Simulation of Unconventional Thermal Engine

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Abstrakt
Na vývoji nové koncepce objemového motoru se presentuje použití simulačních metod v rané fázi konceptuálního výzkumu. Nový motor používá kroužívý píst, avšak vytváří jej spolu se statorem pomocí konchoidního pohybu, tedy ne jako v případě Wankelova motoru. Pětidobý cyklus byl vyvinut tak, že je realizován jak v objemových stupních, tak v objemech vymezených mezi nimi. Cyklus má vlastnosti převyšující současné cykly Stirlingových motorů. Přívod tepla realizovaný mezi druhým stupněm kompresoru a prvním stupněm expandéru probíhá ve vloženém výměníku za řízení objemu a tlaku. Expanze probíhá až na atmosférický tlak.

Abstract
The development of a new engine concept is presented as an example of the use of simulation methods in an early stage of design. A novel positive displacement engine component with a cycling piston using conchoid mechanism (unlike a Wankel engine) is used. The five-period cycle realized inside and between four positive displacement components has been developed. The cycle appears to be competitive to current Stirling engine cycles. The heat supply period can be realized in a heat Exchange at controlable pressure/volume conditions combining changes of volume at the second stage of a compressor side and the first of expander side. Expansion is performed to the atmospheric pressure.

Key words
New Cycling Piston Koncept, Period Cycle, 1-D model, Simulation, Waste Heat Recovery

1.Intoduction

The use of low-temperature heat has become important due to the ever increasing cost of primary energy sources. Both renewable sources or waste-heat utilization from other thermal engines require a simple and efficient thermal machine. It should use external heat supply and air or water steam as a working fluid for the sake of simplicity. High pressure and sufficient expansion ratio create conditions for reasonable thermal efficiency. The use of a reciprocating piston requires a clumsy, unbalanced and expensive cranktrain. Lubricating oil may cause other problems, especially if inlet temperature is higher than 450°C.
The paper is focused on simulation of a five period cycle realization in a positive-displacement engine with cycling piston motion, where the working spaces are created mostly between pistons, i.e., not between a piston and a stator (“cylinder”) of an engine. Atmospheric air has been used as a working fluid at this stage of development.

2. Thermodynamic principles

The ideal Carnot cycle cannot be achieved at limited compression and expansion ratios –[1]. Low heat-supply temperature calls for moderate compression ratio if heat should be supplied by convection in a heat exchanger. The heat rejection at ambient temperature and low temperature difference would require low pressure of the working fluid and big heat exchanger, especially if using air as working fluid.
The analysis of different thermodynamic concepts has resulted in the use of five period cycle, i.e., compression of fresh air between the 1st and 2nd stage to decrease temperature irreversibility during heat supply, heat supply in the 3rd stage between the 2nd and 4th stage, expansion to atmospheric pressure between the 4th and 5th stage and replacement of used air at constant pressure in the 5th stage by the inlet of fresh air into the 1st stage without the

requirement for a low-temperature heat exchanger. Pressure-volume and entropy-temperature diagrams are in Figure 1. The low specific work, due to low temperature of the working gas, is clear from comparison to an IC engine, the efficiency reduction due to isothermal heat supply (without compensation of the same process during heat rejection) is evident from the T-s diagram – [2], [3].

