Methods of increasing thermal efficiency of steam and gas turbine plants

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Abstract. Three new methods of increasing efficiency of turbine power plants are described. Increasing average temperature of heat supply in steam turbine plant by mixing steam after overheaters with products of combustion of natural gas in the oxygen. Development of this idea consists in maintaining steam temperature on the major part of expansion in the turbine at level, close to initial temperature. Increasing efficiency of gas turbine plant by way of regenerative heating of the air by gas after its expansion in high pressure turbine and before expansion in the low pressure turbine. Due to this temperature of air, entering combustion chamber, is increased and average temperature of heat supply is consequently increased. At the same time average temperature of heat removal is decreased. Increasing efficiency of combined cycle power plant by avoiding of heat transfer from gas to wet steam and transferring heat from gas to water and superheated steam only. Steam will be generated by multi stage throttling of the water from supercritical pressure and temperature close to critical, to the pressure slightly higher than condensation pressure. Throttling of the water and separation of the wet steam on saturated water and steam does not require complicated technical devices.

1. Introduction.
Steam, gas and combined cycle turbine power plants are generating major part of the energy used by humanity. Thus increase of their efficiency is important. Steam turbine plants are characterized by high thermodynamical efficiency for achievable maximum temperature of steam but relatively low values of this maximum temperature. Gas turbine plants are characterized by high values of maximum temperature of working substance due to direct combustion of fuel in the stream of the working substance (compressed air which turns into gas in combustion chamber) but high temperature of exhaust gas which means high average temperature of heat removal in the cycle. Combined cycle power plants are incorporating positive features of steam and gas turbine power plants, however transfer of substantial part of the heat from gas to steam causes substantial losses of efficiency due to irreversibility of heat transfer processes.

2. Steam turbine power plant.
We have proposed to increase average temperature of heat supply in the steam turbine power plant which would in turn increase its efficiency. In the first case it is achieved by way of mixing superheated steam with the products of combustion of natural gas in oxygen on the high temperature parts of isobaric processes of initial and intermediate superheating of the steam [1, 2, 3]. In second case average temperature of heat supply is further increased on account of maintaining of steam’s
temperature during major part of its expansion in the turbine close to initial temperature at the turbine’s inlet. To achieve this additional heat is supplied to the steam during its expansion in the turbine [4, 5].

These ideas were tested on the steam turbine power plan K-1200-240 (LMZ); its power output is 1,200 MW, steam’s maximum pressure is 24 MPa and temperature is 540 °C. Thermodynamical efficiency is 53.3% and net efficiency is 47.4%.

2.1. Mixing superheated steam with the products of combustion of natural gas in oxygen.

The essence of the method is described on Figures 1 & 2 – scheme of modified steam turbine power plant and its cycle in the coordinates \( t,s \). Steam after the overheater 2 is directed into combustion chamber 3, in which it is being mixed with the products of combustion of methane in oxygen. Maximum temperature of working substance is 1,430 °C which is equal to the temperature at turbine inlet in GE “H” class gas turbines. Working substance, which consists of 88% H\(_2\)O and 12% CO\(_2\), with pressure 24 MPa and temperature 1,430 °C is expanding in the turbines. CO\(_2\) is removed from condenser by compressor 10.

Due to increasing of maximum temperature from 540 °C to 1,430 °C cycle’s thermodynamic efficiency is increased by 24.8% (from 53.3% to 66.5%) and net efficiency of the plant – by 22.4% (from 47.4% to 58.0%).

Figure 1. Scheme of modified steam turbine plant. 1 – boiler; 2 – overheater; 3 and 5 – combustion chambers; 4, 6 and 8 high, medium and low pressure turbines; 7 – surface regenerative heater of feeding water; 9 – condenser; 10 – compressor for removal of carbon dioxide from condenser; 11 – feed water pump; 12 – air separation plant; 13 and 14 – oxygen and natural gas compressors.
Figure 2. Modified cycle in t,s coordinates. 1-2 – expansion in high pressure turbine; 2-3 – reheating of steam-gas mixture in combustion chamber; 3-4 – expansion in the medium pressure turbine; 4-5 – cooling of steam-gas mixture in the regenerative heat exchanger; 5-6 – expansion in the low pressure turbine; 6-7 – condensing of steam; 7-8 – compression of feed water by the pump; 8-9 – heating of feed water in the regenerative heat exchanger; 9-10 – heating of water and steam in the boiler; 10-1 – superheating in the combustion chamber.

