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Study of a Supercritical CO2 Power Cycle Application in a Cogeneration Power Plant

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Abstract

Cogeneration, or combined heat and power (CHP), is the use of a heat engine or power station to simultaneously generate electricity and useful heat. Cogeneration is considered as a thermodynamically efficient use of fuel [1]. In separate productions of electricity, some energy must be discarded as waste heat, but in cogeneration this thermal energy is put to use. Usually the CHP plant uses steam as the working fluid and has a flexible ratio of cogeneration, which depends on the season.

In turn, supercritical CO2 (S-CO2) power cycles are proposed for future energy applications because of their high thermal efficiency [2]. However, the possibilities of carbon dioxide applications as working fluids for cogeneration power plants have never been published or expanded on.

This paper considers several S-CO2 cycles in terms of application in a CHP plant. Combined steam/S-CO2 cycles are also proposed and described, though the S-CO2 cycles are used as "bottom" cycles which only turn on when heat production is not required. The performance of several standalone cycles and combined steam/S-CO2 cycles are compared with typical steam cogeneration cycle performances. Combined steam/S-CO2 cycles, steam turbine and S-CO2 turbine features are considered.

Introduction

Supercritical CO2 operating in a closed-loop recompression Brayton cycle has the potential of equivalent or higher cycle efficiency versus supercritical or superheated steam cycles at similar temperatures [2]. The current applications of the supercritical CO2 Brayton cycle are intended for the electricity production only and the questions which are related to the building of CHP plants based on Supercritical CO2 technology were not considered yet.

CHP is the concurrent production of electricity or mechanical power and useful thermal energy (heating and/or cooling) from a single source of energy. CHP is a type of distributed generation, which, unlike central station generation, is located at located at or near the point of consumption. Instead of purchasing electricity from a local utility and then burning fuel in a furnace or boiler to produce thermal energy, consumers use CHP to improve efficiency and reduce greenhouse gas (GHG) emissions. For optimal efficiency, CHP systems typically are designed and sized to meet the users' thermal base load demand. CHP is not a single technology but a suite of technologies that can use a variety of fuels to generate electricity or power at the point of use, allowing the heat that would normally be lost in the power generation process to be recovered to provide needed heating and/or cooling. This allows for much greater improvement in overall fuel efficiency, therefore resulting in lower costs and CO2 emissions. CHP's potential for energy saving is vast.

It should be noted that CHP may not be widely recognized outside industrial, commercial, institutional, and utility circles, but it has quietly been providing highly efficient electricity and process heat to some of the most vital industries, largest employers, urban centers, and campuses. While the traditional method of separately producing useful heat and power has a typical combined efficiency of 45 %, CHP systems can operate at efficiency levels as high as 80 % (Figure 1) [1].

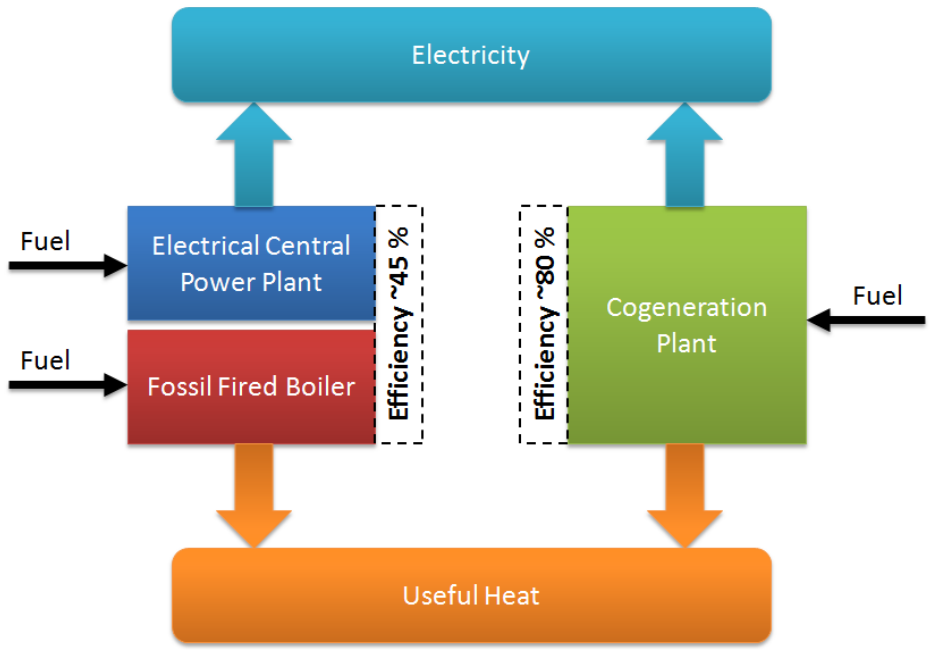


Figure 1. CHP Process Flow Diagram.

Taking into consideration the high efficiency of fuel energy utilization of CHP plants and the high potential of the supercritical CO2 technology, the latter should be also considered as the base of future CHP plants. The comparison with traditional Steam based CHP plants also should be performed.

The study of CHP plant concepts were performed with the use of the heat balance calculation tool AxCYCLE™ [3].

CHP can use a variety of fuels, both fossil- and renewable-based. However the consideration of all varieties of fuel is a quite complex task and for simplicity the only fossil fuel based CHP plants will be considered in the scope of this paper. Other source-based CHP with supercritical CO2 may be the topic of interest of future papers.

