THERMODYNAMIC CYCLES OF POWER AND TRANSPORT GAS TURBINE ENGINES

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Contents

Summary

The considered data show that the GTU of the open and closed cycle have very attractive perspectives of their development for the XXI century energetics as separate units and combined units for different kinds of fuel, including nuclear fuel.

1. Introduction

This article is confined to present the basics of gas dynamic compressor design including the main topics of thermodynamics and aerodynamic theory for them. The volume compressors are only mentioned because they have just evident design and very complicated theory. The reader can get the main information about the axial flow compressors, including their parts such as blade and vane performances, ducts and rotors. The same data the reader gets about the radial flow compressors and the combined ones. It is possible to find some preliminary information on compressor design and estimation of their performance, including their modeling and control. The gas dynamic compressors are the main parts of the gas turbine engines of a wide range of applications- power production on Earth and in Space, transport here and on airplanes, marine units, gas pipeline driving, etc. The advanced readers can widen and deepen their information on the subject with the bibliography to this text.

2. Energy diagrams and main parameters of power, driving and transport GTE (GTU).

GTE (GTU) of the open cycle is important in the study of modern thermal energy systems. They are widely used to drive electric generators on heat power stations and compressors on pipelines of natural gas systems. They also play an important role in transport apart from aircraft applications, where they are used on heavy trucks, buses, military machines, locomotives and ships. It should be noted that GTE is usually associated with transport applications, GTU – with power and driving units. Figure 1 shows a scheme of the simplest one shaft GTU [3,7,11] which consists of a compressor C, combustion chamber CC and gas turbine T.

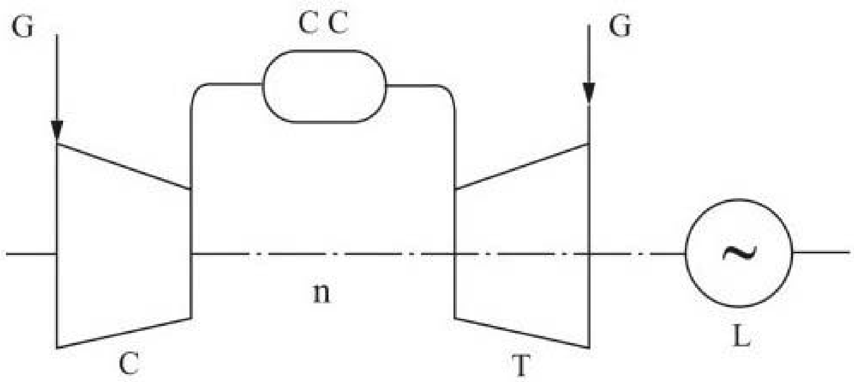

Figure 1. Scheme of the simplest GTE

On the right side one can see the load L i.e. an electric generator or air (water) screw which are usually connected to the GTU shaft through a reducer (not shown) and a gas compressor. CC can be replaced by a heater H to heat the compressed gas. Before C a cleaner can be installed to filter the air, after T – a silencer or exhaust gas cleaner. The main parameters of this GTU are the total pressure ratio $\pi_c=P_2^*(P_1^*)^{-1}$ and the total temperature ratio $\tau=T_3^*(T_1^*)^{-1}$. It is possible to name some GTU parameters: gas mass flow rate G, rotor speed n and GTU power P, though there are definite connections between them. There is a possibility to increase the efficiency of GTU by introducing regeneration of waste-gas heat to heat the air before CC. The scheme of a regenerative one-shaft GTU is shown on Figure 2.

