THE INFLUENCE OF WORKING FLUID PARAMETERS ON THE THERMAL EFFICIENCY OF GAS - VAPOR POWER PLANTS

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Abstract. *In this work, the influence of working fluid parameters (e.g., combustion products, steam) on the thermal efficiency of thermoelectric power plants operating either on a binary or a combined cycle. These are the most utilized thermal cycles currently in the global power generation scenario. It is herein defined that the binary cycle is the gas-vapor thermoelectric power plant cycle with a utilization boiler (GVPPUB) and the combined cycle is the gas-vapor thermoelectric power plant cycle with a complete combustion boiler (GVPPCCB), with the difference that the binary cycle has a utilization boiler that uses the gas turbine exhaust gases and the combined cycle also uses the exhaust gases, but additionally burning the remaining fuel still present in the gas turbine exhaust gases. In the analytical analysis, the influence on power plant thermal efficiency of the following working fluid parameters: the initial combustion products temperature at the gas turbine inle; the thermal efficiency of the thermodynamic cycle of the steam turbine part; the exhaust gases temperature; the excess air value in the exhaust gases. The results make possible the choice of working fluid parameters that are more suitable for the optimization of the power plant cycles analyzed in this study .*

Keywords: binary and combined cycle power plants, thermal efficiency.

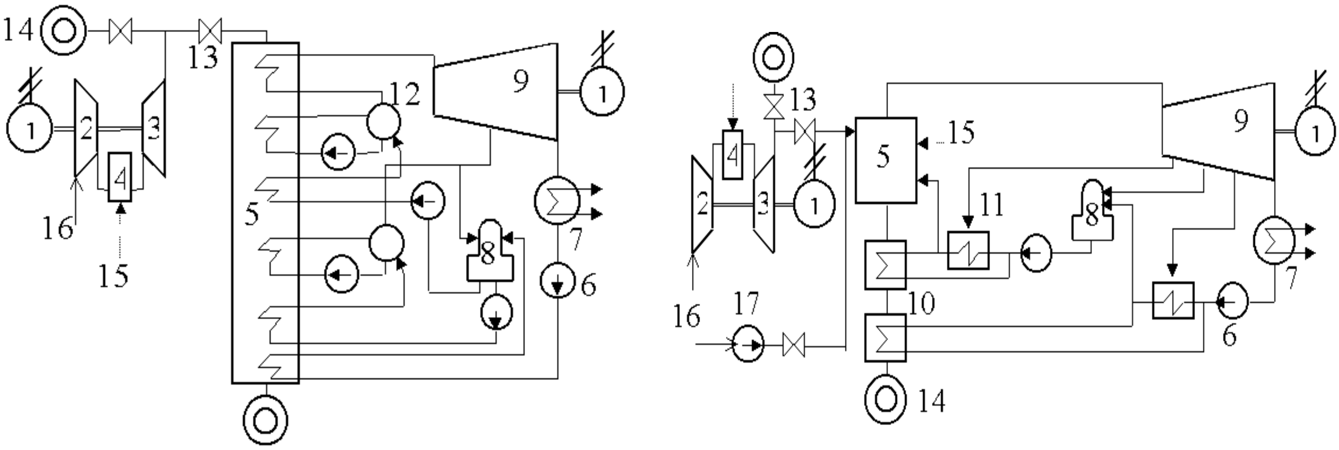
1. INTRODUCTION

The newest Gás-vapor power plants (GVPP) are combined installations of two or more basic thermodynamic cycles, realized by different working fluids (e.g., combustion products, steam) and in different bands of temperature variation. These installations are known too as binary or with cogeneration. The cycles that works with the highest temperature as called as “Higher” and the one’s that works with the lower temperature are called as “Lowers”. Usually the higher cycle is the Gas turbine (Brayton), with the thermal efficiency near 0,35 (Radchenco, 1998). The lower cycle is the vapor turbine cycle (Ranquine), with a thermal efficiency near 0,4. From this combination comes the gas – vapor power plants denomination.

