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Selecting the configuration of inter-stage coolers for a CO_2 compressor

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Abstract

The compression process is one of the most energy-consuming stages of the entire cycle of the carbon dioxide capture and storage. a reduction in the energy consumption of this process may have a significant impact on the overall net efficiency of electricity generation. One method to improve the efficiency of the compression process is to introduce inter-stage cooling, which makes it possible to cause compression to resemble an isothermal process. This paper presents an analysis of the operation of a cooling system of an eight-stage integrally geared compressor working for a 900 MW coal-fired power plant equipped with a CCS system. The research conducted enables rational values of the heat transfer coefficient to be determined for individual inter-stage coolers. a series of calculations is performed in this analysis for different assumptions and for an assumed geometry of the inter-stage coolers. The results obtained allow one to determine both the heat exchange surface area and the pressure drops on the "hot" and "cold" agent sides for individual coolers. Moreover, the presented analysis identifies a number of problems that arise when selecting the configuration and the appropriate parameters of the cooling system.

Keywords: CCS, CO_2 compression, CO_2 compressor, CO_2 inter-stage coolers

1. Introduction

The development of carbon dioxide capture and storage technologies may significantly reduce carbon dioxide emissions. The main technologies in this field are the following: post-combustion capture, pre-combustion capture and oxygen combustion [1–4]. These technologies are extensively described in the reference literature. However, less attention is devoted to the next stages, i.e. to the process of CO_2compression and its further transport to storage sites and to the injection into suitable geological formations.

Long-distance transport of large amounts of carbon dioxide via pipelines is considered to be the most rational method in terms of economy. In this case, the medium should be in its "dense" state, i.e. liquid or supercritical [5], as this causes smaller losses related to friction in the pipeline, compared to CO_2 transport in the gaseous state or as a two-phase mixture. Considering the above and taking account of pressure losses during the transport of CO_2 over long distances, the required outlet pressure after the CO_2 compressor should be 150..200 bar [6, 7]. Moreover, it should be noted that the CO_2 compression process differs from the compression of other liquids due to the large molecular weight, high compressibility and the critical point close to the ambient temperature. Another problem is the large volume flow of the captured CO_2 which is subjected to compression. Therefore, the compression process is one of the most energy-consuming stages of the entire capture cycle. The power of the CO_2 compressor may in this case amount to approximately 6-8% of the rated power of the entire turbine set, which substantially increases the thermal power plant's own needs and, as a consequence, reduces the net efficiency of electricity generation.

For this reason, inter-cooling is introduced in order to improve the compression process efficiency, as it makes it possible to cause compression to resemble an isothermal process. The introduction of inter-stage coolers may in

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this case significantly reduce the required driving power of the compressor. It is worth noting that the optimum selection of the cooling system configuration, which may also take account of the use of waste heat in the thermal power plant cycle or in the CCS system, should be preceded by a thermo-economic analysis, e.g. [8].

At present, the reference literature does not offer any information on the structure or the operating parameters of individual inter-stage coolers of CO_2. This mainly concerns the achievable values of the heat transfer coefficient, which determines the surface area of the heat exchange and, consequently, the cost of individual inter-stage coolers. Moreover, due to the very wide range of pressure values and the resulting variations in CO_2 density, it may be expected that for individual inter-stage coolers the coefficient will vary to a large extent.

The presented factors underscore the need for research aimed at determining rational values of the heat transfer coefficient for individual inter-stage coolers. This is a far from straightforward task since the heat transfer coefficient depends on many parameters—both geometrical and thermal-related. Therefore, a series of calculations is performed in this analysis for different assumptions and an assumed geometry of inter-stage coolers. The results obtained allow one to determine both the heat exchange surface area and the pressure drops on the "hot" and "cold" agent sides for individual coolers. Moreover, the presented analysis identifies a number of problems that arise when selecting the configuration and the appropriate parameters of the cooling system.

2. The selection of the CO_2 compressor

The compression process with inter-stage cooling is possible using the following [9]:

- single-shaft, multi-stage compressors,
- integrally geared compressors,
- shock wave compressors [10].

