USE OF THE AVAIABLE ENERGY IN THE RE-GASIFICATION PROCESS OF LIQUEFIED NATURAL GAS APPLIED TO COMBINED HEAT AND POWER PLANTS (CHP)

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Abstract. *This work evaluates the possibilities of taking advantage of the heat absorbed by the Liquid Natural Gas (LNG) in the regasification process. A Brayton-Rankine Combined Heat and Power Plant (CHP) using environmental air as cold source is the reference point. It is proposed the coupling of this CHP plant to a LNG regasification plant in two cases that use the heat absorbed by the LNG during its regasification as cold source for the CHP plant. In the first case the cold source was used only to lower the temperature in the condenser of the Rankine sub-cycle, and in the second case the cold source from the LNG stream is used in addition to cool down the inlet air in the compressor of Brayton sub-cycle, in both cases an expansion turbine is used for pressure reduction. Assuming the ammount of natural gas fed to the combustion chamber equal for all three cases as comparison reference, an energetic analysis of the systems was made, where the case 1 achieved a total net power generation of 20.6 MW, an efficiency of 41.1% and a NG regasification mass rate of 5.2 kg/s; in the second case the net power generation was 22.3 MW, the efficiency was 44.6% and the NG regasification mass rate was 6.5 kg/s.*

Keywords*: coupled systems, Liquid Natural Gas, energetic analysis, energy recovery, CHP*

1. INTRODUCTION

Natural Gas (NG) is a fossil fuel composed by a mixture of methane, ethane, propane and other heavy hydrocarbons. It displays the lowest Greenhouse Gases emission per tonne of burned gas among usual carbon based fuels, like oil and coal (EIA, 2009),. In order to meet environmental restrictions and also due to its competitive costs, NG use has increased in the past decades.

The usual way of NG transportation is by gas pipelines, but whenever the distances between the end use consumers and the extraction sites are important, the liquefaction of NG for long range transportation can be an alternative. NG volume is reduced by a factor of 600 on its liquid phase and is called as Liquefied Natural Gas (LNG). The process of liquefaction consists on compression and heat removal, which requires a large amount of energy and economic resources, e.g., a liquefaction plant producing 7 million tonnes per year of gas costs approximately US$1.92 billion according to GASNET, 2009.

Before being consumed, LNG needs to be re-gasified. Commonly, regasification plants use sea water as a hot source, which exchanges heat with LNG in a set of heat exchangers. At the end of the process this water is returned to the sea at a lower temperature level. This results in a waste of energy and is also a thermal pollution source affecting the environment near the plant.

It is known that it is possible to improve the efficiency of a Rankine power cycle by lowering the temperature of heat rejection of its working fluid and it is also possible to improve the efficiency of a Brayton power cycle by decreasing the temperature of the inlet air. In conventional power plants both mentioned temperatures are bounded by environmental conditions.

This work proposes the use of LNG as a cold source for a Brayton-Rankine Combined Heat and Power Plant (CHP), whose heat rejection will help on the regasification process of LNG. The energy availability of the process is quite large as the depart state of LNG is cryogenic (temperature about -160 ºC for pressures of 7 atm) and its final dead state is at ambient temperature and pressure.

Hisazumi *et al*, 1998, proposed a Rankine cycle power plant, using a chlorofluorocarbon (Freon) as working fluid and vaporizing LNG throughout the cycle condenser. Their proposed cycle reached a thermal efficiency of 53%. Miayazaki *et al.*, 2000, made a similar experience using a mixture of water and ammonia as working fluid, obtaining an increase of 53% on the thermal efficiency when comparing to a conventional cycle. Acunha Jr. *et al.*, 2008, proposed the use of the LNG for cooling down the inlet air of a Brayton power cycle. An increase of 4% in the power generation was calculated, corresponding to a gain of 1.27 MW.

