THERMODYNAMIC ANALYSES FOR OPTIMIZING THE DESIGN AND PREDICTING THE PERFORMANCES OF OXY FUEL ERICSSON-JOULE CYCLE IN SEMI-CLOSED LOOP, WITH FRAGMENTED COMPRESSION, IC, AND RHE

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The paper is a thermodynamic approach on oxy-fuel technology, in a semi-closed loop GT Ericsson-Joule cycle, allowing integrated CO_2 capture. In a first step the cycle was analyzed with simplified procedures, to delimitate the interest area. In the second, a complex model, using the thermodynamic properties of different gases involved in the process, and rational input data was used for optimizing the compression and predicting the performances. The results show that the thermodynamic efficiency of analyzed cycle is next to the Combined Cycle one, but with the environmental gain of CO_2 capture.

Keywords: oxy-fuel, CO_2 semi-closed loop, gas turbine, inter-cooling, internal heat recovery, compression ratio, thermodynamic efficiency

Nomenclature: GT - Gas Turbine, TPP - Thermal Power Plant, IRHE/RHE - (Internal) Regenerative Heat Exchanger, IC - Inter-Cooling, LHV – Low Heating Value, HHV – High Heating Value, CC - Combined Cycle, ST - Steam Turbine, K - compressor, CA - Combustion Chamber, T - Turbine, G - electric Generator.

1. Introduction

The idea of a thermodynamic analyze on a Ericsson-Joule cycle using oxy-fuel came from the need to understand the implications of changing the usual working fluid for a GT cycle, with a fluid consisting mostly from carbon dioxide (CO_2). The first steps were made using a simplified model in which CO_2 is the working fluid in a closed cycle, providing us the optimum configuration of the gas turbine cycle. Now having this configuration set, we lead our study to a more close to reality case by applying a model based on the real thermodynamic properties of the working fluid (gases mixture) in the different points of the installation. We putted into evidence the need of vapor water condensing during inter-cooling, in order to obtain high purity CO_2 after the compression line, ready for transport and storage. The considered fuel is a usual clean natural gas (mostly methane and no Sulfur content). For different compression ratios we evaluate the

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technical performance indicators and optimum compression ratio.

2. Objectives, assumptions and methods.

Oxy-fuel TPP use CO_2 semi-closed loops, allowing the integrated CO_2 capture [1 to 4]. The paper refers to the situation when CO_2 is the working fluid in a GT cycle, witch pressures does not depend from the atmospheric one. In order to keep clean the working agent it is necessary to use a clean fuel - in this situation the natural gas. ***The first objective*** of the paper is choosing cycle’s configuration. We started from the simple Joule cycle (Fig. 1), adding first an Internal Regenerative Heat Exchanger (IRHE) - Fig. 2, and then fragmenting the compression in 2 to 4 stages, with 1 to 3 Inter-Cooling processes (IC) - see fig. 3 to 5 [5]. In the paper the latest 3 designs will be labeled as Ericsson-Joule cycles.

