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# Hydrogen utilization by steam turbine cycles

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**Abstract**

Based on thermodynamic analysis, the paper presents the utilization of hydrogen in steam turbine cycles. Various configurations (GRAZ, TOSHIBA, WESTINGHOUSE and MNRC) proposed in literature are recalculated using the same software and the same thermodynamic functions, thus comparisons can be made. It is possible to achieve efficiency levels of 60% (HHV based) which is at least 10 percent points higher than the efficiency of the most efficient current power units. The investigated systems are characterized by very high specific power (2,200..4,700 kJ/kg), which is much higher (in extreme cases, by an order of magnitude) than the performance of current gas or steam turbines or combined cycles.

## 1. Introduction

Power generation systems based on hydrogen could be an important alternative to conventional power systems based on the combustion of fossil fuels. The main effort in the field is oriented to the use of hydrogen in fuel cells [1–4] or their combination with gas turbines. Hence, hydrogen as a clean fuel for end-users has attracted the interest of many research institutions all over the world. National and international projects for hydrogen utilization have been developed in several countries. One of the most important programs was in Japan: the International Clean Energy Network Using Hydrogen Conversion—WE-Net [5]. The WE-Net Program predicts the implementation of the Hydrogen-Fuelled Combustion Turbine Cycle (HFCTC) as a new energy source for the power sector. To this end, a configuration and performance study of the HFCTC was conducted. The research was performed in co-operation with the Institute of Energy Utilization, AIST—Tsukuba, Japan. Further on in this paper, the results obtained in respect of hydrogen turbine cycle thermodynamics are presented.

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Thermodynamic analysis of the efficiency of power units that use hydrogen as a fuel as well as basic results of high-pressure $H_2/O_2$-steam-generators experimental modeling is presented in [6]. It is shown that at capacities of power units more than 10 MW, steam-turbine hydrogen units are preferable to fuel cell power units. Experimental $H_2/O_2$-steam generators, with capacity of up to 25 MW(t), were created and the peculiarities of their use in steam-turbine hydrogen power units were experimentally studied.

In light of concerns over $CO_2$ emissions [7–10], in [**?** ] proposed a novel integrated gasification combined cycle (IGCC) system with a steam injected $H_2/O_2$ cycle and $CO_2$ recovery. A new evaluation criterion for comprehensive performance of the IGCC system was also presented. The results show the new system has a lower energy penalty for separating and recovering $CO_2$, at an efficiency decrease of less than 1 percentage point.

In [11] two options for $H_2$ production through fossil fuels are presented and their performances are evaluated when integrated with $H_2/O_2$ cycles. The investigation was conducted with reference to two different schemes, both representative of advanced technology (TIT = 1,350°C) and of futuristic technology (TIT = 1,700°C).

The high cost of hydrogen production [12–15] makes thermal efficiency of over 60% HHV ( i.e. about 71% LHV) of the energy generation cycle a must in the WE-Net system. Nowadays, thermal efficiency of the most advanced combined cycles (natural gas fueled) is close to 60% LHV [16] and comparable to 50.4% HHV if hydrogen is used as a fuel.

It is evident that efficiency should be increased by about 10 percent points, which is the equivalent of about 20% higher than the most efficient contemporary power plant units. Meeting this requirement poses a very serious technological challenge, as it is both a qualitative and a quantitative change. Moreover, when increasing the working medium temperature at the turbine inlet to 1,700°C (currently, c. 1,600°C) it is essential to implement a new, non-traditional approach to both the conceptual design of the system (configuration and working parameters) and detailed construction solutions.

Several concepts of the HFCTC have been proposed to date, of which the most important are the following:

1. Combined Steam Cycle with Steam Recirculation prepared by prof. H. Jericha (Technical University of Graz) by convention called the GRAZ cycle [17–20];
2. Direct-Fired Rankine Steam Cycle (New Rankine Cycle), which was studied in the following variants:
   (a) proposed by Toshiba Co., by convention called the TOSHIBA cycle [19];
   (b) proposed by Westinghouse Electric Co., by convention called the WESTINGHOUSE cycle [21];
   (c) Modified New Rankine Cycle, the authors' own concept, by convention called the MNRC cycle [22].

