SUPERCRITICAL CARBON DIOXIDE CYCLES THERMODYNAMIC ANALYSIS AND COMPARISON

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Abstract

A thermodynamic analysis and comparison of supercritical carbon dioxide cycles have been performed. Analyzed cycles were: simple brayton cycle, pre-compression cycle, recompression cycle, split expansion cycle, partial cooling cycle and partial cooling cycle with improved regeneration. A computer code was developed for each cycle to evaluate all thermodynamic states. Compressor inlet and turbine inlet temperatures were hold constant (32°C and 550°C) and other parameters such as compressor outlet pressure, turbine pressure ratio were varying. From analyses among others has emerged: The pre-compression cycle achieves thermal efficiency ~44% at compressor outlet pressure 10MPa. The re-compression cycle achieves the highest efficiency for an optimal pressure ratio (~45% for 20MPa), which corresponds to compressor inlet pressure about 7.7MPa. With a change of this ratio, the efficiency significantly decreases. The split expansion cycle behaves as the recompression cycle but reaches lower thermal efficiency due to pressure reduction in the reactor. The partial cooling cycle reaches similar efficiency such as re-compression cycle, however its decrease with change of turbine pressure ratio is less significant. The partial cooling cycle with improved regeneration has a potential for proper condition reach high thermal efficiency, but mostly suffers from pinch points within recuperators. From the cycles comparison, has emerged: For compressor outlet pressures up to 20MPa seems to be the most effective improvement to simple brayton cycle an addition of pre-compressor, while for higher pressure gives better results the re-compressor with flow dividing. Via splitting the expansion, can be sufficiently reduced pressure in the reactor, without significant reduction of cycle thermal efficiency.

Key words:
S-CO_2, thermodynamic, power cycle, thermodynamic analysis

1 Introduction

1.1 Why supercritical carbon dioxide?

The supercritical power cycles are taking advantage of real gas behaving in order to achieve high thermal efficiency. There are two main types of supercritical cycles. The supercritical water cycle where a heat addition at super-critical pressures increases the turbine inlet temperature and the supercritical CO_2 (S-CO_2), where main improvement of cycle efficiency comes from compressor work reduction due to a properties change, when it is compressed near the critical point (30.98°C, 7.38MPa). Because low critical temperature, it is possible to use water at ambient temperatures such as a coolant.

Other benefits of CO_2:

- S-CO_2 cycles achieve high efficiency at low temperatures
- High operating pressure allows small size components
- More than twenty years experiences of CO_2 application in nuclear reactors (MAGNOX, AGR)

- Well known thermodynamic properties
- Stability
- Non-toxicity
- Abundance
- Low molecular leak due to higher molecular mass
- Low cost

1.2 Real Gas Behavior

Compression near the critical point requires less compressor input work. It is caused by a rapid increase of CO2 density at pressure slightly above the critical pressure (pseudo-critical pressure). The change of compressor work can be seen on ***Figure 1.1*** where is drawn the work needed to compress the fluid from various inlet pressure to compressor outlet pressure 20MPa. For a comparison on the same figure is drawn turbine work produced by expansion for the same pressures (pressure losses are neglected). It shows that the turbine is acting such as for an ideal gas and the highest net work is achieved for the inlet pressure about 7.7MPa.

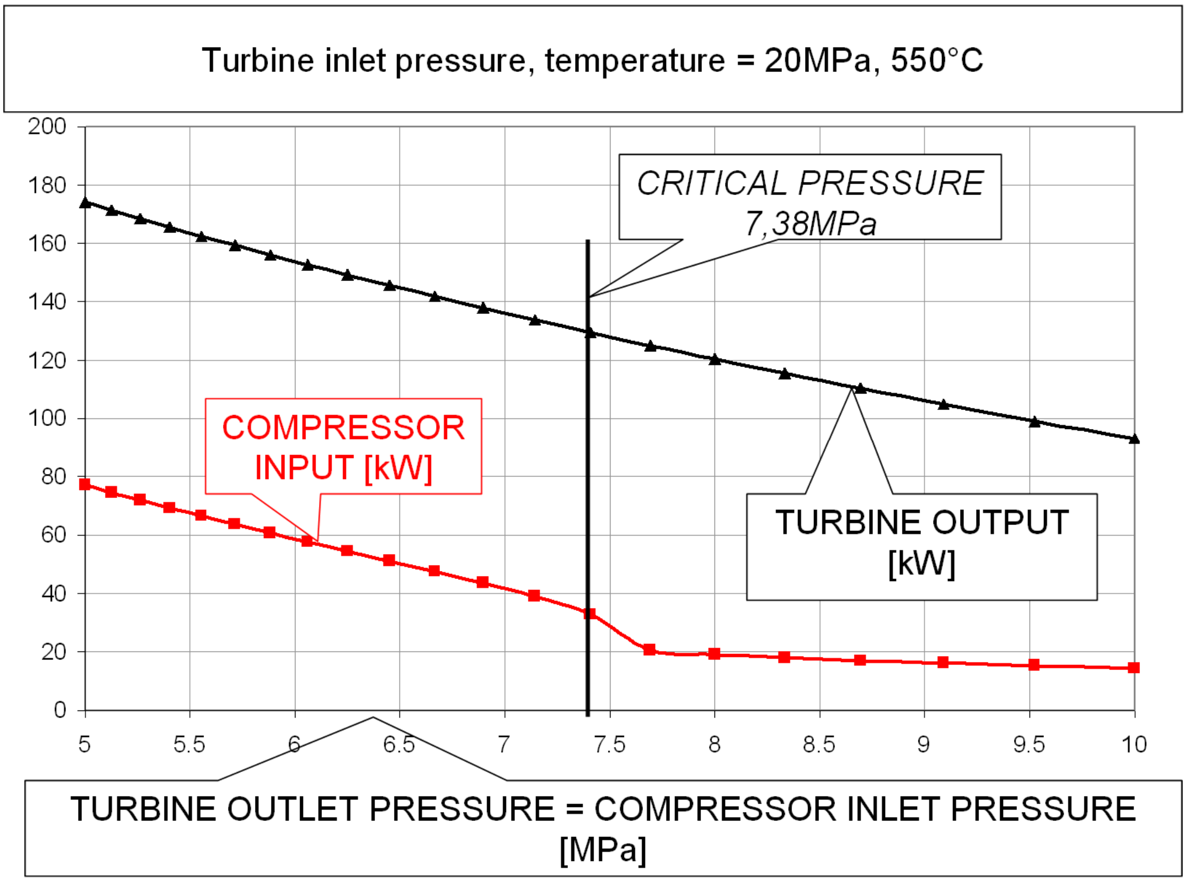

***Figure 1.1** Compressor and turbine work*

Another phenomenon of CO2 is a strong dependency of heat capacity on pressure and temperature (see ***Figure* 1.2**), therefore in certain condition a pinch point can occurs somewhere within the heat exchanger not just on the cold or hot end. Heat capacity plays an important role in design of heat exchangers. As "pinch point problem" is called a place where the minimum temperature difference is not satisfied (see ***Figure 1.3***). In order to reach high cycle thermal efficiency detailed analysis of heat exchanger is necessary to prevent the pinch point problem.

