

# Quasi Isothermal Heat Engine for Concentrating Solar Power Systems

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**Abstract** Recently we developed new type of solar concentrators, which have parabolic dish surface approximated with flat facet mirrors. To create the solar power plant on the base of these concentrators it is necessary to develop efficient heat engine for the temperature difference that can be obtained with new concentrators. For this purpose we propose to create quasi isothermal heat engine based on the Ericsson cycle and designed as rolling piston engine. We present in this paper two types of such engines: one-valve heat engine and valve-less heat engine. The theoretical analysis shows that for high temperature 600 K and low temperature 300 K the thermal efficiency can be obtained 0.46-0.49 for one-valve heat engine and 0.36-0.46 for valve-less heat engines for compression rate 1.2-1.05. To obtain acceptable power-two-weight ratio of the engine it is necessary to maintain the mean pressure of the cycle equal to 100-200 bar as it is made in some Stirling engines.

**Keywords:** *Ericsson cycle engine, quasi isothermal heat engine, rolling piston heat engine, solar power plant, solar energy*

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## 1. Introduction

Recently we developed new type of solar concentrators, which have parabolic dish surface approximated with flat facet mirrors. One prototype of this concentrator is shown in Figure 1. These concentrators have lower weight and must have lower cost than traditional parabolic dish concentrators [1,2,3,4].



Figure 1. Solar concentrator prototype

Solar power plant based on developed solar concentrators needs the energy storage system to obtain continuous supply of electrical energy. There are different systems of energy storage based on the batteries, storage of mechanical energy (compressed air, water elevated to the high level, etc.), thermal energy storages, and others. In general, thermal energy storages (TES) are cheaper than other energy storages. For this reason it is worthwhile to

use TES with low cost solar concentrators to obtain cheap electrical energy. There are some types of TES. At present molten salt is frequently used to store thermal energy for some hours. To store energy for larger periods the molten salt is too expensive. If it is needed another methods are used. One of them is connected with phase change materials that collect the energy and release it at the constant temperature. The cheapest TESs usually are based on the simple heating and cooling of different materials. These TESs are termed sensible TESs. To transform electrical energy to thermal energy and restore electrical energy from thermal energy it is necessary to use external combustion heat engines. For phase change materials usually they use Brayton cycle gas turbines and Rankine cycle vapour turbines [5]. These types of heat engines work well if the temperature difference between hot and cold parts of TES is relatively constant. If temperature difference varies in large diapason it is better to use "hot air" engines, such as Stirling engines [6], and Ericsson engines [7,8,9]. The Brayton cycle gas turbines and Rankine cycle vapour turbines are used in large scale power plants [5]. For a small or medium scale power plant the Stirling engines are proposed [6].

The main advantages of the Stirling engine are: a relatively simple design with small number of moving parts, an absence of the valves, an absence of the noise, the possibility to obtain high thermal efficiency, long life time. The main drawbacks are: the Stirling engine needs high temperature difference between hot and cold sources (the hot source temperature can be 700 – 800° C). It demands to use expensive special materials. The Stirling

engine has a heat regenerator installed in the internal engine space that increases the engine dead volume and decreases its efficiency. Moreover the heat regenerator internal position makes it difficult the Stirling engine lubrication. It is possible to eliminate the main drawback of the Stirling engine using the Ericsson cycle heat engine. Detailed analysis of this engine and comparison with the Stirling engine is given in [7]. The Ericsson cycle engine properties are described in [8,9]. The Ericsson cycle makes it possible to remove the heat recuperator from the engine internal space that decreases the dead volume and permits us to increase the heat recuperator sizes. Large size recuperator has better parameters than small size one.

The Ericsson cycle makes it possible to create a quasi isothermal heat engine that has a level of efficiency close to the ideal (Carnot) cycle [10,11].

Two examples of Ericsson heat engines are presented in [12]. It is shown that good results may be obtained if the engine uses a microchannel heat recuperator. An example of a microchannel recuperator is described, and the scheme of calculation of the recuperator parameters is presented. The quasi isothermal heat engine (QuiHE) described in [12] has a reciprocating structure. In this article, we propose two types of QuiHEs based on the rolling piston principle of design. This principle permits us to decrease the number of valves (one valve for one cylinder or even valve-less cylinder) and to simplify the general structure of the engine.

Two types of rolling piston QuiHEs are described. For the first type, which we term the one-valve rolling piston QuiHE, each cylinder of this engine contains one valve for air flow control. For the second type, which we term the valve-less rolling piston QuiHE, no valves are used. The

analysis and comparison of both types of QuiHE are presented.

Recently, rolling piston compressors and expanders have been extensively studied [13,14,15,16,17]. The advantages of the rolling piston approach are the simple structure and manufacturing process, high mechanical efficiency, and absence of reciprocating moving parts. Moreover, the rolling piston approach has a good possibility to seal the internal space of the engine. These advantages make it possible to develop low cost QuiHEs for solar thermal power plants.

To begin the development of new QuiHE it is necessary to make rough estimations of efficiency of proposed engines. To make this rough estimation we use simple thermodynamical equations. This permits us to make qualitative comparison of the proposed engines and select one of them for prototyping. In the future more detailed analysis can be made to evaluate the influence of friction, hydraulic and other losses. This analysis will be based on finite difference method. The final solution about the properties of the proposed engine can be made on the base of the experimental investigations of the prototypes. We plan to fulfil these works in the future.

## 2. Rolling Piston Expanders and Compressors

Recently many types of rolling piston expanders and compressors are proposed [18,19,20]. In this article we consider two types of rolling piston expanders and compressors.

The scheme of a one-valve rolling piston expander is shown in Figure 2.

