Assessment of the Quota of Recuperative Cooling of the Compressed Gas at Turbocharged Reciprocating Internal Combustion Engines

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***Abstract* – Turbocharging is a method of reducing the specific investment and raising the electrical efficiency of Reciprocating Internal Combustion Engines (RICE). The paper starts with a qualitative analysis, useful in educational purposes, about the energy flows and on the performances indices of RICE used for power generation and cogeneration. The paper continues with a statistical analysis of the manufacturers' data for a RICE with turbochargers, useful in management purposes. The main paper' section contains a computational model of compression and cooling air or air-gas mixture process, intended for Romanian climate. Its results are useful for RICE cogeneration.**

***Keywords* – reciprocating gas engines, cogeneration, turbocharger, air cooling.**

1. Introduction

On the market there is a wide variety of Reciprocating Internal Combustion Engines (RICE). They can be classified according to different criteria, such as:

- type of fuels (liquid, gas);
- thermodynamic cycle and type of ignition (Otto, Diesel, Diesel-gas, etc.);
- number of revolutions per cycle (one or two);
- air–fuel equivalence ratio;
- stroke vs. bore ratio;
- engine rotation speed (slow, medium or rapid);
- degree of turbocharging: with atmospheric supply, Normally Aspirated (NA), respectively Turbocharged & Aftercooled (Tb&A).

Due to the diversity designs and functioning schemes of RICE, and the technological advancements in the field, they are currently utilised in a wide variety of domains (land, sea or air transport, agricultural machines, industrial processes [1]). In the past few decades the performances of RICE lead to their comeback in the power generation sector, including cogeneration applications [2],[3],[4]. Below is a summarised list of the main advantages which lead to the increased interest from energy companies in utilising RICE:

- In "power generation only" applications, the RICE's electrical efficiency at generators clamps reaches, for 5 to 25 MW units, values beyond 49 %. Thus, RICE has become the most efficient mean for thermodynamic generation of electricity in a simple cycle.
- With certain changes into the fuel feeding system [5], RICE can utilise a large variety fuels: liquids (heavy fuel, diesel petroleum fuel, diesel bio fuels, alcoholic biofuels, and others), or gaseous [6],[7],[8] (methane, syngas from coal or organic waste, landfill gas, biogas from digesters, and other hydro-carbonates).
- RICE's start-up is quicker than any other equipment for power generation through thermodynamic means [2],[6], faster than gas turbines (GT). This offers an advantage for the energy systems with significant contribution from renewable sources as sun or wind, due to their intermittent nature. RICE can quickly

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intervene during periods of unforeseen outage of renewables, thus ensuring the reliability of the energy system.

- In the small and medium applications (from 10 kW to 5 MW) [9],[10], RICE's electrical efficiency is still good enough, and is above that of gas turbines of comparable powers. This brings an advantage for remote areas, where connection to external power grid is not possible, or the connections are weak.
- Similarly, to GT [4], RICE [4],[11] can be adapted for cogeneration by recovering its heat losses, without increasing the fuel consumption or reducing the electricity production, with potential overall efficiencies higher than 85%.

When recovering heat losses from any type of RICE, for cogeneration, of particular interest are the heat flows rates associated with [2],[4]:

- technological cooling of the engine block and engine oil (because of low temperature it could be used only for residential consumers);
- sensible heat of flue gas (at medium temperature).

Furthermore, Tb & A type RICE have an additional source of heat loss, associated with cooling the compressed air / gas-air mixture [3]. This loss is, at the same time, a technological cooling and a thermodynamic one. It has a lower thermal level comparative to flue gases, but they can also be recovered for residential consumers, but only partially.

The paper starts with a comparative analysis of the input and output energy flows and performance indices for NA type RICE, and Tb & A type RICE. It continues with a statistical analysis of manufacturers' data on "RICE $_{gas}$" (fuelled with natural gas), used in the energy field, with unit powers between 100 kW and 10 MW. The final paper's section models the process of compression and cooling the air (or air+CH4 mixture), for Tb & A type RICE, in the specific climate of Romania. This allowed us to compute the amount of heat that should be evacuated in the cooling process and determine the quota from this that could be recovered for residential cogeneration, for improving the overall efficiency of Tb & A type RICE.

2. Analysing the energy flows and defining the performances indices for reciprocating gas engines

Figure 1. illustrates the energy fluxes on the RICE's energy balance contour. Zone "I" contents the irrecuperable losses. Zone "II" contents the fluxes that result from the process of converting the fuel's heat into mechanical work and then into electricity. Zone III contents the heat losses that can be recovered, at least partially, for cogeneration. Figure 2. displays the blocks shown in Figure 1. in a tree diagram form, for NA type RICE, both for power generation only and for residential cogeneration. In the power generation regime, the sole indicator of technical performance is the electrical efficiency [12] (1).

$$\eta_{el} = \text{Electric power / Fuel} = P_{el} / Q_{fuel} \qquad (1)$$

In the cogeneration case there is an additional useful effect from the recovered heat [12]. In the Sankey diagram for NA type RICE (Figure 2.) it was considered that:

- heat flow rate from liquid cooling circuit ($Q_{w\,rec}$) is fully recoverable (2);
- heat flow rate recovered from flue gases ($Q_{fg\,rec}$) (3) is a share (β_{fg}) of ΔQ_{fg} (4).