2.2. Isobaric-isothermal heat supply
In the steam turbine’s cycle maximum temperature of the steam is at the turbine’s inlet, during expansion its temperature is decreasing. In order to increase thermodynamic efficiency, temperature of steam during major part of expansion process may be maintained at the levels, which are close to initial temperature. It will require supply of additional heat during expansion. This can be achieved by multi-stage combustion of fuel in the stream of working substance during its expansion in the turbine. Due to supply of the heat, first part of the process of expansion in the turbine will be approximately isothermal as opposed to traditional adiabatic. However, in order to maintain low temperature of het removal, second part of expansion in the turbine should be adiabatic.

Such modification of steam turbine power plant can be achieved by mixing of steam with temperature 540 °C after overheater of the boiler with products of combustion of natural gas (methane) in oxygen, containing excess of methane. For burning this excess of methane, during expansion in the turbine oxygen is being added to the working substance. It is clear from the t,s chart, that in the modified cycle average temperature of heat supply is higher which transpires into increase of the thermodynamic efficiency. Modified cycle is becoming closer to generalized Carnot cycle.
Figure 3. Scheme of modified steam turbine plant. 1 – combustion chamber; 2, 3 and 5 – high, medium and low pressure turbines; 4 – regenerative heater of feed water; 6 – condenser; 7 – compressor for removal of carbon dioxide from condenser; 8 – feed water pump; 9 – air separation plant; 10 and 11 – oxygen and natural gas compressors.

Figure 4. Cycle of steam turbine plant with isobaric-isothermal heat supply in coordinates $t$,$s$. 1-2 – isothermal expansion in high pressure turbine; 2-3 – adiabatic expansion in medium pressure turbine; 3-4 and 7-8 – isobaric heat transfer in regenerative heat exchanger; 4-5 – adiabatic expansion in low
pressure turbine; 5–6 – heat removal in condenser; 6–7 – compression of feed water in the pump; 8–1 – heat supply to the steam in the combustion chamber.

Temperature of the working substance on the isothermal part of expansion in the turbine is substantially higher than temperatures of self-ignition of methane with oxygen, hydrogen with oxygen and CO with oxygen. Consequently chemical reaction between products of incomplete combustion of methane and oxygen will be running in wide range of concentrations without any means of ignition.

Increase of net efficiency is 20.1% (from 60.0% in GE “H” class combined cycle power plant to 72.1%).

3. Gas turbine power plant.
Efficiency of gas turbine power plant with regenerative heat exchange may be increased if regeneration of the heat will be between gases after high pressure turbine and air after compressor. This will increase temperature of air entering combustion chamber with corresponding increase of average temperature of heat supply. Average temperature of heat removal will be decreased because temperatures of gases after expansion in the low pressure turbine will be lower than their temperature at regenerative heat exchanger outlet in conventional cycle [6].

![Figure 5](image.png)

**Figure 5.** Conventional 1-2-5-3-4-6-1 and modified 1-2-7-3-8-9-10-1 cycles in coordinates $T,s$. 

Essence of the method is exhibited on Fig. 5 – conventional and modified regenerative cycles of gas turbine plants in $T,s$ coordinates. Schemes of gas turbine plant WR-21 and modified plant with high temperature heat exchange are shown on Fig. 6.

**Figure 6.** Schemes of conventional - a) and modified - b) gas turbine plants. LPC – low pressure compressor, IC – intercooler, HPC – high pressure compressor, RHE – regenerative heat exchanger, CC – combustion chamber, HPT – high pressure turbine, LPT – low pressure turbine.
Increase of thermodynamic efficiency will be observed if following condition is met:

\[
\frac{L - \Delta L}{Q_i - \Delta Q_i} > 0
\]  \hspace{1cm} (1)

Above equation is always fulfilled if relative efficiencies of turbines and combustion chamber are ignored, i.e. thermodynamic efficiency is increasing for all values of pressure ratio and maximum temperature at high pressure turbine inlet which are typical for modern gas turbine plants. So for practical purposes we may consider that thermodynamic efficiency is always increasing.

Unfortunately, reduction of the gases temperature at low pressure turbine inlet and corresponding reduction of enthalpy values during expansion is reducing power output of low pressure turbine. Work produced in the cycle is reduced by the area 10-9-8-4. This factor is not influencing thermodynamic efficiency but it does affect net efficiency. Net power output, which takes into account efficiencies of compressors and turbines, is equal to:

\[
L_{ef} = L_T \times \eta_T - L_C / \eta_C
\]  \hspace{1cm} (2)

where \( L_{ef} \) – net power output, \( L_T \) i \( L_C \) – theoretical (thermodynamic) power output/consumption of turbines and compressors, \( \eta_T \) and \( \eta_C \) – their efficiencies. Net efficiency of gas turbine plant with high temperature regeneration is increasing for high maximum temperature (above 800 °C) and high efficiencies of the compressors and turbines.

