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FIRST AND SECOND LAW ANALYSIS OF CO_2 OFFSHORE COMPRESSION SYSTEM

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Abstract. *The Oil & Gas floating production, storage and offloading platforms (FPSO) are associated with high-energy consumption and CO_2 emissions, which are related to separation systems, hydrocarbons and water treatment, gas and water overflow injection in reservoirs for storage or secondary elevation, and utilities in general where combustion of fuels is required. This study presents the formulation and simulation to develop a first and second law analysis for a booster and main CO_2 injection compression processes within a FPSO, in order to identify and quantify the irreversibilities of each equipment and propose alternatives to increase the overall exergy efficiency. The analysis is done for a theoretical CO_2 mass flow of 55kg/s, handled by a four stage booster compression train driven by gas turbine, with compression ratio 62,7:1, and a main injection compression train driven by medium-voltage electrical motors, in turn powered by gas turbo-generator electric power outlet, and a compression ratio 1,4:1. Three different values of CO_2 mole fraction were modeled. Compression systems are one of the most energy-intensive processes therefore the results of availability assessments contribute to improve them.*

Keywords: *CO_2 compression, exergy, FPSO.*

1 INTRODUCTION

The Oil & Gas exploration and production in offshore platforms is characterized by using large amounts of energy for physical and chemical processes, usually supplied by gas turbines based generation systems in open cycle. Several studies analyzing the technical, economic and environmental gains to do, first, heat recovery from the exhaust gases (Pierobon et al. 2014), and second, the capture and storage of CO_2 (Kvamsdal et al. 2007) due to the fact that the foretold generation brings the emission of gaseous products of combustion as well. However, the implementation and review of these alternatives in offshore have not been extensively deployed as in other industries (Kuramochi et al. 2012; Berghout et al. 2013).

A second law analysis by simulation was performed by comparing the performance of a platform with CO_2 capture and storage with the case without CCS. Moreover a general process of main separation was modeled including gas compression, oil pumping, gas turbines and CO_2 capture system with MEA (Mono-Ethanol Amine) (Carranza & Oliveira Jr 2014). The analysis concluded that the separation train is the main contributor of exergy destruction with and without CO_2 capture. Also is noteworthy that the implementation of the CO_2 capture system brings a significant increase on the platform exergy destruction. The 75% reduction achieved in CO_2 emissions is penalized with a reduction in exergy efficiency of 2.7%. The system makes the platform more favorable (2.5%) in environmental terms, taking as a basis for comparison the Energy Renewability Index.

Specifically in CO_2 compression, have been developed reviews of compression alternatives about intercooling chain compression and Shockwave compression (Baldwin 2009), as well as their waste heat recovery opportunities and their exergetic performance (Pei et al. 2014).

In the Brazilian scenario, new E&P developments brought an important challenge: The high CO_2 content in the well fluid, requiring separation and compression processes for reinjection into reservoirs. Several studies have been developed analyzing exergy efficiency of offshore platforms, but in cases where the CO_2 content were negligible (Nguyen et al. 2013; Voldsund et al. 2013). In this article, we describe the design of an offshore CO2 compression system, and simulate the mass, energy and exergy balances with the software Aspen Hysys® and EES®. It was possible to quantify the exergy destruction and exergy losses in the system, and to comment some alternatives of improvement and/or integration.

2 UNIT DESCRIPTION

The studied CO_2 compression unit input gas stream comes from a CO_2 removal unit (by membranes) to be compressed in two identical trains operating at 2x50% loading rate. As seen in *Fig. 1* the gas enters in the first stage vertical scrubbers and leaves in the same composition, considering negligible the liquid outlet in the normal operating condition. The CO_2 pressure is raised in a first stage centrifugal compressor, and is cooled later on printed circuit heat exchangers. At this point, the first compression stage is deemed complete, and the process is repeated in the following three stages, each time with a pressure at a ratio of 2.86: 1. Therefore, each train has four vertical scrubbers (V-01-A/B to V-04-A/B), four intercoolers (HX-01-A/B to HX-04-A/B) and a multi-stage centrifugal compressor (C-01-A/B). The compressors are coupled mechanically to two gas turbines (GT-01A/B).