$$Q_{w\,rec} = \Delta Q_m + \Delta Q_{jc} \qquad (2)$$

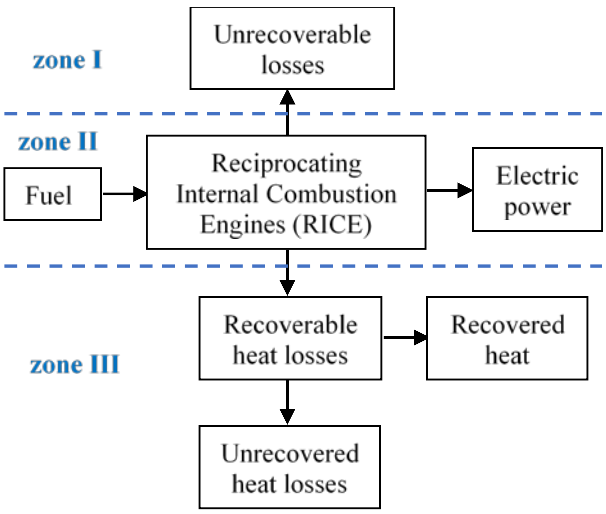
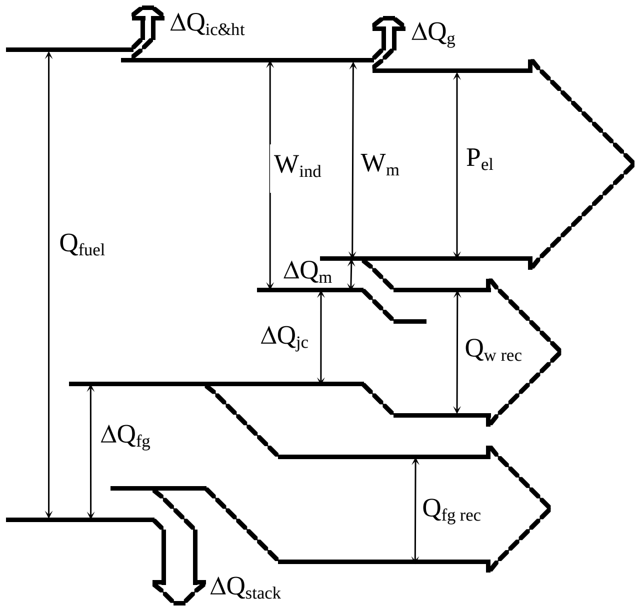


Figure 1. Flows of RICE working in cogeneration



Q_{fuel} – fuel heat flow rate, W_{ind} – indicated work, W_m – mechanical work, P_{el} – power at generator's clamps, $Q_{w\ rec}$ – heat flow rate recovered from liquid cooling circuit, $Q_{fg\ rec}$ – heat flow rate recovered from flue gases, ΔQ_m – mechanical losses, ΔQ_{jc} – jacket cooling losses, ΔQ_{fg} – sensible heat of gases; ΔQ_{stack} – stack losses.

Figure 2. Sankey diagram for NA type RICE, working in residential cogeneration

$$Q_{fg\ rec} = \Delta Q_{fg} - \Delta Q_{stack} \quad (3)$$

$$\beta_{fg} = Q_{fg\ rec} / \Delta Q_{fg} \quad (4)$$

In this case, some additional performance indicators appear. For NA type RICE these are:

- thermal efficiency (η_{th}):

$$\eta_{th} = \text{Recovered heat / Fuel} = (Q_{w\ rec} + Q_{fg\ rec}) / Q_{fuel} \quad (5)$$

- overall efficiency (η_{gl}):

$$\eta_{gl} = (\text{Electric power + Recovered heat}) / \text{Fuel} = (P_{el} + Q_{w\ rec} + Q_{fg\ rec}) / Q_{fuel} = \eta_{el} + \eta_{th} \quad (6)$$

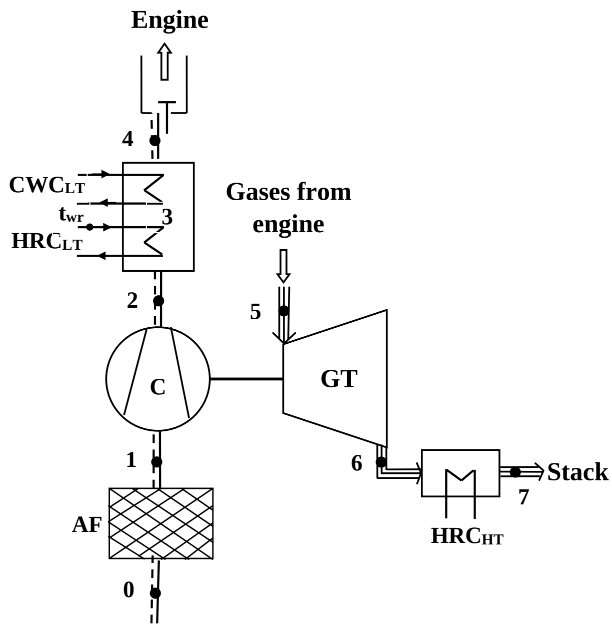
- cogeneration index (y_{cog})-power vs. heat ratio:

$$y_{cog} = \text{Electric power / Recovered heat} = P_{el} / (Q_{w\ rec} + Q_{fg\ rec}) = \eta_{el} / \eta_{th} \quad (7)$$

RICE's supercharging is a way to increase the unit power and electrical efficiency, based on aspirated gas pressure increase. This raises the RICE's indicated mean effective pressure and indicated power, without a corresponding increase in friction losses. The result is an increase in mechanical efficiency, brake power, electrical power, and electrical efficiency.