2. COMBINED HEAT AND POWER PLANTS (CHP)

According to Verbruggen, 2008, every thermal power process rejects fatal heat in the environment. The merit of CHP is to recover part or all of this fatal heat and convert it into 'useful' heat. In Fig. 1, a simplified Brayton–Rankine CHP cycle is proposed in order to explore the LNG heat recovery concept.

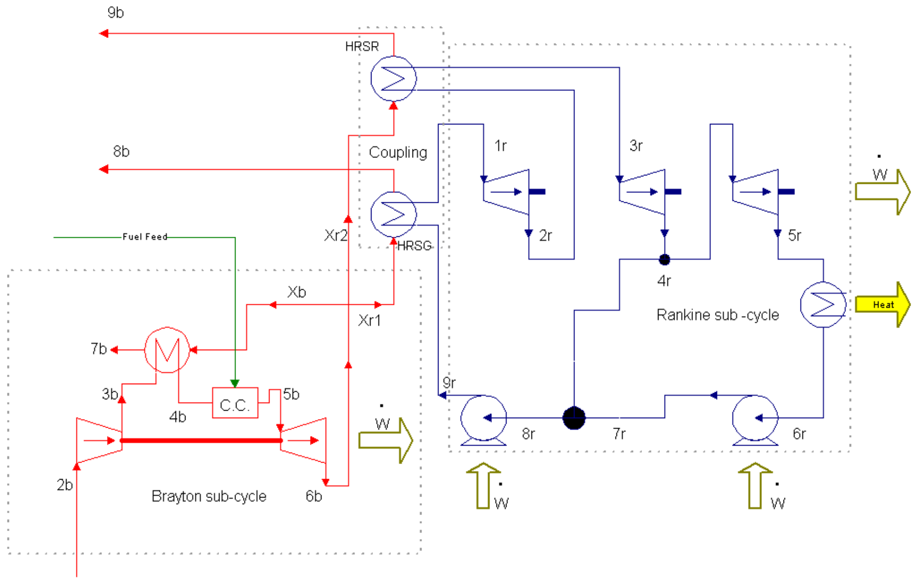

Figure 1. Brayton-Rankine CHP.

The Brayton sub-cycle is pictured with red streams on the dashed line rectangle on the left of Fig. 1, Air is admitted at point *2b* and NG is added at the combustion chamber (CC). The simplified cycle is completed by a compressor, an expansion turbine and a heat recovery exchanger. The Rankine sub-system is presented by blue streams on the right of Fig. 1 in a schematic assembling, based on the essential devices of the cycle. A heat recovery steam generator (HRSG) is the coupling device to the Brayton sub-cycle, along with the heat recovery steam re-heater (HRSR). A simplified three group turbine, two pumps (low and high pressure) and a steam condenser complete the cycle, whose working fluid is water.

In this CHP cycle, the heat is provided solely by NG at the combustion chamber on the Brayton sub-cycle. The flue gases exhausted by this process are splited in three main heating streams: one enters in the HRSG and generates steam for the Rankine sub-cycle, a second heating circuit that goes to the HRSR between first and second stages of the Rankine sub-cycle turbine and a third one preheats air for combustion. Streams 7b, 8b and 9b are flue gases discharged to the environment. Heat rejection of the Rankine sub-cycle takes place at the steam condenser.

The modeling assumptions are as follows: steady state operation; air is considered as an ideal gas and its humidity is neglected; the LNG composition is regarded as pure methane (CH_4); friction losses and pressure drops through pipes and equipments are neglected.

The CHP simulation involves basic thermodynamic relations for modeling the components. Turbines and pumps on both sub-cycles are modeled considering that the power is the product of the mass flow rate to the enthalpy variation.