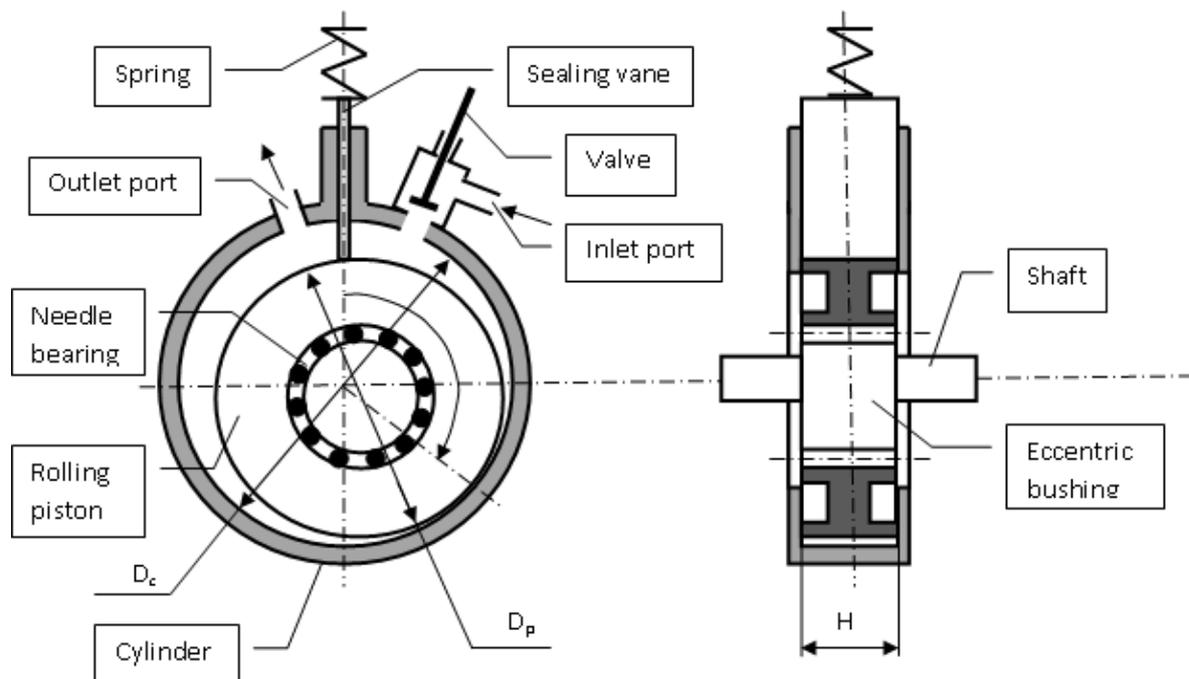


Figure 2. One-valve rolling piston expander

A one-valve rolling piston expander contains a rolling piston that is eccentrically placed in a cylinder. This piston is supported by an eccentric bushing and needle bearing. The eccentric bushing is rigidly connected to the rotating shaft. When the shaft is rotating, the rolling piston rolls on the internal surface of the cylinder. The sealing vane

pressed to the rolling piston with a spring divides the internal free space into two parts: the right part and the left part. When the valve is open, the right part is filled with compressed air due to the inlet port. Simultaneously, the air of the left part is exhausted via the output port. The pressure difference of the right and the left parts produces

the mechanical moment that moves the rolling piston in the clockwise direction. At a certain moment of the cycle, the valve is closed and the air in the right part is slightly expanded to achieve the pressure of the exhausted air.

The compressor has the same design as the expander, but the direction of rotation of the compressor is opposite

to the rotation of the expander. Correspondingly, the outlet port of the expander serves as the inlet port of the compressor, and the inlet port of the expander serves as the outlet port of the compressor. The simplest heat engine can consist of a one-cylinder compressor and a one-cylinder expander (Figure 3).

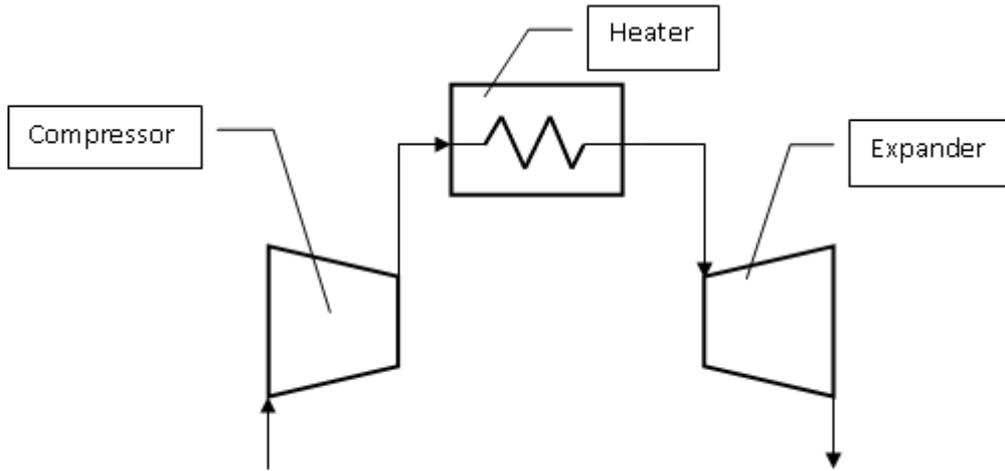


Figure 3. The simplest heat engine

The air from the atmosphere inputs into the inlet port of the compressor. The compressed air travels to the heater and proceeds to the inlet port of the expander. The expanded air travels from the outlet port to the atmosphere. The p-v diagram of this cycle is shown in Figure 4.

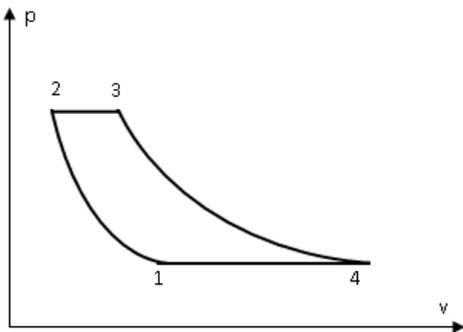


Figure 4. p-v diagram of the cycle

The figure contains the adiabatic compression line 1-2, the isobaric expansion line 2-3, and the adiabatic expansion line 3-4. Air at the input of the compressor can be considered to be isobaric cooled atmosphere air exhausted from the outlet port of the expander. Thus, the cycle contains the isobaric line 4-1.

In this case, we obtain a typical Brayton cycle. To obtain an quasi isothermal cycle heat engine, it is necessary to transform the processes presented by lines 1-2 and 3-4 from adiabatic to isothermal (more precisely quasi isothermal). For these purposes, we use a multi-cylinder expander/compressor. The shaft with eccentric bushings for a 4 cylinder expander/compressor is shown in Figure 5. Each eccentric bushing serves as an installation of one rolling piston placed into one cylinder. The design of all of the rolling pistons and cylinders is the same for all cylinders, excluding the value of the cylinder height  $H$  (see Figure 2).