Figure 3. shows the air – flue gases circuit of a Tb & A type RICE, in a base structure, with one compressor + one turbine + one air aftercooler (also called intercooler, being located between the compressor outlet and the RICE's intake).

During the compression the gas temperature rises. The aftercooler lowers the gas temperature, increasing the gas density and the gas mass flow rate aspirated by the engine. It also avoids accidental ignition during compression.



AF – air filter, C – compressor, HRC_{LT} - heat recovery circuit at low temperature, CWC_{LT} - cooling water circuit at low temperature, GT – gas turbine, HRC_{HT} - high temperature flue gases heat recovery circuit

Figure 3. Integration a single compressor + aftercooler + single turbine turbocharger, in the air-flue gases circuit of a Tb & A type RICE $_{gas}$ for residential cogeneration

Cooling the aspirated air (at least in the warm periods of the year) leads to additional heat outputs at Tb & A type RICE, comparative to NA type RICE. In the analysed circuit, specific for RICE used in cogeneration for residential purposes, we consider the aftercooler split in two sections:

- The first section (HRC_{LT}) is integrated into the Heat Recovery Circuit at Low Temperature. It reduces the heat losses of TB & A type RICE used for residential cogeneration.
- The second section (CWC_{LT}) is part of a distinct low temperature water cooling circuit, without heat recovery.

Figure 4. shows the energy fluxes diagram for Tb & A type RICE. It can be noticed that an additional loss appears in Zone "I", associated to the air cooling in CWC_{LT} without heat recovery, and in Zone "III" there is an additional useful heat flux, associated to the recovered heat from air cooling in

HRC_{LT}.

The additional performance indicators for Tb & A type RICE become (5') to (7'):

- thermal efficiency (η_{th}):

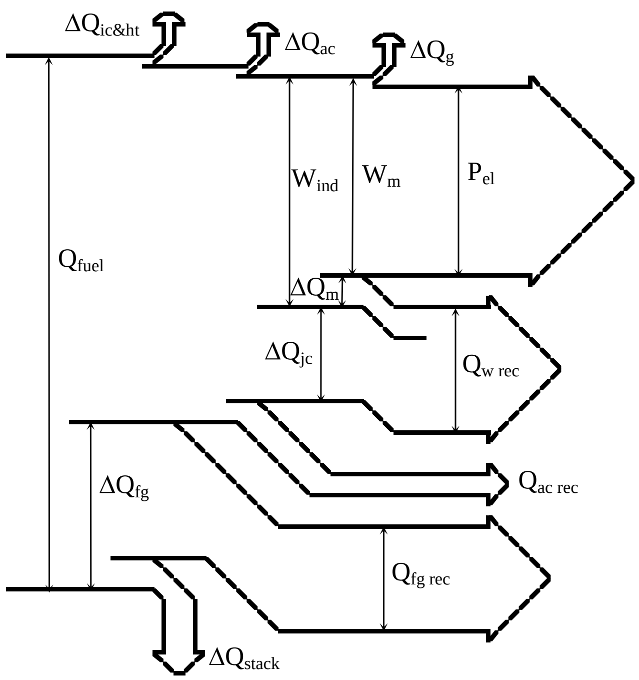
$$\eta_{th} = (Q_{w\,rec} + Q_{fg\,rec} + Q_{ac\,rec}) / Q_{fuel} \quad (5')$$

- overall efficiency (η_{gl}):

$$\eta_{gl} = (P_{el} + Q_{w\,rec} + Q_{fg\,rec} + Q_{ac\,rec}) / Q_{fuel} \quad (6')$$

- power vs. heat ratio / cogeneration index (y_{cog}):

$$y_{cog} = P_{el} / (Q_{w\,rec} + Q_{fg\,rec} + Q_{ac\,rec}) \quad (7')$$



$Q_{ac\,rec}$ – heat flow rate recovered from compressed gas & air cooling, ΔQ_{ac} –heat flow rate unrecovered from compressed gas & air cooling.

Figure 4. Sankey diagram for Tb & A type RICE, working in residential cogeneration

It is worth mentioning that University POLITEHNICA of Bucharest has a thermal power plant equipped with two Tb & A type $RICE_{gas}$ of power P_{el} = 800 kW, which operate in cogeneration. The power plant is used not only for electricity and heat supply, but also for educational purposes, having specialised measuring equipment. The laboratory work carried out by the students inside the power plant consists of calculating the energy fluxes for constructing the Sankey diagram in Figure 4. and determining the performance indicators by using equations (1) to (4), respectively (5') to (7').