The outlet air temperature T_o of the Brayton compressor is given by Cohen *et. al.*, 1996, in (1), where T_i is the temperature at the inlet point, η_c is the compressor isentropic efficiency, r is the compression ratio and γ_c is the specific heat ratio.

$$T_o = T_i\left[1 + \frac{1}{\eta_c}\left(r^{\frac{\gamma_c - 1}{\gamma_c}} - 1\right)\right] \quad (1)$$

The compressor volumetric flow rate Q_V is given by (2) as

$$Q_V = \frac{\dot{m}_{airB}}{\rho_{air}} \quad (2)$$

where $\dot{m}_{airB}$ is the air mass flow rate and ρ_{air} its density, at the inlet temperature.

The net power in the Rankine sub-cycle $\dot{W}_{netR}$ is given by the output of the turbine less the power of the pumps. In the same way, the net power in the Brayton sub-cycle $\dot{W}_{netB}$ is the difference between the total power of the turbine to the power of the compressor. CHP net power $\dot{W}_{net}$ generationis given by these two contributions, as shown in (3):

$$\dot{W}_{net} = \dot{W}_{netB} + \dot{W}_{netR} \tag{3}$$

The heat transferred in each heat exchanger is modeled by the product of the mass flow rate to the enthalpy variation of the fluid.

Heat added to the system is modeled in just as the product of the lower heat value of methane (50 022 kJ/kg) to its mass flow rate (1.0 kg/s). Flue gas mass flow rate is the summation of the mass flow rates of the air stream and the NG stream. The combustion process only increases the enthalpy of air.

The efficiency of each sub-cycle is given by the ratio of the net work produced and the heat consumed. For the CHP cycle, the efficiency is given by the net power produced and the total heat $\dot{Q}_{CC}$ provided by the combustion (4).

$$\eta_{CHP} = \frac{\dot{W}_{net}}{\dot{Q}_{cc}} \tag{4}$$

As mentioned earlier, the use of coupled systems searches to reduce energy waste due to the flue gases from the Brayton exhaust. The Rankine sub-cycle also rejects heat by the steam condenser device, and there is a possibility of taking advantage of this rejected heat by coupling a process that needs a hot source. The purpose of this work is to study two possibilities, among many others, of coupling a CHP plant to a LNG re-gasification plant.

3. CASE STUDIES

3.1. Brayton – Rankine cycle + LNG. Case 1

The first proposed case of CHP performance improvement is the coupling of a LNG regasification plant to the condenser of a Rankine sub-cycle, as shown in Fig. 2.

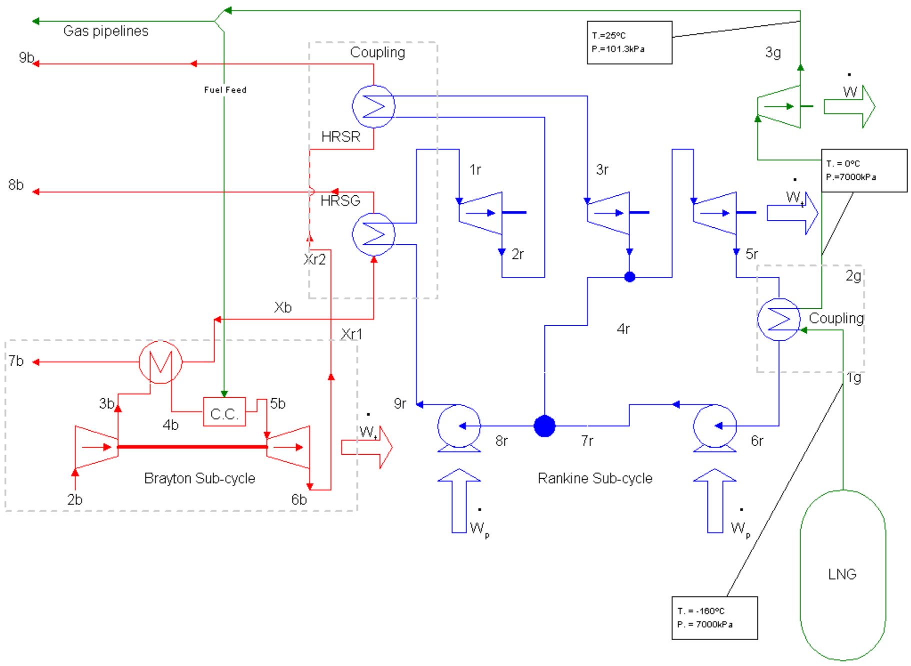

Figure 2. CHP cycle coupled to a LNG re-gasification plant – case 1.