The grouping of the gas turbine cycle with the vapor turbine is an example of joint of installation cycles by increasing the difference between the higher and lower temperatures at the cycle, increasing the installation thermal efficiency. The newst GVPP can reach the power of 750 MW and the thermal efficiency near 0,6. The main fuel used is this power plants is the natural gas, the substitute fuel is a distilled of oil. Using natural gas allows decrease the emission of pollutants and else increasing the operational conditions of the gas turbine, mainly avoiding the corrosion caused by vanadium. The contribution of the gas turbine in the power plant production reaches ~ 70% of all power plant.

Usually the joint of gas and vapor turbines are made in many ways. The thermal schemes are different, the equipments and the characteristics of this schemes are different too. The GVPP thermal scheme determines energetic, economic and ecologic characteristics of the power plant.

Actually, the most used GVPP are called Gas - Vapor thermoelectric power plant with a utilization boiler (GVPPUB) these power plants are known too as Gas – Vapor thermoelectric power plant with recovery boiler. Near 80% off all GVPP built in the world in the last 30 years are GVPPUB with the total installed power of ~ 52000 MW (Bezlepkin, 1997; Vlassov, 2002). The utilization boiler uses the exhaust gases of the higher cycle of the gas turbine. The thermal scheme of this power plant is presented in the Fig. 1a. The exhaust gases of the gas turbine (3) go inside of the steam generator (5) that have two areas, one of low and other of high pressure of the overheated water vapor. The area of high pressure (~ 11 MPa , 540 ^{o}C) with the drum (12) make the overheated water vapor for the turbine (9). The area of low pressure is used to make a better use of the heat in the exhaust gases. The vapor that it makes its used in the low pressure vapor turbine stage and in the deaerator (8). Usually are used gas turbines, the gas turbines of GVPPUB can work without cogeneration, for this situation in the scheme has registers (13) and chimneys (14).



a) b)

Figure 1. – Scheme of thermoelectric power plants with co-generation.
a) – GVPPUB, b) - GVPPCCB
1 – electric generator ; **2** – air compressor; **3** – gas turbine; **4** – combustion chamber; **5** – vapor generator; **6** – water pump; **7** – condenser; **8** – deaerator; **9** – vapor turbine; **10** – heat exchanger - water; **11** – heat exchanger water – vapor; **12** - drum; **13** – register; **14** – chimney; **15** – natural gas; **16** – air; **17** – air fan.

The amount of boilers is equal to the amount of gas turbines. As the temperature of the exhaust gases in the entrance of the gas turbine is increased, the temperature of the gases in the exit is increased too, this allow to increase the parameter of living vapor leaving the boiler. After is used intermediary overheating of the vapor and of the turbine of two or more drums.

The first combined installations where built based in the scheme of a Gas – Vapor thermoelectric power plant with a complete combustion boiler (GVPPCCB), witch the scheme is show in the Fig. 1b. Currently they are ~ 17% of the GVPP. Usually they are build combining the installations of a gas turbine and a vapor turbine that are produced in series by the industry.

At the normal working state both cycles work. The exhaust gases of the gas turbine go through the register (13) opened to the combustion chamber of the vapor generator (5). The temperature of the exhaust gases of a gas turbine varies at $T_4 = 450 - 600\ ^{o}C$ and the volumetric percentage of oxygen in they is 12 - 15%. In the combustion chamber (5) the vapor generator is supplied with additional fuel (15) that burns using the oxygen remaining in the combustion products that came from the gas turbine. The high temperature after the combustion of the additional fuel allows high parameters of the overheated vapor, being able to reach supercritical parameters. Leaving the combustion chamber the gases transfers heat in the heat-exchangers (10) to the feed water and to the condensate. The gas turbines that work in a GVPPCCB produced 15 - 35 % of total power plant power.