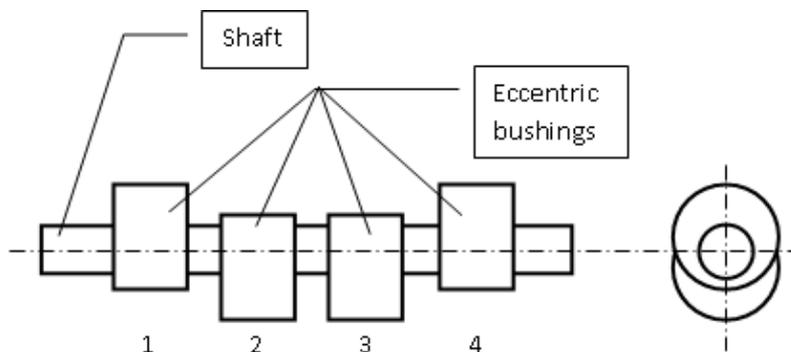


Figure 5. Shaft for a 4 cylinder expander/compressor

The values of the parameter  $H$  for the expander satisfy the conditions:

$$H_1 < H_2 < H_3 < H_4. \tag{1}$$

For the compressor, we have inverse conditions:

$$H_1 > H_2 > H_3 > H_4. \tag{2}$$

To obtain a dynamically balanced expander/compressor, the mass of the rolling piston, including the needle bearing and the eccentric bushing, is equal for each cylinder. The number of cylinders can be more than 4, for example, 6, 8, 10, 12, and so on. The air pass in the multi-cylinder expander is shown in Figure 6, and the air pass in the multi-cylinder compressor is shown in Figure 7.

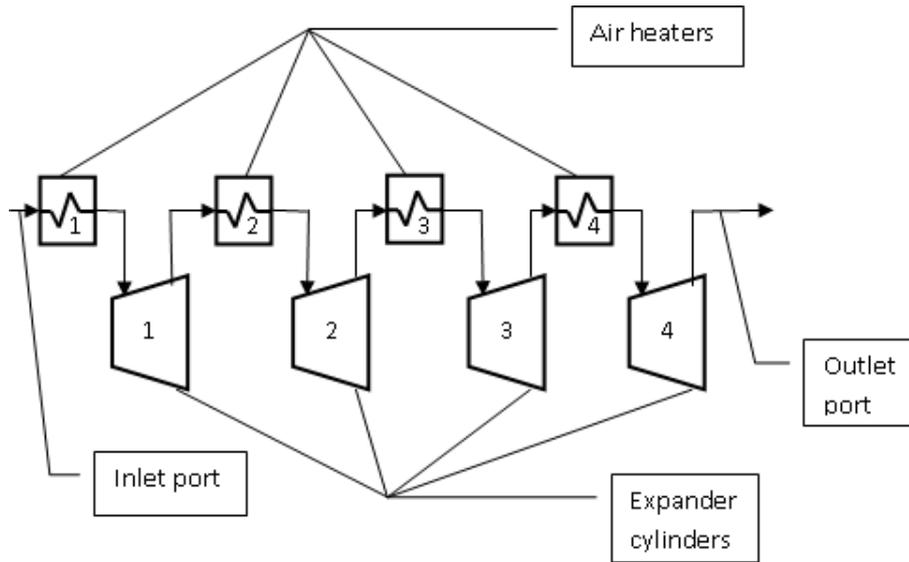


Figure 6. Air pass in a multi-cylinder expander

In a multi-cylinder expander, the air enters the inlet port and is heated in heater 1 to the highest temperature  $T_H$ . Next, the air goes to expander cylinder 1, where it is adiabatically expanded. The temperature of the air slightly decreases to  $T_{H1} < T_H$ . From cylinder 1, the air travels to heater 2, where it is restored to the highest temperature  $T_H$ . This process is repeated in the subsequent cylinders and heaters. The expanded air exits through the outlet port.

In a multi-cylinder compressor, the air enters through the inlet port and is cooled in cooler 1 to the lowest temperature  $T_C$ . Next, the air travels to compressor cylinder 1, where it is adiabatically compressed. The temperature of the air slightly increases to  $T_{C1} > T_C$ . From cylinder 1, the air travels to cooler 2, where it restores the lowest temperature  $T_C$ . This process is repeated in the subsequent cylinders and coolers. The compressed air exits through the outlet port.

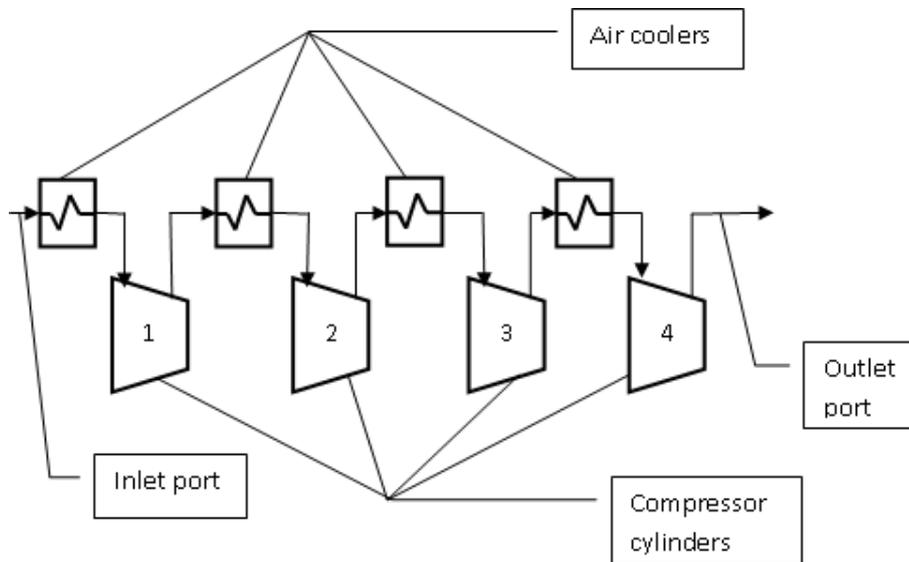


Figure 7. Air pass in a multi-cylinder compressor

### 3. One-valve Quasi Isothermal Heat Engine

The quasi isothermal heat engine (QUIHE) is shown in Figure 8. The engine contains a multi-cylinder compressor, recuperator, and multi-cylinder expander. The compressed air travels from the compressor to the first inlet port of the recuperator, where it is heated from temperature  $T_{C1}$ , which is slightly higher than the lowest temperature  $T_C$ , to the temperature  $T_H^*$ . At this temperature, the air travels to the inlet port of the multi-cylinder expander, where it is

expanded. The shafts of the multi-cylinder expander and the multi-cylinder compressor are connected, and the portion of the work produced by the multi-cylinder expander is used to rotate the multi-cylinder compressor. The excess work produced by the multi-cylinder expander is output for user purposes. The air from the outlet port of the multi-cylinder expander, which has the temperature  $T_{H1} < T_H$ , is input into inlet port 2 of the recuperator, where it is cooled to temperature  $T_C^*$ , which is slightly higher than the lowest temperature  $T_C$ . With this temperature, the air travels to the inlet port of the multi-cylinder compressor.