Fossil-fired Steam Power plants always include a Steam Generator (SG), which represents a complex heat exchanger for the heat transfer from the flue gas to the main working fluid (steam). A slightly simplified scheme of a typical water SG is shown on Figure 2. Consideration of SG as a complex heat exchanger allows to organize the process of heat transfer between the fluids more effectively and to take into the consideration an efficiency of SG heat utilization at the step of the conceptual cycle design.

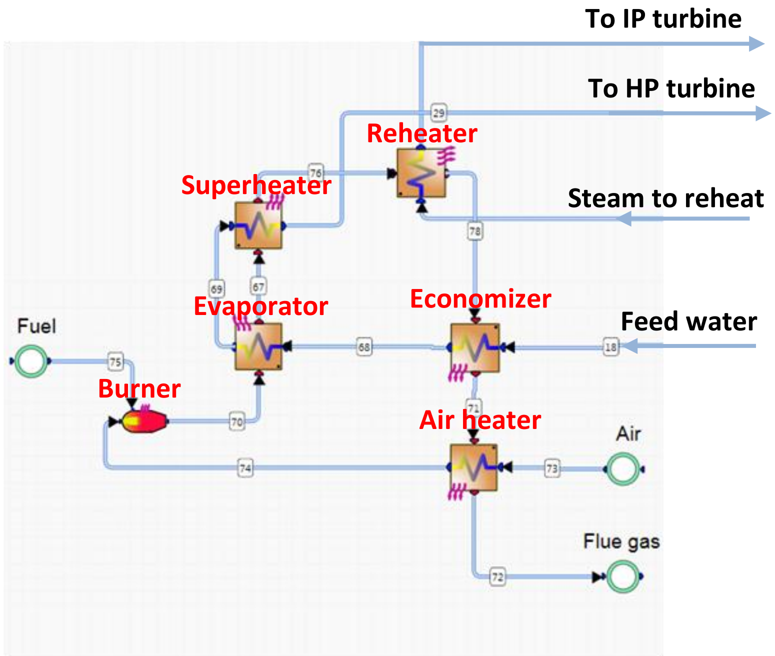


Figure 2. Fossil Fuel Fired Water Steam Generator.

Consideration of SG for a single S-CO2 cycle is especially important because of the small temperature difference in the main heater due to a high degree of the heat recuperation. Such a fact is necessary to take into consideration with S-CO2 power cycle coupling for each specific heat source. Some issues of supercritical CO2 power cycle coupling to the different heat sources were presented in the report [4].

Typical Steam Rankine cycle CHP plants are usually based on a condensation steam power cycle. In such an embodiment the cogeneration is implemented by the adjustable steam extraction for heat production. In dependency with the useful heat purposes the extraction as well as the heated water may have different parameters. For urban centers or campuses heating, the typical steam pressure of the extraction is about 0.2 MPa due to the value of the condensation temperature at this pressure (120.21 °C.) Steam from this extraction is heating the water for the consumers. After the heating the water usually has a temperature around 100°C. The typical pressure of the water for the consumers is about 1 MPa. The mentioned parameters were used as the base in this study.

In order to accommodate supercritical CO2 technology to CHP conception, a lot of configurations and approaches were considered and the following two approaches with two embodiments for each were selected as the most interesting ones:

1. **Steam Rankine cycle CHP plant with bottoming supercritical CO2 cycle**
    a. Combined Complex Steam-S-CO2 CHP Plant
    b. Combined Simple Steam-S-CO2 CHP Plant
2. **CHP plant with single supercritical CO2 working fluid**
    a. Cascaded Supercritical CO2 CHP Plant
    b. Single Supercritical CO2 CHP Plant

The cogeneration steam turbine unit T-250/300-23.5 was taken as a base for comparison. The performance warranty for both electrical power and combined heat and power modes is presented in table 1 [5].

Table 1.T-250/300-23.5 performance warranty.

Parameter	Unit	No Heat Load	Maximum Heat Load
Total heat consumption	MW	790	790
Live steam temperature	°C	540	540
Live steam pressure	MPa	23.5	23.5
Net electrical power	MW	300	253
Heat production	MW	0	384

In order to correctly compare the proposed concepts of S-CO2-based CHP plants with the ordinary CHP unit presented above we use the same total heat consumption and live steam conditions for the combined Steam-S-CO2 schemes. However, for the S-CO2 schemes the live pressure of S-CO2 was optimized and was consequently different. The efficiency of the steam turbine cylinders as well as the CO2 turbine and the compressor were taken as 90 %. The efficiency of the alternators was assigned as 98% and the heat losses for the regenerative heaters of the steam cycle part as well as for the S-CO2 recuperators were defined as 1 % from the transferred heat amounts.

1. Steam Rankine Cycle CHP Plant with Bottoming Supercritical CO2 Cycle

<u>The first concept of CHP is the following:</u> we took the same SG as in the T-250/300-23.5 but the proper scheme of this steam turbine unit was modified by removing of some Steam cycle components. In place of the steam cycle parts that were removed the bottoming S-CO2 cycle was incorporated. Heat production is realized by using some portion of steam after the turbine for water heating which then is directed to the consumer.