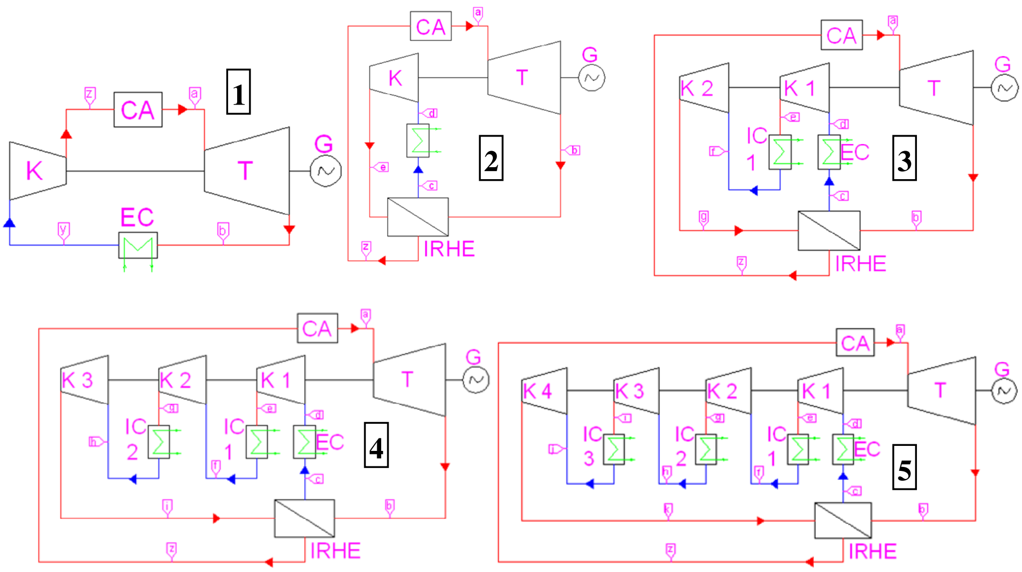

Fig. 1 to 5. Analyzed cycles configurations

The second goal of the study is predicting the performances for the selected configuration. We studied the influence of the compression ratio on the technical performances, and we elected the optimal compression ratio. For these ones, ***the third objective*** is computing the installation’s performances in given conditions, for real thermodynamic processes, and constructing the Sankey diagram of the thermodynamic cycle.

Modeling the real thermodynamic processes used in the first phase is done by analytical, simplified, methods by considering the installation filled with pure CO_2, which acts like a quasi-ideal gas [6], far from its critical point, and works in closed loop; like an outer burning cycle. The irreversibility of real thermodynamic processes (temperature difference at RHE, revolving machines isentropic efficiencies, and pressure drops) were taking into consideration [5]. For the heat exchangers we neglected the heat losses by thermal insulation.

Because the complexity of the process no common linear model was able to fulfill the needs of a good modeling, consequently for the second phase we created an iterative computing program. The simulation is performed only for stationary design loads. We are taking into consideration the real properties of the usual technical gases, and gases mixture [6]. For the water and steam properties where used International Association for the Properties of Water and Steam (IAPWS) equations [7]. We created procedures for numerical modeling of compression, burning, expansion, and cooling. The latest could include condensing the water vapor resulted from burning process. We started with input data based on the first phase results, and we recalculated the cycle and new entry data. The calculus is continuing until the error is small enough.

3. Presentation and interpretation of the obtained results

In the first phase, for the different configurations shown in Fig. 1 to 5 the global compression ratio variation was used for drawing diagrams of the main thermodynamic efficiency indicators. Fig. 6 and 7 show the evolution of the net mechanical work and net efficiency for each configuration.

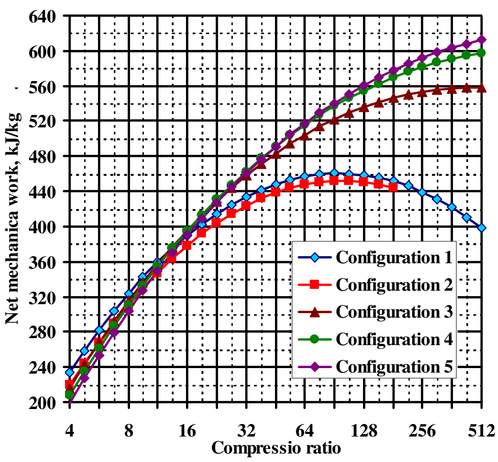

Fig. 6. Net mechanical work vs. global compression ratio, for diverse configurations

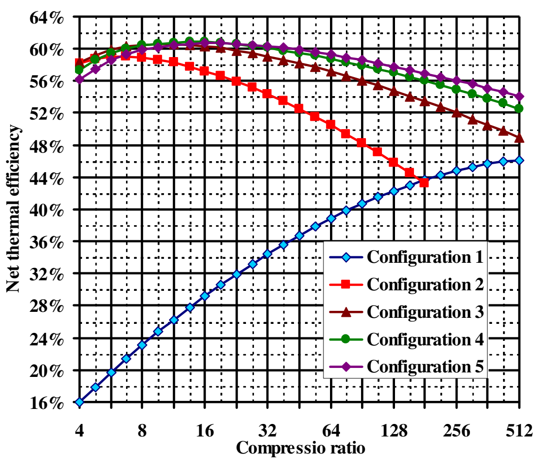

Fig. 7. Net thermal efficiency vs. global compression ratio, for diverse configurations

We found out that the best choice is a semi-closed loop Ericsson-Joule cycle with 3 compression stages and 2 IC heat exchangers (see Fig.8.). For this design we optimized the compression ratio, and we obtained new results regarding the variation of net efficiency and net mechanical work (see Fig. 9 and 10).