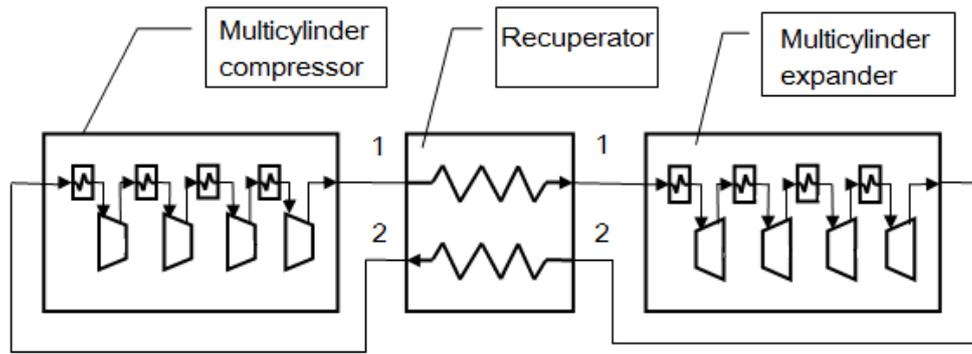


Figure 8. Quasi isothermal heat engine (QuiHE)

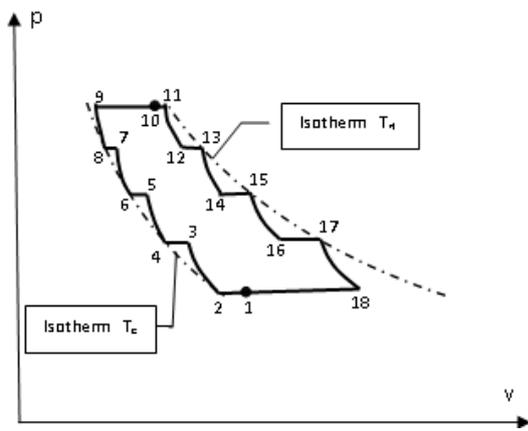


Figure 9. p-v diagram of the multi-cylinder 1-valve heat engine

In the multi-cylinder compressor, the air is compressed with the process that approximates an isothermal compression. In Figure 9, isothermal compression with temperature  $T_C$  is shown with a dashed line. The real compression process is shown with a bold line 1-2-3-4-5-6-7-8-9. It is easy to see that the bold line 2-3-...-9 represents a good approximation of isotherm  $T_C$  (dashed

line) if the number of cylinders in the multi-cylinder compressor is large.

From the last cylinder of the multi-cylinder compressor, the air travels to the recuperator, where it is heated at an approximately constant pressure (bold line 9-10 in Figure 9). Next, the air enters the multi-cylinder expander, where the expansion process approximates the isothermal process  $T_H$ . In Figure 9, this process is shown with the bold line 10-11-12-13-14-15-16-17-18. The expanded air from the multi-cylinder expander enters the second part of the recuperator, where it is cooled at an approximately constant pressure (bold line 18-1 in Figure 9). As in the case of the multi-cylinder compressor, we can improve the quality of the isotherm  $T_H$  approximation by increasing the number of cylinders in the multi-cylinder expander.

#### 4. Valve-less Rolling Piston Heat Engine

Valve-less rolling piston compressors and expanders permit us to simplify the heat engine design and to decrease the number of moving parts. Let us consider the valve-less rolling piston expander (Figure 10).

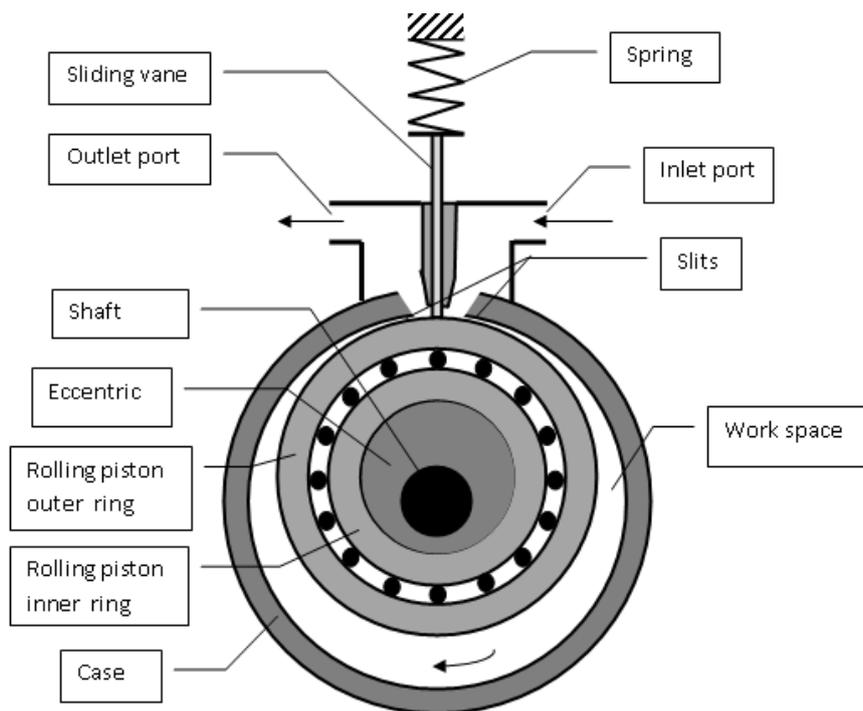


Figure 10. Valve-less rolling piston expander

This expander is similar to the one-valve rolling piston expander (Figure 2), but it contains no valve in the inlet port and the positions of the inlet and outlet ports are placed as close to each other as possible. The corresponding rolling piston compressor has the same design. The gas pressure of the expander in the inlet port is higher than the pressure in the outlet port. The rolling piston compressor has a gas pressure of the inlet port lower than the pressure of the outlet port, which is a practically unique difference between the compressor and the expander.

Let us consider the p-v diagram of the engine that contains only one expander and one compressor (Figure 11). In this diagram, the cycle of the one-valve engine is shown with the lines 5-2-6-4-5. Lines 5-2 and 2-6

correspond to the gas processes in the gas expander. Line 5-2 corresponds to the time period when the inlet valve is open, and line 2-6 corresponds to the time period when the inlet valve is closed. Lines 6-4 and 4-5 correspond to the gas processes in the compressor. Line 6-4 corresponds to the open valve, and line 4-5 corresponds to the closed valve. In Figure 11, the cycle of an ideal isothermal engine is shown with lines 1-2-3-4-1. Lines 1-2-3 correspond to the expander, and lines 3-4-1 correspond to the compressor. In theory, this engine can be achieved with an infinitely slow rotation of the compressor and the expander. If the engine recuperator works without losses, then this cycle corresponds to the ideal Carnot cycle. Therefore, we term the engine with the cycle 1-2-3-4-1 as a Carnot engine.