1.a. Combined Complex Steam-S-CO2 CHP Plant

In the current embodiment instead of the adjustable extraction we fully exclude the Low Pressure (LP) cylinder of the turbine so the steam after the Intermediate Pressure (IP) cylinder is directed to the control splitter which divides the main flow into two parts. The first part is directed to the supercritical CO2 heating and the second part goes to the water heating for the consumers (Figure 3.) In spite of the steam scheme simplification the steam part still has a system of regeneration which includes 1 Low Pressure (LP) heater and 3 High Pressure (HP) heaters as well as a steam reheat after the High Pressure (HP) cylinder. Since we excluded the LP cylinder changes of the heat load does not affect the steam turbine performance.

The considered bottoming supercritical CO2 cycle is quite simple and corresponds to a simple closed recuperated Brayton cycle [6]. The scheme consists of a turbine, a compressor, a recuperator and a cooler. The maximum supercritical CO2 temperature is equal to 120°C according to the steam saturation temperature at the pressure of 0.2 MPa. Actually, the temperature of the CO2 should be slightly lower than 120° C due to underheating (temperature closure). However the underheating was compensated by the heat of the superheated steam before the Intermediate Heat Exchanger (IHX). The bottoming supercritical CO2 cycle has a lower pressure of 7.7 MPa and a compressor outlet pressure of 15 MPa.

The temperature of the supercritical CO2 after the cooler was 32°C and the recuperator efficiency was taken as 95 %.

In such a scheme the steam turbine constantly produces 252.3 MW of net electrical power with any heat load values if all the conditions are kept constant. However, the electrical power produced by the CO2 turbine is varying depending on the heat load. It is obvious that when we direct more steam to the CO2 heating loop we produce more electricity and less heat and vice versa correspondingly.

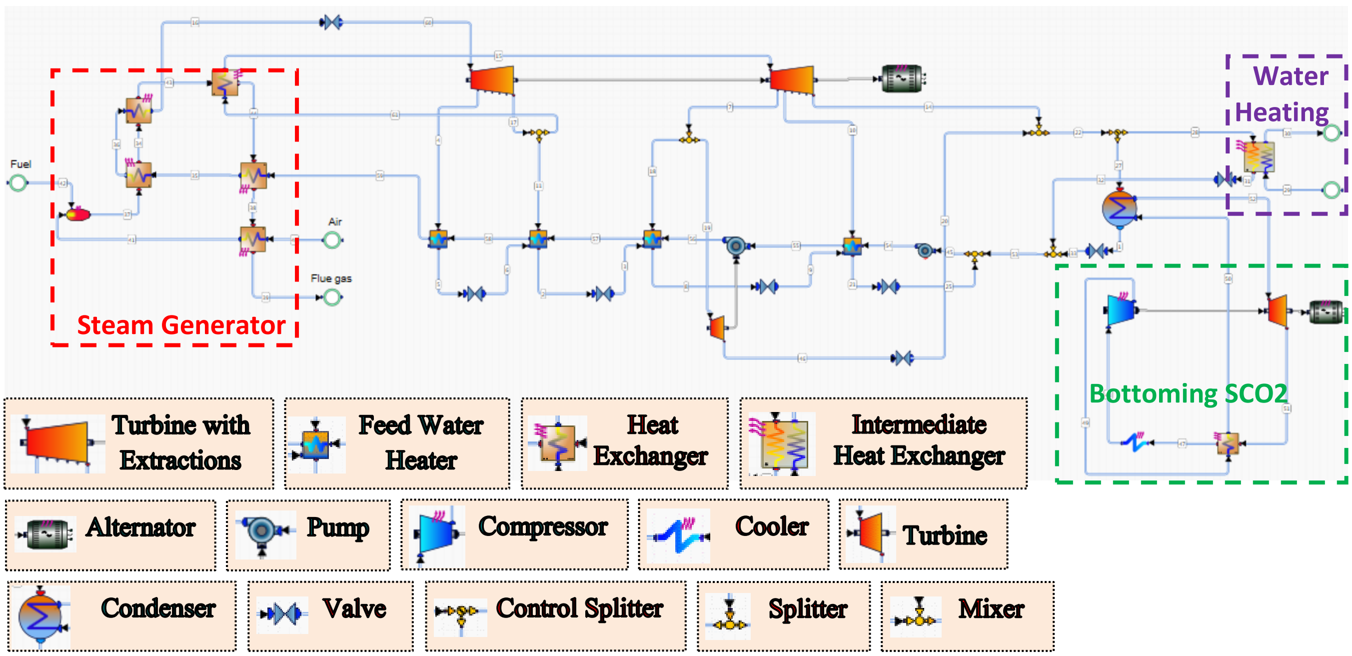


Figure 3. Combined Complex Steam-SCO2 CHP Plant

The obvious advantages of such a concept of CHP plant are that the SG remains the same so it is not necessary to design a new one for the S-CO2 implementation, which in turn can be a quite challenging task. The HP and IP steam turbine cylinders also remain the same as well as the HP and LP heaters of the steam turbine unit regeneration system while featuring the absence of a volumetrically imposing LP cylinder. The condenser will be significantly smaller than an ordinary one because of the high pressure of condensation. The bottoming S-CO2 cycle is extremely simple due to the low temperature of the live CO2, the single recuperator scheme and the low temperature gradient at the recuperator.

1.b. Combined Simple Steam-S-CO2 CHP Plant

The second considered embodiment according to the first approach is the cycle with the simpler diagram of Steam Cycle part than in the embodiment of section 1.a.