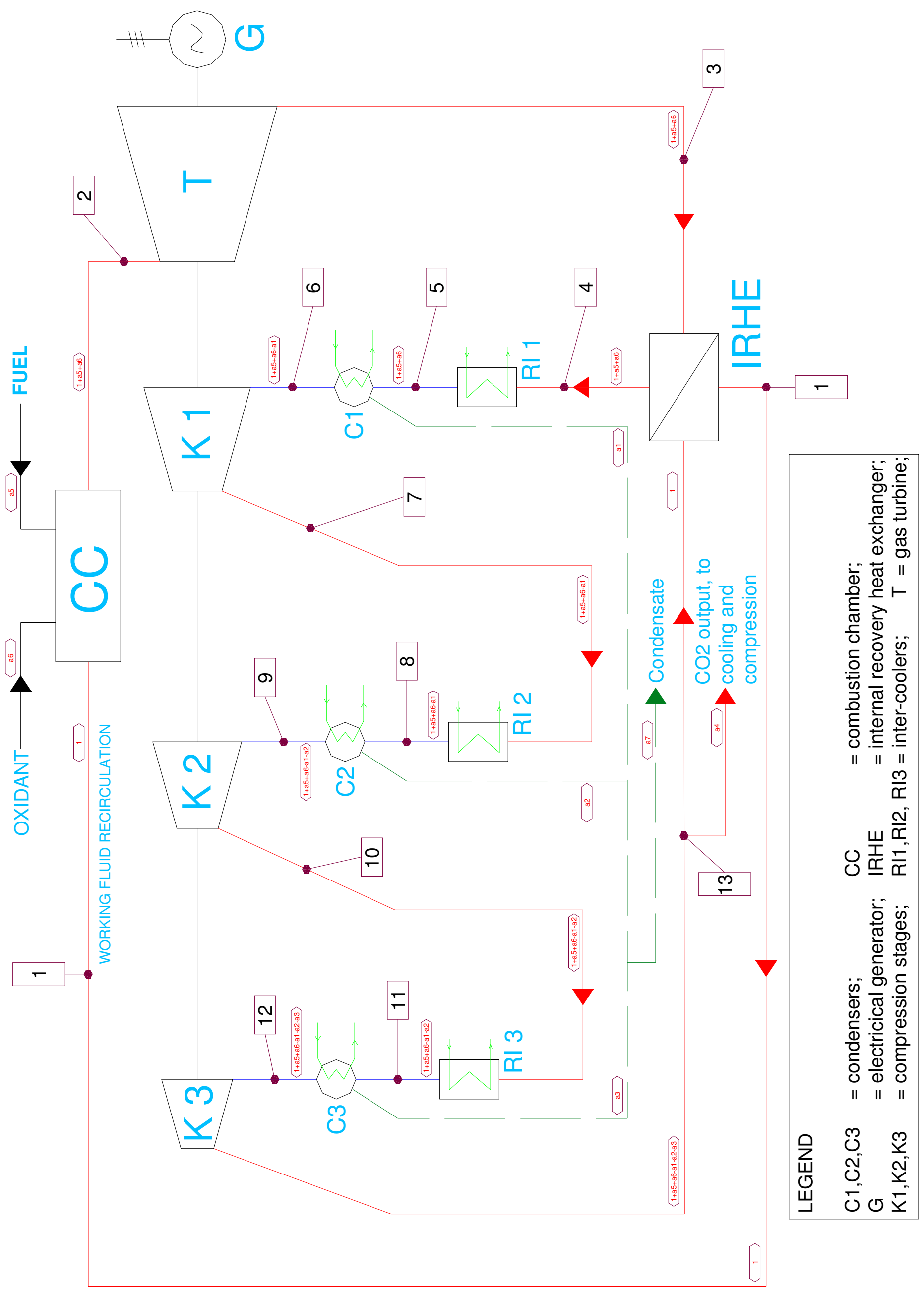

Fig. 8. The optimized configuration of the thermal cycle

Tables 1 and 2 show the inputs data for the fuel (natural gas specific for Romania, with high methane ratio and without Sulfur) and the oxidant (Oxygen with almost 95 % purity, and about 5 % Argon), considered for the real condition.