Since shock wave compressors are not available as yet—because they are still at the experimental stage—the most promising solution at present seems to be to use a multi-stage integrally geared compressor with inter-stage cooling. Integrally geared compressors feature unquestionable merits such as: the possibility of optimum matching of the geometry of subsequent stages to the abruptly changing density of CO_2, selection of the number of revolutions of the subsequent stage pairs (up to 50,000 rev/min) and high polytropic efficiency. The aerodynamic perfection of these machines allows a high pressure ratio of 1.7..2.0 and a wide range of the stage flow indices [11]. With this value of the pressure ratio, the eight-stage integrally geared compressor delivers the required total pressure rise of 100:1. Consequently, this particular compressor type is chosen for further analysis. It is assumed that the CO_2 compressor will work for a 900 MW coal-fired power unit [2] equipped with a capture installation based on the amine absorption process [2]. The basic data assumed for the analysis according to [2, 12] are as follows:

- exhaust gas mass flow: 816 kg/s,
- molar fraction of CO_2: $[CO_2]$ = 0.1818 (18..19%),
- 85% of captured CO_2.

Consequently, the CO_2 parameters at the compressor inlet are as follows [2]:

- mass flow of CO_2 at the compressor inlet: 156.43 kg/s,
- inlet temperature: 35°C,
- inlet pressure: 1.5 bar,
- density: 2.852 kg/m^3,
- moisture content: 3%.

Additional values assumed during the analysis of the compression process are:

- polytropic efficiency—almost linear reduction from 84% in the first compression stage to 56% in the eighth compression stage [13],
- stage average pressure ratio: 1.82,
- range of pressure ratio variability: ~1.7..1.9.

Examples of compressors that can meet the presented criteria are presented in [6, 14]. The analysis of the eight-stage integrally geared compressor is performed using the software package Aspen Plus v7.0 [15]. The real gas equations of state developed by Benedict, Webb, Rubin and Starling (BWRS equations) for the CO_2 pressure range of up to 50 bar and the equations developed by Lee, Kessler and Plocker (LK-PLOCK equations) for pressures higher than 50 bar [15] are used in the calculations. It is assumed that the compression process concerns pure CO_2 without any content of moisture or other gases. For the reference variant it is assumed that the temperature of the cooled CO_2 after individual inter-stage coolers is 35°C,