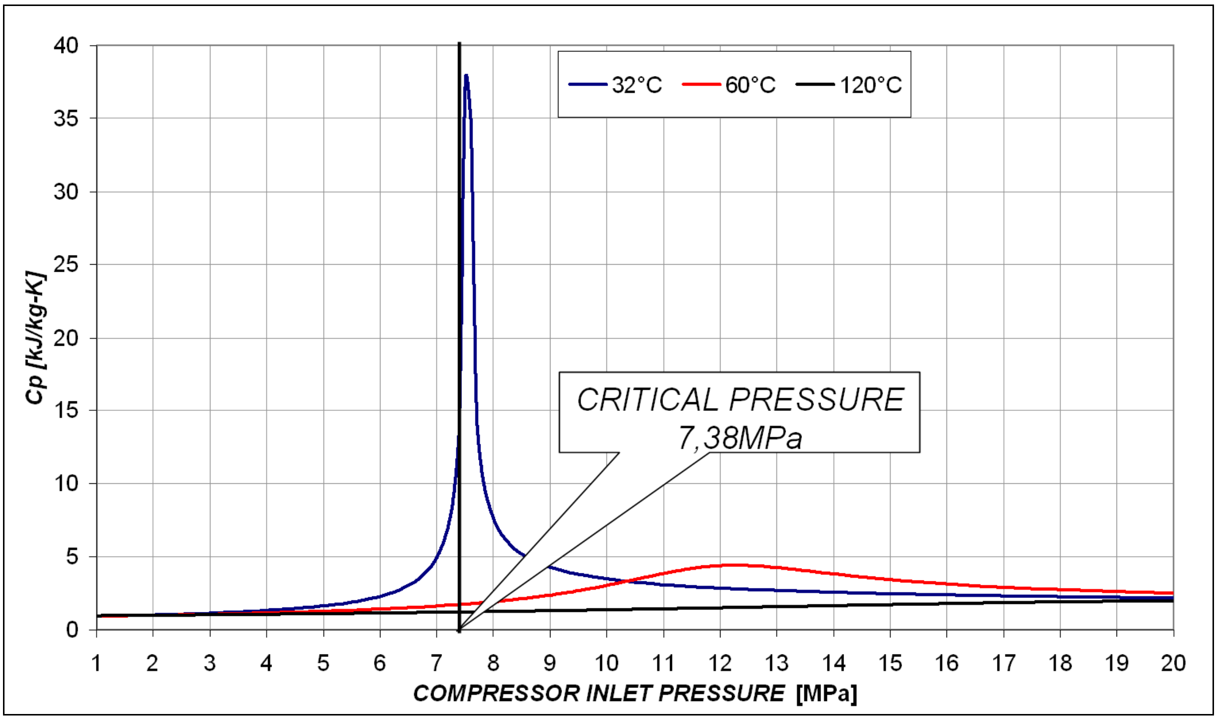

Figure 1.2 *CO2 heat capacity vs. pressure*

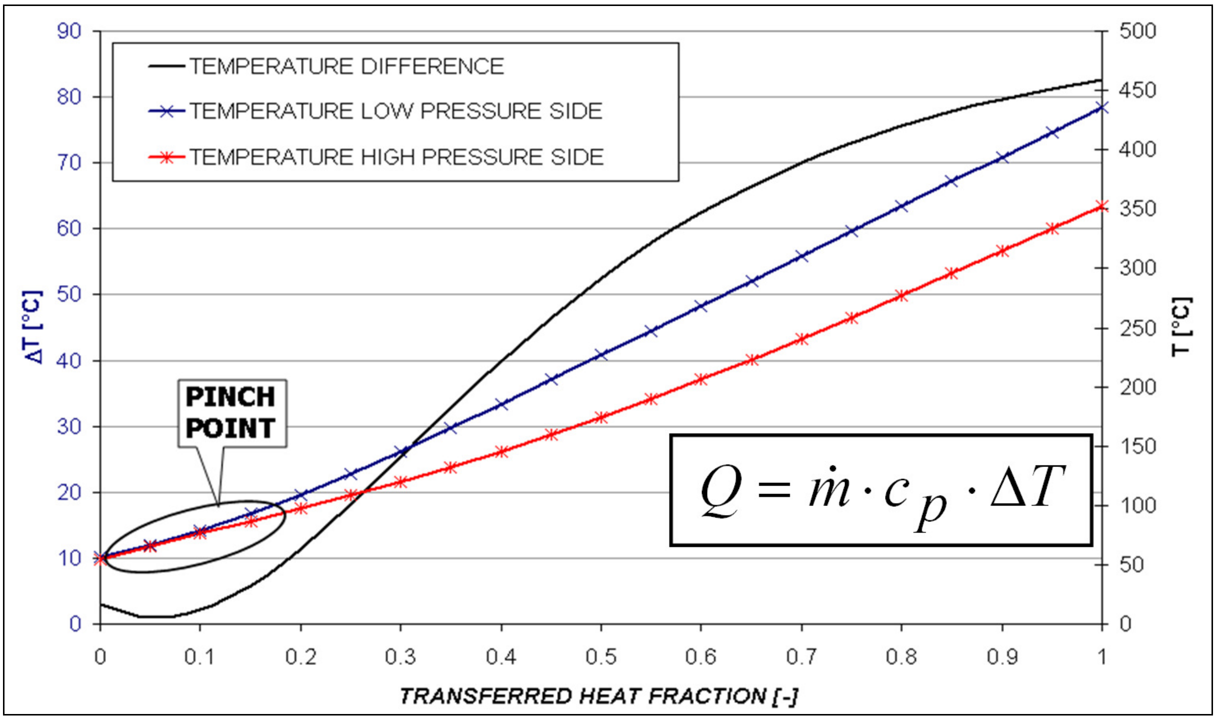

Figure 1.3 *Recuperator pinch point*

1.3 Thermodynamic Cycles

During more than fifty years of CO2 power cycles history were designed several cycle layouts by different authors. Some of them proposed condensation cycles but they left out a problem to cool the CO2 sufficiently and enough to avoid cavitations in a pump. But in fact condensation cycles are also applicable in supercritical region i.e. without condensation. In this work, all cycles are considered to work purely super-critically and the condensation cycles will be termed cooling cycles. Scientists had to make a compromise between high thermal efficiency, material limits and a pinch point problem.