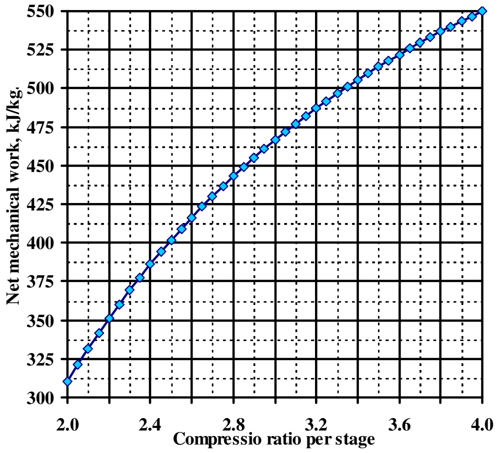

Fig. 9. Net mechanical work vs. compression ratio per stage, for the optimal design

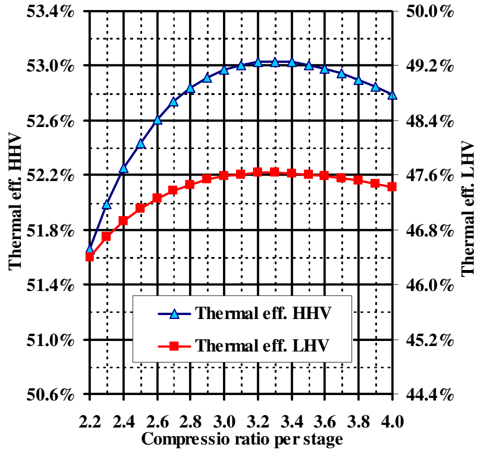

Fig. 10. Net thermal efficiency vs. compression ratio per stage, for the optimal design

Table 1

Fuel parameters

Parameter	Value	Units
Total pressure	43.034	bar
Temperature	50.0	^{0}C
Mixture enthalpy	109.68	kJ/kg
Mixture entropy	-1.436	kJ/kg/K
HHV	39 511	kJ/m^3_N
LHV	35 490	kJ/m^3_N
Mixture density:	26.5655	kg/m^3
Mixture molecular weight	16.521	kg/kmol

Table 2

Oxidant parameters

Parameter	Value	Units
Total pressure	43.034	bar
Temperature	25.0	^{0}C
Mixture enthalpy	22.987	kJ/kg
Mixture entropy	-0.830	kJ/kg/K
Mixture density:	47.992	kg/m^3
Mixture molecular weight	32.123	kg/kmol

Table 3

Working fluid parameters and composition in different points of the process

No	p	t	h	s	ρ	μ	Molecular percentages, %				Weight percentages, %			
	bar	^{0}C	kJ/kg	kJ/kg/K	kg/m^3	kg/kmol	r_{H2O}	r_{N2}	r_{O2}	r_{CO2}	g_{H2O}	g_{N2}	g_{O2}	g_{CO2}
1	35.86	602.7	634.26	0.5502	13.43	42.967	0.423	5.535	0.458	93.59	0.177	3.626	0.341	95.86
2	34.73	1 200	1 537	1.5596	7.108	40.787	9.117	5.065	0.418	85.40	4.027	3.496	0.328	92.15
3	1.12	652.7	815.04	1.6494	0.364	40.787	9.117	5.065	0.418	85.40	4.027	3.496	0.328	92.15
4	1.08	254.5	349.20	1.0029	0.619	40.787	9.117	5.065	0.418	85.40	4.027	3.496	0.328	92.15
5	1.07	45.3	140.85	0.5066	1.01	40.787	9.117	5.065	0.418	85.40	4.027	3.496	0.328	92.15
6	1.05	31.9	75.17	0.2999	1.087	41.956	4.449	5.325	0.439	89.79	1.911	3.573	0.335	94.18
7	3.52	132.7	168.16	0.3231	2.736	41.956	4.449	5.325	0.439	89.79	1.911	3.573	0.335	94.18
8	3.46	54.5	95.25	0.1270	3.335	41.956	4.449	5.325	0.439	89.79	1.911	3.573	0.335	94.18
9	3.41	31.9	41.43	-0.0424	3.631	42.727	1.371	5.497	0.453	92.68	0.578	3.621	0.340	95.46
10	11.41	132.3	132.71	-0.0196	9.15	42.727	1.371	5.497	0.453	92.68	0.578	3.621	0.340	95.46
11	11.23	54.5	61.24	-0.2119	11.14	42.727	1.371	5.497	0.453	92.68	0.578	3.621	0.340	95.46
12	11.05	31.9	31.28	-0.3052	11.89	42.965	0.423	5.549	0.458	93.57	0.177	3.636	0.341	95.85
13	37.03	132.2	122.04	-0.2825	29.96	42.965	0.423	5.549	0.458	93.57	0.177	3.636	0.341	95.85