The thermal scheme of a GVPPCCB allows varying the power of the power plant in a large band. In the partial band, the gas turbine can be stopped and the register (13) can be closed. In this case the atmospheric air (16) is supplied to the combustion chamber of the boiler by a fan (17). It's possible, too, the autonomous work of the gas turbine. In this case, with the help of the register (13), the products of the combustion are lead directly to the chimney. In the hour of maximum consumption of electric energy all turbines in the power plant will be working at the nominal power.

Thermoelectric power plants built in these cycles have a much better possibility of maneuvering of energetic and power characteristics than the conventional thermoelectric power plants. This conditions a great divulging of them in the world energetic that are based in the use of gaseous and liquid fuels.

The efficiency of the GVPP can be determined by the coefficient of thermal efficiency, that show which part of heat liberated by the fuel in the combustion is turned in electric energy. For a power plant using the combined gas – vapor cycle, the coefficient of thermal efficient is determined as:

$$\eta_{et} = \frac{P^g + P^v}{Q_{in}(B^g + B^v)}, \tag{1}$$

where: P^g e P^v - electric power of gas and vapor turbines respectively, in MW;

B^g e B^v - fuel consumption in the gas and vapor turbines, in kg/s;

Q_{in} - fuel's lower heat value, in MJ/kg;

In the work (Vlassov D., Vargas J. V. C., 2005) was demonstrated that the thermal efficient of power plants with combined gas – vapor cycles (GVPP) was:

$$\eta_{et} = \eta_{et}^{g} \frac{\alpha_{esc}}{\alpha^{g}} + \left(1 - \frac{\eta_{et}^{g}\alpha_{esc}}{\eta_{m}^{g}\alpha^{g}} - l_{esc} - l_{outr}\right)\eta_{t}^{v}\eta_{m}^{v} , \tag{2}$$

where: η_{et}^{g} - thermal efficient coefficient of a gas turbine.

η_{et}^{v} - thermal efficient coefficient of a vapor turbine;

η_{m}^{g} -mechanical efficiency, it is calculated as: $\eta_{m}^{g} = \frac{\eta_{et}^{g}}{\eta_{t}^{g}}$;

η_{t}^{v} - thermodynamic efficiency of the vapor turbine cycle;

η_{m}^{v} - mechanical efficiency of electric generator of the vapor turbine;

η_{t}^{g} - thermodynamic efficiency of the gas turbine cycle;

α^{g} - coefficient of air excess in combustion products of gas turbine;

α_{esc} - coefficient of air excess in exhaust gases of vapor turbine;

l_{esc} - a dimensional value of heat loss with the exhausting gases;

l_{outr} - loses of heat due to external cooling of the boiler and pipes, incomplete chemical combustion, and other loses.

The air excess coefficient is calculated as:

$$\alpha = \frac{L}{L_0} , \tag{3}$$

where: L_0 - stichomythic air quantity , in kg of air per kg of gaseous fuel;
L - real air quantity, in kg of air per kg of gaseous fuel.

The a dimensional value of heat loss with the exhaust gases (relative heat loss) is calculated as:

$$l_{esc} = \frac{L_0}{Q_{in}} \alpha_{esc} c_p t_{esc} , \tag{4}$$

where: c_p - specific heat in constant pressure of exhaust gases, in MJ/kgK;
t_{esc} - exhaust gases temperature, in ^{o}C;

In it structure the Eq. (3) has basic parameters of a gas – vapor power plant, allow determining the influence of these parameters in the thermal efficiency coefficient of a gas – vaporing thermoelectric power plant.

The exhaust vapor gases influence analysis on the thermal efficiency of a gas vapor power plant was made for two schemes: with utilization boiler (GVPPUB) and with complete combustion boiler (GVPPCCB) (see Fig. 1). For the influence analysis of a parameter in the whole power plant thermal efficiency other parameters that are not depended of them were fixed. To determine the influence of one parameter in the thermal efficiency of the whole power plant we need to fix some other parameter that do not depend of him.