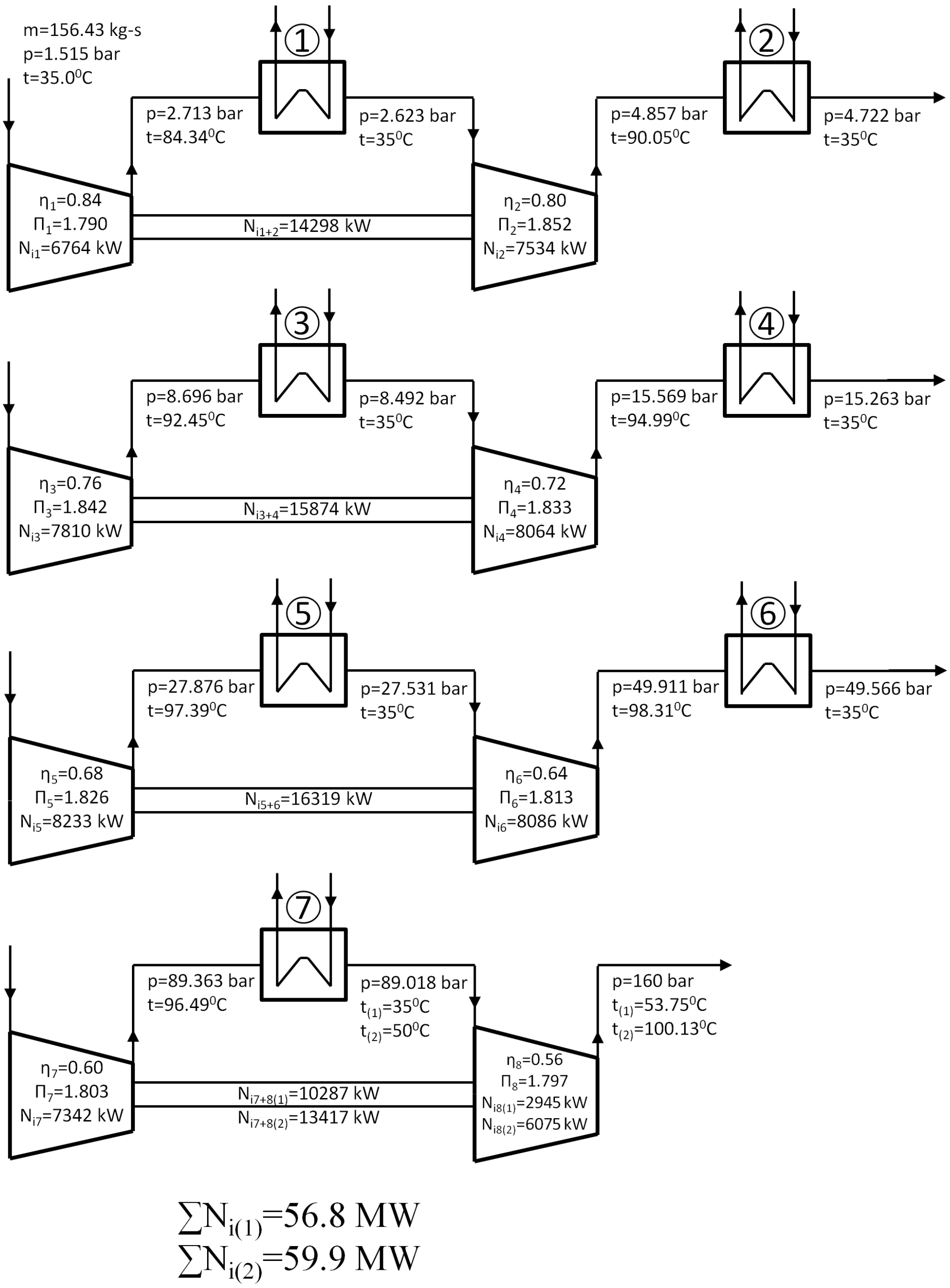

Figure 1: The compression process in an 8-stage integrally geared compressor

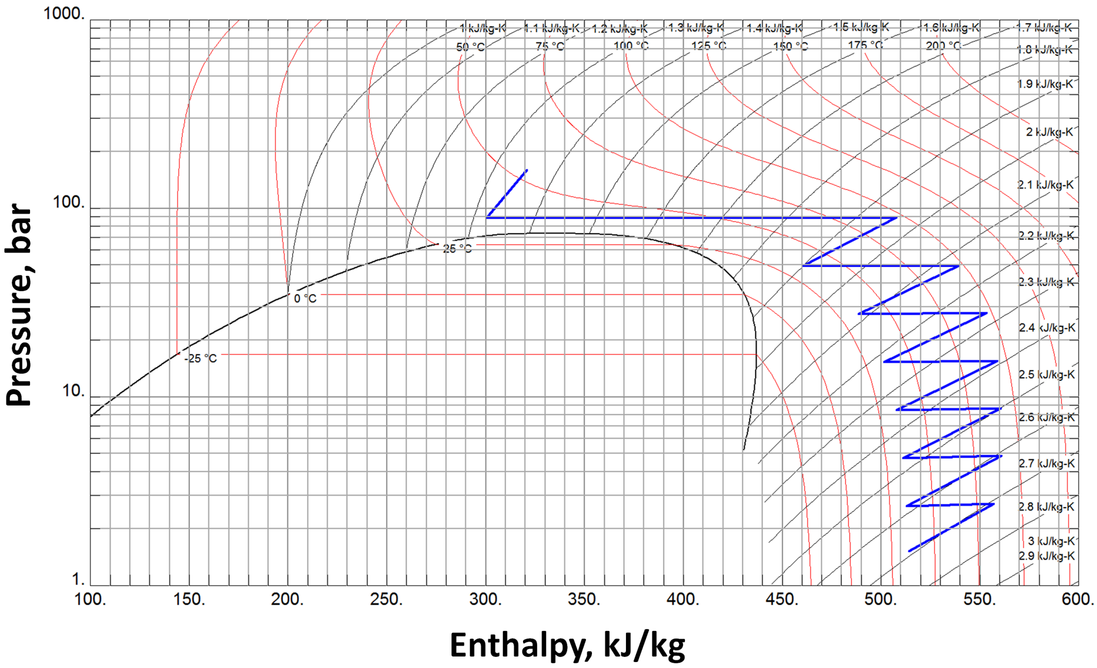

Figure 2: The p-h diagram of the compression process in an 8-stage integrally geared compressor (variant 1)

and the pressure losses are assumed according to the formula (1).
The diagram of the integrally geared compressor together with the basic parameters obtained in the analysis is presented in Fig. 1. Fig. 2 presents the cycle of the compression process transformations illustrated by a $p-h$ diagram. The obtained internal power of the compressor for the reference variant (1) is in this case 56.8 MW. Additionally, increasing the outlet temperature for the last inter-stage cooler leads to a growth in internal power to 59.9 MW (2).