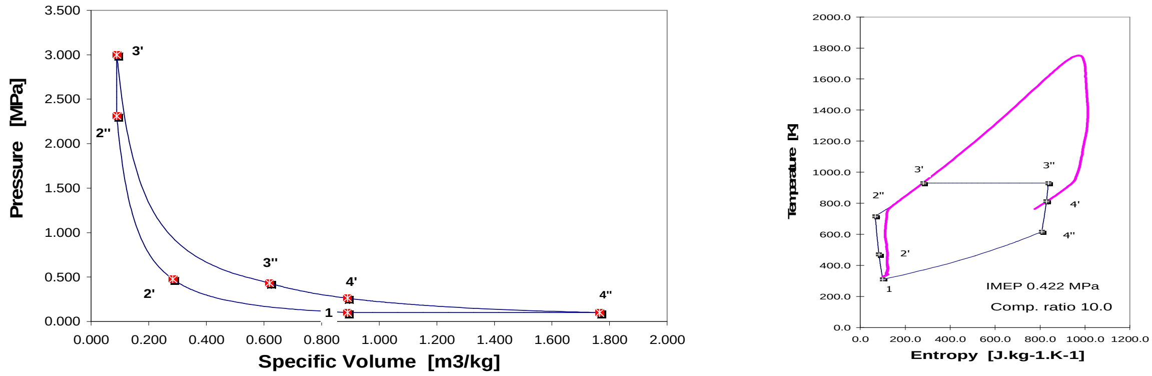

***Figure 1** p-V and T-s diagrams of five period cycle for IMEP of 0.422 MPa. Comparison to SI internal combustion engine of equivalent IMEP 0.85 MPa.*

The results of preliminary thermodynamic analysis have shown that such a cycle with combined isochoric/isobaric/isothermal heat supply and elongated expansion to atmospheric pressure has good potential to reach reasonable 30 – 40% of indicated efficiency.

3. Engine kinematics

A standard reciprocating piston engine is not relevant, evidently, due to contradiction between the requirements of small top-dead-center (TDC) volume at reasonable swept volume (high expansion ratio) and sufficient heat-exchanging surfaces. The solution would be a connection of several pistons using the volumes closed between the couples of them. The kinematic principle should offer small internal volumes at TDC and simple balanced design. This goal may be reached by cycling piston concept.

For the aim stated, the Wankel cycling-piston principle might be applied. Internal-toothed wheel connected to a trochoid-shaped displacer (piston) cycles, being guided by a crank, around standing external-toothed spur gear. For the requirements stated above the basic transmission ratio should be 2, i.e., the stator should feature the diameter of ½ of an internal cycling gear. In such a case, very small TDC volume may be created. The poles of immediate rotation lie at the cylindrical surface of the big internal gear, the connecting line between a pole and the center of a stator spur gear intersects always the center of an outer wheel with internal gears. The obvious disadvantage is the impossibility of a crankshaft use for several cycling pistons in a row, which is hindered by the stator spur gear.

Another solution has been found [4] using the feature of poles and the center of outer circle: conchoid generating principle (Figure 2). A straight line **k** guided by the center of above-mentioned outer cycling internal gear $\mathbf{X_2}$, replaces the gear. It is moved by crank **SO** replacing the previous inner spur gear radius with a center in the middle of **E** vertex. The perpendicular abscissa **p** of **R** half-length intersects **k** in **S** (the endpoint of a crank). It simultaneously always intersects the **O** $\mathbf{X_2}$ straight line in the point $\mathbf{X_1}$, **S** point lying in the crank radius distance from **OB.** The rectangular triangle $\mathbf{X_1X_2S}$ may move (generating cycling motion), whereas **S** stays at crank circle (Thales theorem) and straight line is guided, intersecting $\mathbf{X_2}$ all the time. The vertex **ρ** generates the outer ("cylinder") wall of working space then, which is very similar to the circle of radius **r:**

$$\rho = R - E \cdot \sin \varphi = R(1 - \lambda \cdot \sin \varphi); \quad \text{where} \quad \lambda = \frac{E}{R} \tag{1}$$