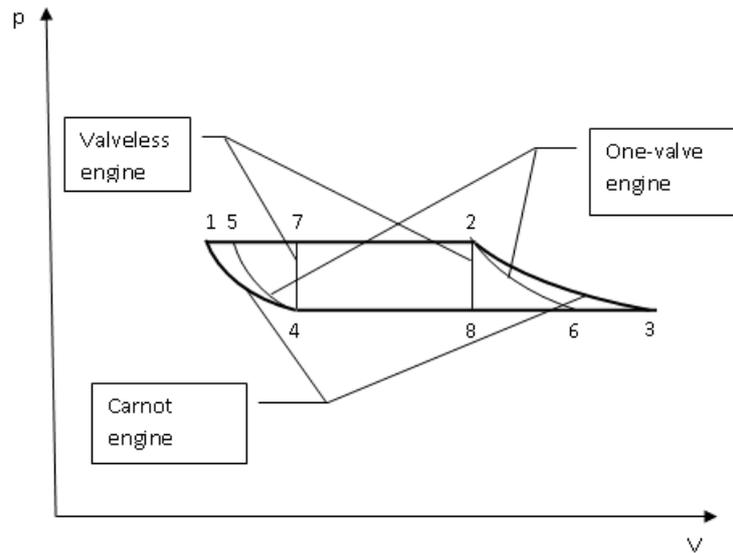


Figure 11. p-v diagram for three types of rolling piston heat engines: Carnot heat engine, one-valve heat engine, and valve-less heat engine

The valve-less engine has one significant difference from the one-valve engine and the Carnot engine: it has a constant pressure in the work space throughout almost all of the cycle, and at the end of cycle, the outlet port of the expander or compressor is opened very rapidly. The discharge process in this engine can be shown with corresponding vertical lines: 2-8 in the expander and 4-7 in the compressor. Therefore, the cycle of a valve-less engine can be shown in p-v diagram with the lines 7-2-8-4-7. The work of the cycle that corresponds to area 7-2-8-4-7 is less than the work of both the one-valve engine area (5-2-6-4-5) and the area of the Carnot engine (1-2-3-4-1). This lower amount of work is the first drawback of the valve-less engine.

The second drawback is the following:

At the upper position of the rolling piston, for a short period of time, two slits appear (Figure 10) that connect the inlet port directly with the outlet port and the portion of the gas passes from the inlet port to the outlet port not producing useful work. This portion must be small because the slits are thin and the time period is short. The losses of energy produced by these slits must be evaluated. If the losses will be significant, then the second version of the rolling piston engine design, shown in Figure 12, must be used.

In this version, the width of the inlet port and the outlet port is equal to the width of the engine cylinder, and the rolling piston “falls in” the outlet and inlet ports so that

the slits do not appear. The drawback of this design is the knocking that is generated with this process. The intensity of the knocking must be evaluated; if the knocking is louder than the knocking of the internal combustion engine valves, another solution to this problem must be sought. One of the solutions is to use lubricating oil to seal the slits in the first version of the engine (Figure 12).

Let us analyse the correlations among the parameters of the Carnot engine, the one-valve engine and the valve-less engine. We consider two parameters as independent variables: the compression/expansion pressure rate  $\gamma$  and high-to-low temperature rate  $\varepsilon$ . In our calculations, we denote the pressure at the  $i$  point of the diagram as  $p_i$ , and we denote the volume of the gas at the  $i$  point of the diagram as  $v_i$ . To simplify our calculations, we accept that  $p_1 = 1 \text{ Pa}$  and  $v_1 = 1 \text{ m}^3$ . In this case, for calculation of the parameters of the Carnot engine, we have the following equations.

$$p_2 = p_1 = 1 \text{ Pa} \tag{3}$$

$$v_2 = \varepsilon \cdot v_1 = \varepsilon \left( \text{m}^3 \right) \tag{4}$$

$$\varepsilon = \frac{T_h}{T_l}, \tag{5}$$

where  $T_h$  is the highest temperature of the cycle and  $T_l$  is the lowest temperature of the cycle.