In this cycle, the LNG re-gasification system is depicted by green lines at the point as heat rejected by the Rankine cycle condensation process is used to promote phase change of the LNG in a heat exchanger. Then the LNG undergoes an expansion at an auxiliary turbine where its pressure is lowered and work is produced. One part of the re-gasified NG feeds the combustion chamber and the remaining is ready to be distributed by a gas duct.

This new cycle is based on the one shown in Fig. 1. An increase on the cycle efficiency and on its net power is expected, not only because of the auxiliary turbine, but also due to the improvement of the CHP plant.

3.2. Brayton – Rankine coupled cycle + LNG. Case 2

In this second proposal a larger improvement in the cycle performance is expected though the addition of a heat exchanger for cooling down the inlet air in the compressor of the Brayton sub-cycle. Figure 3 shows this alternative arrangement, sketched in green lines.

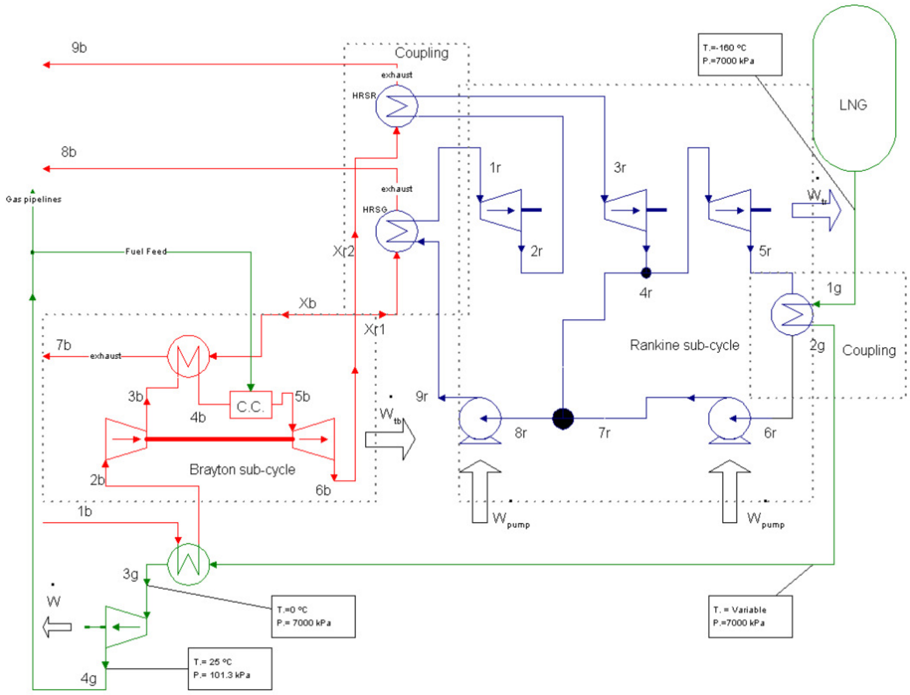

Figure 3. CHP cycle coupled to a LNG re-gasification plant – case 2.

The aim of cooling the inlet air, from points 1b to 2b, is to increase its density and though reduce the compressor volumetric flow rate. A smaller compressor, with lower power consumption, leads to an increase in the efficiency and power generation of this options, if compared to the previous CHP plant.

4. RESULTS AND DISCUSSION

For comparison purposes the mass flow rate of NG burned in the combustion chamber was fixed and identical for the three studied cases, providing the same amount of heat input to all three cycles is equal.