A thermodynamic cycle with the highest efficiency is Carnot cycle.

$$\eta_C = 1 - \frac{T_c}{T_h} \tag{1}$$

In this ideal cycle compression and expansion are reversible and adiabatic and heat addition or rejection is isothermal. In real cycles all the attempts are concentrated to come closer to this condition. First improvement is usually to introduce heat regeneration. It lowers the heat addition as well as heat rejection according to formula for thermal efficiency:

$$\eta_{th} = 1 - \frac{Q_{out}}{Q_{in}} = \frac{W_T - W_C}{Q_{in}} \tag{2}$$

Methods such as split expansion with reheating or split compression with inter-cooling can increase thermal efficiency as well as net work but the cycle will became very complex and it has a negative effect on manufacturing, control, operating, maintenance, cost etc. An increase of turbine inlet temperature and pressure can be also applied but there are material limits. All methods may be combined in an effort to reach optimum solution but one has to aware of complex and uneconomic proposal.

The analyzed cycles are:

- Simple Brayton cycle
- Pre-compression cycle
- Re-compression cycle
- Split expansion cycle
- Partial cooling cycle
- Partial cooling with improved regeneration

1.4 Simple Brayton cycle

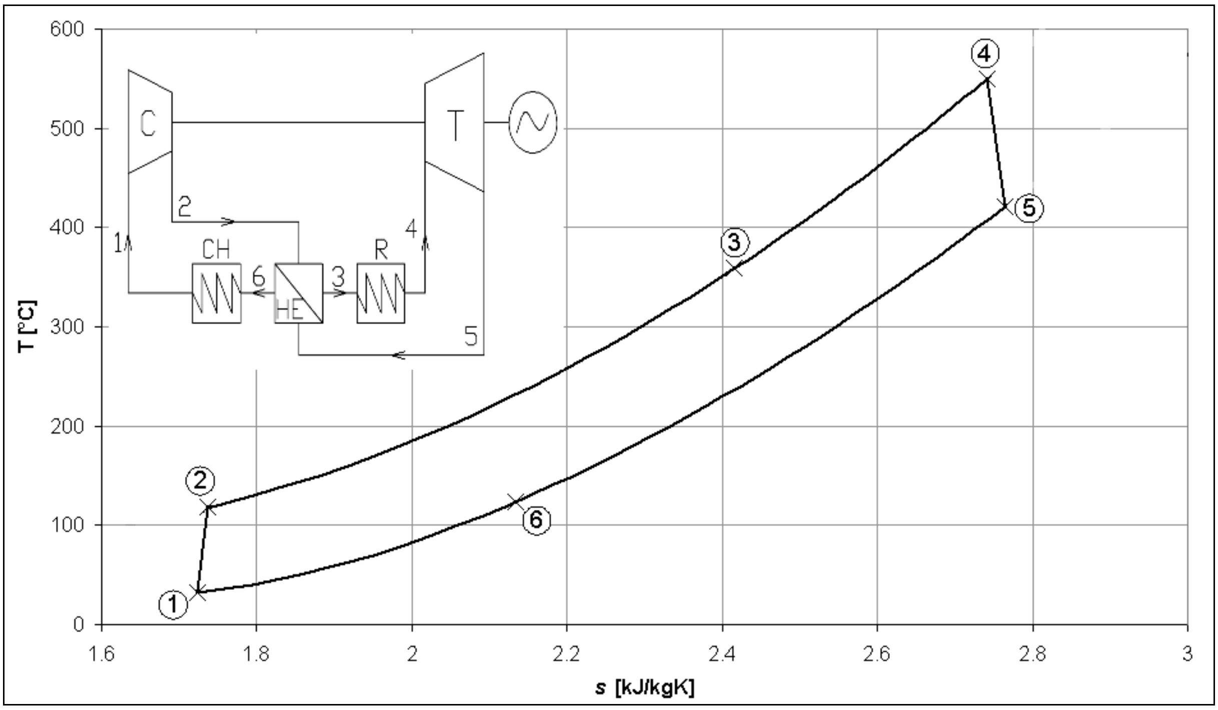

Figure 1.4 *Simple Brayton cycle*

This cycle layout is the backbone of gas cycles. It consists of compressor, turbine, recuperator, heat source (reactor) and chiller. The thermal efficiency is not high; however a comparison various cycles with that of this simple cycle will show clearly a contribution of each arrangement to the cycle thermal efficiency.

1.5 Pre-compression cycle

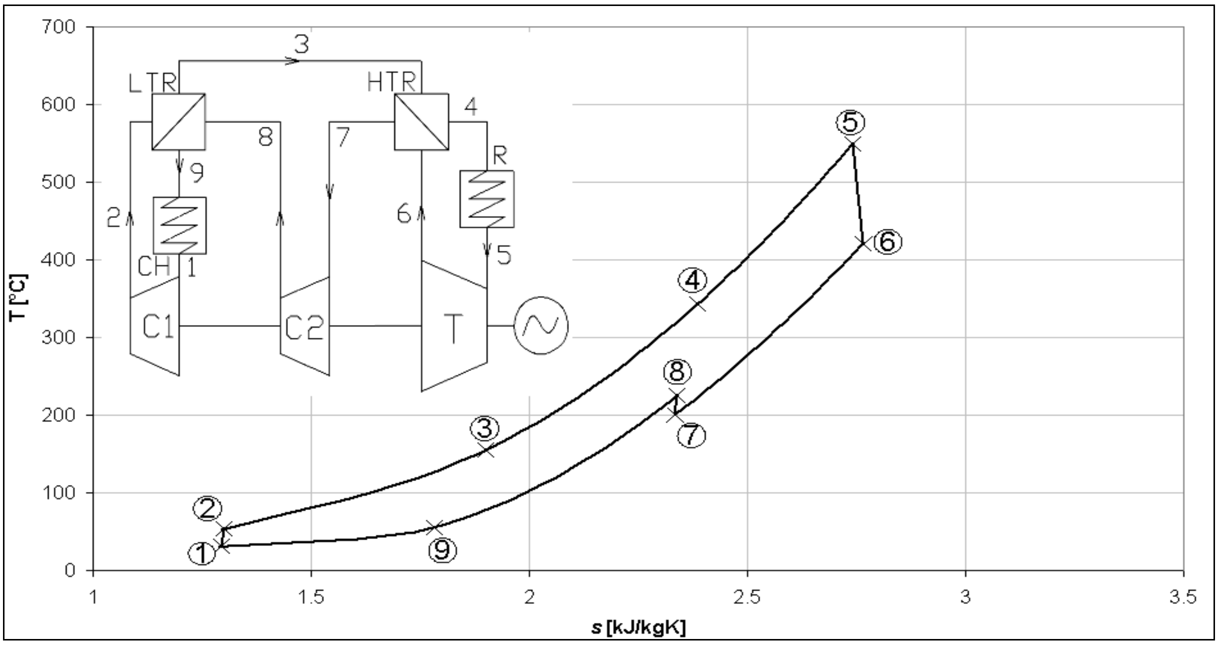

Figure 1.5 *Pre-compression cycle*

This cycle layout improves the Brayton cycle by introducing a pre-compressor between turbine and main compressor in order to make turbine exhaust pressure independent on compressor inlet pressure. Heat exchanger has been split into two to avoid pinch point problem. Basic principle is: when the temperature difference comes close to the minimum an addition compression makes a place for further regeneration.