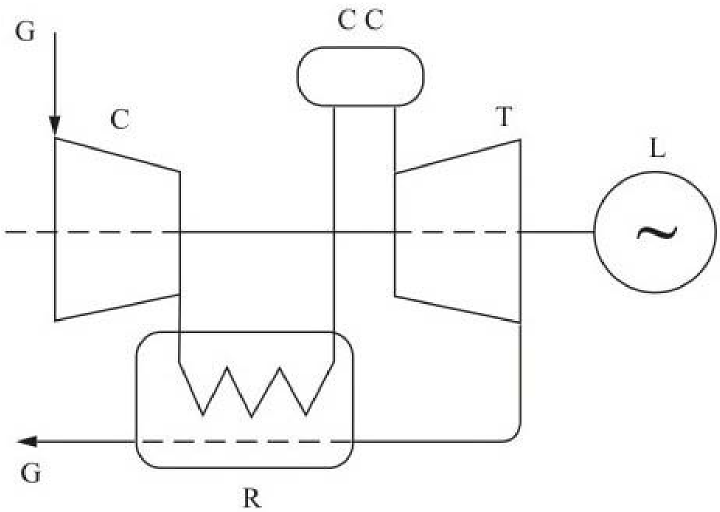

Figure 2. Scheme of the simplest regenerative GTU

It is possible to arrange its scheme with two shafts – with a separate power turbine shaft. The regenerator R is a heat exchanger in which the air is heated by the gas turbine exhaust gases. The heating of air reduces the fuel rate at CC to achieve a certain temperature of gas before the gas turbine T. For this scheme there is one additional parameter - the degree of regeneration, see 1.2. The further efficiency increase of GTU is connected (for the same τ) with the introduction of IC for the compression process and IH for expansion process. These additions, see 1.3 and 1.4, make the cycle looks like the Carnot cycle i.e lead to cycle carnotization. The intermediate cooling of air is usually realized by way of water injection at the IC, the gas intermediate heating - by the additional combustion chambers. The scheme of this one-shaft GTU is shown on Figure 3. IC and IH change the parameters of air and gas. That should be taken into account at a stage of compressor and turbine design. The role of the regenerator should be discussed. It is possible of course to arrange the GTU scheme as a multi-shaft system.

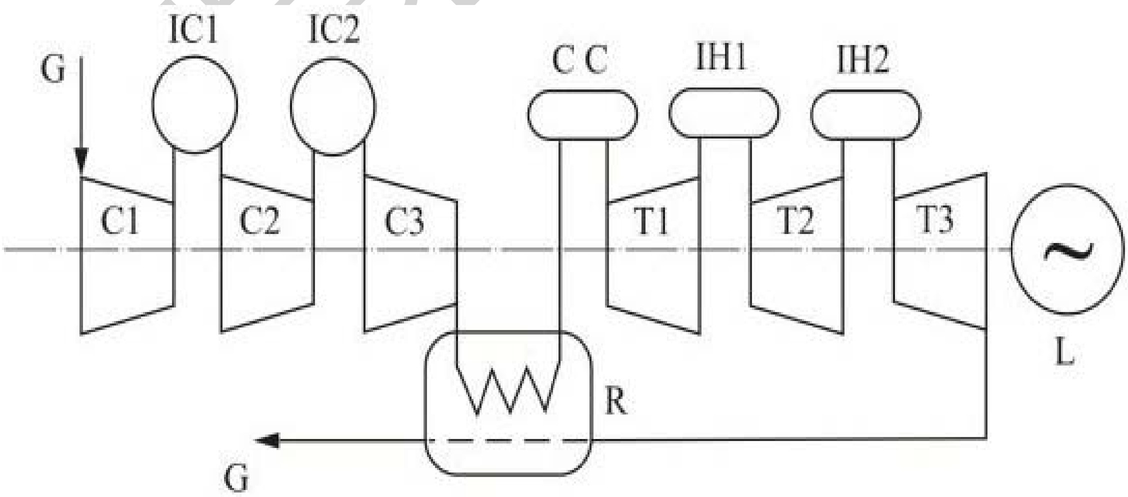

Figure 3. Scheme of GTU with intermediate cooling and intermediate heating

2.1. Thermodynamic cycles of open cycle GTU

The thermodynamic cycle is a process that the working fluid performs in the heat engine. This cycle is usually presented on the p-v or the T-S diagrams. It is a general assumption that the working fluid has constant thermal and other parameters (specific heat, gas constant, viscosity, etc.) for the whole cycle and the mass flow rate is also constant. Actually it is evident that the atmospheric air and the exhaust gas turbine gases have different thermal and other parameters. The exhaust gases disperse into the atmosphere, i.e. the open cycle GTU has no closed cycle in a direct sense, see later. It was decided to start the theory of cycles from an ideal case – constant gas thermal and other parameters and ideal processes in the GTU with inviscous gases, with isentropic processes of compression and expansion.