Each condensate output is recycled upstream of the immediately preceding stage vessel, and there are also facilities for recirculating each compression stage to the previous, but are not considered in the simulation process for being intermittent streams. The compressed CO_2 is led to the injection compression unit.

The injection gas compression unit (*Fig. 2*) has two inlets, one hydrocarbon gas stream discharged from the exportation gas compression unit, and the other coming from the CO_2 compression unit. In the operation mode shown in this article, it is considered that the feed comes solely from the CO_2 compression unit stream. The flow is directed to the vertical suction scrubber (V-05) to be compressed after in the injection centrifugal compressor (C-02), with a compression ratio of 1:39: 1. Once compressed, the gas is cooled with water in the printed circuit heat exchangers (HX-05-A/D) for finally being injected in the Water and Gas Injection (WAG) wells.

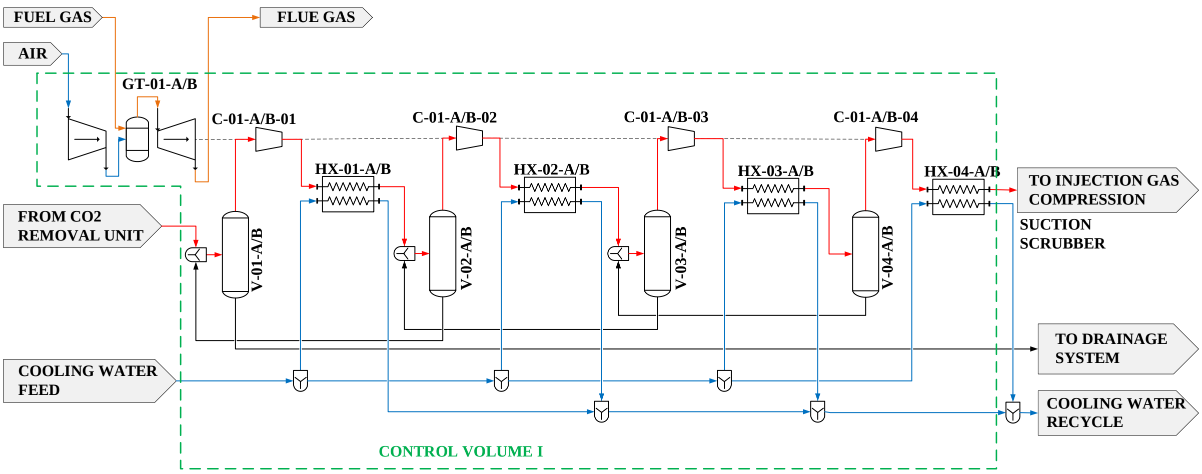

Figure 1. CO_2 booster compression Unit

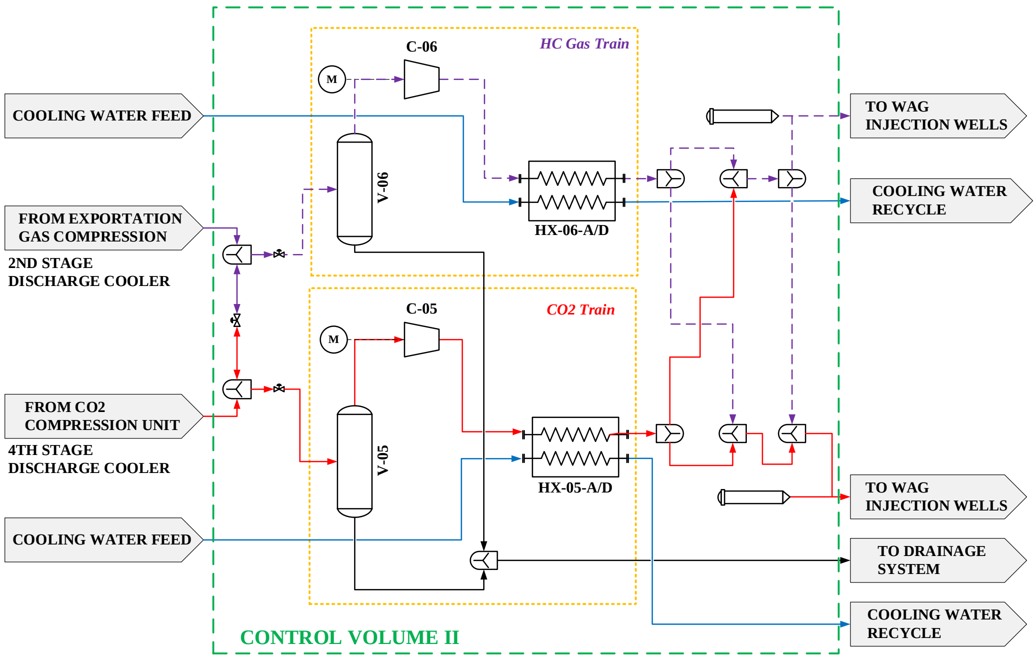

Figure 2. CO_2 Injection compression Unit

3 FLUID CHARACTERIZATION

The compositions and intensive properties of the fluids were obtained from basic project Process Flow Diagrams (PFD) of a FPSO platform. The thermodynamic properties of mixtures, such as enthalpy, entropy, molecular weight, etc., were obtained from the simulation in Aspen Hysys® software and mass, energy and exergy balances were modeled and confronted on EES® software.

Produced CO2: The data correspond to a production scenario with composition and properties of the permeated CO2 gas inlet of the compression system as specified in *Tab. 1*. The compositions correspond to different well production forecasts and at three different stages of field. The mass flow rate, suction and discharge pressures and temperatures of the heat exchangers were kept constant.