1.6 Re-compression cycle

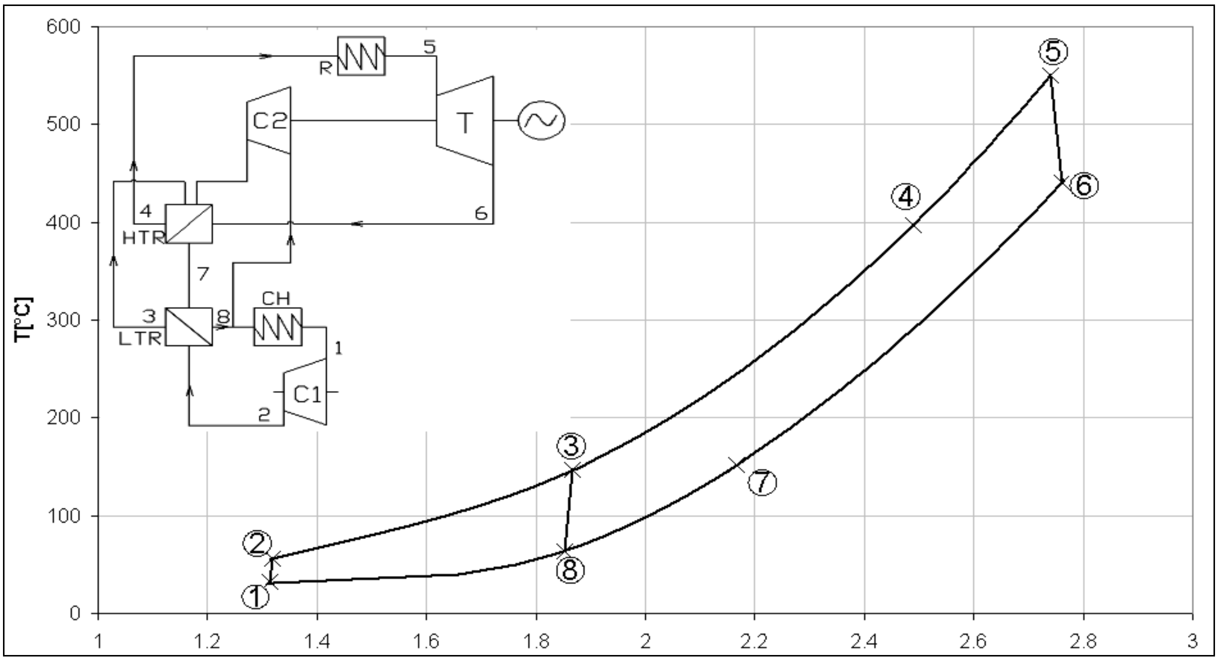

Figure 1.6 *Re-compression cycle*

Recompression cycle has the same number of component such pre-compression cycle but arrangement is different. Before cooling the flow is split into two streams and one goes to recompression compressor since the other one goes through chiller to the main compressor. The flow, which pass the low temperature recuperator should has same pressure and temperature as another one and both streams are mixed again into one stream. The pinch point problem is prevented due to lower mass flow at high pressure side of the low temperature recuperator, hence the heat capacity mass flow weighted on both sides are equal.
This system rejects less heat and because re-compressor input work is lower than saved heat thermal efficiency is improved.

1.7 Split-expansion cycle

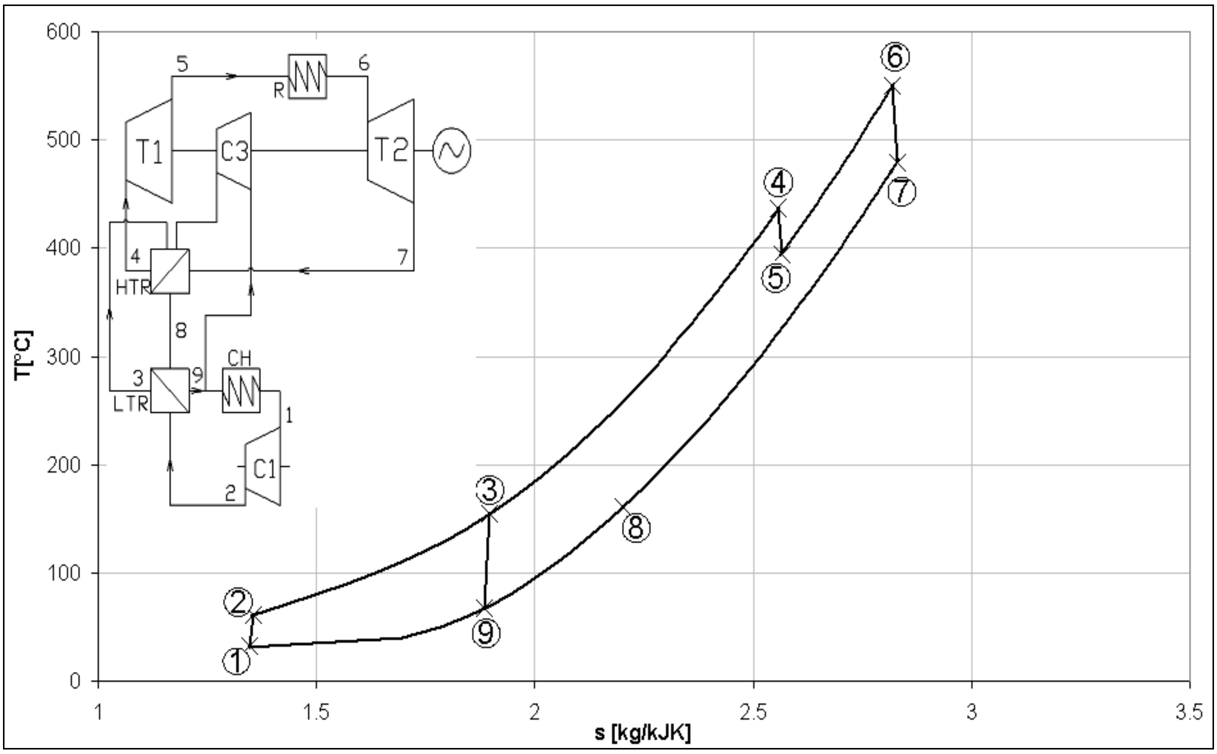

Figure 1.7 *Split expansion cycle*

This cycle come out from recompression cycle and only difference which can bee seen from T-s diagram is divided expansion. Behind layout of this cycle was an effort to reduce a stress in the hottest component of the system. One additional turbine is introduced to split the expansion. Heat is added after expansion from high pressure in first turbine. Arrangement of other components is same as in recompression cycle.