A common feature of the a.m. cycles is that only one working medium (steam) is used for both the topping and bottoming cycle. This is made possible by replacing external firing (as in the Rankine steam cycle [23]) by direct firing (similar to gas turbines or piston engines). The main assumption made here is stoichiometric combustion of a hydrogen and oxygen mixture. The combustion takes place inside a stream of cooling steam, which reduces the combustion temperature to 1,700°C. It is also assumed that hydrogen and oxygen at the ambient temperature are available at a pressure level that allows them to be supplied to the combustor. This means that hydrogen would be provided as a cryogenic liquid and that cryogenic energy could be utilized for pure oxygen production in an air-separator unit.

Table 1: Nominal conditions common to all analyzed cycles

| Parameter | Value |
|---|---|
| compressor stages group internal efficiency, % | 90 |
| turbine stages group internal efficiency, % | 90 |
| combustor efficiency, % | 99 |
| heat exchanger pressure loss, % | 4.3 |
| combustor pressure loss, % | 5 |
| pump efficiency, % | 90 |
| electric generator efficiency, % | 99 |
| cycle overall mechanical efficiency, % | 99 |
| overall power output, MW | 500 |
| temperature after a combustor, °C | 1,700 |
| condenser pressure, MPa | 0.005 |
| condensate temperature,°C | 33 |

The Graz, Toshiba, Westinghouse and MNRC cycles were analyzed in comparable conditions to evaluate their performance. The analysis was undertaken in the same, specific conditions with the same assumptions. The results of research into a.m. cycles published to date do not provide an opportunity for such a comparison due to non-comparable and/or unclear conditions and assumptions used in the research. Cycles without a cooling system were only taken into account in order to create an opportunity for an explicit evaluation.

## 2. Hydrogen fueled steam turbine cycles

Analysis of the HFCTC in nominal conditions was conducted for the following values of parameters as stated in Table 1. Parameters of the working medium were determined using the ITC-PAR calculation routines [24], based on the NIST/ASME steam property tables.

A computer program was created for the purpose of making the cycle performance calculations. Pressure splits between turbine parts were optimized according to the maximum overall HHV thermal efficiency. The efficiency of the analyzed cycles was determined in relation to the Rankine cycle ($\eta_R$), high heating value of fuel (HHV) and low heating value of fuel (LHV) as:

$$\eta_{LHV} = \frac{P}{\dot{m} \cdot LHV} \cdot 100\% \tag{1}$$

where $P$ denotes internal power output of the cycle and $\dot{m}$—hydrogen mass flow rate of hydrogen supplied to the cycle

$$\eta_{HHV} = \frac{P}{\dot{m} \cdot HHV} \cdot 100\% \tag{2}$$

$$\eta_R = \frac{P}{P + \dot{Q}} \cdot 100\% \tag{3}$$

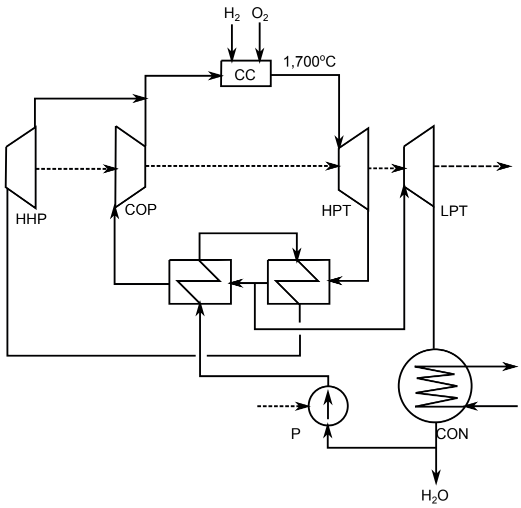


Figure 1: Schematic chart of the Graz cycle

where $\dot{Q}$ denotes the amount of heat which is taken out of the cycle. Efficiencies $\eta_{HHV}$ and $\eta_R$ should be equal (at exact calculations).

In addition, the turbo-set efficiency was determined:

$$\eta_{el} = \eta_{LHV} \cdot \eta_g \cdot \eta_m \tag{4}$$

where $\eta_g$—generator efficiency, and $\eta_m$—overall mechanical efficiency.

Detailed parameters as well as enthalpy–entropy diagrams for all of the presented cycles can be found in the previous work [25]. Here, the figures are for purely information purposes.