Table 1. Composition and properties of inlet stream

	Units	**Case 1**	**Case 2**	**Case 3**
Properties				
Molecular weight	*kg/kmole*	36,13	34,39	39,38
Mass flow	*kg/s*	55,54	55,54	55,54
Pressure	*kPa*	400,00	400,00	400,00
Temperature	*°C*	40,00	40,00	40,00
Specific enthalpy	*kJ/kg*	-8312,46	-8081,38	-8614,05
Specific entropy	*kJ/kg-K*	5,8124	6,1474	5,2525
Composition				
Carbon dioxide (CO_2)	*mole%*	70,81	63,82	83,07
Water (H_20)	*mole%*	0,00	0,00	0,00
Nitrogen (N_2)	*mole%*	0,20	0,55	0,28
Methane (C1)	*mole%*	27,50	33,40	16,35
Ethane (C2)	*mole%*	1,21	1,56	0,14
Propane (C3)	*mole%*	0,21	0,49	0,12
Butane (nC4)	*mole%*	0,03	0,04	0,01
Iso-butane (iC4)	*mole%*	0,02	0,12	0,002
Pentane (nC5)	*mole%*	0,01	0,01	0,00
Iso-pentane (iC5)	*mole%*	0,00	0,01	0,00
C6+	*mole%*	0,00	0,00	0,00

Fuel gas and combustion air: The fuel gas can be associated with production or imported by pipeline. Once it is dehydrated, adjusted its dew point and removed the CO_2, the composition and properties stand as in *Tab. 2*. The combustion air is simulated according to geographical location and meteorological data.

Table 2. Composition and properties of fuel gas and combustion air

	Units	**Fuel Gas**	**Air**
Composition			
Nitrogen (N_2)	*mole%*	0,56	75,629
Carbon Dioxide (CO_2)	*mole%*	3,00	0,030
Water (H_2O)	*mole%*	$3{,}72\text{x}10^{-5}$	2,343
Oxygen (O_2)	*mole%*	-	20,459
Argon (Ar)	*mole%*	-	0,918
Methane (C1)	*mole%*	75,68	-
Ethane (C2)	*mole%*	10,96	-
Propane (C3)	*mole%*	6,65	-
n-Butane (nC4)	*mole%*	0,92	-
i-Butane (iC4)	*mole%*	1,55	-
n-Pentane (nC5)	*mole%*	0,27	-
i-Pentane (iC5)	*mole%*	0,42	-
Properties			
Pressure	*kPa*	4532	101,32
Temperature	*ºC*	70	30
Molecular weight	*g/mole*	21,78	28,62
Density	*kg/m³*	38,20	1,14
Dew point	*ºC*	-	25,08
Relative humidity	*%*	-	75
Low heating value	*kJ/kg*	$4{,}53\text{ x}10^{4}$	