The simulation results obtained for the three cases are displayed in Tab. 1. The Brayton sub-cycle air mass flow rate $\dot{m}_{airB}$ and the Rankine sub-cycle water mass flow rate $\dot{m}_{H_2OR}$ were 54.5 kg/s and 2.7 kg/s, respectively. The heat absorbed by both the HRSG and the HRSR was 10.139 MW.

Table 1. Simulated values of the most significant parameters of the proposed systems (1 kg/s NG mass flow rate).

Parameter	Value		
	Reference Cycle	Case 1	Case 2
$\dot{m}_{NG}$ to pipeline (1)	-	4.2 kg/s	5.5 kg/s
$\dot{W}_{net}$	19.837 MW	20.566 MW	22.298 MW
$\dot{W}_{netB}$	15.576 MW	15.576 MW	17.147 MW
$\dot{W}_{netR}$	4.261 MW	4.332 MW	4.332 MW
$\dot{W}_{netLNG}$	-	0.658 MW	0.819 MW
η_{CHP}	39.7%	41.1%	44.6%
η_B	31.1%	31.1%	34.3%
η_R	42.1%	42.7%	42.7%
Q_V compressor	46 m³/s	46 m³/s	42 m³/s

(1): $\dot{m}_{NG}$ re-gasified minus $\dot{m}_{NG}$ burned in the combustion chamber.

Alternatives 1 and 2 take the advantage of the regasification of LNG against the heat rejection of the Rankine cycle. That strategy allows for lowering the condensing pressure and temperature of the working fluid, and delivered an extra 71 kW of net power on both cases. The heat exchanged for cooling the inlet air of the Brayton system was even more effective, with an increase of 1.57 MW for case 2. The use of the auxiliary turbines as expansion devices can bring an extra net power (0.658 MW and 0.819 MW for cases 1 and 2), but the output NG will be delivered at ambient pressure.

Case 2 presents the highest efficiency when compared to the reference cycle; it was possible to gain 2.46 MW in power generation with efficiency 11% superior than the reference cycle, burning the same amount of NG. With this arrangement it was possible to re-gasify 6.5 kg/s of NG, instead of 5.2 kg/s re-gasified in case 1.

By analyzing the parameters and the results obtained in all three simulations it is possible to identify that two factors are responsible for the largest gains in the cycle presented in case 2: the air inlet temperature and the exhaust pressure of the Rankine cycle low pressure turbine.

The influence of air inlet temperature in the power generation and efficiency is shown in Fig. 4.

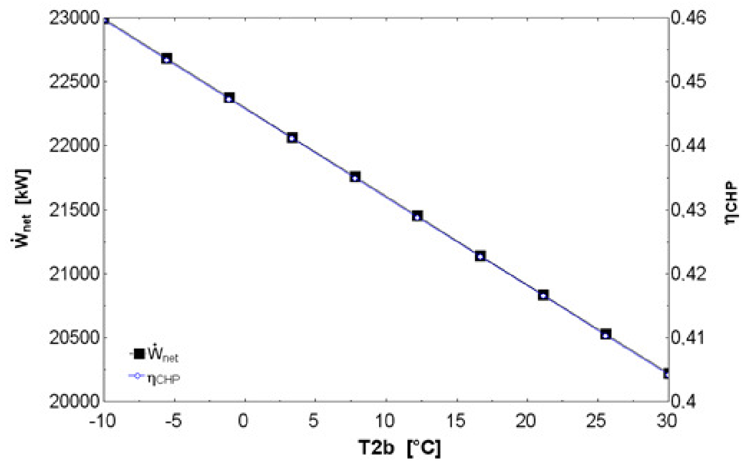

Figure 4. Effect of the inlet air temperature in power generation and efficiency - case 2.

The influence of the exhaust pressure of the Rankine sub-cycle low pressure turbine in the power generation and efficiency is shown in Fig. 5.