### 2.1. Graz cycle

The Graz cycle—proposed by prof. H. Jericha from Technical University of Graz—is an original combination of the Joule and Rankine cycles (see Fig. 1). In the high parameters area the Joule cycle is utilized in a semi-closed configuration, coupled with the Rankine cycle, which operates in the low parameters area. The Rankine cycle plays here simultaneously the role of heat sink for the Joule cycle. The hydrogen combustion chamber is the high temperature source of heat. The original idea which distinguishes the Graz cycle from other cycles, is applying the extraction of the partially cooled working medium from the Joule cycle and using it as a working fluid in the Rankine cycle. An efficiency rise is obtained here due to a significant decrease in the compression work of the working medium in the Joule cycle (steam compressor.)

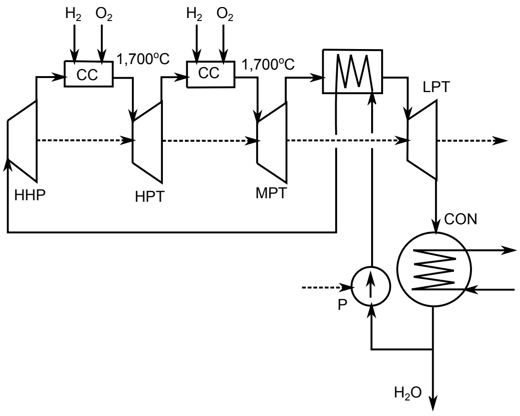


Figure 2: Schematic chart of the TOSHIBA cycle

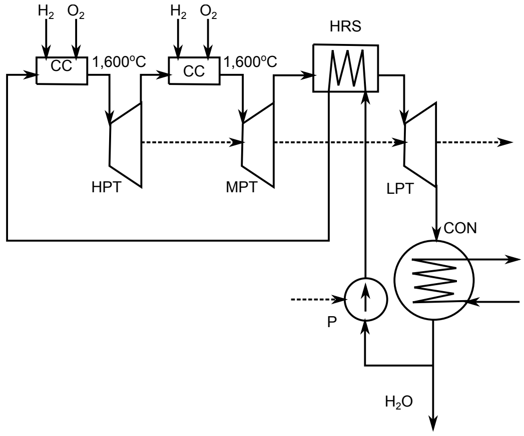


Figure 3: Schematic chart of the Westinghouse cycle

### 2.2. Toshiba cycle

The Toshiba cycle is one of a group of steam cycles with direct combustion—direct hydrogen fired Rankine steam cycles. The Toshiba cycle is also called the MORITS cycle—Modified Rankine Cycle Integrated Turbine System. A schematic chart of the cycle and the calculation results are shown in Fig. 2. It consists of four turbine parts, where two of them—the first and the last one (HHP and LPT)—do not have combustion chambers in front. The regenerator (heat recovery boiler) is located before the last turbine part, where superheated steam is produced.

### 2.3. Westinghouse cycle

The Westinghouse cycle is a variant of the steam cycle with direct combustion of hydrogen with oxygen (for the schematic chart and the calculation results see Fig. 3). Compared to the Toshiba cycle, it does not possess the turbine part used in the highest pressure region. So, it can be classified as a kind of new Rankine cycle with single reheat.

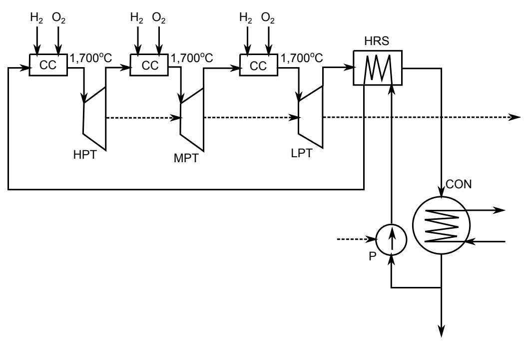


Figure 4: Schematic chart of the MNRC cycle

### 2.4. MNRC cycle

The MNRC cycle is our own concept, proposed after analysis of the above mentioned and published cycles. In contrast to the other cycles, the heat recovery steam generator (HRSG) is placed after the last turbine stage group in the low-pressure zone. This concept was mentioned earlier in the paper [21]. The layout of the cycle allows another reheat stage to be added before the low-pressure turbine stage group. As a result, a very high thermal efficiency of the cycle is achieved. A flow diagram and calculation results of the MNRC cycle are shown in Fig. 4.

### 2.5. Comparative performance of the cycles

The main parameters of the analyzed cycles are shown in Table 2. Comparison and evaluation of these cycles could be done from different points of view. A basic criterion here is maximum overall thermal efficiency, at least 60% HHV for the 500 MW class unit.