Colling water: Exists a cooling water circuit for gas and CO_2 compression facilities. The mass flow is adjusted by control action, keeping constant the temperatures. The properties of that water are shown in *Tab. 3*.

Table 3. Cooling water properties

Property	**Units**	**Value**
Pressure	*kPa*	882,6
Inlet Temperature	*ºC*	35
Return Temperature	*ºC*	55
Molecular weight	*g/mole*	18,02
Density	*kg/m³*	955,6

4 MASS, ENERGY AND EXERGY BALANCES

The following assumptions for process modeling were taken:

- All processes are considered in steady state.
- The stream compositions were obtained from design flow diagrams and the properties obtained from the simulation in software Aspen Hysys®, considering the gas as real fluids and using the equation of state (EOS) BWRS (Benedict-Webb-Rubin-Starling), recommended for gas compression applications. The gas turbines were modeled with EOS Peng-Robinson.
- The recycle lines of equipment or entire processes are not considered in the balance, because its operation is not continuous.
- The variation of potential and kinetic energy are considered negligible.
- The liquid outlet (when considered) of the separator vessels was considered continuous, without assessing the action of valves and level control instrumentation.
- The characteristics of the gas turbine was obtained from the Thermoflex® database.
- The convention adopted for signs of heat and work components are:
    - $Q>0$ when it is made from the surroundings to the system and $Q<0$ otherwise;
    - $W>0$ when it is made from the surroundings to the system and $W<0$ otherwise.

The mass balance has the form (Klein & Nellis 2012):

$$\sum_{i=1}^{\#inlets} \dot{m}_{in,i} = \sum_{i=1}^{\#outlets} \dot{m}_{out,i} + \frac{dm}{dt} \tag{1}$$

Where $\dot{m}_{in,i}$ and $\dot{m}_{out,i}$ are the mass flows entering and leaving the system, crossing the surface control, and dm/dt is the mass variation rate of the system. The mass flow of the gas remains constant in the compression and inter-cooling stages without water or liquid hydrocarbons condensation in separator vessels. It is presented below a summary of the mass flows of fuel gas in turbines to achieve the power demand of compressors and water mass flows in inter-cooling heat exchangers required to achieve the gas discharge temperature between compression stages.

Table 4. Fuel gas and cooling water mass flow

Equipment	**Mass flow [kg/s]**		
	Case 1	Case 2	Case 3
HX-01-A/B	79,39	83,55	72,93
HX-02-A/B	86,82	90,90	80,63
HX-03-A/B	111,60	113,86	110,69
HX-04-A/B	133,50	133,18	136,34
HX-05-A/D	24,75	25,04	24,04
Total water	*436,06*	*446,53*	*424,63*
GT-01-A/B	1,58	1,67	1,44
Total fuel gas	*1,58*	*1,67*	*1,44*

**The notation A/B indicates two identical equipment working in parallel. The values correspond to the total mass flow.*

For the first four exchangers, which have similar enthalpy change, the cooling water flow increases as the pressure increases in all three cases. The flow of fuel gas, linked to the energy consumption of compressors is higher in Case 2, which has higher methane content.

