STIRLING ENGINE

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Summary

The thermodynamic basis of Stirling engine's action, its main constructional schemes, control systems and also the history of its development, its present-day situation, and future perspectives are presented briefly in this chapter.

1. Introduction

Stirling engines are heat engines with external supply of heat that operate under the closed regenerative cycle. The engine was named after the Scottish clergyman R. Stirling who was the originator of regenerative heat exchangers. In 1816 he invented a first regenerative engine with a closed cycle that works by the hot air. In 1818 the engine was constructed and it was used for pumping water out of stone quarries, but the engine's working cycle was open.

2. Thermodynamics

Thermodynamic cycle forms the basis of the engine's action. The cycle is shown in Fig.1 in $P-V$ and $T-S$ coordinates and it is also named after R. Stirling. To understand the basic principles of the engine's action it is better to examine it in detail.

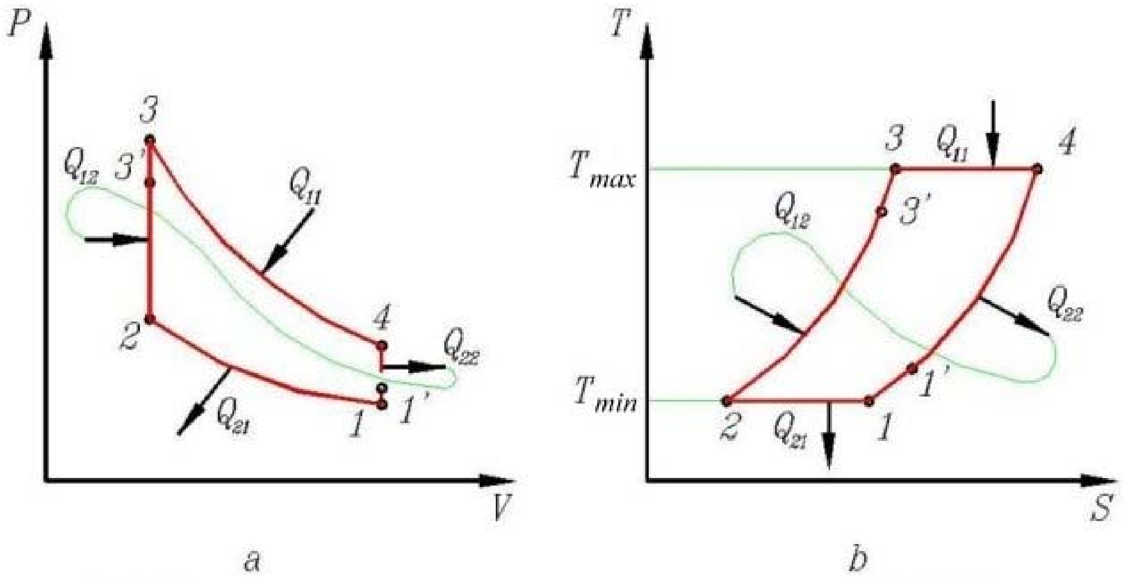

Figure 1. Thermodynamic cycle of Stirling engine

The cycle consists of isothermal compression (1-2) and expansion (3-4) processes and also of isochoric processes of heat's regenerative input 2-3 and output 4-1. The isothermal compression occurs at a low temperature (T_{MIN}) and it is accompanied by the output to the heat absorber (cooler) Q_{21}. The isothermal expansion takes place at a high temperature T_{MAX} with the input from the heat from the heater Q_{11}. As a rule, heat exchangers of recuperative type are used as a heater and a cooler.

Since the working fluid cools down from T_{MAX} to T_{MIN} in the process 4-1, and it heats from T_{MIN} to T_{MAX} in the process of heat input 2-3, the quantity of the output heat Q_{22} in the process 4-1 can be regenerated; in other words it can be received by heat accumulator packing and return to the working fluid in the process 2-3 of its heat.

In case of curves 4-1 and 2-3 equidistat, the heats Q_{22} and Q_{12} will be equal. Then the working fluid's temperature in the cycle will increase from T_{MIN} to T_{MAX} without the use of external energy source's heat and it will decrease from T_{MAX} to T_{MIN} in the process 4-1 without heat output to the surroundings. We can achieve it if we build regenerative heat exchanger into the engine design. In this process regenerative heat exchanger, heater and cooler are considered to be integral parts of any variant of a Stirling engine.