1.8 Partial cooling cycle

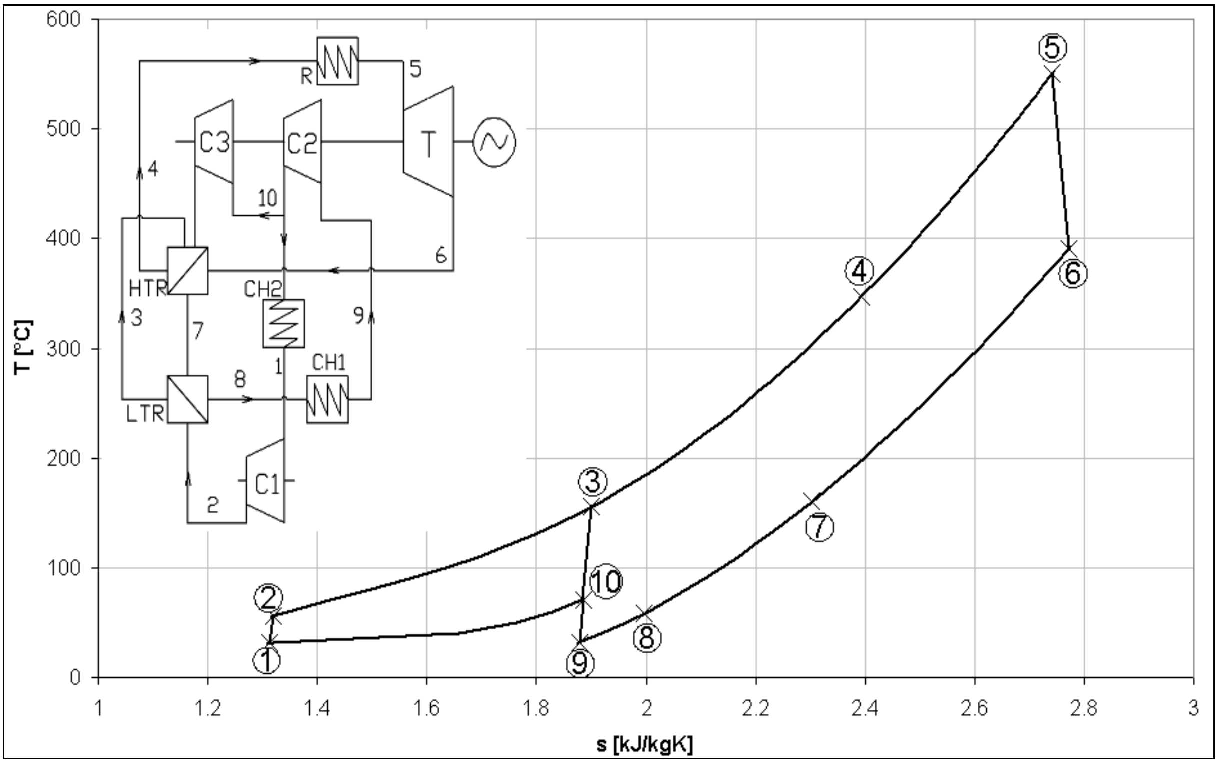

Figure 1.8 *Partial cooling cycle*

The partial cooling cycle combines the pre-compression cycle with the re-compression cycle. This improvement takes advantage of turbine exhaust pressure independency on the compressor inlet pressure; moreover the flow splitting helps to increase the cycle efficiency due to bypassing the chiller as well as to cope with pinch point problem by reducing the flow in high pressure side of the low temperature recuperator.

1.9 Partial cooling with improved regeneration

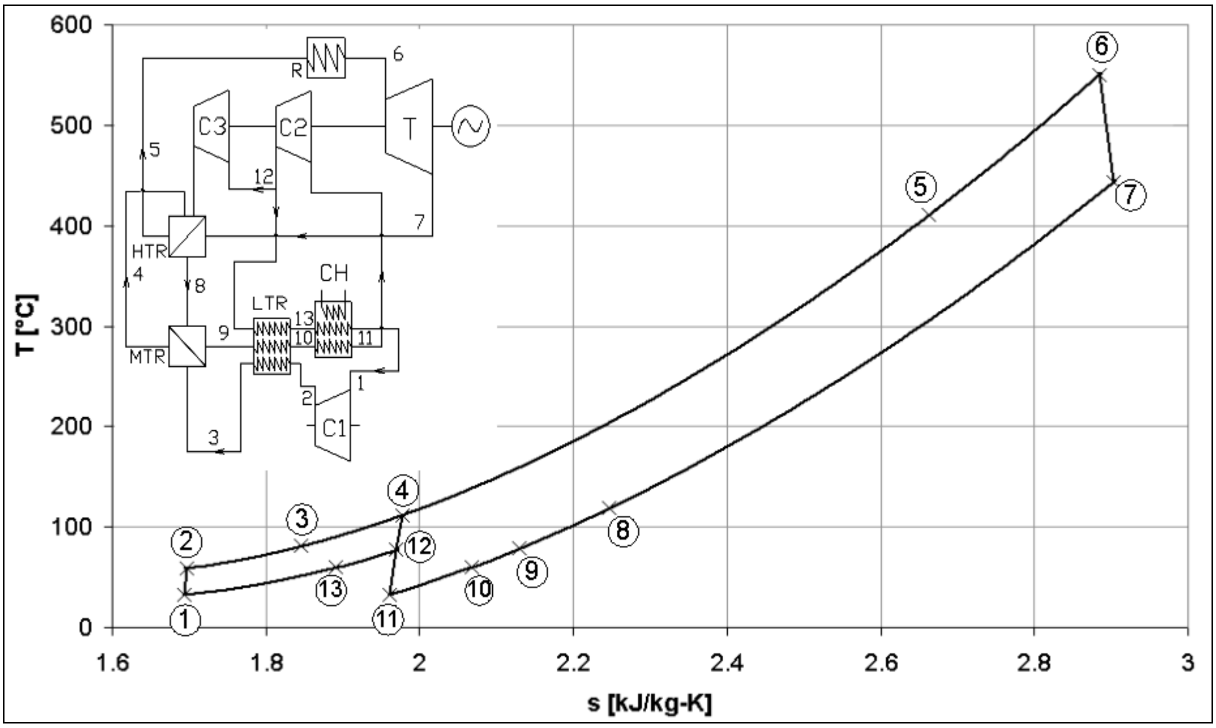

Figure 1.9 *Partial cooling cycle with improved regeneration*

For particular cases when the pre-compressor outlet temperature is above the main compressor outlet temperature there is an available heat to regenerate and the partial cooling cycle efficiency can be further improved. By introducing a third recuperator with three streams can be this heat regenerated as well. However three streams recuperator can be substituted by two common recuperators parallel connected.