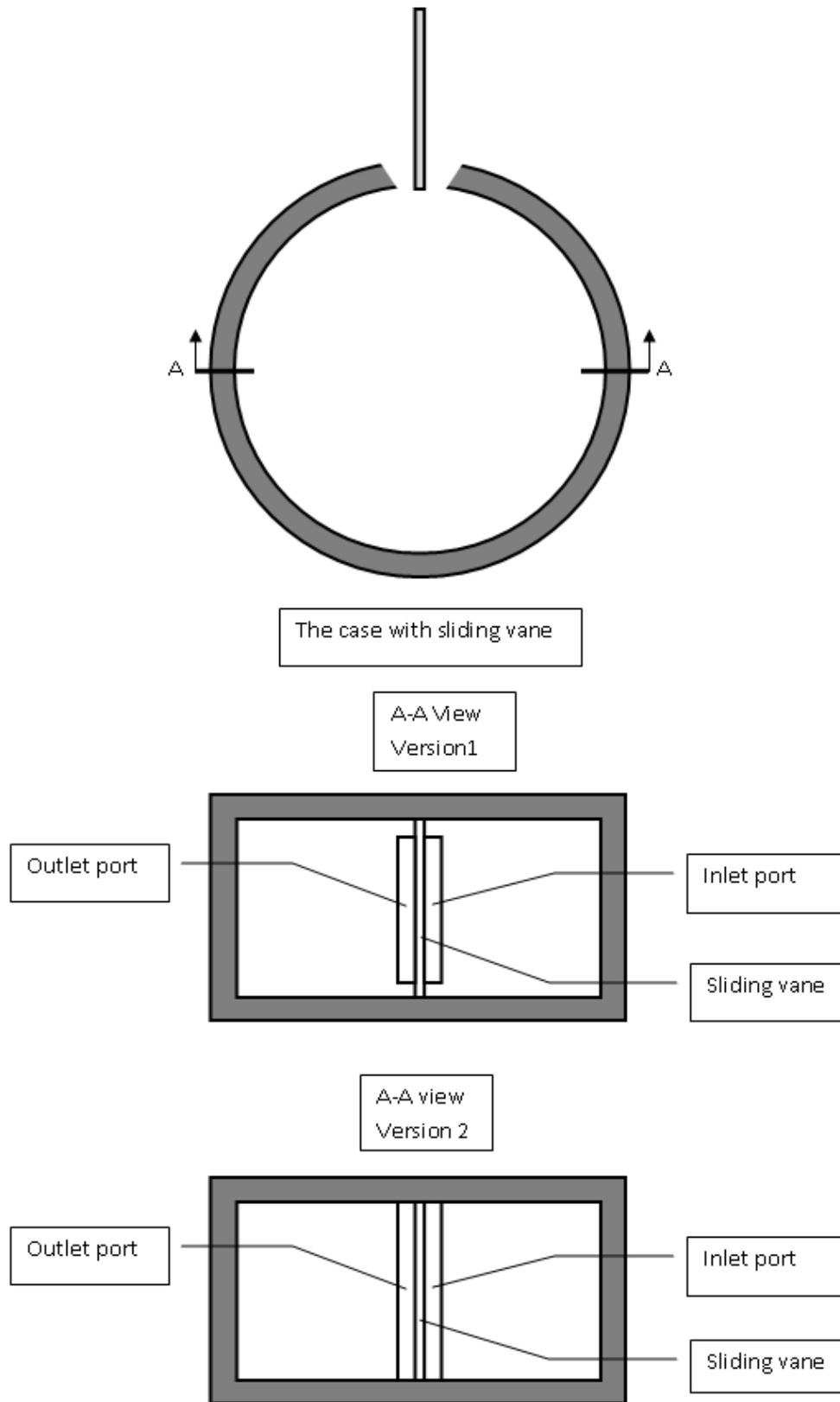


Figure 12. Two versions of the valve-less rolling piston heat engine design

$$p_3 = \frac{p_2}{\gamma} = \frac{1}{\gamma}, \tag{6}$$

$$v_4 = \frac{v_3}{\epsilon} = \frac{v_2 \cdot \gamma}{\epsilon} = \gamma. \tag{9}$$

where  $\gamma$  is the compression/expansion pressure rate.

$$v_3 = v_2 \cdot \gamma, \tag{7}$$

$$p_4 = \frac{1}{\gamma}, \tag{8}$$

The useful work  $A$  of each period of the cycle equals to the area of the  $p$ - $v$  diagram between the corresponding line of the diagram and the  $v$ -axis. For the Carnot engine, we have:

$$A_{1-2} = p_1 \cdot (v_2 - v_1), \tag{10}$$

$$A_{2-3} = p_2 \cdot v_2 \cdot \ln \frac{v_3}{v_2} \quad (11)$$

$$A_{3-4} = p_3 \cdot (v_4 - v_3) \quad (12)$$

$$A_{4-1} = p_4 \cdot v_4 \cdot \ln \frac{v_1}{v_4} \quad (13)$$

For the one-valve engine, we can use the following equations for the parameter calculations.

$$p_6 = \frac{p_2}{\gamma} = \frac{1}{\gamma}, \quad (14)$$

$$p_5 = p_1 = 1Pa, \quad (15)$$

$$v_5 = \frac{v_4}{\gamma^{1/k}} = \frac{\gamma}{\gamma^{1/k}} = \gamma^{\frac{k-1}{k}}. \quad (16)$$

The values of the useful work are as follows:

$$A_{5-2} = p_5 \cdot (v_2 - v_5) = v_2 - v_5 \quad (17)$$

$$A_{2-6} = \frac{\varepsilon}{k-1} \left[ 1 - \left( \frac{v_2}{v_6} \right)^{k-1} \right], \quad (18)$$

$$A_{6-4} = \frac{1}{\gamma} \cdot (v_4 - v_6) \quad (19)$$

$$A_{4-5} = \frac{1}{k-1} \left[ 1 - \left( \frac{v_4}{v_5} \right)^{k-1} \right] \quad (20)$$

Valveless engine.

$$p_7 = p_1 = 1Pa \quad (21)$$

$$v_7 = v_1 \cdot \gamma = \gamma \quad (22)$$

$$p_8 = \frac{p_1}{\gamma} = \frac{1}{\gamma} \quad (23)$$

$$v_8 = v_2 = \varepsilon \quad (24)$$

Useful work:

$$A_{7-2} = p_7 \cdot (v_2 - v_7) = \varepsilon - \gamma \quad (25)$$

$$A_{2-8} = 0, \quad (26)$$

$$A_{8-4} = p_8 \cdot (v_4 - v_8) = \frac{1}{\gamma} (\gamma - \varepsilon) \quad (27)$$

$$A_{4-7} = 0. \quad (28)$$

Useful work for the following engines:

Carnot engine

$$A_{Carnot} = A_{1-2} + A_{2-3} + A_{3-4} + A_{4-1} \quad (29)$$

One-valve engine

$$A_{OneValve} = A_{5-2} + A_{2-6} + A_{6-4} + A_{4-5} \quad (30)$$

Valve-less engine

$$A_{Valveless} = A_{7-2} + A_{2-8} + A_{8-4} + A_{4-7} \quad (31)$$

These equations were used to calculate the parameters of the one-valve and the valve-less engines. The results are presented below.

## 5. Relative Efficiency

We term the relationship of the work of cycle  $X$  to the work of the Carnot cycle as the relative efficiency of cycle  $X$  and denote it as  $\eta_r^{(X)}$  (sometimes the term “second law efficiency” is used in this case).

One-valve engine:

$$\eta_r^{(One-valve)} = \frac{A_{One-valve}}{A_{Carnot}} \quad (32)$$

$$\eta_r^{(Valveless)} = \frac{A_{Valveless}}{A_{Carnot}} \quad (33)$$

Using these equations, we calculated the relative efficiencies of the one-valve and the valve-less engines for different values of the independent variables  $\gamma$  and  $\varepsilon$ .

The variable  $\varepsilon$  has three different values: 2, 1.5, and 1.25. This variable depends on the hot temperature at the engine input and the low temperature at the engine output. For example, a solar thermal power plant containing solar concentrators that give a temperature of 327 °C (600 K) and a cooler with 27° C (300K) has  $\varepsilon=2$ . If a solar thermal power plant has thermal energy storage (TES) based on hot water with a high temperature of 90° C (363K) and a cooler of 17° C (290K), then the engine that works with this TES has  $\varepsilon=1.25$ .

The variable  $\gamma$  depends on the engine design. The smaller values (1.05, 1.1) correspond to the engines that have high pressure inside the external casing. The higher values (1.15, 1.2) can be accepted for engines with smaller pressure inside the external casing.

The results of the calculations are presented in [Table 1](#).