The displacer may be shaped above and below the abscissa **p** arc-wise to create a piston while these shapes can almost touch the stator upper wall in a TDC position (Figure 3).
The machine may feature very high internal compression ratio, it may be fully balanced and a common crank may be used for the row of linked cylinders. No expensive gears have to be manufactured.
The angular speed of rotation of abscissa **p** around an intersection $\mathbf{X_1}$ is the same like that of line **k** around point $\mathbf{X_2}$ but unlike angular speed of a crank. The crank rotates by a double angular speed (two crank revolutions are needed to bring back the line **p** to the initial position).
The geometry of a single-stage machine is presented in Figure 3. 3 parts of in-cylinder volume are visible as areas in this view: two working variable volumes and inner displacer area. The area of a piston $\mathbf{A_p}$ may be determined by simple integration.
The real piston has to be sealed in radial direction (towards cylinder wall) by straight-line lips, pre-loaded by radial oriented springs. Another possibility is to use a small gap after running-in the piston edges towards a cylinder. The lid planes has to be sealed by analogy to piston rings, forced to the lids by distributed flexible thrust components. A slight lubrication of lid surfaces is advisable.

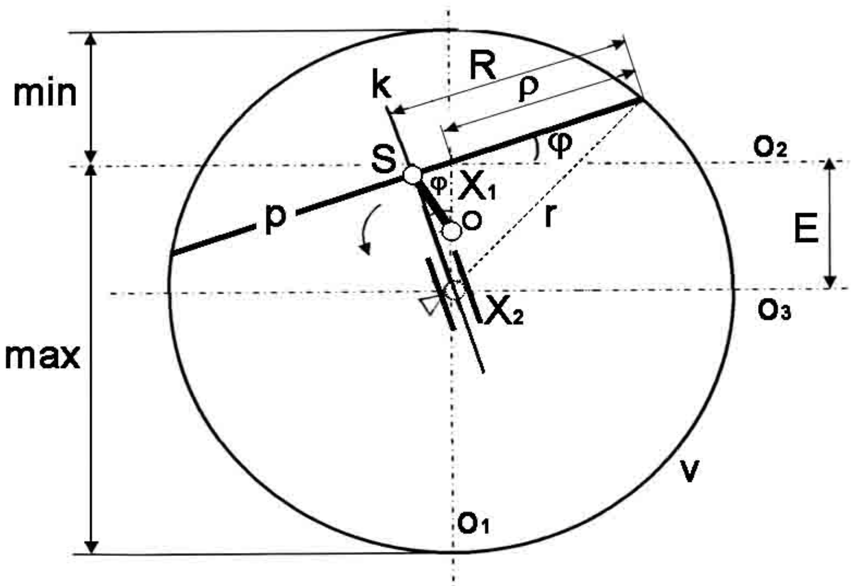

Figure 2 *The kinematic principle of a conchoid generation of a cycling double-acting piston/cylinder*