1.10 Summary

Five cycles were selected for analysis plus simple Brayton cycle. Two of them are simple arrangement (pre-compression. re-compression) and three are more complex (split expansion, partial cooling and improved regeneration). A method of preventing recuperator against pinch point problem is flow dividing at all cases except for the pre-compression cycle where additional compression is applied.

1.11 Expected Results

An Italian scientist Angelino has made great research work in the area of CO2 thermal cycles (all of the analyzed cycles were proposed by him) and had drawn a comparison of some cycles from point of losses to the ideal Carnot cycle. He assumed turbine inlet temperature 700°C and select different compressor inlet pressures for each cycle (re-compression 75atm, partial cooling 20atm (i.e. with condensation), and partial cooling with improved regeneration 20atm) and a turbine inlet pressure was varied. On his figure we can roughly see what to expect from the analysis. The re-compression cycle has the lowest losses from the Carnot since the turbine inlet pressure is above 200atm the cycle thermal efficiency grows with increasing pressure. Partial cooling cycle has similar but not so steep trend otherwise the losses are higher. Partial cooling with improved regeneration cycle seems to be independent of turbine inlet pressure and its thermal efficiency is till 250atm the highest at higher pressures is overcame by the re-compression cycle.

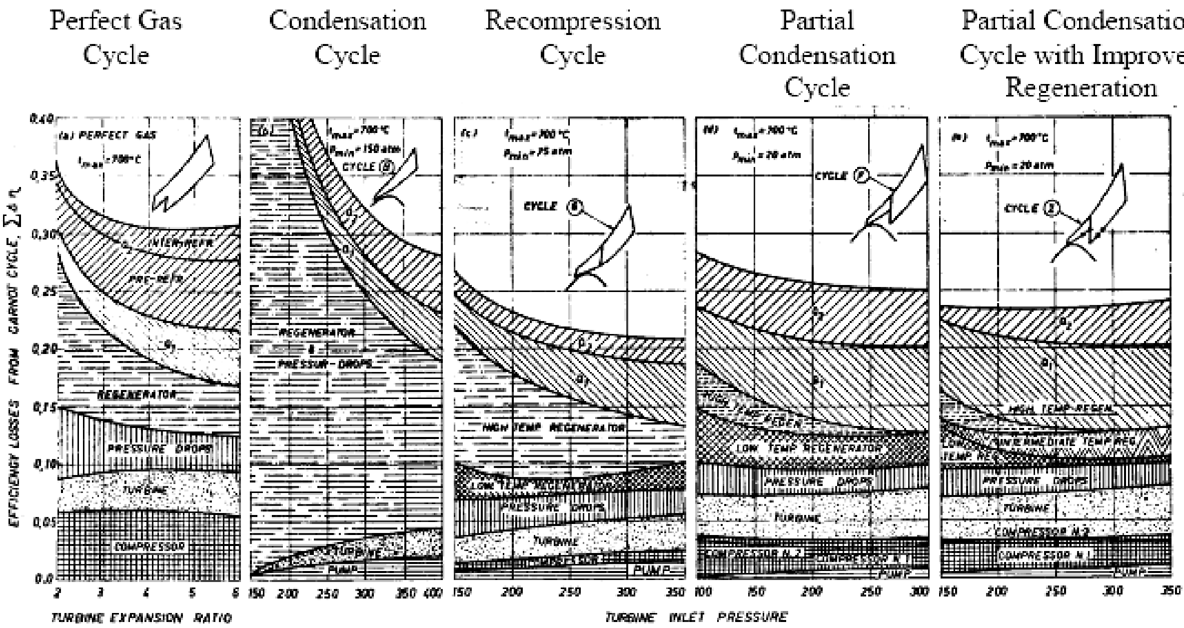

__Figure 1.10__ Angelino's comparison

2 Analysis

Because the compressor inlet pressure and temperature are fixed for reason to minimize the compressor work as was explained in the introduction chapter, an analysis of certain parameters influence on cycle efficiency was performed.
The parameters correspond to relevant cycle:

- turbine pressure ratio *rT* (all cycles)
- turbine inlet pressure (all cycles)
- turbine inlet temperature (all cycles)
- ratio of pressure ratio *rpr* (pre-compression, re-compression, partial cooling)
- dividing pressure (split expansion)
- pre-compressor inlet temperature (pre-compression)
- recuperator efficiency (improved regeneration)

In this basic design others parameters were hold constant:

- compressor efficiency 89%
- turbine efficiency 90%
- recuperator efficiency 95%
- turbine inlet temperature 550°C
- compressor inlet temperature 32°C
- pressure losses are neglected

A mathematical model of each cycle was worked out for purpose of analysis. The calculation itself was made in code of VISUAL FORTRAN6.5 with usage NIST subroutines for evaluating of CO2 thermodynamic properties. Computed data were elaborate and plotted in MS EXCEL.

2.1 The Cycles Comparison

From results of each cycle analysis were selected for comparison those with high thermal efficiency and wide range of pressure ratios. An effect of varying compressor outlet pressure and pressure ratio was observed. The results from analysis for particular pressure 10, 15, 20, 25MPa of each cycle are plotted into a graph and correspondent comment is below the graph.