3. The cooling system configuration

Correct selection of the surface area, the structure and the configuration of inter-stage coolers is a key issue which may substantially improve the efficiency of the compression process. With regard to the selection and structure of inter-stage coolers, the main problems are as follows:

- the large mass flow of cooled CO_2,
- the low density of cooled CO_2 in the first low pressure coolers,
- the high pressure of cooled CO_2 in the last inter-stage coolers,
- the content of other gases and moisture in the mass flow of cooled CO_2.

The temperature of the cooling water has a decisive impact for the cooling system, as it may improve the effectiveness of the whole process. The cooling water temperature depends on many factors, but mainly on the type of cooling system used and on the climatic zone in question. The cooling water average annual temperature varies across a wide range depending on average air temperature in the given area. Therefore, the assumed reference literature data are included in a rather wide range of 12..29°C [13]. It is assumed in the calculations that the cooling water temperature is 19.1°C . The value is taken based on detailed design calculations of a cooling tower for a 900 MW coal-fired power plant which will operate in Poland [16].
The heat transfer coefficient is a value which decisively affects the heat exchanger dimensions. Rational determination of it is a key task that opens the way to determining the heat exchange surface area of individual inter-stage coolers. According to [17], the value of the heat transfer coefficient for liquid-gas exchangers, depending on the range of pressure under analysis, may be: 15..70 W/(m^2·K) (1 bar), (200..400) W/(m^2·K) (250 bar). However, the presented data do not concern any particular type of gas or liquid exchanging heat and, being of indicative nature, they are offered for general information only.

The value of the permissible pressure drop in inter-stage coolers on the side of the cooled CO_2 is assumed according to the formula presented in [18], except that pressure is expressed in bars:

$$\Delta p = \frac{(14.504 \cdot p_{CO2})^{0.7}}{10 \cdot 14.504}\ \text{bar}, \tag{1}$$

but not higher than 0.344 bar

The values obtained according to formula (1) are the permissible limit of the pressure drop for individual inter-stage coolers and they constitute a design parameter. A potential increase in permissible pressure losses leads to a rise in velocity of the flowing CO_2 and, consequently, to a more intense heat exchange, which in turn entails a reduction in the necessary surface area of the heat exchange. Therefore, optimum selection of the size of pressure losses should take account of their impact on the dimensions of the heat exchanger itself.

Table 1 presents the assumed values of pressure losses on the side of the cooled CO_2. For Variant (A) they are assumed according to formula (1), and for Variant (B) they are doubled compared to Variant (A). Additionally, Table 1 presents two different versions concerning the calculations of the seventh inter-stage cooler. For version 1, the CO_2 cooling outlet temperature is assumed to be the same as for the other coolers: 35°C. For version 2, it is raised to 50°C.

4. The assumptions and the assumed structure of the inter-stage coolers

The design calculations of individual inter-stage coolers are performed using the commercial software package Aspen Exchanger Design. This particular module is intended for use in designing and modelling the operation of heat exchangers and is a component of the AspenTech software [15]. The heat exchanger computational algorithm used in this software is based on the American standard TEMA (The Tubular Exchanger Manufacturers Association, Inc.) [19].

The basic parameters of the heat exchanging agents are obtained using thermodynamic functions implemented in AspenTech software. As in the compressor calculations, the BWRS and LK-PLOCK (>50 bar) procedures are used in the analysis. The cooling water parameters are determined based on the IAPWS-97 procedures [20].

The operating parameters of individual inter-stage coolers are assumed according to Fig. 1, and the preliminary value of pressure losses is assumed according to Table 2.