The energy balance is of the form (Klein & Nellis 2012):

$$\sum_{i=1}^{\#inlets} \dot{m}_{in,i}\left[h_{in,i}\right]+\sum_{i=1}^{\#heat\ terms}\dot{Q}_i=\sum_{i=1}^{\#outlets}\dot{m}_{out,i}\left[h_{out,i}\right]+\sum_{i=1}^{\#work\ terms}\dot{W}_i \tag{2}$$

It is considered the internal and external energy associated with the mass that crosses the surface control, and the system's possibility of interaction with the surroundings as heat or work. Are presented as results the power required by the compressors, the heat transferred in the heat exchangers between gas and cooling water streams and the isentropic efficiency of the compressor in the following Table:

Table 5. Energy balance for the three cases

Equipment	**Energy Streams**					
	Work/Heat [MW]			Efficiency [%]		
	Case 1	Case 2	Case 3	Case 1	Case 2	Case 3
Compressors						
C-01-A/B-1ª Stage	6,21	6,56	5,67	74,74	74,40	74,97
C-01-A/B-2ª Stage	6,18	6,53	5,62	73,31	73,01	73,48
C-01-A/B-3ª Stage	5,81	6,18	5,24	72,34	72,15	72,23
C-01-A/B-4ª Stage	4,99	5,27	4,69	69,51	72,00	61,77
C-02	1,30	1,23	1,49	71,74	83,86	51,57
Total Work	*24,49*	*25,77*	*22,71*	-	-	-
Heat exchangers						
HX-01-A/B	6,53	6,87	5,99	-	-	-
HX-02-A/B	7,14	7,47	6,63	-	-	-
HX-03-A/B	9,18	9,36	9,10	-	-	-
HX-04-A/B	10,97	10,95	11,21	-	-	-
HX-05-A/D	2,03	2,06	1,97	-	-	-
Total Heat	*35,85*	*36,71*	*34,90*			
Turbines						
GT-01-A/B	-23,20	-24,54	-21,22	32,36	32,38	32,49

**The notation A/B indicates two identical equipment working in parallel. The values correspond to the total mass flow.*

The largest energy demand (heat + work) was found in Case 2, followed by cases 1 and 3 respectively, in descending order of methane content in the mixture. The difference in isentropic efficiency becomes more sensitive the greater the pressure was, evidenced in the early compression stages where efficiencies were very close to each other. The fuel gas composition was kept constant, therefore the thermal efficiency of the turbines only changed by the change in energy demand.

For the calculation of exergy, the total flow of exergy of a stream is defined by the following equation:

$$\dot{B}=\dot{B}_k+\dot{B}_p+\dot{B}_{ph}+\dot{B}_{ch} \tag{3}$$

Which has contribution of kinetic, potential, physical and chemical exergy. The first two, according to the assumptions made for the mass and energy balances can be neglected. The physical exergy in their specific thermal and mechanical components may be defined as follows (Ghannadzadeh et al. 2012):

$$b^{\Delta T}=h(T,P,z)-T^0 s(T,P,z)-\left[h\left(T^0,P,z\right)-T^0 s\left(T^0,P,z\right)\right] \tag{4a}$$
$$b^{\Delta P}=h\left(T^0,P,z\right)-T^0 s\left(T^0,P,z\right)-\left[h\left(T^0,P^0,z\right)-T^0 s\left(T^0,P^0,z\right)\right] \tag{4b}$$

Being the total physical exergy the sum of thermal and mechanical ones (Kotas 1985):

$$b_{ph}=b^{\Delta P}+b^{\Delta T}=h(T,P,z)-T^0 s(T,P,z)-\left[h\left(T^0,P^0,z\right)-T^0 s\left(T^0,P^0,z\right)\right] \tag{5}$$

Where *h* is the enthalpy, *s* is the entropy and the subscript 0 indicates the dead state. The chemical exergy of a mixture, by mole of gas is given by:

$$b_{ch}=\sum_i x_i b_{ch}^i+\bar{R}T_0\sum_i x_i \ln \gamma_i x_i \qquad (6)$$

Where x_i is the molar fraction of the i-th component, b_{ch}^i is the standard chemical exergy of the component and γ_i is its activity coefficient (Szargut et al, 1988).