The new scheme is presented on Figure 4 (The designations are the same as on Figure 3). The simplification is done through the removing of the IP cylinder and all of the steam heat regeneration system. Thus in this scheme the steam after the HP turbine flows to the control splitter which divides the main flow into two parts. The first part is directed to the supercritical CO2 heating and the second part goes to the water heating for the consumers. In such a scheme the steam turbine constantly produces 140,4 MW of net electrical power with any heat load values at *ceteris paribus*. As in the previous embodiment the electrical power produced by the CO2 turbine is depends on the heat load.

Structurally, the bottoming supercritical CO2 cycle was the same as in the scheme of Figure 3. However, some of parameters were different. Namely, since we removed the IP cylinder of the steam turbine the pressure after the turbine became 3 MPa. Accordingly, the live supercritical CO2 temperature was equal to 225°C. It should be noted that the saturation temperature at the pressure of 3 MPa is 233.8°C so the underheating was equal to 8.8 °C which is different from the previous scheme because it could not be compensated by steam superheating due to its small magnitude. The bottoming supercritical CO2 cycle has a lower pressure of 7.7 MPa and an upper pressure of 25 MPa. The temperature of the supercritical CO2 after the cooler was 32°C with a recuperator efficiency of 95 %.

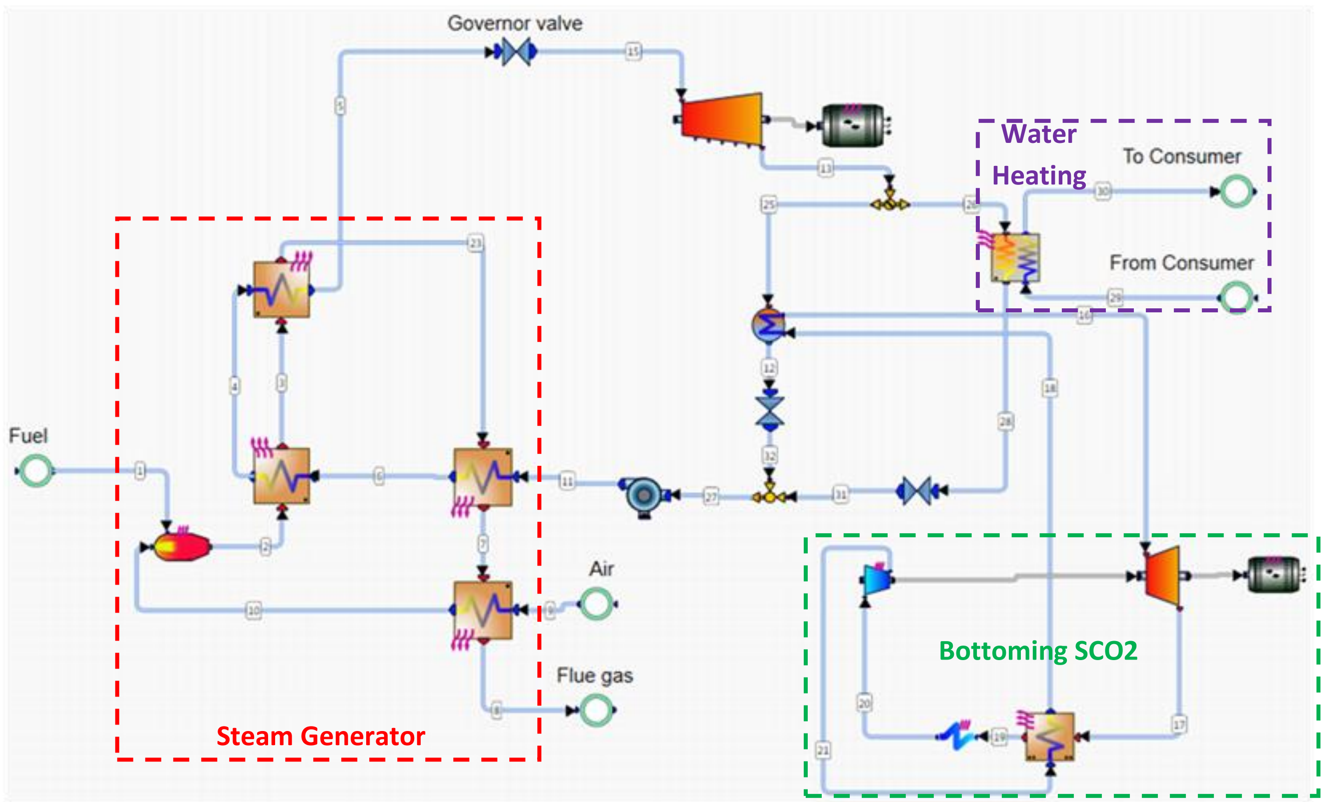


Figure 4. Combined Simple Steam-SCO2 CHP Plant

2. CHP plant with single supercritical CO2 working fluid

The second concept of CHP is the concept according to which only the supercritical CO2 fluid is used and the heat from the fossil fuel burning is transferred to the supercritical CO2 directly to the complex heater. It should be pointed out that strictly speaking the fossil fuel burner cannot be named as Steam Generator for this scheme because it does not generate steam but instead warms up the supercritical CO2 fluid. For this reason, it is named here as Complex Supercritical CO2 Heater (CSCO2H). CSCO2H is of the same type as the Water SG but with accommodated internal heaters to the supercritical CO2 features. The heat production is realized by water heating in the supercritical CO2 coolers or directly in the complex heater.