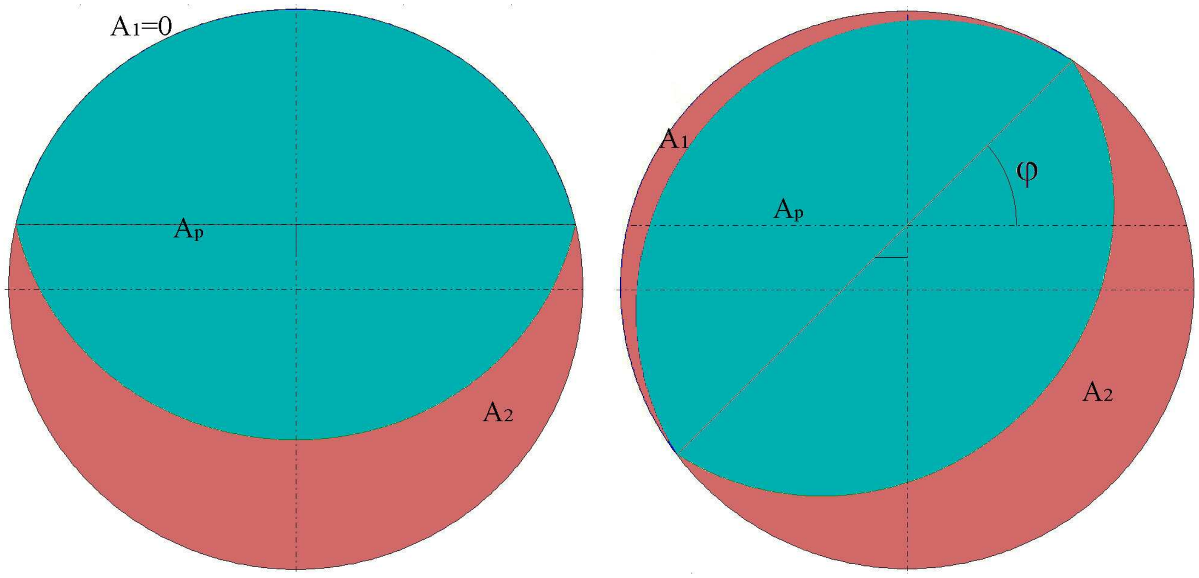

Figure 3 *Working spaces, at the right side in general position, at left the upper one in engine TDC and the other one in BDC*

4. Gas exchange ports

Two types of ports can be used at cycling piston machines, as shown in Figure 4, namely radial (in a cylinder wall) or axial (created in frontal lids of a working space). The radial ports feature a big advantage in speed of opening and closing because of the piston circumferential velocity. Moreover, the time of radial port opening lasts longer than 360° of crank rotation. If two radial ports are used overlapping period occurs: in the same time both of them are opened. It should be avoided if the pressures in both ports are not approximately the same otherwise throttling irreversibilities take place, which is often associated with intensive temperature increase at the side of low pressure. One-way (e.g., automatic reed) valves in at least one of the ports or a combination of radial and axial ports should be used in such a case. Another possibility is to de-axe the ports from the piston centerline in a TDC position to the top of the cylinder drawn in Figure 4. The short-circuit flow is sealed by a small gap between a piston and a wall in the TDC position then. All these possibilities have to be carefully evaluated taking undesired throttling into account.

The free areas of ports can be calculated using the geometry of a piston and cylinder, described above. Axial port area is best determined by numerical integration.

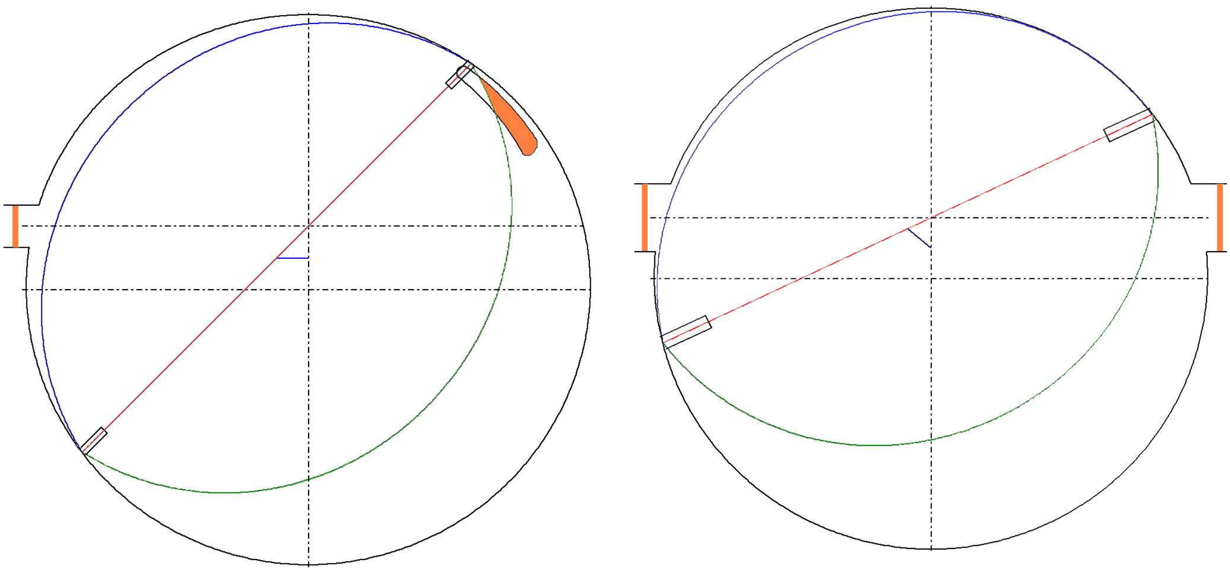

Figure 4 *Gas transfer ports: left combined radial and axial ones, right radial only*