3. Analysis on performances of Tb & A type RICE starting from producer' data

Usually the manufacturers' data is defined: ***a)*** in power generation mode, ***b)*** for operation on CH_4 as fuel, and ***c)*** for ISO air external conditions (101.3 kPa, 15 °C, 60 % humidity). Generally, the provided data is: ***1)*** P_{el}, ***2)*** electrical efficiencies (η_{el}), ***3)*** exhaust flue gases mass flow rates ($\dot{m}_{fg}$), and ***4)*** exhaust flue gases temperatures ($t_{fg@ex}$). Out of these:

- The main quality indicator of the RICE performance for power generation only is η_{el}.
- Exhaust flue gases temperature is also a quality indicator, but only for RICE working in cogeneration. Higher values of $t_{fg@ex}$ indicate the opportunity of recovering heat at higher thermal levels, and the possibility of achieving a higher recovery factor β_{fg}. On the other hand, $t_{fg@ex}$, and $\dot{m}_{fg}$, enable determining the thermal flux that can be recovered from the flue gases' sensitive heat and preliminarily establishing the technical performance of RICE in cogeneration.

It was identified in the Thermoflex software database a number of 260 $RICE_{gas}$, with power at generator's clamps (P_{el}) in the range (0.1÷10) MW. 36 of them are NA type, all with P_{el} < 1 MW. Out of the 224 TB & A type, 144 of them have P_{el} < 1 MW, respectively 80 have P_{el} > 1 MW.

Table 1. shows the results of a statistical analysis on two significative data (η_{el}, and $t_{fg@ex}$) of the 224 TB & A type $RICE_{gas}$, divided into the above defined power groups. This table indicates:

- The characteristic values of the analysed technical indicators on the studied domains: Min, Average, Max, standard deviation (σ). This allows identifying the intervals in which the analysed indicators can be found, with a probability of 70.7% (Average-σ÷Average+σ).
- The linear correlation coefficient (R^2) between the indicators and a parameter derived from P_{el} (kW): $Log_{10}(P_{el}/100)$. The aim was to establish the degree of correlation between the values of the indicators and P_{el}.

Figure 5. shows (as markers) the values of η_{el}, respectively the trend lines for P_{el}< 1 MW and P_{el} > 1 MW. The analysis of η_{el} trends variation reveals a small scale effect for P_{el}< 1 MW (P_{el} increase leads to an η_{el} gain) and a bigger one over 1 MW. For P_{el} <1 MW, $\eta_{el\,average} \cong 33.7$ % with

$\sigma = \pm$ 2.3 %. The correlation of η_{el} with P_{el} is weak ($R^2 < 5$ %). For $P_{el} > 1$ MW the scale effect is stronger than in the previous field, $\eta_{el\ average}$ is higher (≅ 39.1 %) with $\sigma = \pm$ 5.2 %, and the correlation with P_{el} is better (R^2 = 68.5 %). The main cause of the increase in electrical efficiency with power is associated with the increase of TB & A type RICE $_{gas}$ turbocharging.

Figure 6. shows the variation of the exhaust flue gas temperatures ($t_{fg@ex}$) with P_{el}, and the associated trendlines. Here it is noticeable the downward trend of $t_{fg@ex}$ with P_{el}.

According to Table 1., for P_{el} <1 MW, $t_{fg@ex\ average}$ = 461 °C, with $\sigma = \pm$ 79 °C. The correlation of $t_{fg@ex}$ with P_{el} in this field is neglectable ($R^2 \cong 3$ %). For $P_{el} > 1$ MW, $t_{fg@ex\ average}$ (439 °C) is lower than the previous one, and the correlation with P_{el} is better (R^2 = 30.5 %).

Table 1. Statistical analysis of η_{el} and $t_{fg@ex}$ using manufacturers' data

	Indicators:	**Electrical efficiencies, %**		**Exhaust flue gas temperatures, °C**	
	Power unit (P_{el})	**P_{el} < 1 MW**	**P_{el} > 1 MW**	**P_{el} < 1 MW**	**P_{el} > 1 MW**
Characteristic values	**Min**	28.70	29.60	310	320
	Average-σ	31.39	33.92	381	362
	Average	33.66	39.14	461	439
	Average+σ	35.92	44.37	540	516
	Max	39.70	49.90	634	627
	± σ	2.26	5.23	79.34	77.18
	R^2	4.91	68.49	2.98	30.53