The useful work of such a cycle will be determined by the difference between input and output heats Q_{11} and Q_{21} and the cycle's area. The value of thermal efficiency will be determined only by T_{MAX} and T_{MIN} values, as it is in Carnot's ideal cycle. Thus Stirling's cycle has got the highest possible efficiency in complete reversible regenerative heat exchange side by side with Carnot's cycle. It doesn't depend on the various types of working fluids and it is determined by a well-known relation:

$$\eta_{St} = \eta_K = 1 - \frac{T_{MIN}}{T_{MAX}} \tag{1}$$

The cycle's work and mean cycle's pressure will be determined by the following equations:

$$L_t = \eta_{St} \cdot R \cdot T_1 \cdot \ln\varepsilon , \tag{2}$$

$$P_t = P_1 \cdot \frac{\varepsilon}{\varepsilon - 1} \cdot \frac{\ln\varepsilon}{\tau} \cdot \eta_{St} , \tag{3}$$

where: R - gas constant; ε - volume compression ratio, equal to the ratio of volume max to volume min; τ - T_{MIN} to T_{MAX} cycles ratio.

Theoretically, the existence of a great number of regenerative thermodynamic cycles with isothermal compression and expansion processes is possible. Here the value of thermal efficiencies will be equal to the value of efficiency of Carnot's cycle, where the difference of T_{MAX} and T_{MIN} is given. For this, regenerative heat exchange process in the cycle must be reversible and equidistant, affording the equality of stored up and given up heats. Cycle of Ericsson can be taken as an example. This cycle differs from cycle of Stirling. In cycle of Ericsson the isochoric processes are replaced with isobaric one's. It is necessary to notice that, a lot of attempts of practical realization of such a cycle were made, but they failed to create a heat engine with high efficiency.

For values of polytropic index of regenerative processes 4-1 and 2-3, falling in the range from $-\infty$ to 0, the cycle's work will be determined by the equation:

$$L_t = (1-\tau) \cdot P_{\max} \cdot V_{\max} \cdot \frac{\tau^{\frac{1}{n-1}}}{\varepsilon} \cdot \ln\left(\frac{\varepsilon}{\tau^{\frac{1}{n-1}}} \right) , \tag{4}$$

where: P_{MAX} and V_{MAX} - maximum values of pressure and volume.

Let us find the volume compression ratio ε_L, corresponding to the max value of cycle's work. We'll equate to zero the first derivative of the equation (4) and get the following equation as a result:

$$\varepsilon_L = e \cdot \tau^{\frac{1}{n-1}} \tag{5}$$

For Stirling cycle $n = -\infty$ and the value ε_L will be equal to the natural logarithmic base e. We will get the same result while examining the range of polytropic index of regenerative process change from 0 to $+\infty$. In other words, the cycle's maximum in Stirling's cycle can be reached in a low value of compression ratio, in which connection it is considerably lower than the values the other famous thermodynamic regenerative cycles have.

From a practical point of view, the advantages of engines without large volume compression ratio are obvious. It gives an opportunity to have sufficient volume for placement of heat exchangers with relatively small exchanger surfaces. It will help to minimize hydraulic losses in overflowing of the working fluid. As a result, a medium increase of the pressure ratio in the cycle will provide a fair change of loads to the components of engines design.

Thus, the ideal regenerative cycle of Stirling engines can be characterized as a cycle with maximum possible thermal efficiency and low values of compression ratio and medium increase in the pressure ratio. It favorably distinguishes it from the Carnot's cycle. The Carnot's cycle is the cycle consisting of two isothermal processes and two isentropic processes. Its realization requires very high values of maximum pressure in the cycle. If we apply heat regeneration process for any cycle, it will inevitably lead to internal losses, because heat exchange process in the regenerator is irreversible. To maintain heat transfer in the regenerator, it is necessary to have a finite difference of temperatures between the working fluid giving back (receiving) heat and the heat stored regenerator's packing, receiving (giving back) the heat.

That is why the working fluid's temperature at the end of regenerative cooling process $T_1^{'}$ will be higher than T_1. But at the end of regenerative heating up process $T_3^{'}$, it will be lower than T_3. It will be necessary to carry off the heat to the surroundings, where the heat will be proportional to the area under the curve $1'-1$ (Fig.1b) and also to supply the working fluid with an additional quantity of heat, proportional to the area under the curve $3-3'$ (Fig.1b). As a result the cycle's work will not change, but the thermal efficiency will decrease.