The thermal efficiency was analyzed admitting the starting data:

- The Fuel is the Bolivian natural gas that was removed its heavy hydro-carbons (Vlassov, 2001). After cleaning the heavy components the volumetric composition was:CH_4=97,73%; N_2=1,11%; CO_2= 1,13%; O_2=0,03%. From the gas composition were calculated: the lower heat value of the fuel Q_{in} =47,53 MJ/kg and the theoretician quantity of air L_0 = 16,27 kg/kg;

- The air excess coefficient in the exhaust gases of the boiler α_{esc} =1,03;

- The exhaust gases temperature is 100 °C, and its specific heat is c_p =1,37 kJ/kgK;
- The mechanical efficiency of a vapor and a gas turbine were assumed as: $\eta_m^g = \eta_m^v$ = 0,97;
- Others heat loses l_{outr} = 0,02 ;
- For the vapor turbine thermodynamic cycle efficiency were assumed the values η_t^v = 0,32; 0,36 e 0,40.

2. MOTOR FLUID PARAMETERS INFLUENCE ANALYSIS IN THE THERMAL EFFICIENCY OF A GAS-VAPOR POWER PLANT

2.1. Gases temperature influence in the turbine entrance

The gases temperature at the turbine entrance is a basic parameter that exerts a bigger influence on the GVPP thermal efficiency. The thermal efficiency of a gas turbine (Kostuk, 1979) depends of many factors and is calculated as:

$$\eta_t^g = \frac{\tau\left(1-\dfrac{1}{\beta^m}\right)\eta_i - \left(\beta^m - 1\right)\dfrac{1}{\eta_c}}{\tau - 1 - \left(\beta^m - 1\right)\dfrac{1}{\eta_c}}\eta_{cc}\,, \tag{5}$$

where: τ - increasing temperature tax, $\tau = \dfrac{T_1}{T_3}$;

T_1 - absolute temperature of the exhaust products in the entrance of the gas turbine installation, in K;

T_3 - absolute temperature of the air in the compressor entrance, in K;

β - increasing pressure tax in the compressor;

m - exponent, $m = \dfrac{k-1}{k}$;

k - adiabatic index, k = 1,4 for the air, k = 1,33 for the combustion products;

η_i - gas turbine installation internal efficiency;

η_c - compressor efficiency;

η_{cc} - gas turbine combustion chamber efficiency;

Graphical interpretation of Eq. (5) is shown in Fig. 2.

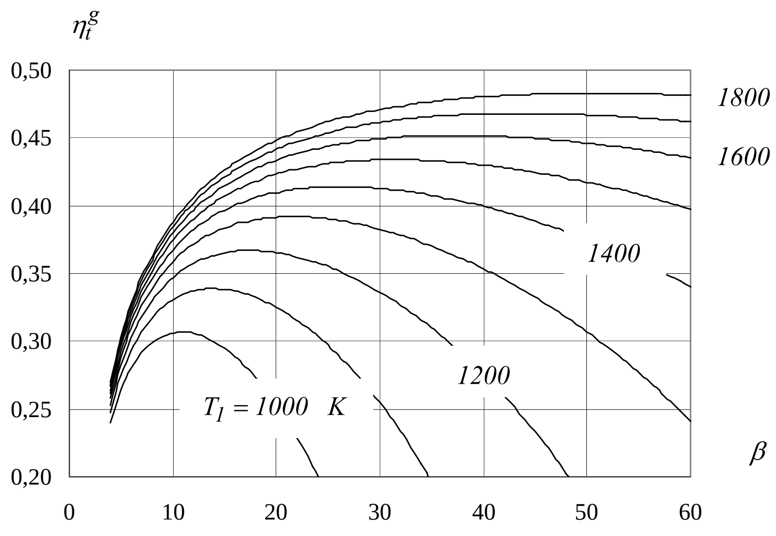


Figure 2. - Variation of gas turbine thermal efficiency dependent of compression proportion of compressor and gases temperature in turbine inlet.