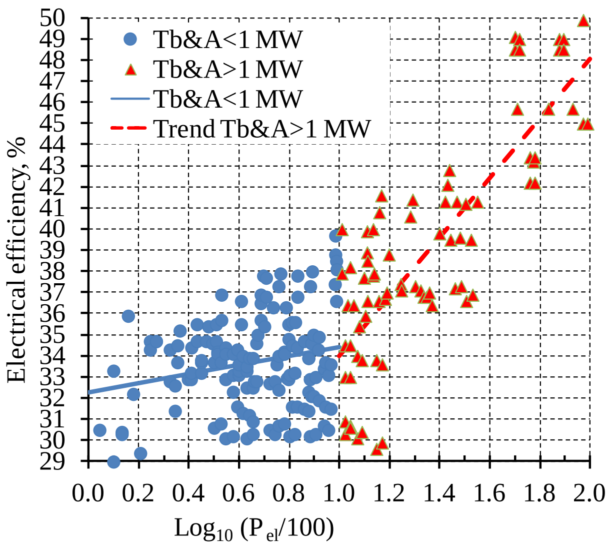


Figure 5. Electrical efficiency vs. RICE power

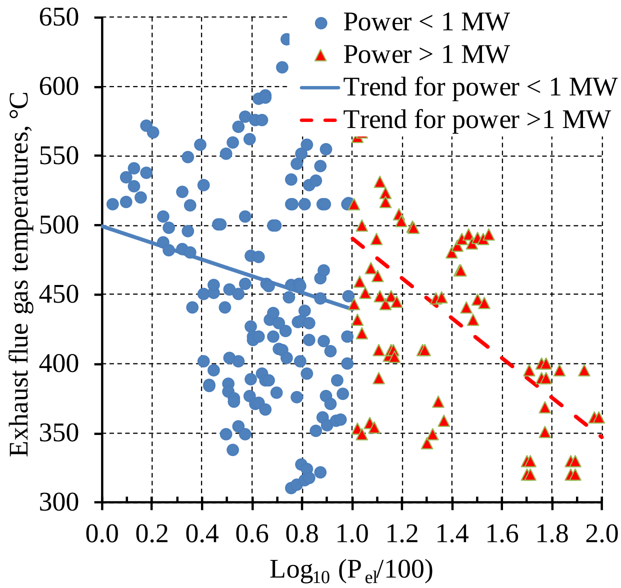


Figure 6. Exhaust flue gas temperatures (°C) vs. RICE power

Based on the information above it can be established that both the numerical values of Table 1. and the graphs in Figures 5. and 6. are useful tools for the managerial staff when purchasing a RICE. These enable a performance comparison of the equipment on the list of offers to what's available on the market.

When choosing an engine with a given P_{el}, regardless of its future utilisation, either for power only or cogeneration, it can be first checked in Figure 5. if the value of η_{el} guaranteed by the manufacturer is above the trendline. If RICE is to be used in cogeneration, among two engines with similar η_{el}, the choice will be in favour of the one with a higher thermal potential and better ratio of heat recovery from the sensible heat of the exhaust flue gases ($t_{fg@ex}$ guaranteed by the manufacturer should be above the trendline in Figure 6.).

4. Computing the recovered heat from the aftercooling process

This paper analyses the compression and cooling processes for the design given in Figure 3., but the models developed here are also applicable to more complex schemes.

Figure 7. shows the flow-chart for modelling the compression and aftercooling process and the recovered heat from air. The main hypotheses used for developing the models are summarised below:

- the air is taken directly from the atmosphere (outside of the engine room), with a range of ambient temperatures $t_0 \in (-20 \div 40)$ °C, typical to the climate of Romania;
- the quasi-adiabatic compression, without heat exchange with the ambient, is in fact non-isentropic (isentropic efficiency is 80 %);

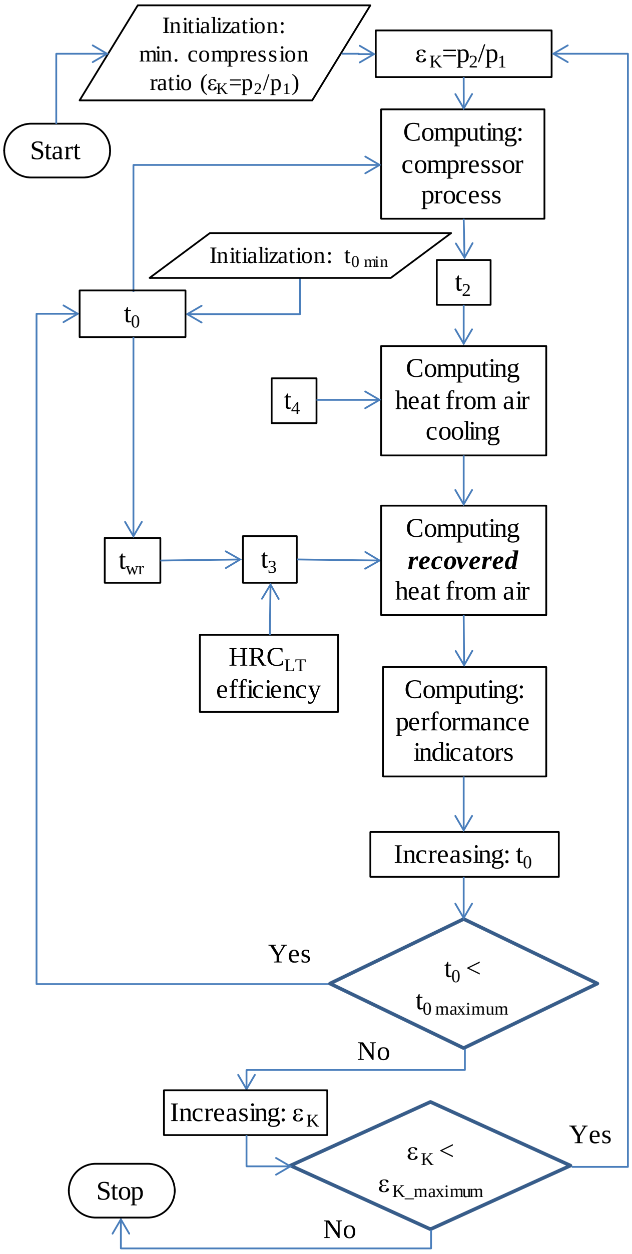


Figure 7. Flow-chart for modelling the compression and aftercooling process and the recovered heat from air