It is assumed that all analyzed inter-stage coolers are shell-and-tube heat exchangers. In this case, the cooling agent (water) flows in the tubes, whereas the carbon dioxide is in the shell of the exchanger. In order to intensify the heat flow, baffles (usually 2-6) are used to change both the velocity distribution in the individual cross-sections of the exchanger and the character of the flow of the heat exchanging agents. The typical parallel flow is transformed into cross flow.

Moreover, it is assumed that all analyzed inter-stage coolers are made of stainless steel SS304. Compared to other steel grades, this steel features has relatively low thermal conductivity coefficient of approximately 16.2 W/(m·K) [21]. Stainless steel is used due to the fact that the stream of carbon dioxide captured in the CCS installation contains a large amount of moisture (approximately 3%) and some other gases, which may cause corrosion [1, 2].

It is assumed for all analyzed inter-stage coolers that the external diameter of the tubes is 16 mm and the tube thickness is 1.6 mm. Making the assumption of constant tube thickness is justified, because this analysis focuses on thermal rather than strength calculations of the heat exchanger. Hence, a change in tube thickness does not have a significant effect on the value of the heat transfer coefficient due to the high value of the steel thermal conductivity coefficient.

The calculations are performed for two cases—for smooth and finned tubes, respectively. The height of the finned tubes is assumed as 1.5 mm, the thickness as 0.3 mm and the distribution density as 748 fins/m. These parameters are assumed according to the series of types of finned tubes [15].

It is assumed in the calculations that minimum cooling water velocity is 1 m/s [22]. Lower velocity values result in very small values of the heat transfer coefficient on the side of the cooling agent, which significantly increases the required surface area of the heat exchange and provides favourable conditions for deposits to accumulate on the cooling water side [22]. At the same time, higher cooling water velocity leads to a reduction in the total cross-section of all cooler tubes. Consequently, keeping the same tube pitch, a smaller diameter of the whole exchanger is obtained. This in turn translates into a significant increase in the velocity of the flowing CO_2, which causes overly high pressure losses on the cooled carbon dioxide side. This problem concerns the first inter-stage coolers in particular, due to the low density of the cooled CO_2 and, consequently, large CO_2 volume flows. Therefore, a relatively large pitch value is assumed for these

Table 1: Assumed pressure losses on the side of the cooled CO_2 taken for the analysis of individual inter-stage coolers.

Inter-stage cooler	1	2	3	4	5	6	7 (1)	7 (2)
A. $\Delta p_{CO2\ assumed}$, bar	0.090	0.136	0.204	0.306	0.344	0.344	0.344	0.344
B. $\Delta p_{CO2\ assumed}$, bar	0.180	0.272	0.408	0.613	0.688	0.688	0.688	0.688

Table 2: The assumed tube pitch value for the calculations of inter-stage coolers

Inter-stage cooler	1	2	3	4	5	6	7
Pitch s/d	1.875	1.875	1.625	1.5625	1.5625	1.5625	1.25

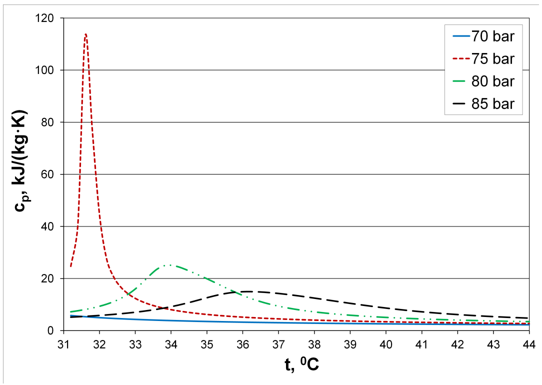

Figure 3: The curve of the CO_2 specific heat capacity depending on temperature for different values of pressure near the critical point (based on LK-PLOCK procedures)

coolers. The pitch is reduced in subsequent inter-stage Coolers (Table 2). The presented values are included in the range of permissible ratios of 1.25..2.5 [22]. According to the presented parameters, the increment in cooling water temperature is selected for individual inter-stage coolers.

The calculations concerning the seventh inter-stage cooler, which operates in the supercritical region, are performed for two different values of CO_2 outlet temperature. For case (1), as for the other inter-stage coolers, the temperature is 35°C , whereas for case (2) the temperature is increased to 50°C . This is related to the fact that the CO_2 thermodynamic parameters, especially the specific heat capacity near the critical point, rise significantly.