2.1.1. Ideal cycle

The ideal cycle of the simplest GTU has the name of Joule or Brighton cycle[13] and is shown on p-v and T-S diagrams by two isentropes (compression 1-2 and expansion 3-4) and two isobars, Figure 4 (a, b). The heat Q_1 is added along the isobar (2-3) and the heat Q_2 is taken away along the isobar 4-1. It should be noted that the gas velocities at the cycle points are not shown, so they can be supposed to be zero. In most cases p-v and T-S diagrams deal with the parameters of isentropically stagnated gas flow p^*,T^* and ρ^* or $v^*=(\rho^*)^{-1}$. It should be noted that the heat Q_1 can be added along isochoric line v = const, . The work of the ideal cycle p = const is equal to $H_{i.c.}=Q_1-Q_2$, where $Q_1=C_p(T_3-T_2)$, $Q_2=C_p(T_4-T_1)$ and is proportional to the areas of the cycle on the diagrams. This work is equal to the difference of the works of the ideal turbine and the ideal compressor

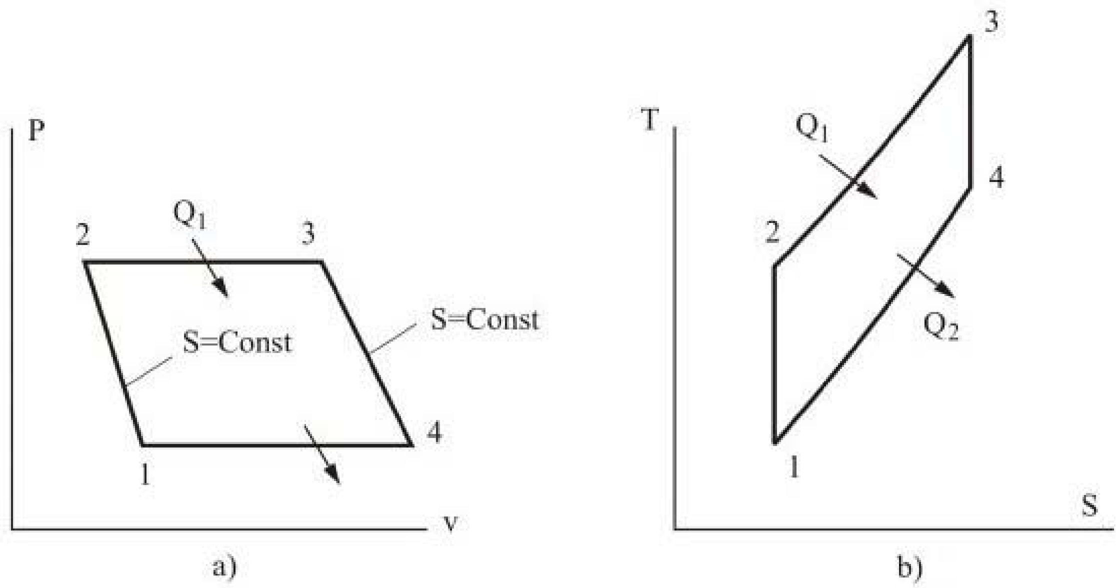

Figure 4 Ideal Joule or Brayton cycle

$H_{i.c.}=H_T - H_C$, where $H_T=C_p(T_3-T_4)=C_pT_3[1-T_4(T_3)^{-1}]$

$=C_pT_3\{1-[p_4(p_3)^{-1}]^{(k-1)k\,^\wedge(-1)}\}$, $H_c=C_p(T_2-T_1)=C_pT_1[T_2(T_1)^{-1}-1]=C_pT_1\{[p_2(p_1)^{-1}]^{(k-1)k\,^\wedge(-1)}-1\}$ or

$$H_{i.c.}=C_p\{T_3[1-\pi_c^{(1-k)k^\wedge(-1)}]-T_1[\pi_c^{(k-1)k^\wedge(-1)}-1]\}=C_pT_1\{\tau[1-\pi_c^{(1-k)k^\wedge(-1)}]-[\pi_c^{(k-1)k^\wedge(-1)}-1]\}, \quad (1)$$

where $\pi_c=p_2(p_1)^{-1}=p_3(p_4)^{-1}$, $\tau=T_3(T_1)^{-1}$.