- the compression ratios (p_2/p_1) have typical values for single-stage diagonal subsonic design, $p_2/p_1 \in (1.12 \div 2)$, in geometric progression with the ratio $10^{1/20}$;
- air Filter (AF) and aftercooler pressure losses were taken into consideration;
- the limit value of t_4 was set to 50 °C;
- ***if*** computed $t_2 < t_4$, ***then*** it is not necessary the aftercooling, consequently $t_2 = t_3 = t_4$;
- ***if*** computed $t_2 > t_4$, ***then*** the air cooling is mandatory, with or without the heat recovery;
- the limit value for the temperature of return water from the residential heat network (t_{wr}) is correlated with the ambient temperature;
- the value of t_3 is correlated with t_{wr} and the HRC_{LT} efficiency;
- ***if*** $t_2 > t_3$, ***then*** it is possible to recover a quota of heat from the air' cooling process, ***else*** the HRC_{LT} is bypassed.

Figures 8. to 12. show the results of modelling the compression and cooling processes on the entire range of values mentioned above for p_2/p_1 and t_0.

Figure 8. shows that the necessary mechanical work for compression per kg varies almost linearly with $\log_{10}(p_2/p_1)$ and has little influence from t_0. On the other hand, the air temperature at the compressor outlet (t_2) grows linearly with t_0 and $\log_{10}(p_2/p_1)$, reaching values above 70 °C, where heat recovery for cogeneration is possible, but also values below 50 °C, where cooling is not necessary (Figure 9.).

Figure 10. shows the total amount of heat per 1 kg extracted during air aftercooler, when cooling the air down to 50 °C. In fact, only a quota of this cooling process can be done with heat recovery, the remaining, necessary for technological aims, is irrecoverable.

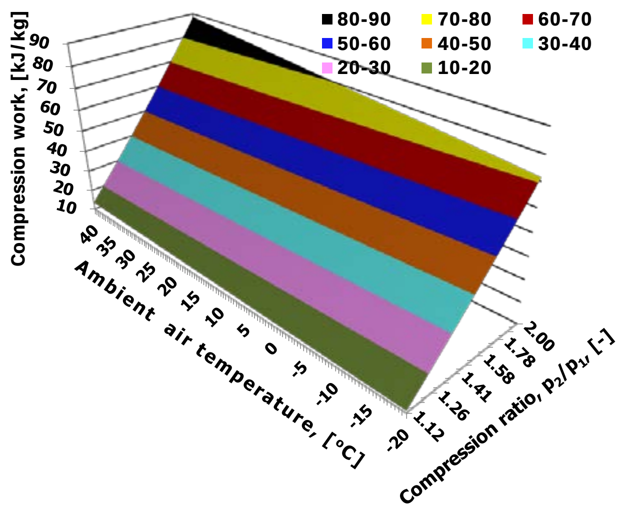


Figure 8. Compression work (kJ/kg) vs. ambient air temperature (°C) and compression ratio (-)

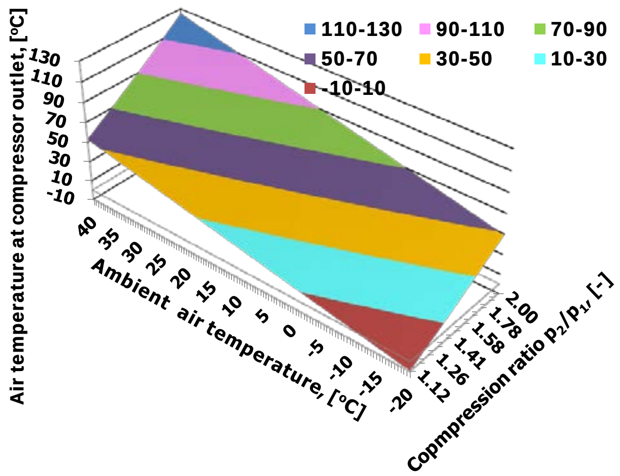


Figure 9. Air temperature at compressor outlet, versus ambient air temperature, and compression ratio

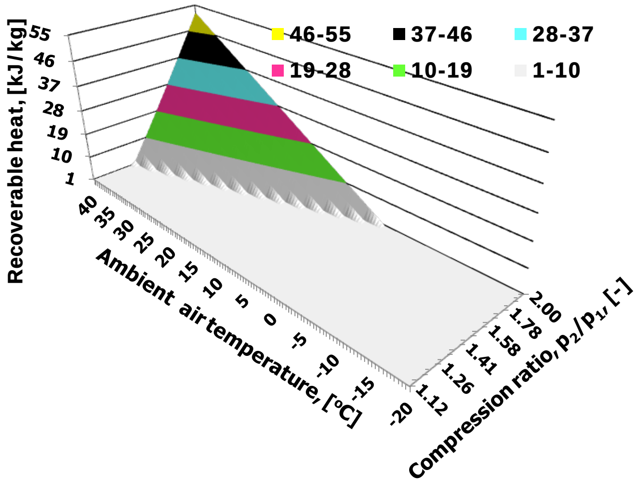


Figure 11. Recoverable heat from air aftercooling process per 1 kg, versus ambient air temperature, and compression ratio