The thermal efficiency determined in Eq. (5), depends of atmospheric air temperature T_3, the efficiency of: compressor η_c; turbine η_i; and combustion chamber η_{cc}. The highest influence in the efficiency is the increase of pressure tax in compressor β and the temperature of the gases in the entrance of the turbine T_1. The graphic shown in

Fig. 2 was built for the gases temperature of 1000 to 1800 K at the turbine entrance, and fixing some values: T_3 =293 K; $\eta_c = \eta_i$ =0,89 e η_{cc} =0,99. Fig. 2 shows that for each gases temperature value at the turbine entrance have the best value for the increase of pressure in the compressor, and this will result in the installation maximum thermal efficiency η_t^g . The best way to improve the gas turbine installation efficiency is the increase of the gases temperature at the turbine entrance.

It's possible to see in Fig. 2 the correlation between installation thermal efficiency and gases temperature at the turbine entrance. From data treatment of Fig. 2 we can show this correlation as:

$$\eta_t^g = 0{,}2993 \cdot \ln T_1 - 1{,}7567\ . \tag{6}$$

The thermal efficiency of gas – vapor power plants combined cycles (GVPP) by Eq. (2) and using Eq. (6) is graphically represented at Fig. 3.

The analysis of the curves allow making some conclusions:

- The thermodynamic thermal efficiency of gas – vapor power plants, of binary cycle (GVPPUB) and combined

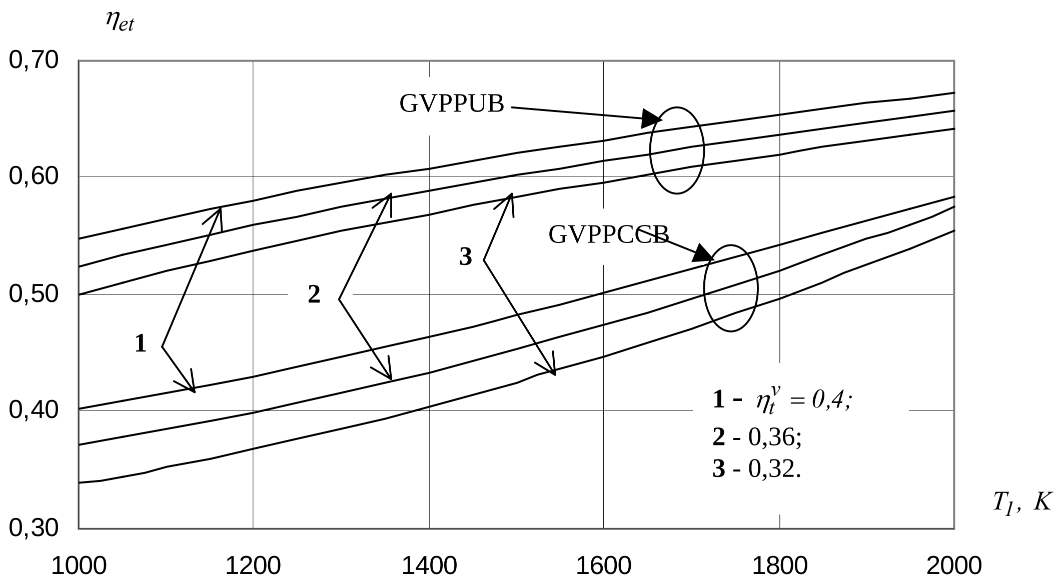


Figure 3. Influence of gases temperature at the turbine entrance and thermal efficiency of gas – vapor Turbine cycle of GVPPUB and GVPPCCB.

- (GVPPCCB) is bigger than the basic cycles that they are made of.

- In all gases temperature variation at the turbine entrance, always the efficiency of thermoelectric power plants of binary cycles (GVPPUB) is bigger than thermoelectric power plants of combined cycles (GVPPCCB). The curves will coincide (and the thermal efficiencies will be the same) in the limit case, when the fuel combustion in gas turbine will be stichomythic. But in this case the turbine wings will support the combustion products temperature, for example to support the theory temperature of stichomythic combustion of gas natural from Bolivia (α^g =1,0 and not counting with the dissociation) it's of T_1 =2435 K. Increasing the temperature at the entrance of the gas turbine from 1200 to 1800 K the efficiency of both gas – vapor cycles increases gradually. For example, at the binary cycle (GVPPUB) with vapor cycle thermal efficiency of 40%, the thermal efficiency can increase to 58-66% In the combined cycle (GVPPCCB), and in same conditions, the thermal efficiency can increase to 43-50%