The thermal efficiency of this ideal cycle (S-P) is the ratio of $H_{i.c.}$ to Q_1 or

$\eta_t=H_{i.c.}(Q_1)^{-1}=1-(T_4-T_1)(T_3-T_2)^{-1}=1-T_1(T_2)^{-1}[T_4(T_1)^{-1}-1][T_3(T_2)^{-1}-1]^{-1}$, where

$T_4(T_1)^{-1}=T_3(T_2)^{-1}$ as $P_2(P_1)^{-1}=P_3(P_4)^{-1}$, hence $\eta_t=1-\pi_c^{(1-k)k^\wedge(-1)}$. (2)

So the work of the ideal cycle is a function of two parameters π_c and τ, the efficiency is a function of only one parameter π_c. The higher is π_c, the higher is η_t. Another ideal cycle is a cycle of two isobars and two isotherms (cycle P-T or Zotikov cycle), Figure 5 (a, b). This ideal cycle is the limit situation of the cycle, Figure 5 (b), which corresponds to a GTU with increasing number of IHs and ICs. Figures 5 (a, b) show the difference between the works of these ideal cycles, $(H_{i.c.})_{J(Br)}<((H_{i.c.})_{zot}$. Figure 5 (c) shows the Carnot cycle for the same temperature level differences T_1=const and T_3=const. The Carnot cycle (T-S) 1g2e corresponds to the Zotikov cycle aocd. The Carnot cycle has the efficiency $\eta_{car}=1-T_1(T_3)^{-1}$.

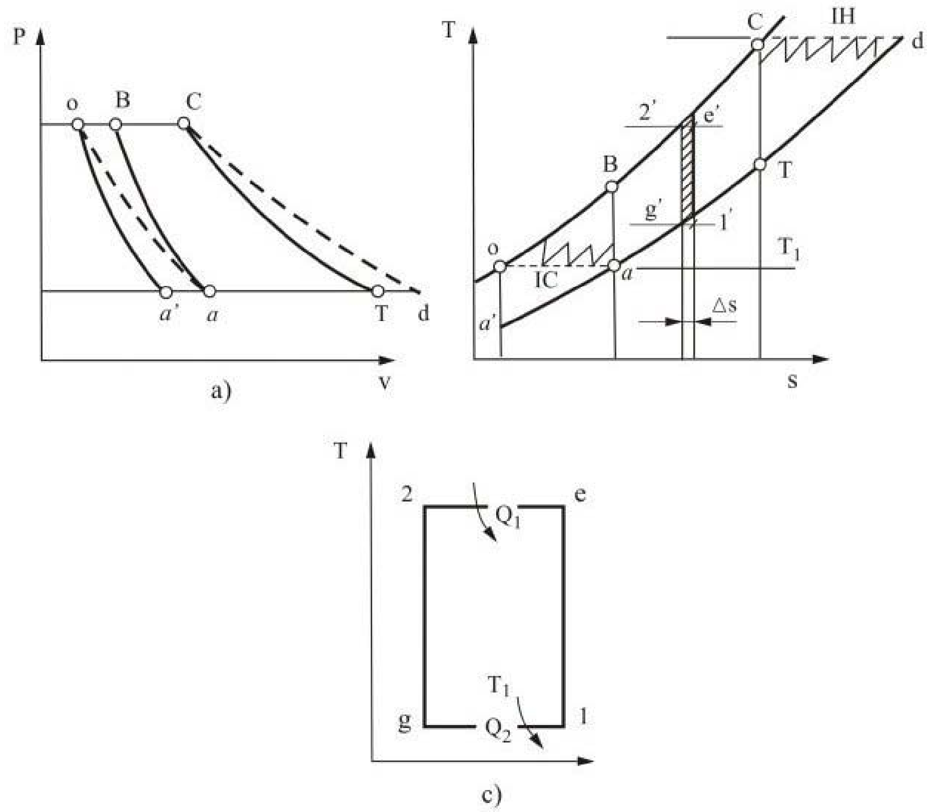

Figure 5 Ideal cycle with isothermal compression and expansion processes

The Brayton cycle efficiency is $\eta_{car}=1-\pi_c^{(1-k)k\wedge(-1)}=1-T_1(T_2)^{-1}$ or $\eta_{t(Br)}< \eta_{car}$.