In [Table 1](#) we present also an example of absolute thermal efficiency of two engines. For the case  $\frac{T_{High}}{T_{Low}}=2$ ,

when lower temperature  $T_{Low}$  is 27° C (300 K) and higher temperature  $T_{High}$  is 327° C (600 K) we have Carnot

efficiency  $\eta_C = 0.5$ ; for the case  $\frac{T_{High}}{T_{Low}}=1.5$ ,  $T_{Low}=300$  K,

$T_{High}=450$  K and  $\eta_C=0.33$ ; for the case  $\frac{T_{High}}{T_{Low}}=1.25$ ,  $T_{Low}$

$=300$  K,  $T_{High}=375$  K and  $\eta_C=0.2$ . Two last columns

show the absolute efficiency for the both engines. It is necessary to mention that the temperature difference  $T_{High}$

$- T_{Low}$  is maintained not with compression ratio (that is true for internal combustion engines) but with work of recuperator, heaters and coolers that are installed between

compression and expansion stages of an engine. Theoretically, this engine has to work with mechanically uncompressible working fluid that has a thermal

expansion property. The real gases used as working fluids have approximately these properties if they have very high mean pressure of the cycle but low pressure difference

(compression rate). For this reason this type of engine

sometimes is placed in high pressure vessel to obtain mentioned conditions. For example, some Stirling engines work with mean pressure of the cycle about 100 – 200 bar

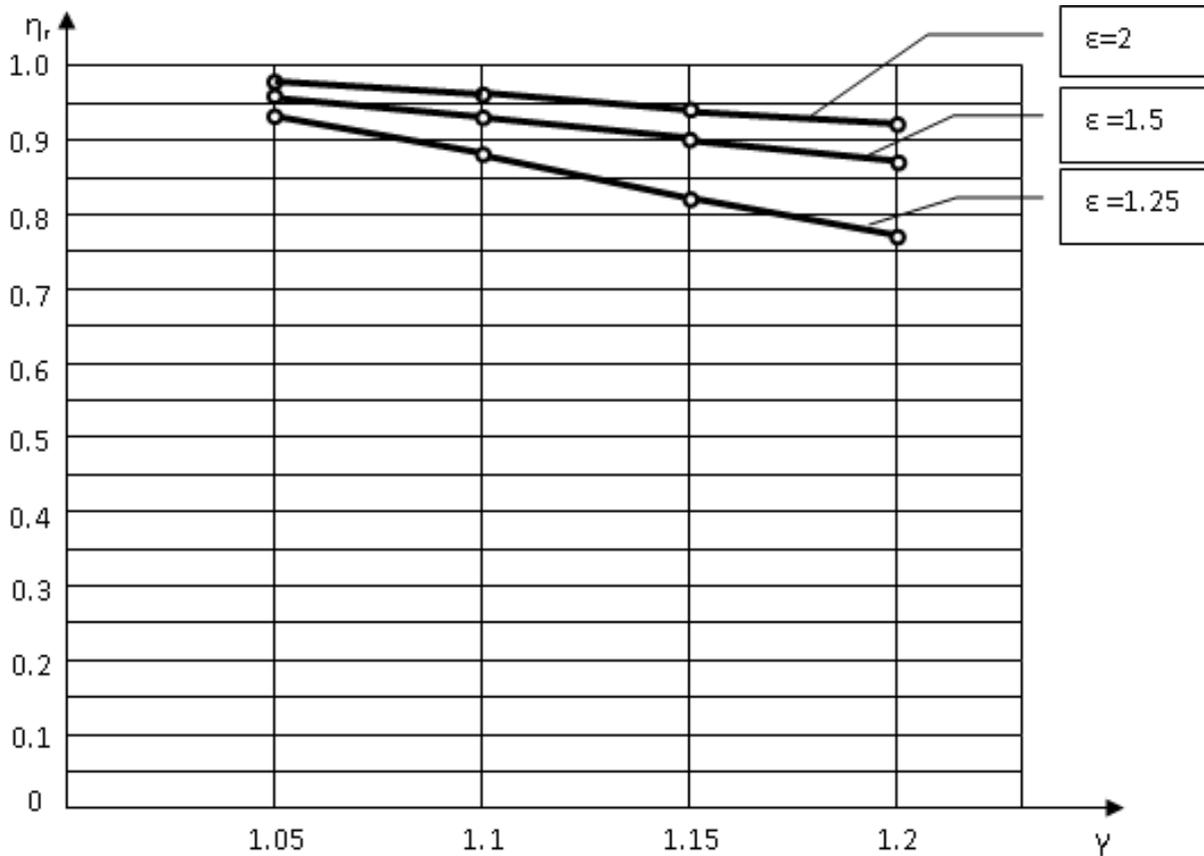
and have pressure differences only 10-20 bar. Similar conditions are convenient for Ericsson engine. New version of Table 1 is presented below.

**Table 1. Relative efficiencies of the one-valve and the valve-less engines**

$\gamma = \frac{P_{High}}{P_{Low}}$	$\varepsilon = \frac{T_{High}}{T_{Low}}$	Thermal relative efficiency $\eta_r$		Absolute thermal efficiency $\eta_{abs}$	
		One-valve engine	Valve-less engine	One- valve engine	Valve-less engine
1.05	2	0.979	0.927	0.49	0.46
1.1	2	0.959	0.858	0.48	0.43
1.15	2	0.940	0.793	0.47	0.4
1.2	2	0.922	0.727	0.46	0.36
1.05	1.5	0.964	0.878	0.32	0.29
1.1	1.5	0.932	0.763	0.31	0.25
1.15	1.5	0.900	0.653	0.3	0.22
1.2	1.5	0.870	0.548	0.29	0.18
1.05	1.25	0.936	0.780	0.19	0.16
1.1	1.25	0.878	0.572	0.18	0.11
1.15	1.25	0.821	0.373	0.16	0.07
1.2	1.25	0.766	0.182	0.15	0.04

The corresponding curves are presented in Figure 13 and Figure 14. The values presented in Table 1 and in

Figure 13 and Figure 14 correspond to the thermal relative efficiency.



**Figure 13. Relative efficiencies of the one-valve engine**

To obtain the final relative efficiencies, it is necessary to multiply these values by the mechanical efficiency of the rolling piston engine and the efficiency of the heat recuperator. The mechanical efficiency of the rolling

piston engine is accepted to be 0.93 (see [21,22]), and the efficiency of the heat recuperator is accepted to be 0.96. In Table 2, we present the final relative efficiencies of the one-valve and the valve-less engines.