Thus, carrying out the regenerative heat exchange in Stirling cycle inevitably will lead to the appearance of specific source of losses and, as a result, lead to the internal irreversibility. The less the value of the input heat in the cycle Q_{11}, the stronger the influence of heat loss on the reduction of cycle efficiency.

For any regeneration degree μ_R (as a quantitative performance of deviation of non-ideal process of regeneration from the ideal) the equation for determination cycle of Stirling efficiency will be the following:

$$\eta_{St}^{*} = \frac{\eta_{St}}{1 + \dfrac{1-\mu_R}{\gamma - 1} \cdot \dfrac{\eta_{St}}{\ln \varepsilon}} \tag{6}$$

where: η_{St} is determined from equation (1), and the regeneration degree μ_R - from the following equation:

$$\mu_R = \frac{T_3^{'} - T_2}{T_3 - T_2} = \frac{T_4 - T_1^{'}}{T_4 - T_1} \qquad (7)$$

It is obvious that when μ_R is less than unity, the thermal efficiency depends on the kind of the working fluid. In real engines of Stirling the value of regeneration, as a rule, exceeds 0.9. The increase of ratio of specific heat of the working fluid (γ) leads to the increase of the value of cycle's thermal efficiency. That is why in modern Stirling engine configurations, the working fluids with a low molecular weight (hydrogen, helium) are preferable. It also gives an opportunity for the same value of cycle's work to use less of the working fluid in the internal engine's outline. The last mentioned circumstance helps to reduce hydraulic losses during the working fluid flows across the head exchangers.

For practical realization of isochoric regenerative processes, R. Stirling advised to place two reciprocating pistons in one of the cylinders, whose displacements are mutually synchronized. Later on, one of these pistons was called a power piston, and the other - a displacer piston. This allowed dividing functions of working fluid's displacement, compression and expansion between pistons. In this case the main purpose of a displacer piston is to move the working fluid from the hot end of the cylinder to the cold one across the heat exchangers and vice versa.

A small amount of work is required, since the pressure difference between the two sides of the displacer piston is insignificant. Besides, the displacer piston performs the function of a heat barrier between hot and cold spaces of the cylinder. The cylinder's head is at a high temperature and the opposite end - at a low temperature.

Let us examine the conditions of practical realization cycle of Stirling in such a design, schematically shown in Fig. 2.

At the beginning of the process of isothermal compression 1-2, the displacer piston (1) is kept stationary at its top dead center, and the power piston (2) starts its movement from its bottom dead center to the top dead center. In this case, the total volume occupied by the working fluid is reduced and the temperature T_2 in the cold space (3) is maintained constant by rejecting the heat to the cooler space (4), that extracts during the process of heat compression to the cooler space .

Further, in the isochoric process of heat input 2-3 to the working fluid in the regenerator (5), the power piston (2) is kept stationary at the top dead center, and the displacer piston (1) is moving towards its bottom dead center. The total volume occupied by the working fluid remains constant.