The obvious advantage of such a concept is that the only one working fluid is used.

2.a. Cascaded Supercritical CO2 CHP Plant

The first embodiment of the second concept is based on the cascaded supercritical CO2 approach [4] in the context of CHP technology (cf. Figure 5. The designations are the same as on Figure 3). According to the approach, the cascade of supercritical CO2 recompression cycles [2] is used for an electrical power production. The cascade of supercritical CO2 cycles is reasonable because of the small temperature difference in the IHX due to the high degree of supercritical CO2 cycle heat recuperation and allows utilizing heat from the burning of the fossil fuel more effectively. The CSCO2H differs from the water SG by that here the economizer and the reheater are absent and by that the two main IHXs will also in all probability differ from the evaporator and the heater in the ordinary SG.

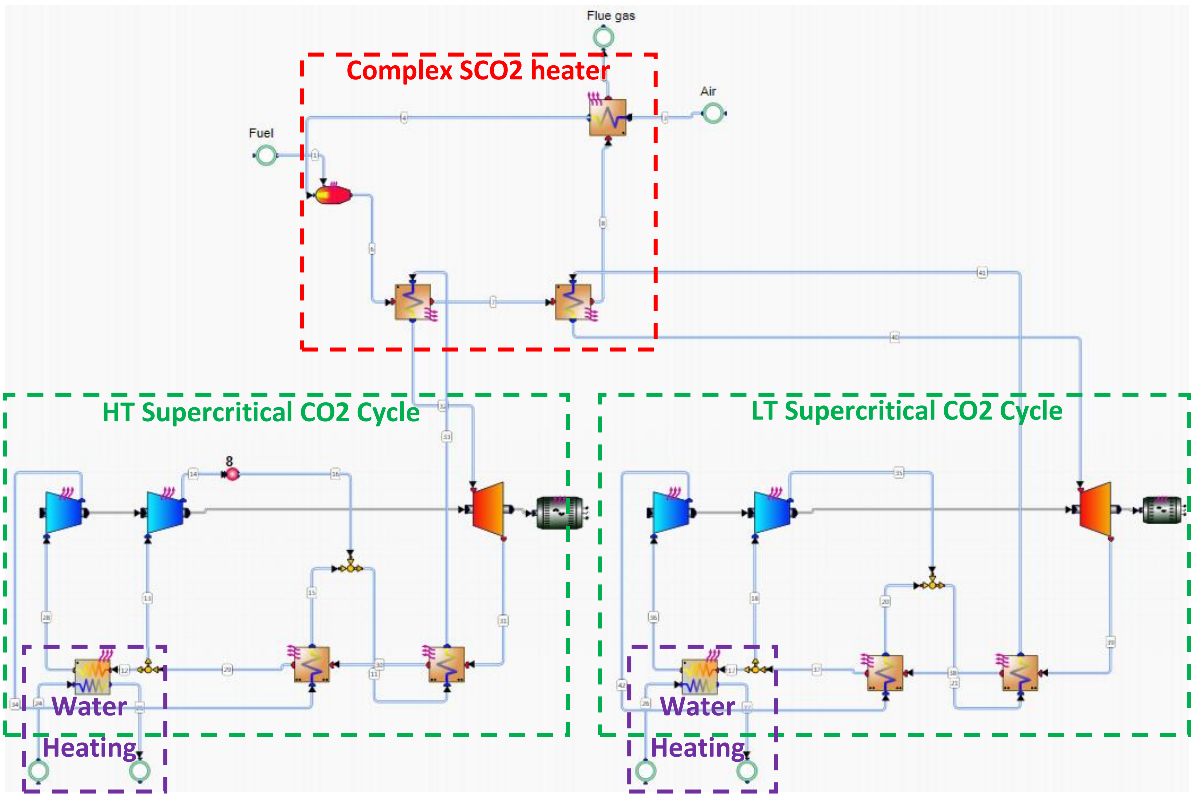


Figure 5.Cascaded Supercritical CO2 CHP Plant

The temperature of the live carbon dioxide in the HT cycle of the cascade was equal to 540°C while it was equal to 300°C for the LT cycle. The lower pressure magnitude of both cycles was assigned as 7.7 MPa while the compressor outlet pressure magnitudes were optimized and defined as 21 MPa and 20 MPa for the HT and LT cycles, respectively. The recuperators‘ efficiency equal to 95 %.

In order to get the possibility to heat the water for the consumers up to 100°C the ratio of mass flow rates between the main compressor and the recompressor was optimized for both the HT and LT cycles of the cascade to finally obtain a temperature of carbon dioxide after the LT recuperator equal to 115°C for HT cycle and 110 °C for LT cycle respectively. In addition, to avoid negative pinch on the water heater (SCO2 cooler) the temperature of the supercritical CO2 after the cooler was increased from 32 °C to 37 °C.

It should be pointed out that the heat load in such a CHP plant can be adjusted by dividing the cooling water into two parts. One part should be directed to the consumers while the other should go to the cooling tower which is not shown on the presented Figure. However, the total amount of cooling water should be the same to preserve the mode of turbines operation and the power production.

In such a scheme both turbines together constantly produce 311.6 MW of net electrical power with any heat load values if all the rest of the conditions are kept constant.