As can be seen, the efficiencies of the Graz and Toshiba cycles are practically at the limit of the WE-Net program requirements. The Westinghouse cycle fulfills these requirements with some overlap and the MNRC cycle goes far beyond. It is possible to increase the efficiency of the Graz cycle by adding to the basic configuration a compressor, inter-stage cooling and heat recuperation (high-temperature regeneration) [18, 26].

All cycles discussed here can achieve very high specific power (related to maximum mass-flow in the cycle). Those values are much higher (in extreme cases, by an order of magnitude) than the ones achieved by current heavy-duty gas turbines (450..500 kJ/kg), combined steam-gas cycles (600..700 kJ/kg) or steam turbo-sets (1,200..1,400 kJ/kg). It would be possible then to build extremely compact power units with minimal usage of construction materials.

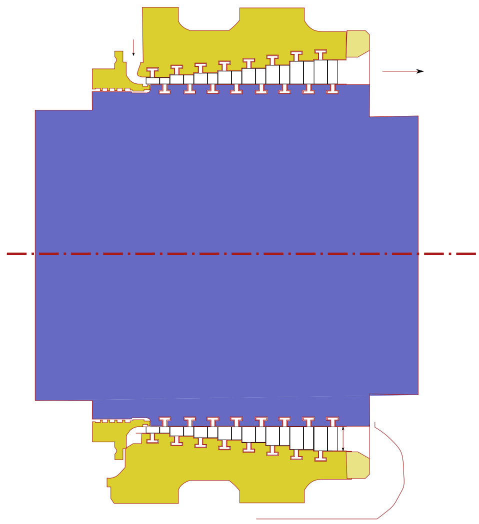

Figure 5: Concept of the HPT of the hydrogen fueled steam turbine

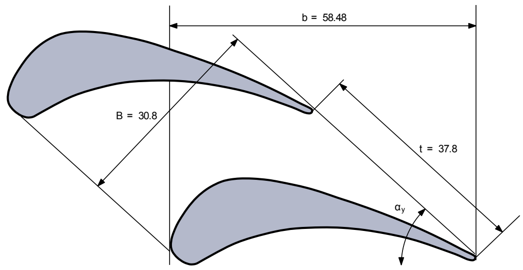


Figure 6: General view of the stator and rotor blade profiles used

### 2.6. Conceptual design of the High Pressure Turbine part

An example of the conceptual design of the HPT part is shown in Fig. 5. All blades are based on the similar blade profiles shown in Fig. 6 (stator and rotor). The blade system was designed as fully reactionary ($\rho$=0.5), thus the same profiles were applied.

Velocity triangles of the HPT part are presented in Fig. 7. The presented conceptual design is not an optimal design, but is shown for the purpose of judging the possibilities of producing this part of the turbine. The presented design of the HPT is that of a high speed unit 7,000..12,000 rpm, with a stage number of 5..10, mounted in a single body with two or three covers. It could be a very compact unit with a diameter of less than 1 m and length of between 1.5 and 2 m.

It should be noted that very high steam parameters make it possible to increase the stages load—isentropic enthalpy drop in the range 160..230 kJ/kg, with relatively low Mach numbers (0.4..0.5)—compared to standard units: 40..80 kJ/kg. This gives a stage power of around single MW. In contrast to standard steam turbines, there is no drastic temperature decrease through

Table 2: Main parameters of the cycles

| Cycle Parameter | GRAZ | TOSHIBA | WESTINGHOUSE | MNRC |
|---|---|---|---|---|
| $p_{max}$, bar | 350 | 380 | 250 | 250 |
| $t_{max}$, °C | 1,700 | 1,700 | 1,700/1,600 | 1,700 |
| Gross power, MW | 513 | 513 | 513 | 513 |
| $\eta_{LHV}$, % | 70.8 | 71.2 | 74.0/72.8 | 79.0 |
| $\eta_{HHV}$, % | 59.5 | 59.8 | 62.2/61.2 | 66.4 |
| Specific power, kJ/kg | 2,202 | 3,331 | 3,489 | 4,706 |
| Net (electric) power, MW | 500 | 500 | 500 | 500 |
| $\eta_{el,LHV}$, % | 69.0 | 69.4 | 72.2/71.0 | 77.0 |
| $\eta_{el,HHV}$, % | 58.0 | 58.3 | 60.6/59.7 | 64.7 |
| Temperature at the most thermal loaded point, °C | 1,700 | 1,700 | 1,700/1,600 | 1,700 |
| Pressure at the most thermal loaded point, bar | 50 | 73 | 250 | 250 |
| Pressure at the most pressure loaded element, bar | 350 | 343 | 277 | 277 |
| Temperature at the most pressure loaded element, °C | 650 | 876 | 517 | 463 |
| Quantity of heat exchanged (HRSG heat load), MW | 315 | 329 | 256 | 165 |