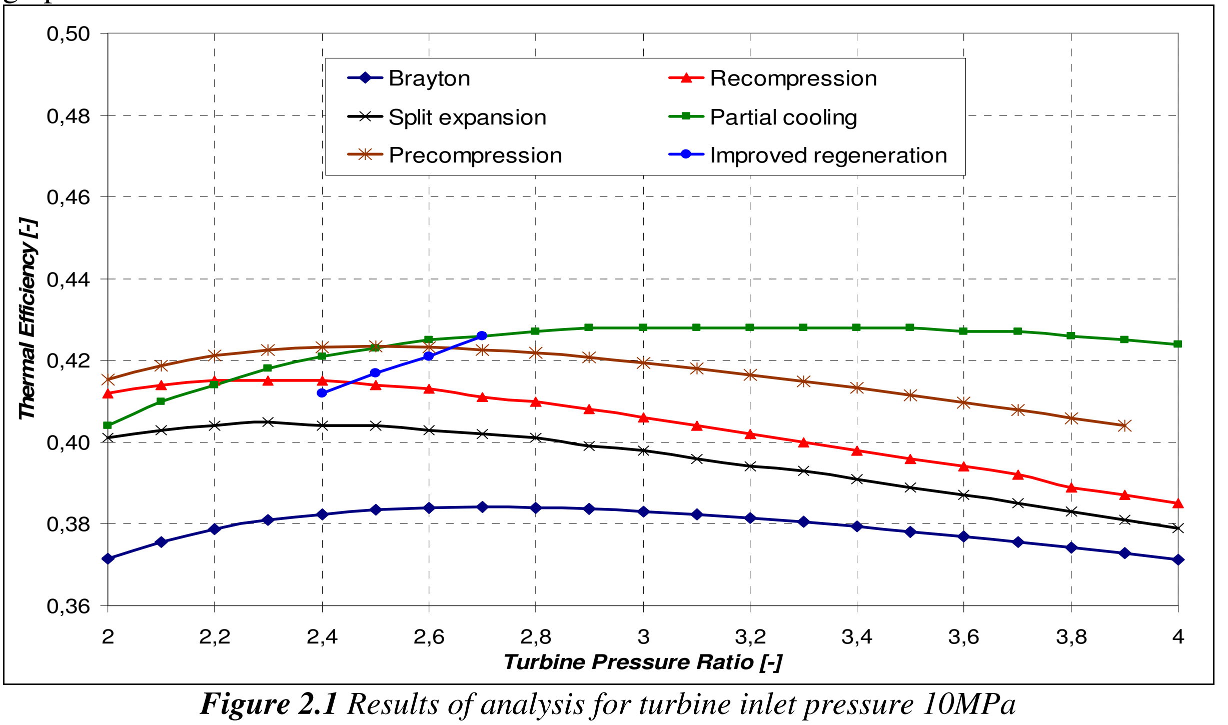

Figure 2.1 *Results of analysis for turbine inlet pressure 10MPa*

The partial cooling cycle achieves the highest thermal efficiency 43% for turbine pressure ratio 3.0; moreover, when the turbine pressure ratio exceeds 2.7 further increasing of the ratio has only slight effect on the efficiency. Interesting is pre-compressor improvement – about four percentage point to the simple Brayton cycle. Improved regeneration has very limited turbine pressure ratio operating range and the thermal efficiency characteristic does not exceed the others. Recompression and split expansion cycles have identical trend as due to their similarity, but achieved thermal efficiency falls behind the others.

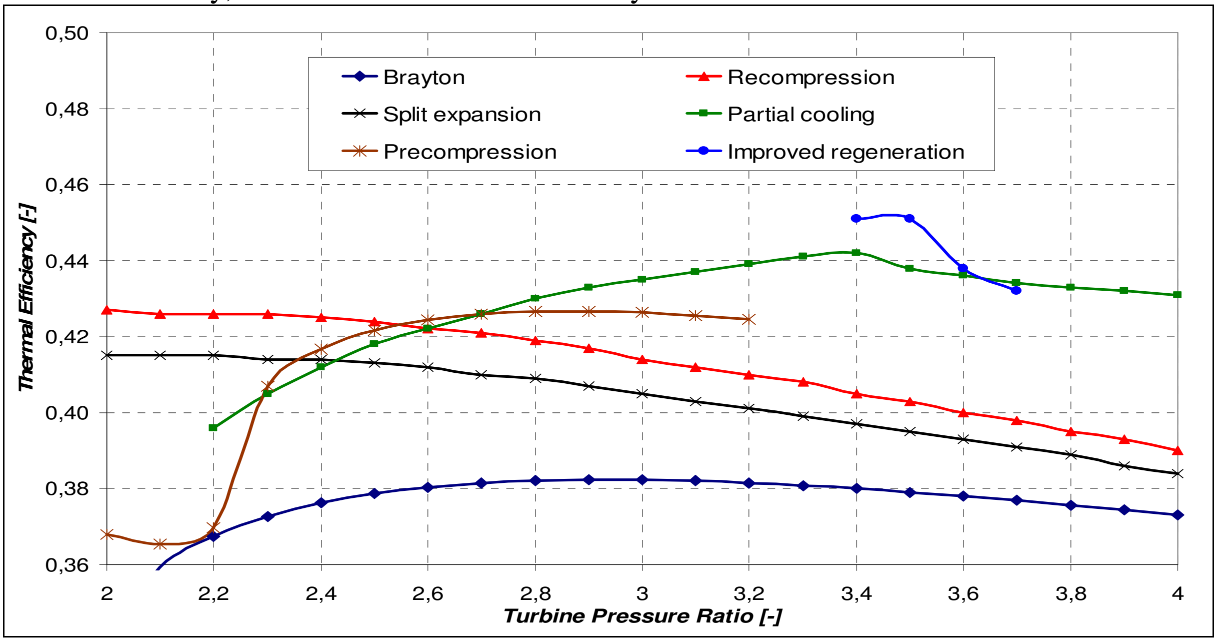

Figure 2.2 *Results of analysis for turbine inlet pressure 15MPa*

Improved regeneration exceeds the partial cooling only for special condition – in this case for turbine pressure ratio 3.3 and 3.4, when the value of thermal efficiency is more than 45%. For turbine pressure ratio between 2.0 and 2.5 the re-compression cycle performs the best since pre-compression achieves similar thermal efficiency for the ratio from 2.6 to 3.2.