It should be noted that the first turbine in the cascade produces much more electrical power in comparison with the second one (297.7 MW vs. 13.9 MW, respectively). Thus, to simplify the scheme the additional water heater was used instead of the second supercritical CO2 cycle. Such a scheme is considered below.

2.b. Single Supercritical CO2 CHP Plant

The second embodiment of the second concept differs from the previous one by that the second supercritical CO2 cycle (LT) was substituted by an additional water heater, as pictured on Figure 6 (The designations are the same as on Figure 3).

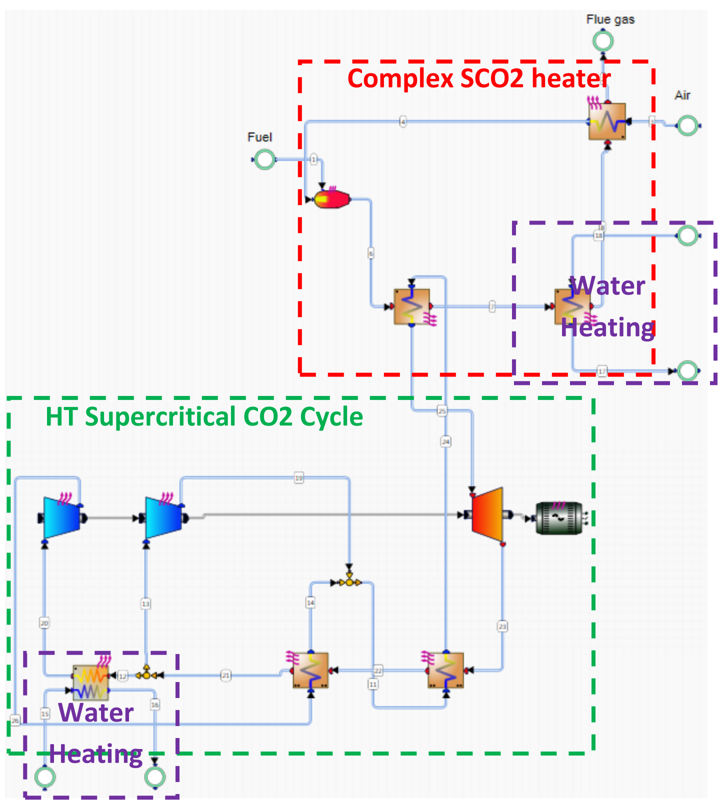


Figure 6. Single Supercritical CO2 CHP Plant

The temperature of the live carbon dioxide in the HT cycle of the cascade was equal to 540°C. The minimum pressure of the cycle was assigned as 7.7 MPa. In turn, the maximum pressure was defined as 21 MPa. The temperature of the supercritical CO2 after the cooler was 37 °C. The recuperators' efficiency was taken as 95 %.

As in the previous case, the ratio of mass flow rates between the main compressor and the recompressor was optimized to obtain the temperature of carbon dioxide after LT recuperator equal to 115 °C to provide sufficiently warm enough water to the customers.

The heat load adjustment as in the previous case can be realized by the subdivision of the heated water between the consumers and cooling tower (again not shown on the presented Figure). The total amount of cooling water should be the same due to the same reasons as for the cascaded S-CO2 CHP embodiment.

In such a scheme, the supercritical CO2 turbine constantly produces 297.7 MW of net electrical power with any heat load values if the HT S-CO2 cycle parameters are left unaltered.

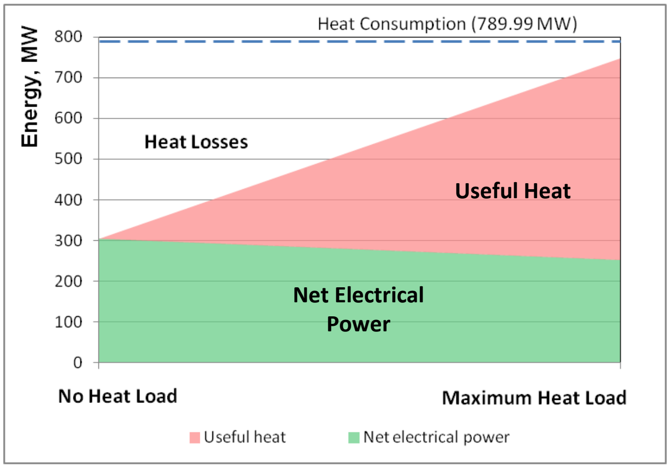
Calculation results and concepts comparison

The performances of all the presented CHP plant concepts were calculated with different heat loads (HL) and compared with the performances of the T-250/300-23.5 unit. The main results are summarized in Table2 and shown on Figure 7.

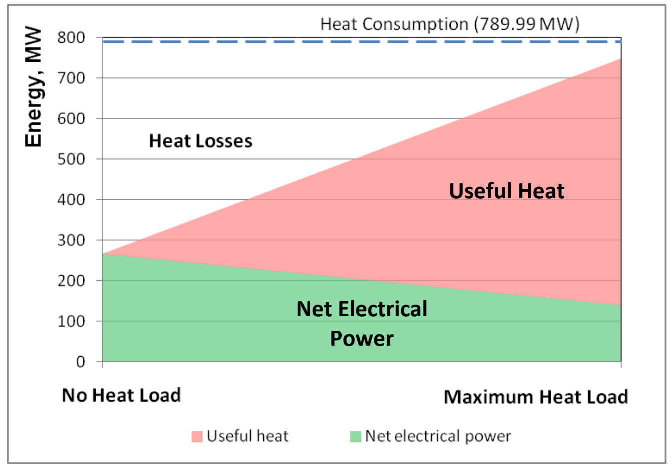
Table2. Calculated Performances of the Considered CHP Plants