If one considers an elementary cycle with isothermal compression $1^/g^/$, Figure 5 (b), with isothermal expansion $2^/e^/$ and use a well known approach to represent the cycle as the sum of elementary Carnot cycles with
$\eta_{el.car}=1-T_1^/(T_2^/)^{-1}$,

see Figure 5 (b), the efficiency of any cycle can be written as [7]

$$\eta_t=(\sum\Delta Q_{1\ el.car}\eta_{el.car})(\sum\Delta Q_{1\ el.car})^{-1} . \qquad (3)$$

If the gas temperature T_4, Figure 4, is higher than T_2 it is reasonable to use regeneration of the heat. The regeneration is estimated by the degree of regeneration r – the ratio of the actual warming to the limit one, so, see Figure 6,

$$r=(T_5-T_2)(T_4-T_2)^{-1} \qquad (4)$$

The simplest regenerative GTU cycle is shown on Figure 6, Figure 2 shows its scheme, line 4-6 is cooling of gas after the turbine, line 2-5 is the heating of air after the compressor. The ideal cycle with two ICs and two IHs is shown on Figure 7. Considering this cycle with increasing number of IC and IH one can get saw-shaped processes instead of the lines T=const.

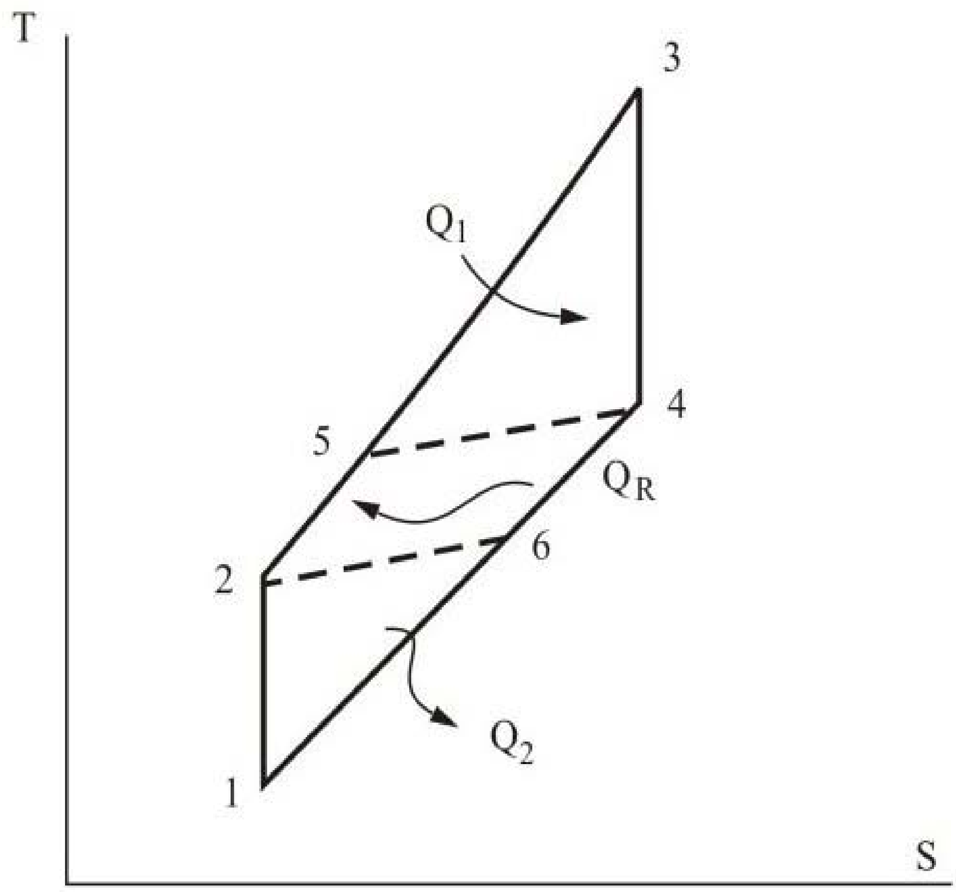

Figure 6 Ideal Joule or Brayton regenerative cycle

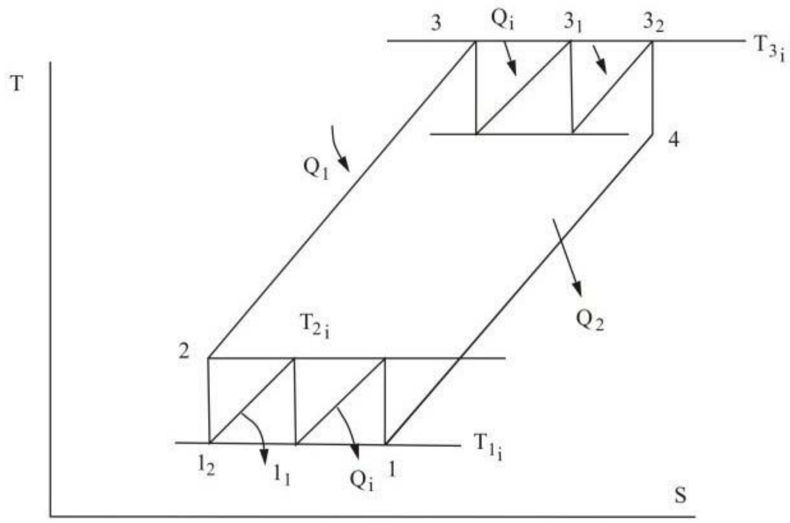

Figure 7 Ideal cycle with two intermediate coolers and two intermediate heaters

The efficiency of this cycle $\eta_t=(Q_1-Q_2)(Q_1)^{-1}$, where $Q_1=C_p(T_3-T_2)+ C_p(T_3-T_4)(\zeta -1)$, $Q_2=C_p(T_4-T_1)+ C_p(T_2-T_1)(z-1)$, where ζ is the number of turbines, z – the number of compressors. As a result of some rearrangements one can get

$$\eta_t=\{\zeta\tau[1-\pi_c^{(1-k)(k\zeta)^\wedge(-1)}]-z[\pi_c^{(k-1)(kz)^\wedge(-1)}-1]\}\{ \zeta[1-\pi_c^{(1-k)(k\zeta)^\wedge(-1)}] + [1-\pi_c^{(k-1)(kz)^\wedge(-1)}\tau^{-1}]-[1-\pi_c^{(1-k)(k\zeta)^\wedge(-1)}]\}^{-1}\tau^{-1} \quad (5)$$