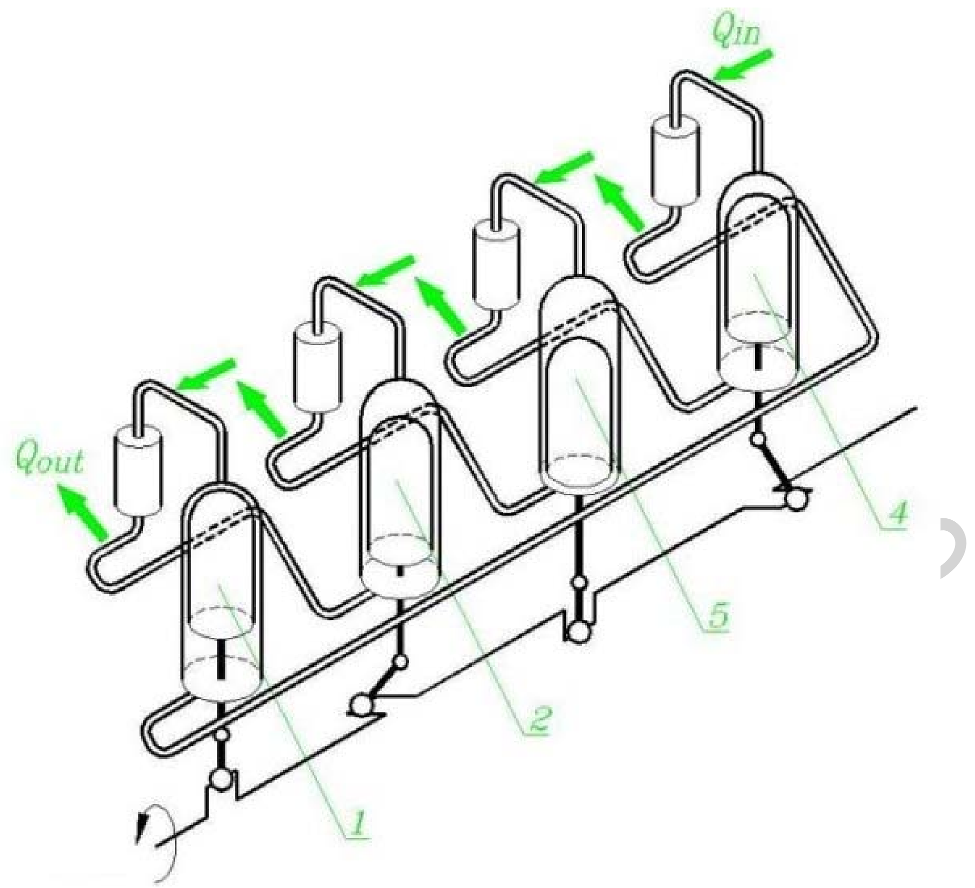

Figure 2. The simplified scheme of Stirling engine's work.

During the isothermal process of expansion 3-4, the power piston (2) and the displacer piston (1) are moving jointly towards their bottom dead centers and the total volume occupied by the working fluid increases. In this case the hot space temperature is maintained constant and equal to T_1. It can be done by using the heat input to the working fluid in the heater (7).

And, finally, in the process of heat output 4-1 from the working fluid in the regenerator (5), the power piston (2) is kept constant at the bottom dead center, and the displacer piston (1) is moving towards its top dead center, displacing the working fluid from the hot space (6) to the cold one (3). In this case, total volume occupied by the working fluid, remains constant.

Since, it can be noticed that for realization of thermodynamic Stirling's cycle piston's moving must be not only mutually coordinated but discontinuous and in spite of this, while the displacer piston must lead the power piston in phase, moving in order to fulfill working fluid's displacement to the corresponding spaces, having maintained the heat input and output during the necessary periods of time.

At present there is no information about driven devices that could implement the discontinuous reciprocating piston's motion; that is why practical realization of the thermodynamic cycle of Stirling is not possible.

In real designs of engines of Stirling the volumes of hot and cold spaces and the total working volume vary smoothly and continuously and for providing the law of mass distribution between spaces, necessary for work's production, the volume's change of a hot space leads in time the corresponding volume's change of a cold space.

Besides, compression and expansion processes in reality, are not isothermal and the heat regeneration takes place during the total working engine's volume changes; in other words it does not represent the isochoric process.

In the ideal thermodynamic cycle of Stirling the heat exchangers of the internal loop (the heater, volumes of the regenerator and the cooler) are not taken into account. It is assumed that the whole working fluid mass is divided between the cold and hot spaces.

The mass also takes part in every process consistently and the process makes up a cycle that includes compression (regenerative heating) and expansion (regenerative cooling).

That is why the thermodynamic process, proceeding in the engine's total working volume, can not be shown graphically in the thermodynamic coordinates. We also can not show the thermodynamic process, happening separately in each of the five mentioned spaces because their working fluid masses are different in time.

That is why it will be better to speak about engines, working according to 'Stirling '.

The influence of the great number of factors (besides the factors mentioned above, there are the non-isothermal of expansion and compression processes, finite and little time of cycle's realization, non-ideal property of the working fluid, heat losses in components of engine's design, non-gas-tight seals and etc.) causes considerable difference of the cycle of the engine's work from the ideal thermodynamic cycle of Stirling and above mentioned equations for the efficiency and for the work (Equations 1-6) can not represent a complete rule of how the process works.

In fact, from Eq.(6) follows continuous increasing of the efficiency with rising of volume compression ratio; though in practice efficiency optimum is reached in quite specific volume compression ratio, a bit less than two.