The exergy physical and chemical fractions and its flow by each equipment were calculated, finding for each of them the exergetic efficiency and irreversibilities. In the literature, various formulations have been proposed for exergetic efficiency. A more general expression, called *total* or *universal* in the literature, is the relationship between all exergy inputs and outputs as follows:

$$\varepsilon=\frac{\dot{B}^{out}}{\dot{B}^{in}} \qquad (7)$$

However, it does not provide a clear view of cases where a significant amount of wastes or losses are produced. It is then proposed the functional exergetic efficiency, which is the ratio between the desired exergy output and exergy used. It allows to evaluate the performance of the processes used in the CO_2 compression and injection system.

$$\psi=\frac{Desired\ Exergetic\ Effect}{Used\ Exergy}=\frac{\Delta B_{desired\ output}}{\Delta B_{used}} \qquad (8)$$

Where $\Delta B_{desired\ output}$ is the result produced by the system and depends on the equipment function, and ΔB_{used} represents the net resources invested in order to generate that effect.

Table 6 summarizes the expressions to calculate the numerator and denominator of exergetic efficiencies for equipment considered in the process (Ghannadzadeh et al. 2012). The scrubbers are not considered as separators, due to the simulation results of the mass balance, which does not show separation of liquid flow in them, and because in case it happens, it would be accumulated until be emptied by a control action under intermittent flow. It was not calculated exergy efficiency in "virtual" mixing and separation processes because these processes are performed in constant pressure and temperature and there is no exergy destruction in them.

Table 6. Exergy efficiency for the simulated unit operations

Unit Operation	$\boldsymbol{\Delta B_{desired\ output}}$	$\boldsymbol{\Delta B_{used}}$
Expansion: Gas turbine, shaft work.	$\dot{W}_{shaft}$	$\dot{B}_{in}-\dot{B}_{out}$
Adiabatic compression: exergy increment	$\dot{B}_{out}-\dot{B}_{in}$	$\dot{W}_{shaft}$
Heat exchange between two streams: cooling	$B_{hot\_in}^{\Delta T}-B_{hot\_out}^{\Delta T}$	$(B_{cold\_out}-B_{cold\_in})$ $+(B_{hot\_in}^{\Delta P}-B_{hot\_out}^{\Delta P})$

Are presented in *Tab. 7* and *Fig. 3* the external and internal irreversibilities of the equipment, and exergetic efficiency associated to each one:

Table 7. Irreversibilities and exergy efficiency of unit operations

Equipment	**Irreversibilities [kW]**			**Exergy efficiency [%]**		
	Case 1	Case 2	Case 3	Case 1	Case 2	Case 3
Internal						
C-01-A/B-1ª Stage	1160,74	1242,43	1049,73	81,32	81,06	81,50
HX-01-A/B	913,63	962,39	837,16	12,62	12,42	12,99
C-01-A/B-2ª Stage	1212,95	1297,98	1096,59	80,36	80,13	80,50
HX-02-A/B	881,29	925,47	814	10,14	9,61	11,05
C-01-A/B-3ª Stage	1175,99	1258,82	1062,63	79,78	79,63	79,72
HX-03-A/B	1025,74	1058,85	983,94	23,66	21,54	28,71

Equipment	Irreversibilities [kW]			Exergy efficiency [%]		
	Case 1	Case 2	Case 3	Case 1	Case 2	Case 3
C-01-A/B-4ª Stage	1133,38	1096,5	1340,08	77,31	79,18	71,44
HX-04-A/B	1100,98	1108,06	1113,6	46,53	42,24	55,66
C-02	333,63	180,25	661,39	74,34	85,38	55,84
HX-05-A/D	25,95	26,70	25,13	39,50	37,21	42,20
GT-01-A/B	31155,04	32899,00	28238,08	30,84	30,86	30,96
SUBTOTAL	*40119,38*	*42056,41*	*37222,33*			
External						
GT-01-A/B (Exhaust gases)	21065,51	22279,97	19273,65	-	-	-
HX-01/05-A/B (Water return)	3947,59	4081,47	3773,83			
TOTAL	**65132,42**	**68417,89**	**60269,81**			