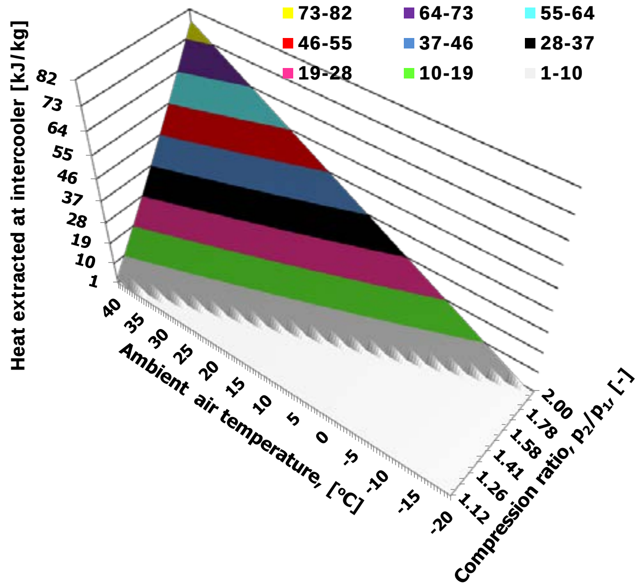


Figure 10. The amount of heat taken over at aftercooler per 1 kg air, versus air temperature, and compression ratio

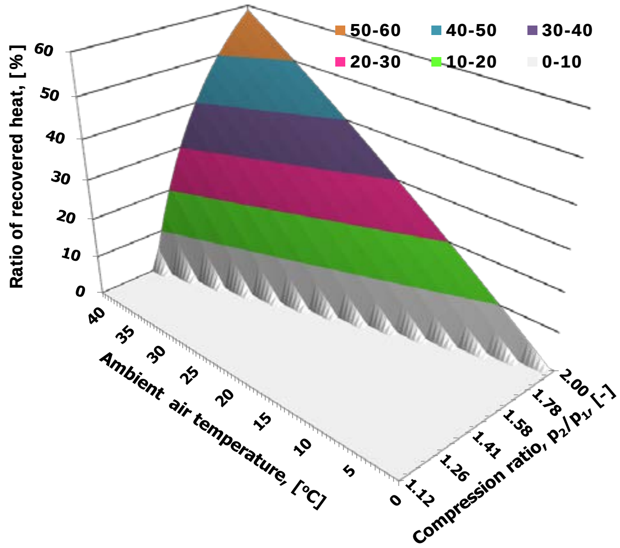


Figure 12. Ratio of recovered heat from air aftercooling process vs. air temperature, and compression ratio

Figure 11. highlights the fraction of the total heat amount extracted at air aftercooler, recoverable by cooling down to 70 °C.

Figure 12. emphasises the dependency on t_0 and $\log10(p_2/p_1)$ of the ratio of heat recoverable that needs to be evacuated at cooling. The maximum values of the recoverable ratios (around 60 %) are reached for the highest ambient air temperatures and the bigger compression ratios.

On the other hand, it must be considered that the ambient air temperatures have various probabilities of occurrence on the maximum domain of values.

Figure 13. shows the probabilistic distribution of ambient temperatures in the Romanian climate, illustrated by a Gaussian curve. This distribution helped to establish 100 temperature intervals of equal probability.

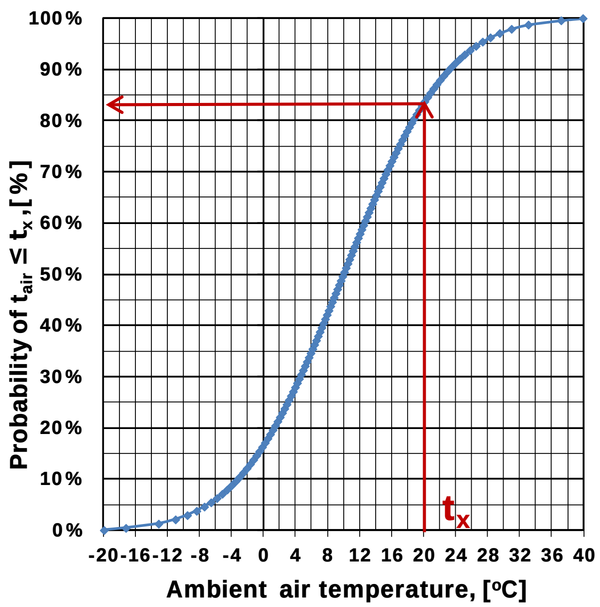


Figure 13. Probabilistic distribution of ambient air temperatures in the average climate conditions of Romania

For the central temperature of each interval the values of the indicators in Figures 8. to 12. were calculated. By averaging them, it was possible to obtain the annual mean values for the mechanical work for compression, total amount of heat that should be extracted at aftercooling, the recoverable heat from previous, and the share of recovery (Figure 14.).

It can be observed (Figure 14.) that for compression ratios under 1.33, it is not necessary to use the cooling. For p_2/p_1 between 1.33 and 1.41 the cooling process became necessary, but the heat recovery is negligible. When the ratio p_2/p_1 exceeds 1.41, the amount of recoverable heat from aftercooling, for cogeneration, continuously increases, but it remains significantly lower than the total amount of heat that must be evacuated for cooling the air (Figure 14.).