The graph in Fig. 3 presents the change in the specific heat capacity depending on temperature for individual isobars in the range of 70-85 bar (the CO_2 critical point is 73.8 bar and 31.1°C). The presented curves show that with pressure and temperature values close to the critical point there is an abrupt increase in CO_2 heat capacity. The presented results are obtained based on LK-PLOCK procedures. However, very similar behaviour of CO_2 may be observed by analyzing the latest procedures for determining the CO_2 thermodynamic parameters, which are presented e.g. in [23].

This is why an increase in temperature to the value of 50°C results in a CO_2 heat capacity value which is similar to the values obtained for the other inter-stage coolers operating in the subcritical range, which very significantly reduces the heat flux exchanged in the cooler. It also leads to a substantial reduction in the required stream of cooling water and, as a consequence, reduces the surface area of the heat exchanger itself. Moreover, a rise in temperature after the seventh inter-stage cooler causes a rise in the compressor total internal power by more than 3.1 MW and an increase in outlet temperature after the compressor to 100°C, a possible solution in this case is to remove the seventh cooler entirely and to introduce a heat exchanger after the compressor. The heat obtained in this way may be used in the condensing power unit regenerative system owing to the rather high temperature of CO_2 at the compressor outlet of approximately 170°C.

5. The calculation results

The obtained results of the calculations comprise the analysis of the two variants (A and B) listed in Table 1. For both these variants additional calculations are performed, taking account of the effect of the application of finned tubes on the basic operating parameters of individual inter-stage coolers. The basic data for the calculations are taken according to Fig. 1. Table 3 and Table 4 present the results of the calculations for Variant A, assuming smooth and finned tubes, respectively.

The assumed increment in cooling water temperature for all analyzed variants is included in the range of 1.9-5.9°C. The low values of the cooling water temperature increment are caused by the need to obtain minimum cooling water velocity within the limits of 1 m/s, without exceeding the permissible pressure drops on the side of the cooled CO_2 at the same time. Assuming a low incre-

Table 3: Variant A. Calculation results obtained for individual inter-stage coolers

No	$\triangle p_{CO2}$	$w_{CO2\ in}$	m_{H2O}	$t_{H2O\ out}$	$\triangle p_{H2O\ in}$	$w_{H2O\ in}$	Q	$\triangle t_m$	U	A
	bar	m/s	kg/s	°C	bar	m/s	kW	°C	W/(m^{2}K)	m^{2}m^2
1	0.088	19.8	874	21	0.064	1.1	6921	34.53	234	937
2	0.135	17.9	648	22	0.063	1.0	7836	36.1	292	792
3	0.157	14.8	514	23	0.061	1.0	8363	36.54	403	575
4	0.228	12.3	445	24	0.086	1.3	9102	36.94	581	431
5	0.225	7.5	418	25	0.076	1.2	10295	36.93	720	391
6	0.259	5.2	510	25	0.194	1.9	12548	36.11	755	470
7 (1)	0.137	2.7	1290	25	0.137	1.3	31734	25.44	571	2207
7 (2)	0.339	6.3	571	25	0.172	1.8	14049	45.49	742	420

Table 4: Variant A + fins. Calculation results obtained for individual inter-stage coolers

No	$\triangle p_{CO2}$	$w_{CO2\ in}$	m_{H2O}	$t_{H2O\ out}$	$\triangle p_{H2O\ in}$	$w_{H2O\ in}$	Q	$\triangle t_m$	U	A	A_F
	bar	m/s	kg/s	°C	bar	m/s	kW	°C	W/(m^{2}K)	m^{2}m^2	m^{2}m^2
1	0.085	18.8	874	21	0.108	1.6	6921	34.54	199	718	1721
2	0.125	20.1	648	22	0.199	2.3	7836	36.1	275	367	879
3	0.180	18.2	514	23	0.158	2.1	8363	36.53	330	300	718
4	0.236	13.6	445	24	0.148	2.0	9102	36.94	467	244	586
5	0.175	7.9	418	25	0.152	2.0	10295	36.93	498	253	606
6	0.266	7.3	510	25	0.191	2.2	12548	36.11	462	319	765
7(1)	0.075	2.4	1290	25	0.188	2.0	31734	25.44	516	1027	2553
7(2)	0.089	3.8	571	25	0.185	2.4	14049	45.49	553	238	570