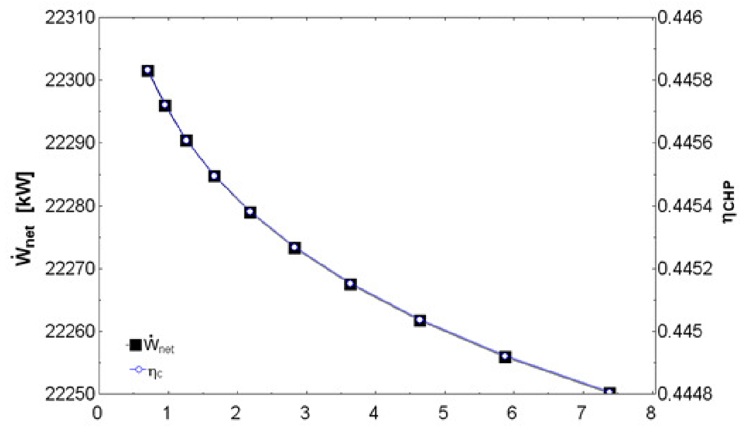

Figure 5. Effect of the outlet pressure of Rankine sub-cycle low pressure turbine on the energy production and efficiency - case 2.

The exhaust temperature of the low pressure stage of the turbine was limited due to the liquid – solid phase transition of the working fluid. To be able to work with lower temperatures a more suitable working fluid must be selected.

Finally, the behavior of the CHP cycle is more sensitive to the air inlet temperature change than any other parameter. The lower the inlet air temperature, the lower the power consumption of the compressor, as it can be noticed in Fig 6,

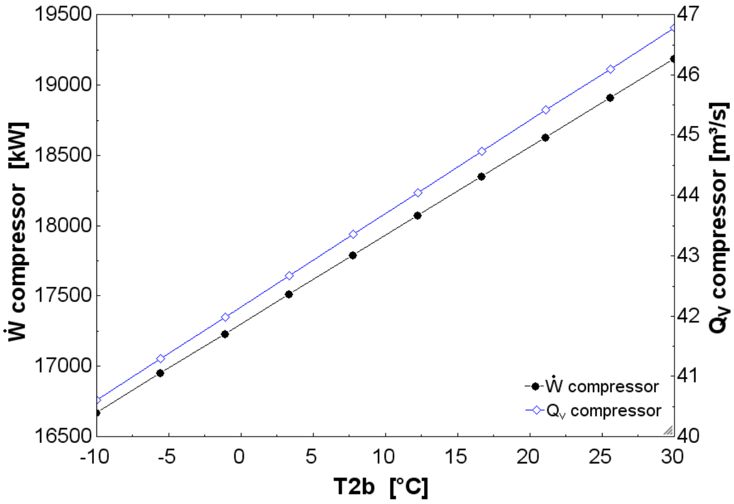

Figure 6. Effect of Brayton sub-cycle inlet air temperature on the compressor volumetric flow rate and power consumption.

The linear behavior of both the power generation and the efficiency against the temperature change is due to the linear behavior of the enthalpy change in this same interval. A complete set of data used to perform the simulations are displayed on Table A.1, at the end of this paper.

5. CONCLUSIONS

CHP plants coupled to a LNG regasification plant allow for the use of a larger part of the available energy of this last system. As a secondary gain, the energy needed for performing LNG phase change, from liquid to gas, was avoided and with it, some environmental hazardous impacts, produced by the cooling down of seawater and other thermal pollution in the environment nearby. Besides, working with LNG regasification allows for working with lower heat rejection temperatures (heat sinks), making possible to have gains due to the increase in the temperature difference between the hot and cold reservoirs. It is also possible to have performance improvements on thermal equipments due to the lowering of the working fluid density.

Two energy recovery arrangements were proposed, the first one exchanging heat in the Rankine sub-cycle condenser of a CHP plant and the other exchanging heat also for cooling the inlet air of the Brayton sub-cycle. In both options the LNG was expanded in a turbine and then distributed to the gas pipeline. The objective of both arrangements was to improve the power recovery of the LNG stream.

