	0.79	0.83	0.85	0.88	0.90	0.91	0.92
M_{reltip}	-	1.12	0.85	0.73	0.65	0.60	0.56	0.53	0.50	0.48	0.47
PR	-	1.52	1.37	1.28	1.23	1.19	1.17	1.15	1.14	1.13	1.12

CONCLUSION

This paper presents the conceptual design of a mid-sized, semi-closed dual-pressure oxy-fuel combustion combined cycle and the conceptual design of a SCOC turbine and compressor. The SCOC-CC model was based on the heat and mass balance code IPSEpro. The physical properties of the gases were calculated using values determined by NIST software REFPROP.

The designed SCOC-CC process has a net power output of 108 MW and 47 % efficiency. Note that all values exclude parasitic losses and step-up transformer losses. The air separation unit reduces the net efficiency by 10 percentage points while the CO_2 compression reduces it by another 2 percentage points.

The overall compressor pressure ratio is 39 which is higher than the operating pressure ratio for a conventional industrial gas turbine. The combustion outlet temperature is set to 1400 ^{o}C. The turbine requires an efficient cooling system to maintain blade metal temperature within the limits. The total calculated cooling flow is 49.8 kg/s or 26.2 % of the compressor inlet flow.

A four-stage, single-shaft gas turbine configuration was chosen for simplicity. The rotational speed selected was 5200 rpm. The turbine exit Mach number was maintained below 0.60. The compressor was designed with 18 stages. The current value of AN^2 was found to be $40 \cdot 10^6$.

The last stage loading issue together with the relative low compressor rotor inlet Mach number suggests that a twin-shaft unit is a better candidate for optimal turbomachinery operation. Furthermore a twin-shaft design would allow the gas generator to operate at a higher speed level making the compressing system more compact as well as providing an optimized power turbine at a lower speed level. However, the advantages of a twin-shaft design must always be weighed against retaining the inherent simplicity and low cost of the simple cycle single-shaft design.

ACNOWLEDGEMENTS

This research has been funded by the Swedish Energy Agency, Siemens Industrial Turbomachinery AB, Volvo Aero Corporation and the Royal Institute of Technology through the Swedish research code TURBO POWER, the support of which is gratefully acknowledged. The authors would also like to acknowledge David Olsson and Jonas Svensson for developing the LUAX-T turbine design tool. The detailed steam turbine calculations would not have been possible without the assistance from Specialist Åke Göransson (Siemens).

REFERENCES

[1] Metz, B., Davidson, O., Coninck, H., Loos, M., Meyer, L., 2005, "IPCC Special Report on Carbon Dioxide Capture and Storage", Cambridge University, ISBN-13 978-0-521-86643-9.

[2] Sanz, W., Jericha, H., Bauer, B., Göttlich, E., 2005, "Qualitative and Quantitative Comparison of Two Promising Oxy-Fuel Power Cycles for CO2 Capture", Journal of Engineering for Gas Turbines and Power, **130**.

[3] Bolland, O., Saether, S., 1992, "New concepts for natural gas fired power plants which simplify the recovery of carbon dioxide", Energy Convers.Mgmt, **33**, (5-8), pp. 467-475.

[4] Woollatt,G., Franco, F., 2009, "Natural gas oxy-fuel cycles-Part 1: Conceptual aerodynamic design of turbo-machinery components", Energy Procedia, **1**, pp. 573-580.

[5] Rezvani, S., Bolland, O., Franco, F., Huang, Y., Span, R., Keyser, J., Sander, F., Mcllveen-Wright, D., Hewitt, N., 2009, "Natural gas oxy-fuel cycles-Part 3: Economic Evaluation", Energy Procedia, **1**, pp.565-572.

[6] Bolland, O., Kvamsdal, H.M, Boden, J. C., 2005, " A compison of the efficiencies of the oxy-fuel power cycles water cycle, Graz cycle and Matiant cycle", Carbon dioxide capture for storage in deep geologic formations, **1**.

[7] Bolland, O., Undrum, H., 2003, "A novel methodology for comparing CO2 capture options for natural gas-fired combined cycle plants", Advanced in Environmental Research, **7,** pp. 901-911.

[8] Hammer, Th., Keyser, J., Bolland, O., 2009, "Natural gas oxy-fuel cycles-Part 2: Heat transfer analysis of a gas turbine" Physics Procedia, **1,** pp. 557-564.

[9] Bolland, O., Mathieu, P., 1998, "Comparison of two CO_2 removal options in combined cycle power plants", Energy Convers. Mgmt, **39**, (16-18), pp. 1653-1663.

[10] Cengel, Y., Boles, M., 2002, "Thermodynamics an engineering approach", 4^{th} ed, ISBN 0-07-112177-3.

[11] Allam, R., White, V., 2005, "The oxyfuel baseline: revamping heaters and boilers to oxyfiring by cryogenic air separation and fuel gas recycle", Carbon dioxide capture for storage in deep geologic formations, **1**.

[12] IPSEpro, 2003, SimTech Simulation Technology (SimTech), Graz, Austria.

[13] REFPROP, Version 8, Standard Reference Data Code, National Institute of Standard and Technology.

[14] Aungier, R., 2003, "Axial-Flow Compressor", ISBN 0-7918-0192-6.

[15] Mallen, M., Saville, G., 1977, "Polytropic processes in the performance prediction of centrifigual compressors", Institution of mechanical engineers, London, United Kingdom, pp. 89-96.

[16] Walsh, P.P., Fletcher, P., 1998, "Gas Turbine Performanc" ISBN 0-632-04874-3.

[17] Moustapha, H., Zelesky, M F., Baines, N., Japikse, D., 2003, "Axial and Radial Turbine", Concepts ETI, ISBN 0-933283-12-0.

[18] Hartsel, E., "Prediction of effects of mass transfer cooling on the blade row efficiency of turbine airfoils", General Electric Company, Evendale, ohio.