Parameter	Embodiment 1.a		Embodiment 1.b		Embodiment 2.a		Embodiment 2.b		T-250/300	
	HL 0%	HL 100%	HL 0%	HL 100%	HL 0%	HL 100%	HL 0%	HL 100%	HL 0%	HL 100%
Heat consumption, MW	789.998									
Net electrical power, MW	304.416	252.283	266.685	140.4	311.56	311.56	297.66	297.66	300	250
Useful heat, MW	-	494.627	-	607.627	-	421.011	-	436.166	-	384.100
Electrical efficiency	38.53%	31.93%	33.76%	17.77%	39.44%	39.44%	37.68%	37.68%	37.97%	31.65%
Heat utilization factor	0.385	0.945	0.338	0.947	0.394	0.927	0.364	0.929	0.380	0.803
Potential of Useful heat, MW	-	-	-	-	421.011	-	436.166	-	-	-

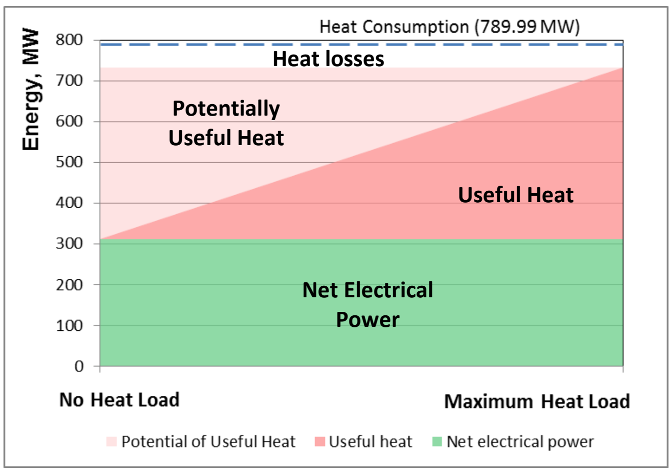
It is obvious from Figure 7 that for all the considered cycles (except for the T-250/300-23.5 cycle) the electrical power varies linearly with the heat load. Note that such a dependency is the consequence of the fact that we keep constant the efficiency of the bottoming CO2 cycle components. The characteristics for the installed unit (T-250/300-23.5) were built using data at the modes with zero and maximum heat loads from the literature [5]. In reality, the T-250/300-23.5 characteristics are slightly different from being strictly linear but it does not significantly affect the results of the current study.



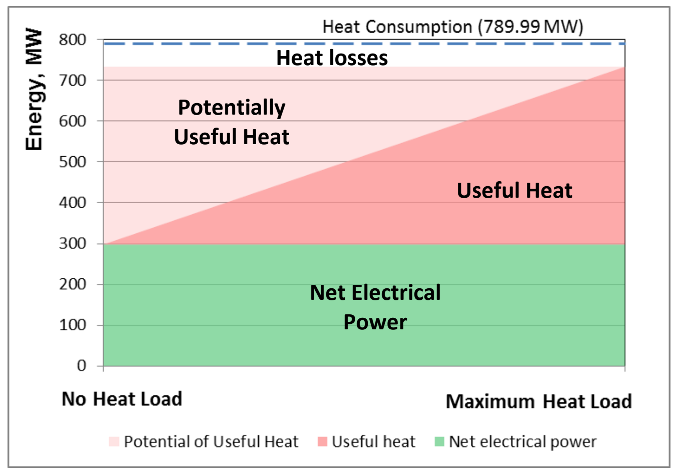
1.a



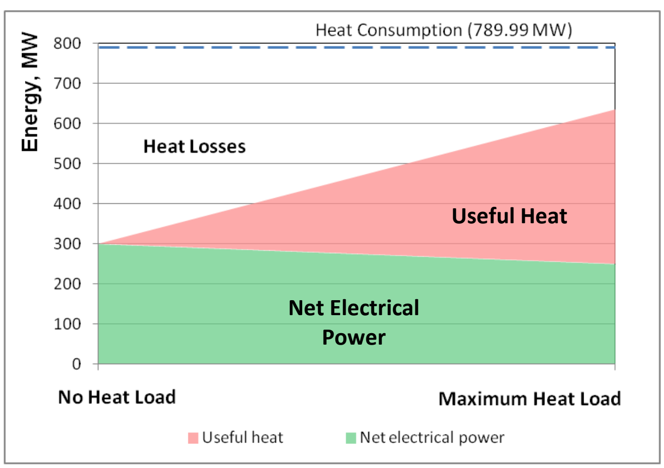
1.b



2.a



2.b



T-250/300-23.5

Figure 7. Cycles performances at different heat loads.

As one can observe from Figure 7, the diagram **1.b** (Simple Steam-SCO2) is the worst in terms of electric power generation with the maximum heat load as well as without it. However, this flow diagram is the simplest and this unit has the highest Heat Utilization Factor (HUF – the ratio of a sum of a Net

Electrical Power and a Useful Heat to a Heat Consumption) at the modes with the 100% heat load due to a huge amount of useful heat. The complex Steam-SCO2 cycle (**1.a**) includes more power equipment but this diagram is more advantageous than the previous one due to the higher electricity generation and the quite high heat utilization factor (see Figure 7 and Table2).