Table 3 shows the parameters and composition of the working fluid in different points of the process. With these values the approximate T-s diagram shown in the Fig. 11 was constructed.

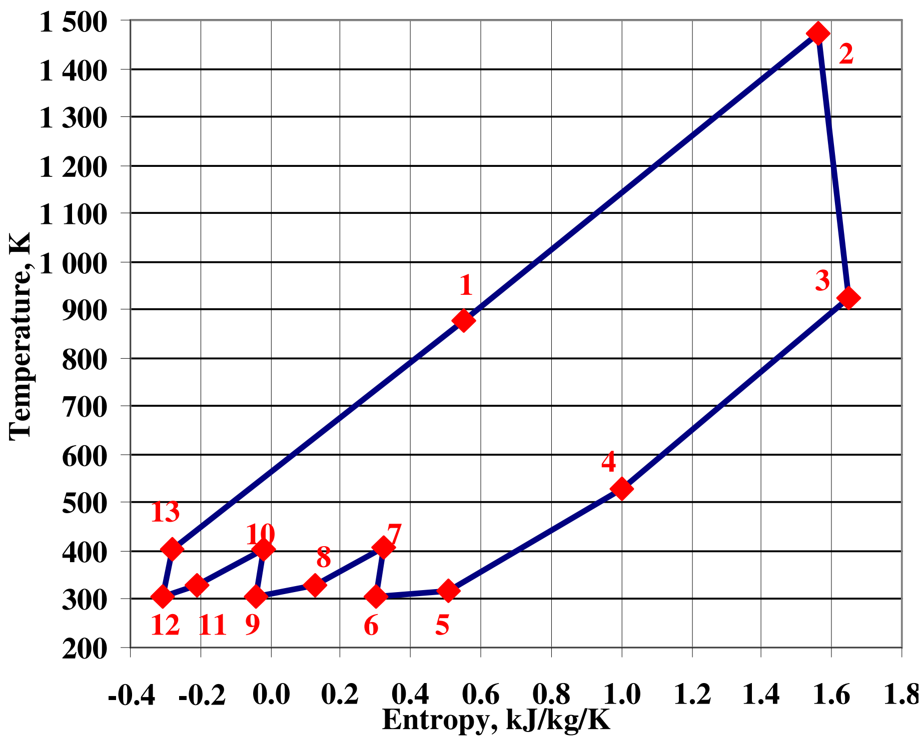

Fig. 11 T-s diagram

Table 4

The main thermal cycle performance indicators in the optimized configuration

Parameter	**Notation**	**Value**	**Units**
Mechanical work produced by the gas turbine	L_T	793.91	kJ/kg
Mechanical work consumed by the compressor	L_K	292.87	kJ/kg
Net mechanical work	L_{net}	501.04	kJ/kg
Relative compressor energy consumption	$C_{r\,K}$	36.89	%
Heat received from the hot source for HHV of the fuel	Q_{1HHV}	1 051.87	kJ/kg
Heat received from the hot source for LHV of the fuel	Q_{1LHV}	944.83	kJ/kg
Exhausted heat to the cold source of the cycle	Q_2	542.19	kJ/kg
Internal heat recovery	Q_{rec}	512.23	kJ/kg
Heat recovery ratio for HHV of the fuel	$C_{Q\,R\,HHV}$	48.70	%
Heat recovery ratio for LHV of the fuel	$C_{Q\,R\,LHV}$	54.21	%
Thermodynamic cycle efficiency for the fuel's HHV	η_{t_HHV}	47.63	%
Thermodynamic cycle efficiency for the fuel's LHV	η_{t_LHV}	53.03	%

Putting all together we completed the work by calculating the main performance indicators (Table 4), and representing the energy flows in a Sankey diagram (Fig. 12).