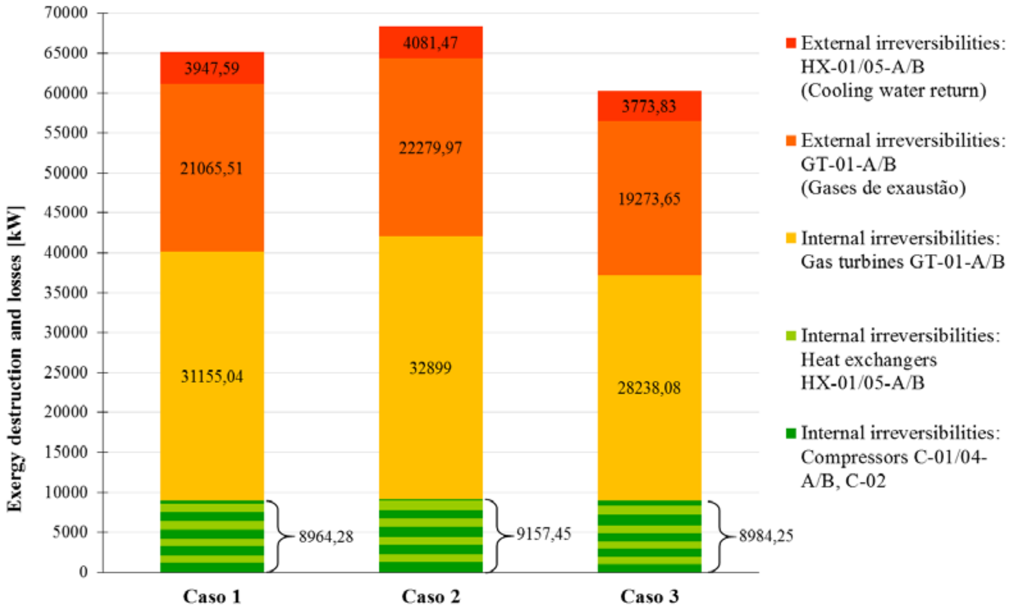

Figure 3. Irreversibilities of unit operations

It was observed that gas turbines are the main contributors of exergy destruction and losses in the process. The exergy destroyed in the turbines (internal irreversibilities) are around three times the sum of the irreversibilities of compressors and heat exchangers. Internally, the process of compressing the combustion air is responsible for about 9.5% of irreversibilities, combustion chamber 72.5% and the expansion itself of 18%. The discharge of exhaust gases (external irreversibilities), in turn, is of the order of twice the sum of the irreversibilities of compressors and heat exchangers.

The exergy efficiency, considered as isentropic is also more sensitive to the composition of the mixture at higher pressures, showing significant difference among the three cases only on the third compression stage.

It was quantified the exergy losses of atmospheric discharge of exhaust gases and return of cooling water with a higher temperature. Both losses could be reduced or eliminated by integrating that streams to processes with needs of heating, for example using steam or organic Rankine cycle power modules in an optimized design. Studies with guidelines for the design of high-efficiency, cost-competitive and low-weight power systems for offshore installations are available (Pierobon et al. 2014).

5 CONCLUSION

A first and second law analysis of a multi-stage CO_2 compression system with inter-cooling has been developed, quantifying the energy consumption as heat and power to compress the CO_2 stream for reinjection, and its variations with fluid composition. It was concluded that the driving unit (gas turbine) is the principal source of internal and external irreversibilities in the process. An alternative to changing drive unit for electric motors could mean reduction of irreversibilities, if is integrated into the main platform generation system, reviewing the load balance with consideration of gas and CO_2 flow decline with time.

Exergy destruction correspond to 61% and exergy losses to 39% of the total irreversibilities, being the gas turbines solely, responsible of 48% approximately. The selection of high efficiency turbine packages in nominal and partial load is of big importance to the exergy performance. Heat transfer with the environment is almost negligible.

An important aspect to be studied in future studies is the integration of CO_2 compression associated to oil production with a capture and sequestration system. Due to the need of sizing the compression train for the maximum CO_2 mass flow, could be avoided partial load operation, reduce irreversibilities and CO_2 emissions by CCS, not only of turbo compressors exhaust gases but also of the turbo generator's one.

6 ACKNOWLEDGEMENTS

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