5. Engine layout

The individual positive displacement machine described above can be used as an individual compressor or expander. The small minimum volume in a TDC piston position is of great advantage, limiting the volumetric efficiency drop at high pressure ratio of a compressor and throttling losses from pressure unbalance during admission period of expander.

In the case of five period cycle for low-temperature heat utilization, several machines should be combined to realize the individual changes – [5], Figure 5. The compressor 1 receives the air at atmospheric pressure. The compression is realized in the first stage and between the stages 1 and 2 after the port between them is opened - ***Figure 7***, Figure 9. At the end of compression the connection to a heat exchanger is opened and the air is transferred through the heat exchanger 3 to the first expander stage 4. Simultaneously, it is heated up with some pressure increase following, if a small change of volume between pistons 2 and 4 is present, which was not the case of engine simulated until now. The pressurized and heated air is transferred into the volume between pistons 4 and 5, where expansion follows. After transfer port 4-5 is closed, the rest of expansion is realized in 5. The exhaust of air to atmosphere finishes the whole process.

The good function of the engine depends on appropriate phasing of volume changes and optimized timing of all gas exchange processes. Especially in the case of the compact machine described here, the tuning of gas exchange in an already manufactured specimen would be extraordinary difficult because of the lack of adjusting tools (like adjustable cams, etc.). Therefore, careful simulation-based optimization is advisable.

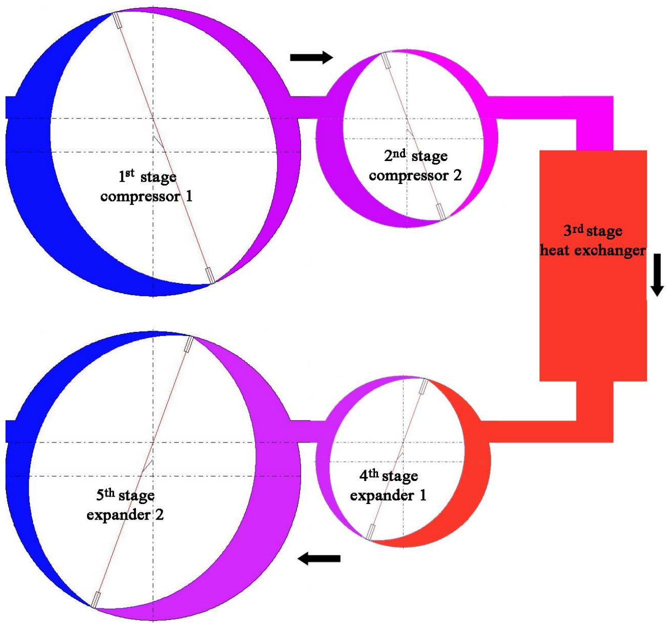

Figure 5 *The core of engine consisting of two stages creating a compressor, a heat exchanger and two stages creating an expander. Work is produced in the last stage and main compression takes place in the first one. The smaller stages between them realize gas exchange control mainly (Figure 9).*