where $\pi_c=p_2(p_1)^{-1}$ – the total pressure ratio of the cycle. For $z=\zeta=\propto$ the expression (5) gives the indeterminancy of $0(0)^{-1}$ type. Using the L'Hospital rule one can get instead of (5) the formula

$\eta_t=[(\tau-1)\ln\tau][\tau(1+\ln\tau)-1]^{-1}$, where $\tau=\pi_c^{(k-1)k^\wedge(-1)}$, which coincides with the efficiency of (T-P) cycle.

The (T-P) cycle with the regeneration r has the efficiency

$$\eta_t=[(\tau-1)\ln\tau][(\tau-1)(1-r)+ \tau\ln\tau]^{-1}, \quad (6)$$

for the case r=0 one can get the previous formula, for the case r=1 one can get a formula for the Carnot cycle (T-S) efficiency $\eta_t=\eta_{car}=1-(\tau)^{-1}$.

A detailed analysis of the GTU cycles led to an isothermic-isentropic-isobaric cycle (T-S-P) or Uvarov cycle, Figure 8, (a), which is fairly close to the Carnot cycle. The Uvarov cycle has a practical value of the pressure ratio $[p_2(p_1)^{-1}]$ in comparison with the Carnot cycle $[p_a(p_b)^{-1}]$. The line $11^{/}2$ – compression with intermediate cooling, the line $33^{/}4$ – expansion with intermediate heating. Sometimes this approach is referred to as cycle carnotization.

For the ideal cycle, Figure 8, (b), the efficiency η_t and the work of the ideal cycle $H_{i.c.}$ are correspondingly:

η_t=1-[ln$\pi^{(k-1)k^{\wedge}(-1)}_{isot\ c}$]$(\tau-x)^{-1}$, $H_{i.c.}=C_pT_1[\tau-x-\ln\pi^{(k-1)k^{\wedge}(-1)}_{isot\ c}]$, (7)

where $\tau=T_3(T_1)^{-1}$, $x=T_2(T_1)^{-1}$, $\pi_{isot\ c}=p'_1(p_1)^{-1}$. The cycle, Figure 8, (c), 1acb is a binary cycle, its efficiency η_t and work $H_{i.c.}$ are:

$\eta_t=1-(\tau-1)(\tau\ln\tau)^{-1}$, $H_{i.c.}=C_pT_1[\tau\ln\tau-(\tau-1)]$, (8)

where $\tau=T_3(T_1)^{-1}$. The cycle, Figure 8, (d), 1acb, is a ternary cycle. The efficiency η_t and work $H_{i.c}$ of the cycle, Figure 8, (e), are respectively:

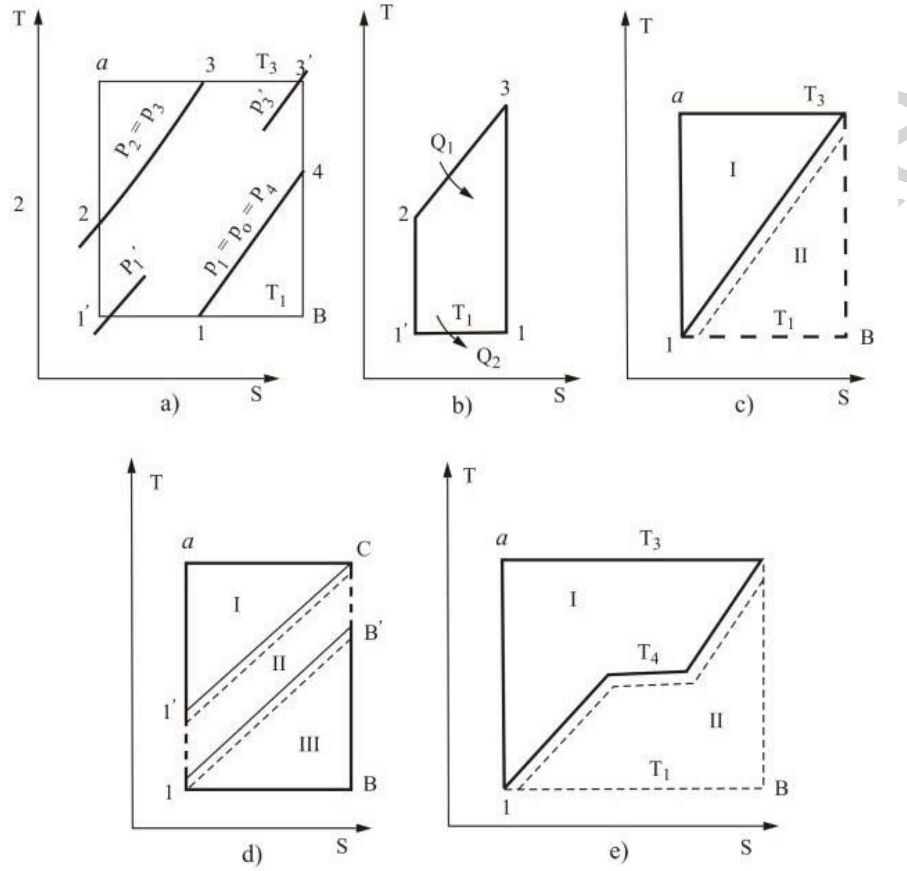

Figure 8 Ideal Carnot cycle (a), ideal Uvarov cycle (b)

$\eta_t=\{(\tau-\tau_1)\ln\pi_{\Sigma c}^{(k-1)k^{\wedge}(-1)}+\tau_1\ln\tau-(\tau-1)\}\{[\tau\ln\pi_{\Sigma c}^{(k-1)k^{\wedge}(-1)}\}^{-1}$,
$H_{i.c.}=C_pT_1\{(\tau-\tau_1)\ln\pi_{\Sigma c}^{(k-1)k^{\wedge}(-1)}+\tau_1\ln\tau-(\tau-1)\}$, (9)

where $\tau_1=T_4(T_1)^{-1}$, $\tau=T_3(T_1)^{-1}$, $\pi_{\Sigma c}=p_a(p_c/)^{-1}$

It should be noted that the cycle, Figure 8, (e), or its variant with a cutout left upper corner is preferable to a combined binary steam-gas turbine design for the Rankine cycle. One should also bear in mind that the temperatures T_{3i}, T_{4i}, T_{1i}, T_{2i}, Figure 7, being variable, can be optimized by the designer.

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Biographical Sketch

Victor S. Beknev graduated from the Moscow Bauman State Technical University (1951) and the Lomonosov Moscow State University (1954). He is the author of more than 160 publications in the field of fluid mechanics, turbomachinery and gas turbine engines of different applications. Dr. Beknev has delivered a number of papers at ASME meetings.