ment in cooling water temperature causes a significant increase in cooling water mass flow which, depending on the cooler in question, is included in the range of 418-874 kg/s (cooler 7 (1) excluded). The value of the heat flux exchanged in individual inter-stage coolers rises for subsequent coolers from 6.9 MW to 14.1 MW. For cooler 7 (1) the flux of exchanged heat reaches a value exceeding 31.7 MW. Such a high rise in the flux of exchanged heat is mainly related to the rise in the CO_2 heat capacity near the critical point (Fig. 3). At the same time, this also causes a very significant increase in the heat exchange surface area. For this reason, the CO_2 outlet temperature for the cooler 7 (2) is raised from 35°C to 50°C. This results in a reduction in the flux of exchanged heat from 31.7 MW to 14.1 MW, which causes a reduction in the heat exchange surface area from 2207 m^2 to only 420 m^2 for Variant A. The results obtained with the additional application of fins give a reduction in the heat exchange surface area from 1027 m^2 to 238 m^2, respectively (ignoring the surface area of the fins themselves).

For Variant A (Table 3), the obtained values of cooling water velocity are included in the range of 1.0-1.8 m/s at the cooled CO_2 velocity in the range of 5.2-19.8 m/s (cooler 7 (1) ignored). These values depend mainly on the assumed increment in cooling water temperature, on the assumed tube pitch and on the number of baffles in individual inter-stage coolers.

The lower velocity values obtained after the application of fins (Table 4) are caused by the smaller number of baffles (typically two) used in this variant, which is related to the smaller length of individual inter-stage coolers. The obtained values of the heat transfer coefficient are slightly higher for smooth tubes and they fall in the range of 234-742 W(m^2·K). For finned tubes the values are in the range of 199-553 W(m^2·K), respectively.

Fig. 4 presents the calculated pressure drop in individual inter-stage coolers for Variants A and B, for which calculations are made that take account of the effect of the finned tubes. The results do not exceed the limit values presented in Table 2. For Variant A, the pressure drops range from 0.088 bar to 0.339 bar (Table 3).

Fig. 5 illustrates the value of the calculated heat transfer coefficient for all variants under analysis. The results show that a double rise in the permissible pressure drop does not have a significant impact on the increment in the heat transfer coefficient for individual inter-stage coolers. Fig. 6 presents the percentages of the reduction in the heat exchange surface area of individual inter-stage coolers with reference to the base Variant A. The presented heat exchange surface area in the case of finned tubes

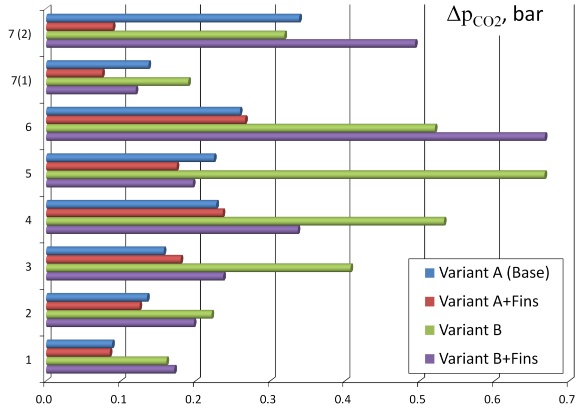

Figure 4: The pressure drop on the side of the cooled CO_2 for individual inter-stage coolers

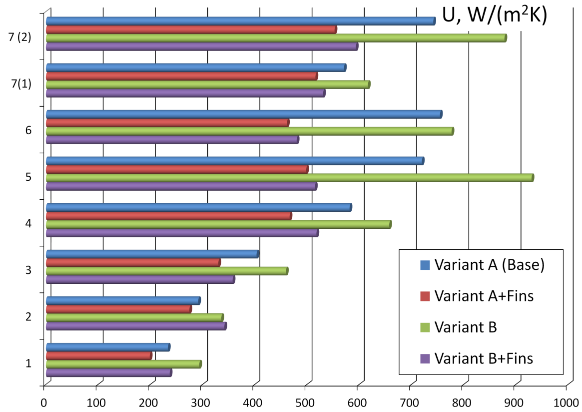

Figure 5: The heat transfer coefficient obtained for individual inter-stage coolers

comprises the surface area of the tubes only, without the surface area of the fins themselves (Table 3, Table 4). The reduction in the heat exchange surface area related to the increase in the values of permissible pressure drops (Variant B) falls in the range of 4-26%, depending on the cooler in question. The biggest reduction in surface area occurs in this case for cooler 1 and cooler 2.