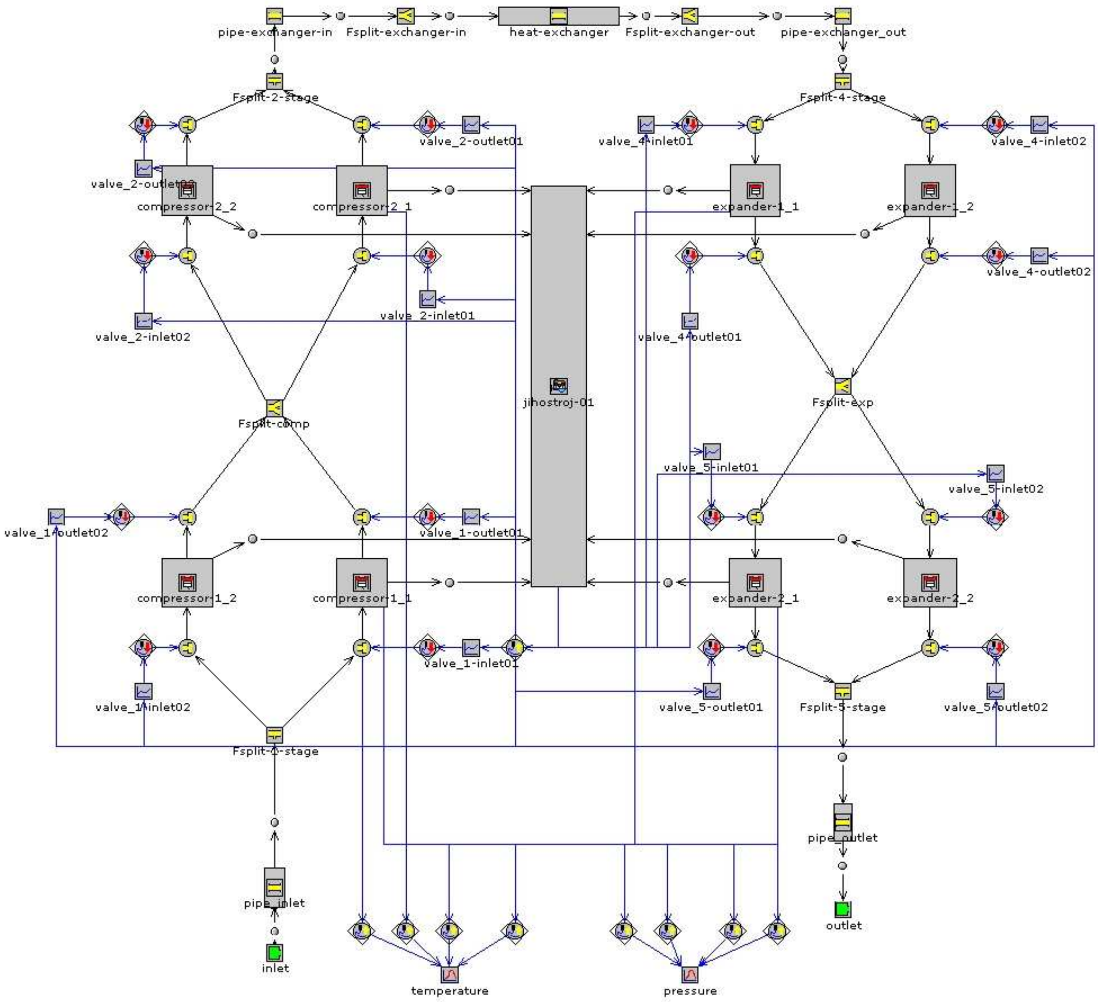

Figure 6 *GT Power model for simulation of the engine with double-acting stages 1, 2 – compressors and expanders*