- As the gases temperature at the entrance of the gas turbine increases from 1000 K, the difference of thermal efficiency between GVPPUB and GVPPCCB increase gradually and reach a maximum of 15 – 17% when T_1 = 1050 – 1250 K. The thermal efficiency difference is much higher as lower is the thermal efficiency from the vapor cycle, to η_t^v = 0,4, the difference is 15% and to η_t^v = 0,32 the difference is 17%. The posterior increase in the gas temperature at the gas turbine entrance make the difference decrease and reach zero when stichomythic combustion, at the gas turbine occur.

It's important to notice that when the gases temperature at the gas turbine entrance is higher than 1600 K the oxygen percentage at the combustion gases becomes lower than 11 %, and not allow the fuel's complete burning.

Therefore for a GVPPCCB the value of η_{et}^{g} have only an illustrative character. But for GVPPUB the increase of gases temperature at the gas turbine entrance higher than 1600 K still have advantages.

2.2. Vapor turbine thermal efficiency influence.

The gas turbine thermal efficiency influence at GVPP is too much considerable. This influence can be analyzed in Fig. 2. The influence from the increase in gas turbine thermal efficiency is bigger at GVPPCCB. For a gas temperature at turbine entrance of T_1 = 1400 K the increase of the cycle's thermal efficiency is of η_t^v = 0,32 to η_t^v = 0,4 and this make the thermal efficiency of GVPPCCB increase from 0,404 to 0,464, a variation of 14,85% and in GVPPUB from 0,69 to 0,608,a variation of 6,85%.

2.3. Escape gases temperature influence.

The escape gases temperature influence in the GVPP thermal efficiency can be seen at Fig. 4. In it is shown relative heat loses l_{esc} of GVPPUB and GVPPCCB calculated with Eq. (4).

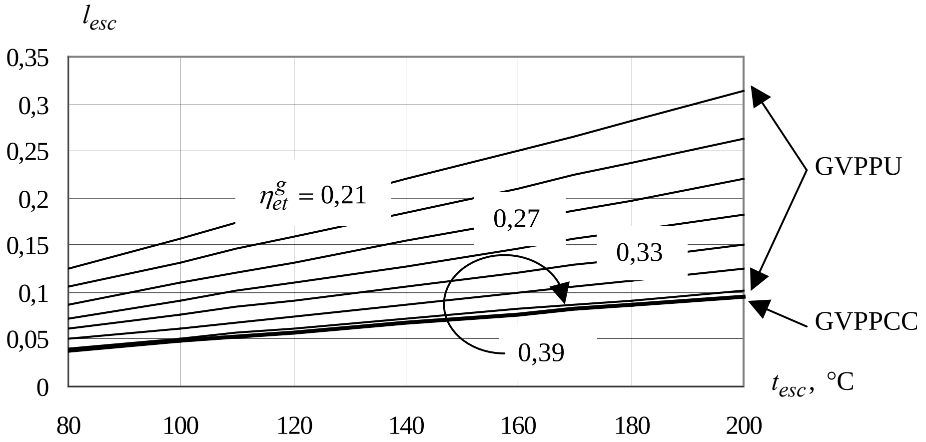


Figure 4.Escape gases temperature influence in fuel's relative heat loses.

From Fig. 4 is possible to see that the heat loses due to escape gases from GVPPCCB, with the coefficient of air excess in escape gases (for a gaseous fuel can be $\alpha_{esc} = 1{,}03$), is determined only by the escape gases temperature. Increasing the escape gases temperature from 80 to 200 ^{o}C the relative loses value in GVPPCCB increase from 3,9% to 9,65%.