The performances of the considered cycles exceed the ones of the T-250/300-23.5 at any heat load. Also the cycle **1.a** has a high unification degree with the existing steam turbine plants. To build it, we can use the boiler, HP and IP cylinders, heat exchangers and other equipment from the existing steam cycles. Considering the above Complex Steam-SCO2 cycle, the cycle **1.a** can be recommended as the initial step to use SCO2 technologies for CHP (for new plants and for the modernization of existing ones).

In the cycles **2.a** and **2.b** the electricity generation does not depend on the heat load of the plant for fixed fuel consumption (electrical efficiency is a constant). As it is evident from Table2 and Figure 7 the diagram **2.a** has the best performance (a higher electrical efficiency), but this scheme is more complicated in comparison with the embodiment **2.b**. To make a choice between the cycles **2.a** and **2.b** the value of the capital costs and the specific needs of the customer in the thermal and electrical energy should be considered. The values of the electric efficiency and the useful heat for the cycle **2.a** are superior to the similar parameters of the existing T-250/300-23.5 unit. According to the authors, the cycle **2.a** is the best of all the above.

Conclusions

- Taking into consideration high efficiency of fuel energy utilization of CHP plants and the high potential of the supercritical CO2 technology the latter should be also considered as the base of future CHP plants.
- In order to accommodate supercritical CO2 technology to CHP conception numerous configurations and approaches were considered and the concepts were selected as the most interesting ones: 1 - Steam Rankine cycle CHP plant with bottoming supercritical CO2 cycle where the heat production is realized by using some portion of the steam after the turbine for water heating which is then directed to the consumers; 2 - CHP plant with a unique supercritical CO2 working fluid where the heat production is realized by water heating in supercritical CO2 coolers or directly in the complex heater.
- The complex and simple embodiments were considered for each of the supercritical CO2 CHP plant conception: 1.a - Combined Complex Steam-S-CO2 CHP Plant; 1.b - Combined Simple Steam-S-CO2 CHP Plant; 2.a - Cascaded Supercritical CO2 CHP Plant; 2.b - Single Supercritical CO2 CHP Plant.
- The obvious advantages of the "Steam Rankine cycle CHP plant with bottoming supercritical CO2 cycle" conception are that the SG remains the same so it is not necessary to design a new one for the supercritical CO2, which in turn can be a quite challenging task. The HP and IP steam turbine cylinders also remain the same as well as the HP and LP heaters of the steam turbine unit regeneration system while featuring the absence of the huge LP cylinder. Also the condenser will be significantly smaller than the ordinary one due to the high condensing pressure. The bottoming SCO2 cycle is extremely simple due to the low temperature of the live CO2, the single recuperator scheme and the low temperature gradient at the recuperator.
- The obvious advantage of the "CHP plant with single supercritical CO2 working fluid" concept is that a single working fluid is used.

- In terms of electrical efficiency almost all the embodiments with the supercritical CO2 working fluid beats the ordinary Steam CHP unit T-250/300-23.5 except for the embodiment “Combined Simple Steam-SCO2 CHP Plant”. The Cascaded Supercritical CO2 CHP Plant was found to have the best electrical efficiency.
- The electrical power production depends on the heat load of the plant for the "Steam Rankine cycle CHP plant with bottoming supercritical CO2 cycle" concept and does not depend on it for the "CHP plant with single supercritical CO2 working fluid" frame.
- In terms of Heat Utilization Factor the best embodiment is the “Combined Simple Steam-SCO2 CHP Plant” but taking into account that the electrical power is more valuable than the heat the “Cascaded Supercritical CO2 CHP Plant” is preferable. However, to make a choice between the considered embodiments the value of capital costs and the specific needs of the customer in the thermal and electrical energy should be considered.

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References

[1] Anna Shipley, Anne Hampson, Bruce Hedman, Patti Garland and Paul Bautista, "Combined Heat And Power - Effective Energy Solutions for a Sustainable Future", Report for the U.S. Department Of Energy, Oak Ridge National Laboratory, Oak Ridge, Tennessee 37831-6283, December 1, 2008

[2] Vaclav Dostal, Pavel Hejzlar, Michael J. Driscoll, “The Supercritical Carbon Dioxide Power Cycle: Comparison to Other Advanced Power Cycles”, Nuclear Technology Vol. 154, pp. 283-282, June 2006.

[3] Oleg Guriev, Yuriy Govoruschenko, Leonid Moroz, Supercritical CO2 Brayton Cycle. S-CO2 represents an advance in power conversion technology, Turbomachinery International. The Global Journal of Energy Equipment, January/February 2013.

[4] Gregory A. Johnson, Michael McDowell, "Issues Associated with Coupling Supercritical CO2 Power Cycle to Nuclear, Solar and Fossil Fuel Heat Sources", Report, Hamilton Sundstrand, Energy Space & Defense - Rocketdyne.

[5] Truhnij A., Lomakin B., Cogeneration steam turbines and units: manual for University, MPEI press, 2002. ISBN 5-7046-0722-5

[6] Vaclav Dostal, Pavel Hejzlar, Michael J. Driscoll, “A Supercritical Carbon Dioxide Cycle for next generation nuclear reactors”, Advanced Nuclear Power Technology Program MIT-ANP-TR-100, March 10, 2004.