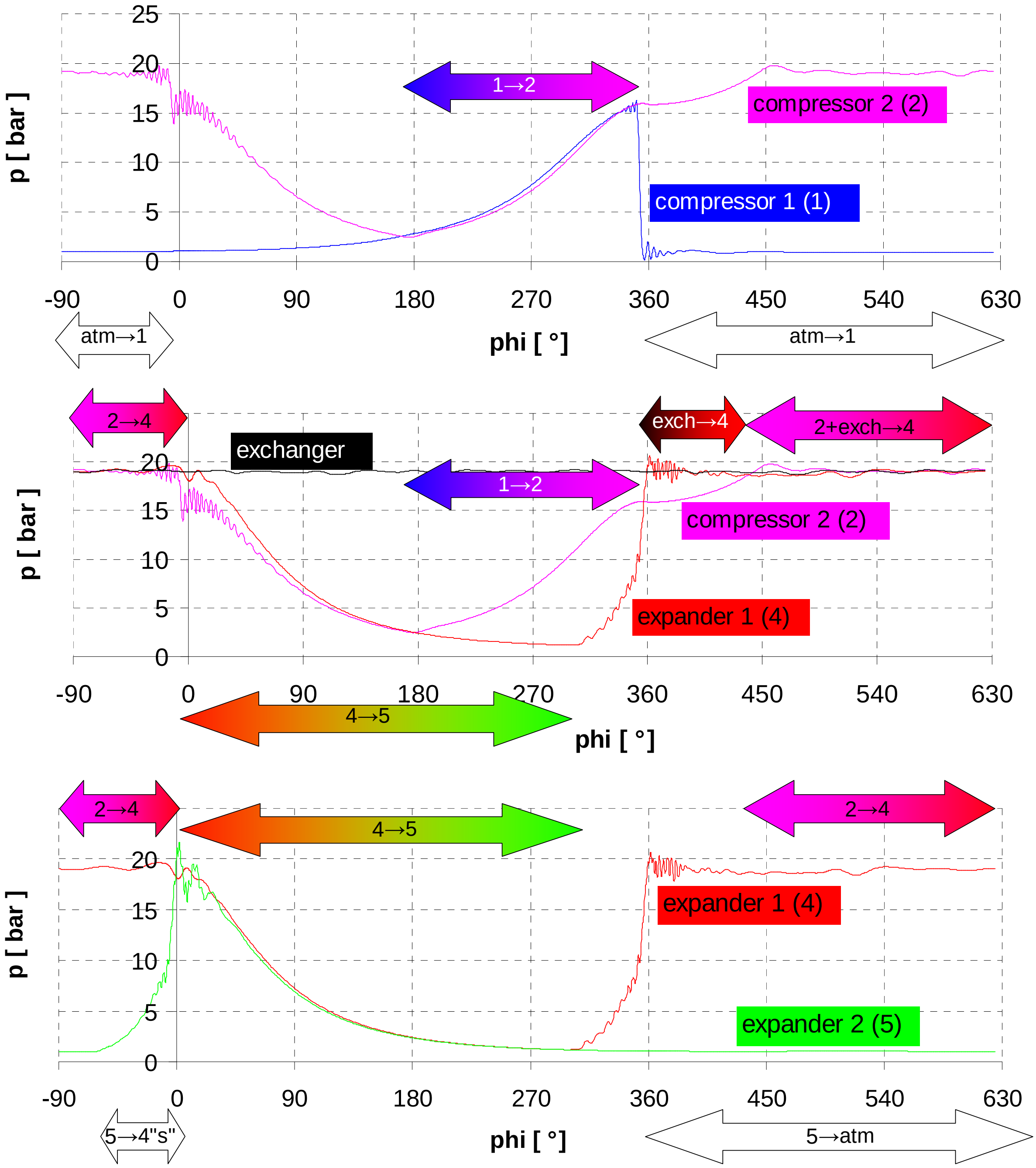

Figure 7 *Gas exchange timing and pressure traces for all compressors, a heat exchanger and expanders*

6. GT Power simulation

The novel engine simulation poses an issue: it is no routine task for existing simulation codes. On the other hand, the unsteadiness of processes inside an engine calls for a detailed description of engine gas dynamics.