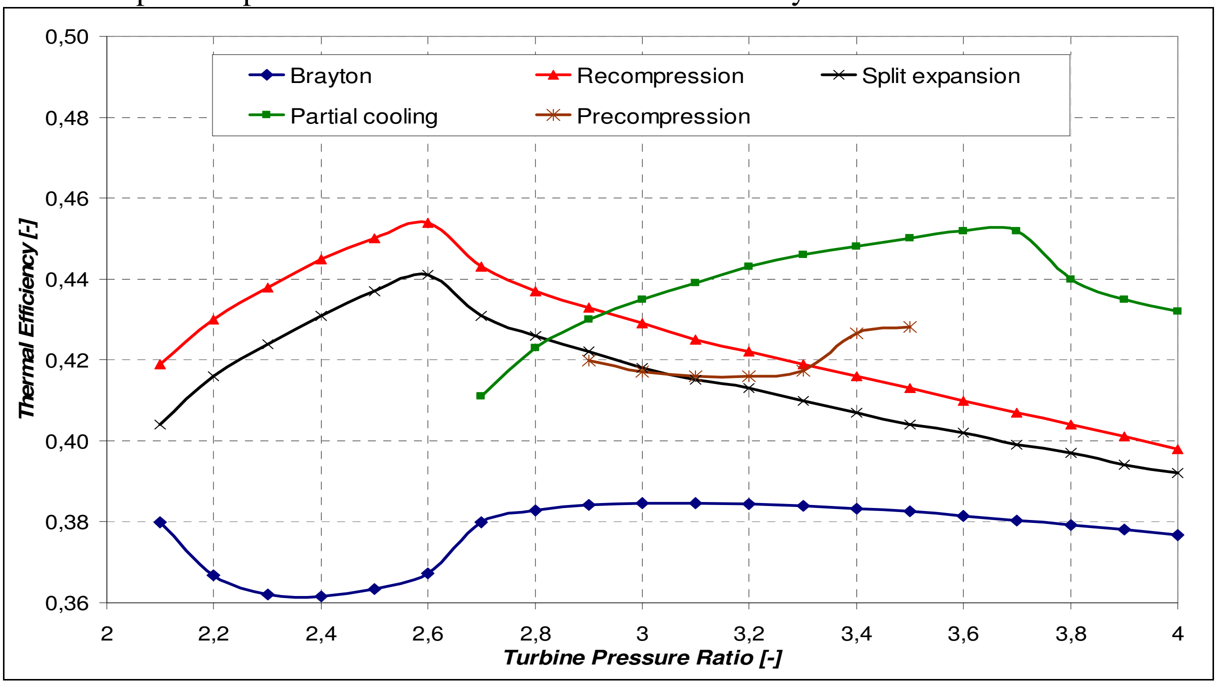

Figure 2.3 *Results of analysis for turbine inlet pressure 20MPa*

This figure shows interesting behaving of the partial cooling, the cycle has similar trend (but not so steep) as the re-compression placed at higher turbine pressure ratios. Both cycles characteristics maximum is almost equal. The pre-compression cycle trend is similar to the simple Brayton cycle, likewise partial cooling is placed at higher turbine pressure ratios.

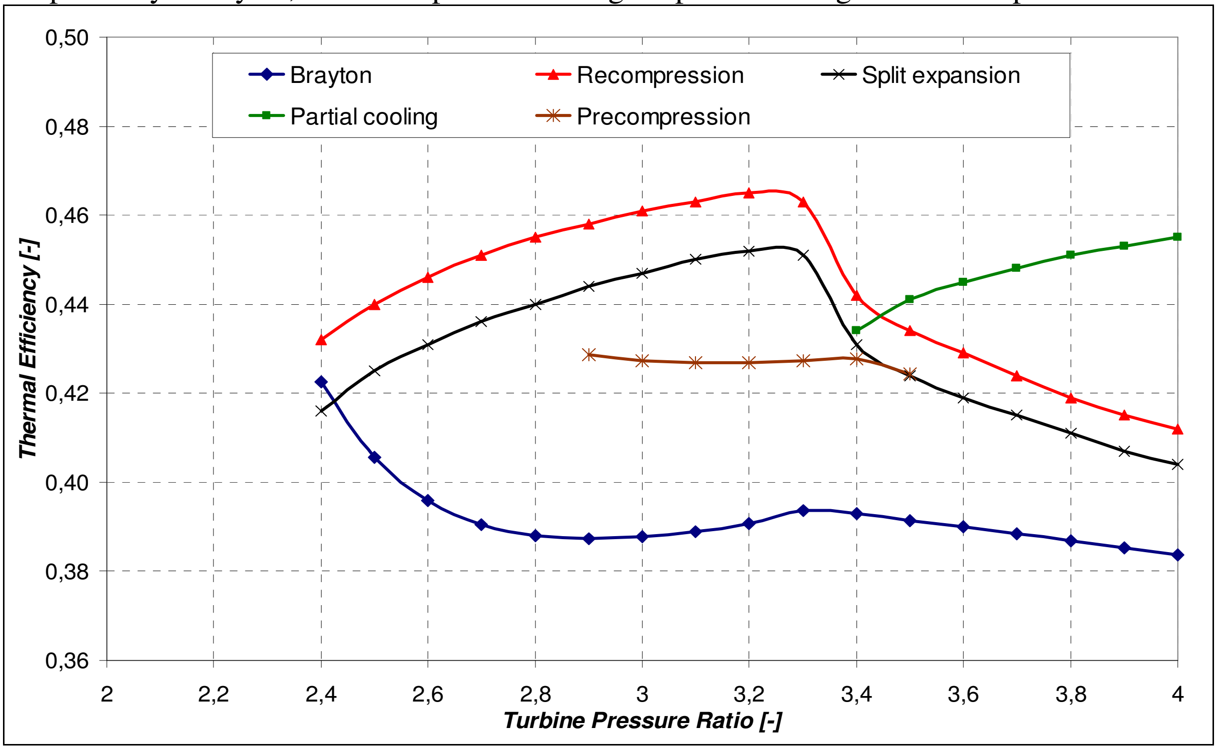

Figure 2.4 *Results of analysis for turbine inlet pressure 25MPa*

The re-compression thermal efficiency trend has moved, together with split expansion and partial cooling, to higher turbine pressures ratio and became less steep; furthermore higher value of thermal efficiency is achieved. The pre-compression thermal efficiency has little increase and the characteristic trend is rather flatter.

3 Conclusion

From the cycles analysis, has emerged:

- Operating close to the critical point is not improving much the simple brayton cycle thermal efficiency. Only for compressor outlet pressure about 25MPa is visible an increase but it falls rapidly with some change in the pressure ratio.
- The pre-compression cycle can reach high efficiency for compressor outlet pressure 10MPa (see Figure III.2.2), which mainly depends on the precompressor inlet temperature and ratio of pressures ratio.
- The re-compression cycle achieves the highest efficiency for optimal pressure ratio, which corresponds to compressor inlet pressure about 7.7MPa. With a change of this ratio, the efficiency significantly decreases.
- The split expansion cycle behaves as the re-compression cycle but reaches lower thermal efficiency due to pressure reduction in the reactor.
- Partial cooling cycle reaches higher efficiency for high ratio of pressures ratio, likewise pre-compressor cycle. For increasing compressor outlet pressure are the characteristics moving to higher turbine pressure ratio.
- The benefits of partial cooling cycle with improved regeneration higher thermal efficiency, pale in overall cycle complexity, which causes pinch point problems within recuperators.

From the cycles comparison, has emerged:

- The benefit from higher thermal efficiency of partial cooling cycle with improved regeneration pales in overall cycle complexity, which causes pinch point problems within recuperators.
- Splitting the expansion, can be sufficiently reduced pressure in the reactor, without significant reduction of cycle thermal efficiency.

It is difficult to judge, which cycle is generally the best. For instance: When an application at compressor outlet pressures about 20MPa is considered, from one point of view the re-compression cycle is the best due to respect of simplicity and high thermal efficiency. While, from point of load control the partial cooling cycle is better, due to less steep characteristic, which means lower change of the cycle thermal efficiency in part load operating. If high thermal efficiency is in the focus, the partial cooling cycle with improved regeneration is the best, but only for compressor outlet pressure 10 to 15MPa, and proper combination of recuperators efficiencies. In the future research, more detailed analysis; which would consider dynamic behaving of the system, technical economic aspects, performance controls and material requirements should be performed.

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