Table 3: Main parameters of the HPT of the hydrogen fueled steam turbine

| Parameter | Inlet | Outlet |
|---|---|---|
| Rotational speed, rpm | 9,000 | 9,000 |
| Blade length, mm | 16 | 54 |
| Shaft diameter, mm | 737 | 737 |
| Mass flow, kg/s | 74.2 | 74.2 |
| Temperature, °C | 1,700 | 1,250 |
| Pressure, MPa | 25 | 3.6 |

the HPT part, which requires a cooling system for the whole length of the HPT part. The differences in blade length are not large (inlet/outlet ratio 3.5..4).

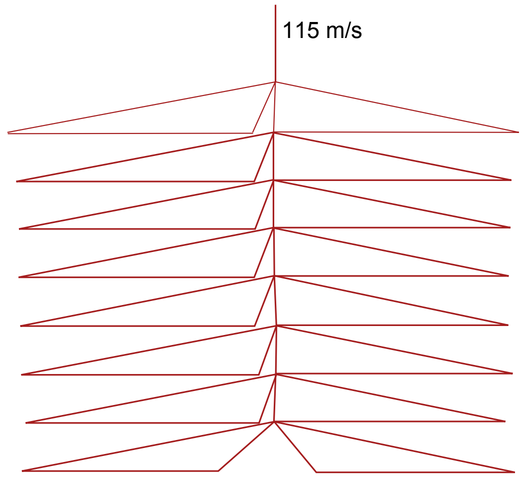


Figure 7: Velocity triangles of the HPT of the hydrogen fueled steam turbine

## 3. Conclusions

All HFCTC cycles with direct firing which have been studied as part of this research (500 MW class units) show very high thermal efficiency (in the range 58..64% HHV), which is far higher than the performance of the known cycles. It has been fully confirmed that it is possible to achieve efficiency levels at least 10 percent points higher than the efficiency of the most efficient current power units. Similarly, all HFCTC cycles have very high specific power (2,200..4,700 kJ/kg), which is many times higher (in extreme cases, by an order of magnitude) than the performance of current gas or steam turbines or combined cycles.

Consequently, extremely compact power units with minimal usage of construction materials could be built—see Fig. 5. Considering the fact that HFCTC cycles almost totally eliminate $CO_2$ and $NO_x$ emissions, this solution could be viewed as an interesting alternative for future development compared to conventional power technologies.

The possibility of technical implementation of the systems needs to be commented on from viewpoints of elements operating in zones of the highest steam parameters. These ranges are summarized in Table 4. It is seen that for MNRC and Westinghouse (highest efficiencies) systems, the highest temperatures occur simultaneously with the highest pressures of the working medium. This applies to high pressure combustion chambers and turbines, and they just seem to be the "critical" elements here. Accordingly, in the Toshiba system the high pressure combustion chamber is abandoned, but at the cost of very high pressure, the introduction of high temperature heat exchangers and lower

Table 4: The most loaded elements of the systems

| Parameter\System | | GRAZ | TOSHIBA | WESTINGHOUSE | MNRC |
|---|---|---|---|---|---|
| Element most loaded | $t$, °C | 1700 | 1700 | 1700/1600 | 1700 |
| by temperature | $p$, bar | 50 | 73 | 250 | 250 |
| Element most loaded | $t$, °C | 350 | 343 | 277 | 277 |
| by pressure | $p$, bar | 650 | 876 | 517 | 463 |
| HRSG heat load | MW | 315 | 329 | 256 | 165 |

system efficiency. An important advantage of the Graz system is relatively low pressure in the area of the highest temperatures, as in the "classical" system of the gas turbine. The undoubted disadvantage of this system is the presence of a steam compressor—a new unrecognized device, an expanded system of heat exchangers, and in general considerable complexity of the system.

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