In power plants with utilization boiler (GVPPUB) the heat loses value due the escape gases depends of vapor turbine installation thermal efficiency. For example, for escape gases temperature of 100 ^{o}C with η_{et}^{g} = 0,39 the relative loses is of α_{esc} =5,1% changing η_{et}^{g} = 0,24 the loses increases to 13,2%.

The heat loses with escape gases are always bigger in GVPPUB than in GVPPCCB. The heat loses difference between GVPPUB and GVPPCCB decrease as the escape gases temperature decreases. For example: for η_{et}^{g} =0,27 and t_{esc} = 200 ^{o}C this difference is 12,4% and the reduction of escape gases temperature to t_{esc} = 80 ^{o}C decrease this difference to 4,95%. From here we can see why the decrease of escape gases temperature increase more the thermal efficiency in GVPPUB than in GVPPCCB.

2.4. Escape gases excess air coefficient influence.

The escape gases excess air coefficient, have influence in GVPP thermal efficiency. This influence can be seen in FIG. 5. Calculating the thermal efficiency of a GVPP with Eq. 2 the escape gases temperature and living vapor parameters are constant.

The Figure 5 show, for a constant gas turbine thermal efficiency value $\eta_{et}^{g} = Const$, the increase of escape gases excess air coefficient (less fuel burn in the boiler) increase the GVPP thermal efficiency. The escape gases excess air coefficient variation in the vapor part is determined by the quantity of extra fuel is burn in GVPPCCB.

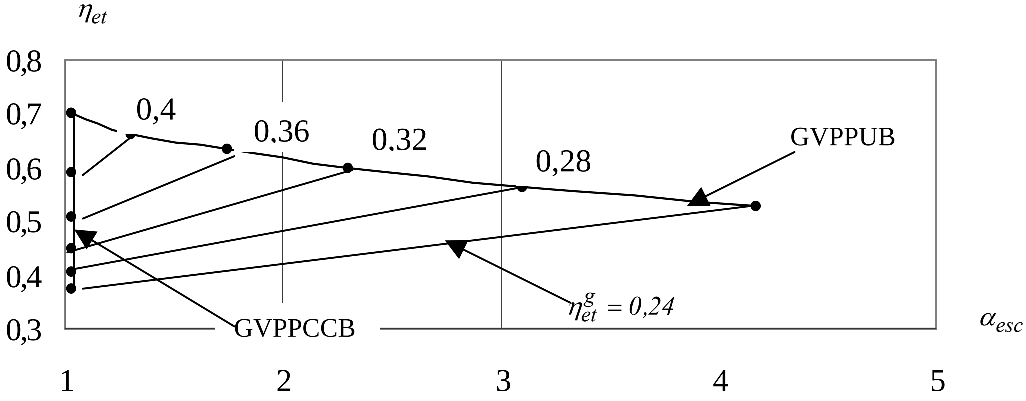


Figure 5. Escape gases excess air coefficient influence and gas turbine installation thermal efficiency on the GVPP thermal efficiency.

For example, for a gas turbine thermal efficiency $\eta_{et}^{g} = 0,32$ the GVPPUB efficiency will be η_{et} = 0,6 and the excess air coefficient gas turbine exit and from boiler will be α_{esc} =2,35. The burning of extra fuel in the boiler decrease the exit air excess and decrease, too, the thermal efficiency of all power plant. In case of α_{esc} = 1,03 the GVPPCCB have a thermal efficiency of only η_{et} =0,46. As bigger is the efficiency of gas turbine installation more sensible become the power plant efficiency to the excess air coefficient variation, in Fig. 5 they have a bigger inclination of the curves $\eta_{et}^{g} = Const$, that join GVPPUB and GVPPCCB. This show why the excess air coefficient $\Delta\alpha_{esc}$ of a GVPPUB with thermal efficiency of $\eta_{et}^{g} = 0,4$ will be bigger than one with $\eta_{et}^{g} = 0,36$. But in some times the extra fuel burning at the boiler is used to increase the parameter of living vapor at the turbine entrance, and this increase the vapor cycle thermal efficiency.

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