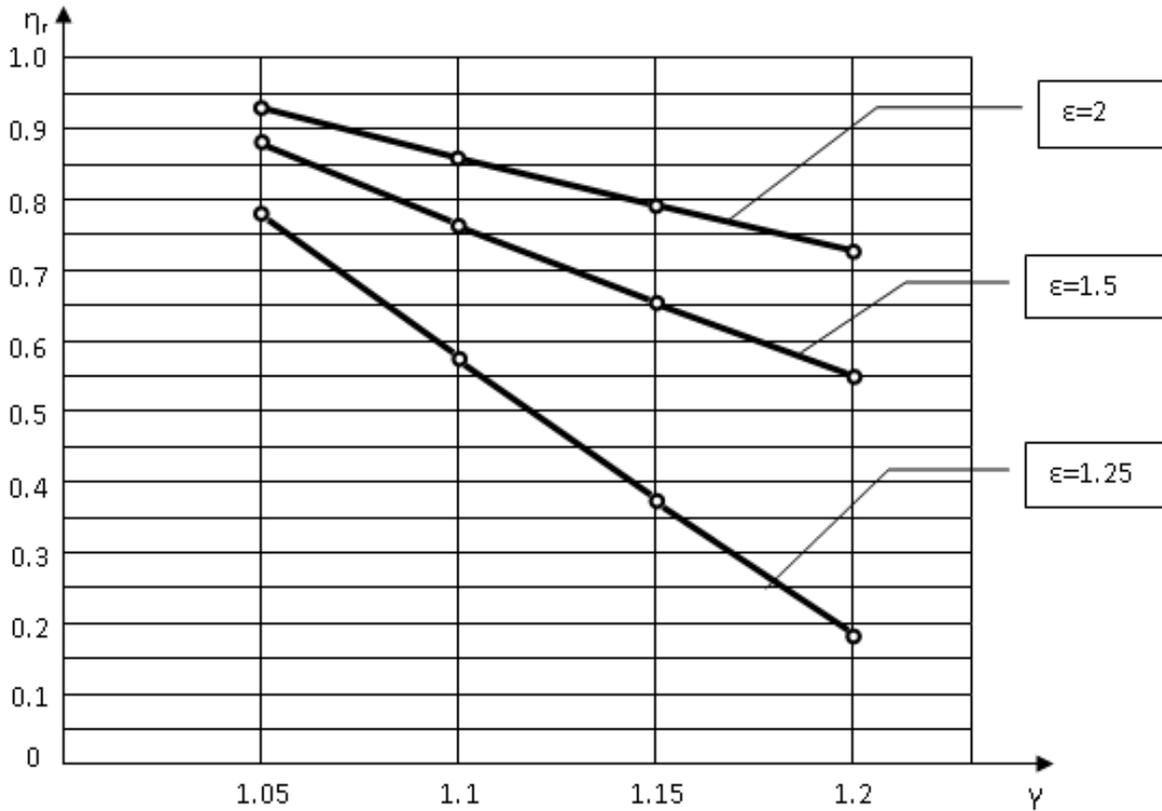


Figure 14. Relative efficiencies of the valve-less engine

Table 2. Final relative efficiencies of the one-valve and the valve-less engines

$\gamma = \frac{P_{High}}{P_{Low}}$	$\varepsilon = \frac{T_{High}}{T_{Low}}$	Final relative efficiency $\eta_r$	
		One-valve engine	Valve-less engine
1.05	2	0.874	0.828
1.1	2	0.856	0.766
1.15	2	0.839	0.708
1.2	2	0.823	0.649
1.05	1.5	0.861	0.784
1.1	1.5	0.832	0.681
1.15	1.5	0.804	0.583
1.2	1.5	0.777	0.489
1.05	1.25	0.836	0.697
1.1	1.25	0.784	0.511
1.15	1.25	0.733	0.333
1.2	1.25	0.684	0.163

### 6. Discussion

Quasi isothermal heat engine for concentrating solar thermal power plants is proposed. Theoretical analysis of heat engine characteristics is made.

To obtain acceptable cost of each compressor or expander stage it is necessary to use high average pressure of the working cycle. For example, if the expander stage produces 1 KW at the average pressure of 1 bar, it will

produce approximately 100 KW at the average pressure 100 bar. The average pressure increasing can lead to the cost rise much less than 100 times. So, the Ericsson cycle engine permits us to obtain acceptable cost of compressor and expander stages using high average pressure of the cycle. The same explanation is correct for the Stirling cycle.

The number of compressor and expander stages for Ericsson cycle can be made as low as needed. For example, if we use only one compressor and one expander, the engine will also have the cycle very close to the Ericsson cycle. The thermal efficiency of the cycle practically does not depend on the pressure rate, but only on the thermal expansion rate in the recuperator and heater. The one-stage Ericsson engine is very similar to the Stirling engine but has no regenerator in internal space and must have, for this reason, better efficiency. The penalty for this advantage is the need of valves (in one-valve configuration) and more complicated design of heat recuperator in comparison with heat regenerator of Stirling engine, but the possibility to enlarge the sizes of heat recuperator makes it possible to decrease hydraulic losses in the gas channels and to increase the efficiency of the whole engine.

In this article we consider the relation  $\varepsilon = \frac{T_{High}}{T_{Low}}$  not more than 2. This means that for the ambient temperature  $T_{Low} = 300K$  the corresponding high temperature  $T_{High} = 600K$  or  $327^\circ C$ . In our case oxygen-free gases can be used as a work fluid. For these conditions synthetic oil can be used for lubrication without any problems.

The lubrication problems appear in the Stirling engines because in the Stirling engines it is needed to use high

values (up to 700 - 800°C) of the high temperature  $T_{High}$ . Elimination of the regenerator from internal space of the engine permits us to decrease  $T_{High}$  up to 300 -350° C because the sizes of heat exchanger in this case can be made almost as large as necessary without increasing the dead volume and decreasing the efficiency.

Special attention is paid to the relative efficiency of the engine. The final relative efficiency of the heat engine is very important for power plants that have thermal energy storage. In general, the energy input to the thermal energy storage can be realized with some types of heat pumps, and the energy output can be realized with a heat engine. The total input-output efficiency is the product of the final relative efficiencies of the input heat pump to the output heat engine. If we use the same engine for the input and the output devices, we have an input-output efficiency of the TES as a second power of the final relative energy efficiency of the heat engine. Thus, for example, the input-output efficiency of hot water thermal energy storage that has  $\varepsilon = 1.25$  and a compression rate of engine  $\gamma = 1.05$  (Table 2) is 0.836. The second power of this value is 0.699, that is, approximately 0.7. The corresponding input-output efficiency for electrochemical batteries is 0.8. The input-output efficiency of a conventional heat pump and heat engine is approximately 0.36. Therefore, the rolling piston heat engine with Ericsson cycle exhibits good potential to be used in solar power plants with thermal energy storage.

## 7. Conclusions

The theoretical analysis of two types of Ericsson heat engines indicated that it is possible to design these engines for solar power plants with thermal energy storage systems. The input-output efficiency of the storage based on hot water can be increased to 70%. The rolling piston engine that has one valve for each cylinder has advantages in comparison with the valve-less rolling piston engine from the point of view of its final relative efficiency. The valve-less engine has advantages from the point of view of design simplicity. Further analysis is necessary to define the areas of application for each engine type.

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