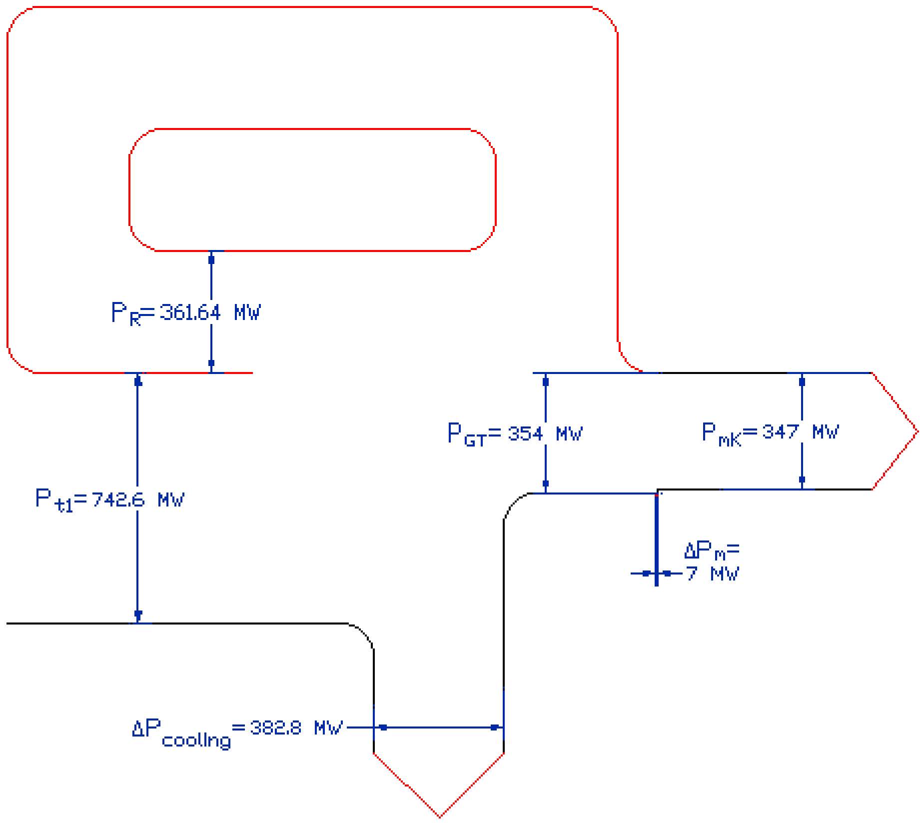

Fig. 12. Sankey diagram

where:

P_{t1} = fuel thermal power (HHV)
P_{GT} = internal GT net mechanical work
ΔP_m = mechanical losses
P_R = re circulated heat
$\Delta P_{cooling}$ = exhausted heat
P_{mK} = mechanical power at GT couple

4. Conclusions

The general guidelines for increasing the performances of Ericsson-Joule cycles are: ***A)*** reducing the irreversibilities and choosing a design correlated with them; ***B)*** increasing the CC's outlet temperature. Because of processes irreversibilities, the analyzed cycle's efficiency augment is gradually lower when n_{IC} ascend from n_{IC}=1 to n_{IC}=2, becoming inferior for n_{IC}=3 than for n_{IC}=2. That's why we were choosing the design with 3 compression stages and 2 ICs.

The thermodynamic efficiency of Oxy-fuel GT TPP using CO_2 semi-closed loop, resulted for the fuel's LHV, is comparable with those of CC TPP. The environmental gain is that the analyzed cycle allows the integrated CO_2 quasi-total capture [8, 9]. The economic limitation is that, it is necessary using a clean fuel - natural gas - with a higher cost.

The procedures created in this paper for numerical modeling the real properties of usual technical gases and gases mixture, and their real thermodynamic processes could be employ for other similar purposes, like analyzing oxy-fuel processes in a CC gas-steam, or ST TPP burning coal, using CO_2 in semi-closed loops, with ST and without GT.

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