Calculations shows that application of high temperature regeneration of heat in gas turbine plant Rolls Royce WR-21 will increase thermal efficiency by 20.1% (from 55.1% to 66.6%) and net efficiency by 12.8% (from 43.6% to 49.2%).

4. Combined cycle power plant.

Efficiency of combined cycle power plant may be improved by reducing internal irreversibility of processes in the thermodynamic cycle of the power plant. It is necessary to transfer between working substances of gas and steam cycles considerable part of the heat, supplied in the gas cycle. This imposes restrictions on the shape and parameters of the steam cycle. As a consequence thermal efficiency of the steam cycle in combined cycle power plant is substantially lower than in the ordinary steam power plant – 36% and 53% respectively for the same values of maximum steam temperature.

\[\text{Figure 7.}\] Thermodynamical cycle of steam part of combined cycle power plant in \( T,s \) coordinates. 1-2, 3-4, 16-5 – expansion in turbines, 7-8-14, 9-10-13, 11-12-17 – heat supply in steam generators, 17-1, 15-3 – heat supply in steam over heaters; 13-15 and 2-15; 14-16 and 4-16 – mixing of steam after
partial expansion in the turbine with saturated steam produced in the steam generator; 6-7, 8-9, 10-11 – feed water compression in pumps.

Typical modern combined cycle power plant is GE S107FA with power output 263 MW. Thermodynamical cycle of the steam part of this plant is exhibited on Fig. 7. No improvements of gas part are suggested.

**Figure 8.** Thermodynamical cycle in $T,s$ coordinates of the steam part of combined cycle power plant with multi stage throttling of working substance.1-2, 3-4, 5-6, 8-9, 10-11, 12-15 – expansion in turbines; 16-17, 18-19 – compression of water in pumps; 20-21, 22-24, 25-27, 28-30, 31-33, 34-36 – throttling; 21-22 and 21-23, 24-25 and 24-26, 27-28 and 27-29, 30-31 and 30-32, 33-34 and 33-35, 36-37 and 36-38 – separation of wet steam on saturated liquid and saturated steam; 23-1, 7-8 – heat supply in steam over heaters; 19-20 – heat supply in water heater, 2-3 and 26-3, 4-5 and 29-5, 6-7 and 32-7, 9-10 and 35-10, 11-12 and 38-12 – mixing of steam after partial expansion in the turbines with saturated steam; 17-18 and 37-18 – mixing of feed water with saturated water after last stage of throttling.

Combined cycle power plants takes advantage of the best features of both gas and steam power plants however they have one serious drawback. Substantial part of the heat supplied in the gas cycle, should be transferred to the working substance of the steam cycle. Due to fundamental difference in the shape of isobars of gas versus water - wet steam – superheated steam, there are high losses in the steam generator due to irreversibility of heat transfer processes. In modern plants these loses are reduced by generation of steam at several stages of pressure. However number of steam generation stages is limited as each stage requires separate system of evaporation and lowering tubes, water and water-steam drums, pump. This complicates the plant so in modern most sophisticated combined cycle power plants (including GE S107FA) number of stages of steam generation is limited to three.

If processes of heat transfer from gas to wet steam will not be used and heat will be transferred only from gas to liquid and from gas to saturated or superheated steam, then losses, related to irreversibility of heat transfer may be reduced. Steam will be generated in the multi-stage throttling of liquid from super-critical pressure and near-critical temperature to the pressure, slightly above condensation pressure. Since throttling of liquid and separation of wet steam on saturated liquid and steam do not require complicated technical devices, number of steam generation stages may be more
than in conventional combined cycle power plant. Such increasing of number of steam generation stages will not complicate the plant [7].

**Figure 9.** Scheme of steam part of combined cycle power plant with multi-stage throttling of working substance. HPT, LTP – high and low pressure turbines; SH1, SH2 – steam overheaters; CD – condenser; WP1, WP2 – feed water pumps; WH – water heater; TV1…TV6 – throttling valves; S1…S6 – separators of wet steam.

Introduction of multi-stage throttling increase net efficiency of steam part by 4.1% (from 31.5% to 32.8%) and net efficiency of the whole combined cycle power plant by 1.0% (from 57.2% to 57.8%).

**5. Conclusions.**

New methods of increasing efficiency of steam, gas and combined cycle power plants are being proposed. Calculations shows increase of net efficiency 20.1% for steam turbine power plant, 12.8% for gas turbine power plant and 4.1% for steam part of combined cycle power plant.

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