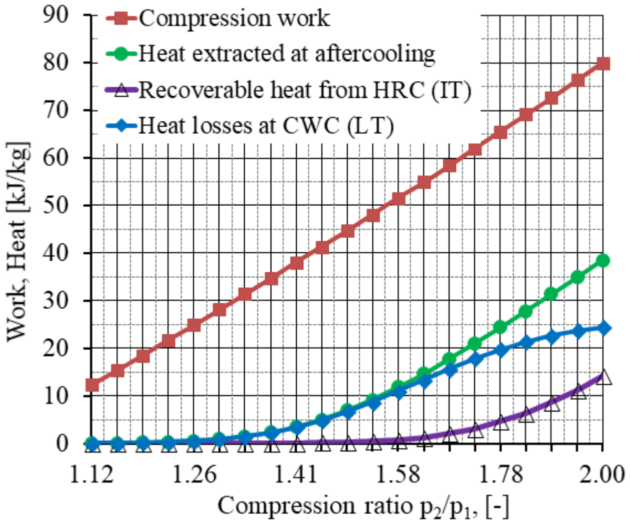
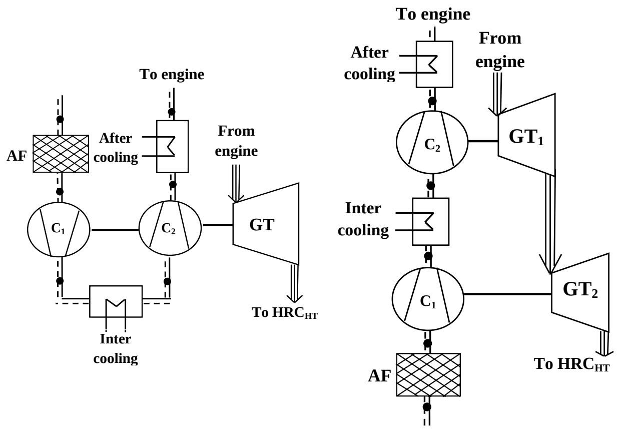


Figure 14. Average annual values of compression work, heat extracted at aftercooling, recoverable heat from HRC_{LT}, and heat losses at CWC [kJ/kg]. (HRC cu LT)

At $p_2 / p_1 = 2$, the maximum value for which the modelling was made, from 38.5 kJ / kg total heat extracted at aftercooler, only 14.1 kJ / kg (about 36.6%) can be recovered for cogeneration. The remaining heat of 24.4 kJ / kg should be discharged into the environment via CWC_{LT} constituting losses.

Higher power Tb & A type RICE, with higher air compression ratio, uses superchargers with two series-connected compressors, driven by a single turbine or separate turbines, and two air coolers (an intercooler between the first stage and the second one, and an aftercooler, between the second compressor and the air intake into RICE) (Figure 15.a. and 15.b.).



a) 2 stage compression turbocharger with a single turbine and 2 compressors

b) 2 stage compression turbocharger with 2 GT and 2 compressors

C_1 – low pressure compressor, IC – Intercooler, C_2 – high pressure compressor, AC – Aftercooler, GT_1 – high pressure turbine, GT_2 – low pressure turbine

Figure 15. Air flow diagrams for superchargers with two-stage compression, intermediate cooling, and aftercooling

In both designs, due to the intermediate air cooling between the compression stages:

- for the same global compression ratio, the mechanical work for compression is smaller compared to the single compressor design;
- for the same mechanical work for compression it is possible to increase the compression ratio.

The design with two heat exchangers for air' cooling (Figure 15.) increases the amount of heat that must be extracted from air at intercooling and aftercooling. Part of this heat can be recovered, and the remaining will contribute to the global RICE's heat losses, reducing the global efficiency in cogeneration.

These aspects will be analysed in a future paper.

5. Conclusion

This paper started by presenting a qualitative diagram of the energy fluxes of RICE. This enabled a qualitative comparison of the fluxes and performance indicators for various types of RICE, with or without turbocharging, respectively for power only or cogeneration. RICE are adapted for cogeneration by recovering heat losses, consequently, the overall efficiencies in cogeneration in these cases depend on the size of heat losses and on their recovery factors. The paper put into evidence the supplementary flows at cogeneration Tb & A type RICE compared to NA type RICE. This paper also presented a case of educational use of the Sankey diagram for turbocharging RICE working in cogeneration.

The paper continues with a statistical analysis, starting from manufacturers' data for Tb&A type RICE used in the energy field, burning natural gas, and having electrical outputs between 100 kW and 10 MW. The graphs presented in the paper can be particularly useful for the managerial staff when choosing a RICE solution for urban cogeneration power plants.

In the 2nd section of the paper, the authors developed a computational methodology that allows determining:

- the amounts of heat per 1 kg air which must be extracted for cooling the air before the aspiration of Tb & A type RICE, for different compression ratios and air temperatures, in known climate conditions, and their average annual values;
- the quotas of recoverable heat, for residential cogeneration, and of non-recoverable heat, from the amount of heat that must be extracted through after cooling.

The method was applied for Romanian weather condition. The results show that, for $p_2/p_1=2$, the yearly average value of total heat that should be extracted at aftercooler is 38.5 kJ/kg $_{air}$.

From this amount, 63.4 % (24.4 kJ/kg) is non-recoverable, while 36.6 % (14.1 kJ/kg) can be recovered for residential cogeneration. This leads to a gain in overall efficiency of about 1%, compared to a Tb & A type RICE without heat recovery at the aftercooler.

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