Best results were found in case 2, as expected. This system allows for achieving a gain of 2.46 MW in power generation with an efficiency of 44.6%, which is 1.11 higher than the reference cycle efficiency. This case also had a better performance when compared with Case 1. The comparison with Case 1 was necessary to underline the importance of cooling down the compressor inlet air.

The mass flow rate gain in the re-gasified gas in Case 2 compared to Case 1 is due to the increase in the total heat absorbed by the LNG stream during the vaporization process.

As highlighted before, the air inlet temperature in the Brayton sub-cycle compressor is the parameter that has major effect on the CHP power generation and efficiency. The inlet air cooling allows both the increase of the compressor performance and the reduction of the equipment size.

Note that, albeit the use of the LNG in the Rankine sub-cycle presents a lower influence in the performance of the plant, this is necessary for the improvement of the re-gasified natural gas mass flow.

For further works it would be interesting to make a second law study, doing an exergetic analysis together with economics (or profit) analysis.

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ANNEX

Table A.1. Fixed parameters used in the three proposed cases.

Point	Parameter	Value		
		Reference Cycle	System 1	System 2
-	$\eta_{turbines}$	90%	90%	90%
-	η_{pumps}	65%	65%	65%
-	$\eta_{compressor}$	70%	70%	70%
-	$\eta_{hx\ (heat\ exchangers)}$	70%	70%	70%
1b	Pressure	-	-	101.3 kPa
1b	Temperature	-	-	25 ºC
2b	Pressure	101.3 kPa	101.3 kPa	101.3 kPa
2b	Temperature	25 ºC	25 ºC	0 ºC
3b	Pressure	810.4 kPa	810.4 kPa	810.4 kPa
4b	Pressure	810.4 kPa	810.4 kPa	810.4 kPa
4b	Temperature	400 ºC	400 ºC	400 ºC
5b	Pressure	810.4 kPa	810.4 kPa	810.4 kPa
5b	Temperature	1200 ºC	1200 ºC	1200 ºC
6b	Pressure	101.3 kPa	101.3 kPa	101.3 kPa
6b	Mass Fraction X_B	50%	50%	50%
6b	Mass Fraction X_R	50%	50%	50%
7b	Pressure	101.3 kPa	101.3 kPa	101.3 kPa
8b	Pressure	101.3 kPa	101.3 kPa	101.3 kPa
8b	Temperature	150 ºC	150 ºC	150 ºC
9b	Pressure	101.3 kPa	101.3 kPa	101.3 kPa
9b	Temperature	300 ºC	300 ºC	300 ºC
1r	Pressure	8000 kPa	8000 kPa	8000 kPa
1r	Temperature	620 ºC	620 ºC	620 ºC
2r	Pressure	800 kPa	800 kPa	800 kPa
3r	Pressure	800 kPa	800 kPa	800 kPa
3r	Temperature	600 ºC	600 ºC	600 ºC
4r	Pressure	300 kPa	300 kPa	300 kPa
5r	Temperature	50 ºC	5 ºC	5 ºC
5r	Quality	1	1	1
6r	Temperature	50 ºC	5 ºC	5 ºC
6r	Quality	0	0	0
7r	Pressure	300 kPa	300 kPa	300 kPa
8r	Pressure	300 kPa	300 kPa	300 kPa
8r	Quality	0	0	0
9r	Pressure	8000 kPa	8000 kPa	8000 kPa
1g	Pressure	-	7000 kPa	7000 kPa
1g	Temperature	-	-160 ºC	-160 ºC
2g	Pressure	-	7000 kPa	7000 kPa
2g	Temperature	-	0 ºC	-
3g	Pressure	-	101.3 kPa	7000 kPa
3g	Temperature	-	25 ºC	0 ºC
4g	Pressure	-	-	101.3 kPa
4g	Temperature	-	-	25 ºC