For the heat engine, working according to the closed cycle (in contrast to internal combustion engine) the thermodynamic cycle that can equivalently describe the working process has not still been suggested, though attempts were made time and again.

In particular, some investigators (Berchowitz D.M., Gross M., Rallis C.J., Reader G.T., Urielli I. and others) proposed several alternative schemes of the thermodynamic cycle's realization. It appears that the most successful are the cycles, that are shown in coordinates $P-V$ (Fig.3), The first was called 'Pseudo-Stirling cycle', and the second - 'cycle of Rallis'.

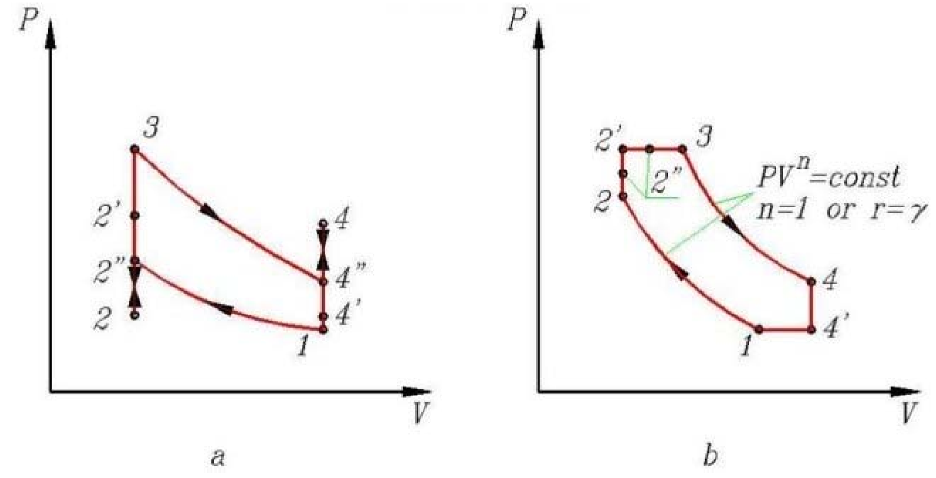

Figure 3. Pseudo-Stirling cycle and cycle of Rallis.

In Pseudo-Stirling cycle, the isothermal process are replaced by polytropic ones and in the limit - by isentropic ones. Ideal isothermal process $1-2$ is substituted by the adiabatic process $1-2''$ and the isochoric process of the following cooling $2''-2$. The analogous process $3-4$ is replaced by the adiabatic one $3-4''$ and by the isochoric heating $4''-4$. The regenerative working fluid's pre-heating starts at the point 2 and finishes at the point $2'$. The regenerative cooling starts, accordingly, at the point 4 and finishes in the point $4'$.

It was found, that the results obtained in the process of Pseudo-Stirling cycle's employment (Fig.3a) are in conformity with natural laws and real engines features. In particular, for the given temperature ratio and regeneration degree, the thermal efficiency possible can be reached when the value of compression ratio is quite specific, in which case, the efficiency and the volume compression ratio are close to the real values.

However, the artificiality of such an approach causes a certain element of non-compliance. If the presence of heat exchangers is taken into account, the transfer of energy in cold and hot spaces of changing volume in real engines is small in comparison with the transfer of energy in mentioned heat exchangers. Interchanging gas process in working spaces will be similar to adiabatic ones or in the ideal case - to isentropic processes. In addition, it is established experimentally that the engine's heat exchange occurs not only in isochoric but also in isobaric conditions (Tresca H.). That is why the cycle, proposed by Rallis C.J. and consisting of six processes will be, obviously, physically trustworthy (Fig.3b).

Thermal efficiency volumes for Pseudo-Stirling cycle and for Rallis cycle give comparable values in for small (close to the actual) values of volume ratio. It should be remembered that the ideal cycles, however successful they may be for one or another heat engine, can be useful only in the capacity of approximate models real working conditions. They scarcely can be a basis for accurate analytical models of working process calculation. In particular, for different configurations of Stirling engine there are different versions of thermodynamic cycle.

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Biographical Sketch

Gaivoronsky Alexander Ivanovich (borne in 1958) graduated from Bauman Moscow Higher Technical School in 1981 (speciality "Piston and Combined Engines"). Assoc.Professor "Piston Engines" Department of the Bauman Moscow State Technical University, Ph.D.. Autor of more than 40 publications in the field of heat engines and power installations.