The application of finned tubes leads to a very significant reduction in the heat exchange surface area (the surface area of fins excluded). The reduction in the heat exchange surface area varies from 23 to 55%. The rise in the permissible pressure drop in the case of finned tubes does not have a significant impact on the heat exchange surface area. Cooler 1 is an exception, however, as there an increase in the permissible pressure drop from 0.09 bar to 0.18 bar leads to a reduction in the heat exchange surface area of 23% (Variant A + fins) to more than 50% (Variant B + fins).

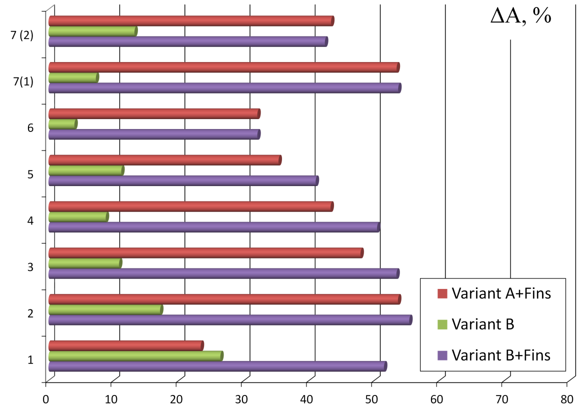

Figure 6: Reduction (as %) in the ht exchange surface area of individual inter-stage coolers with reference to Variant A

6. Summary

This paper presents a detailed analysis of the operation of the cooling system of a 56.8 MW CO_2 compressor operating with a CCS system of a 900 MW coal-fired power unit. 7 inter-stage coolers are applied in this case, operating in the pressure range of 2.7-89.4 bar. For the range of pressure values under analysis, the heat transfer coefficient varies in a wide range depending on the inter-stage cooler in question and on the analyzed variant. The obtained values of the heat transfer coefficient fall in the range of 200-930 W/(m^2·K) depending on the variant under analysis.

The application of finned tubes in the structure of inter-stage coolers may lead to a very significant reduction in their dimensions. The reduction in the heat exchange surface area (the surface area of the fins excluded) may in this case vary between 23 and 55%.

The operation of inter-stage coolers in the range of the CO_2 critical point may cause a very significant rise in the flux of the exchanged heat and, consequently, an increase in the heat exchange surface area of individual coolers. For the analyzed inter-stage cooler working at the pressure of 89.4 bar, the rise in temperature from 35°C to 50°C translates into a change in the flux of the exchanged heat from 31.7 MW to 14.1 MW. This is caused by the abrupt rise in the CO_2 heat capacity near the critical point (Fig. 3).

The analysis makes it possible to determine the heat transfer coefficients and the dimensions of individual inter-stage coolers of the CO_2 compressor. The results obtained may be used to develop an optimum structure for the cooling system, taking account of thermodynamic and economic criteria.

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Nomenclature

η	polytropic efficiency
π	pressure ratio
A_F	surface area including additional area of fins, m^2
c_p	specific heat, kJ/(kgK)
m_{H2O}	cooling water mass flow, kg/s
N_i	isentropic power, kW
$t_{CO2,H2Oin,out}$	inlet and outlet temperature of CO_2 and cooling water, °C
t_{mt}	logarithmic temperature difference, ° C
$w_{CO2,H2Oin}$	inlet CO_2 and cooling water velocity, m/s
$\triangle p_{CO2,H2O}$	CO_2 and cooling water pressure drop, bar
A	surface area excluding area of fins, m^2
d	diameter, m
h	enthalpy, kJ/kg
p	pressure, bar
Q	heat duty, kW
s	distance between the centers of tubes, m
U	overall heat transfer coefficient, $W/(m^2\ K)$