The solution was found using authors' accumulated experience with the very flexible GT Power simulation tool [6], although originally focused on a single-acting reciprocating ICE. The flexibility in describing of the engine volume, heat-transfer surface and port/valve area changes enables the authors to set up a model, which describes all involved phenomena, including gas dynamics in connecting pipes and inside a multiple-pipe heat exchanger. The model is presented in Figure 6. This tool was calibrated using the first experimental data from individual machines and used for optimization purposes then.

7. Gas transfer timing optimization

The process described above is controlled by timing of transfer ports. The examples of pressure traces and port timing are presented for all 4 positive displacement machines in ***Figure 7***. Radial ports are applied in all cases, from that reason the angle rate of opening and closing is quite high unlike ICE valves.

The timing was pre-optimized from the point of view of internal (indicated) efficiency. It determines effective both compression and expansion ratios and sets up conditions of pressure equilibrium at times of connected volumes. The optimization was carried out at constant temperature of heat exchanger walls of 600°C and at different power outputs. The speed of crank was 6 000 min^{-1}.

The understanding to phenomena of the rather complicated engine needs to find a specific tool presenting the results.

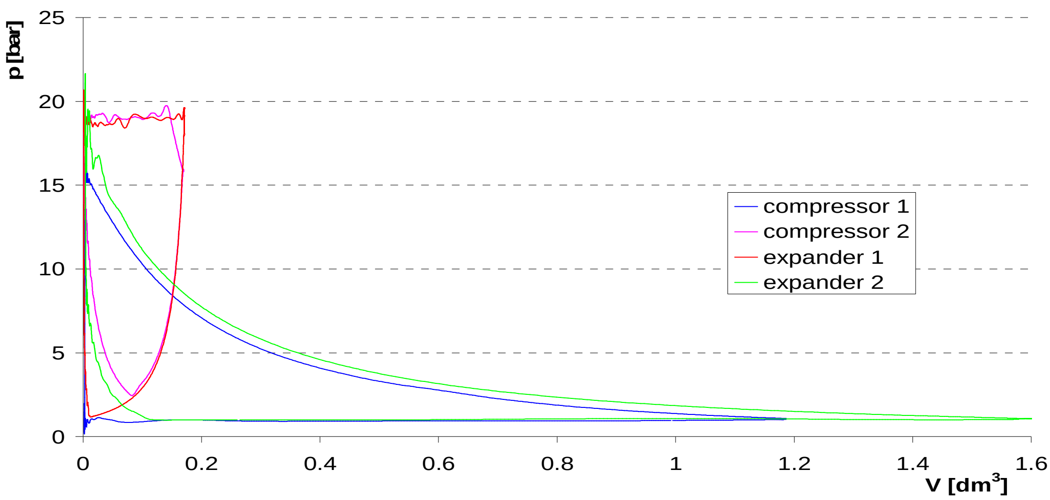

Figure 8 *Pressure diagrams of all positive displacement machines drawn against the volumes of individual machines. The useful work is produced mainly in the last machine 5 with slight contribution of 4. Compression takes place in 1 followed by some pressure increase in 2.*

The examples of results achieved in the form of pressure traces after optimization are presented in ***Figure 7***, in the form of the standard p-V diagram they are presented in ***Figure 8***. The work (not the efficiency!) is visible in p-V diagram. Nevertheless, the standard p-V diagram gives no idea on interaction between stages and the loss of work due to pressure differences. The external work is produced by stages 4 and 5 but most of it is consumed in the stages 1 and 2 and due to flow pressure losses. Pressure traces of both stage 1 and 2 are run counter-clockwise, while the positive work of stages 4 and 5 is produced by clockwise series of changes.

The standard p-V diagram is not transparent enough because of overlaps of volumes changed in both directions. Better insight can be achieved by cumulative volume diagram in ***Figure 9***. All cycles are run counter-clockwise, the expander side work (at negative volume axes) being positive. The left and right parts of individual cycles describe the same changes. The space unfilled by the cycle work areas represents unused potential of positive-displacement machine to produce work. This unused work is not necessarily efficiency loss. However, if there is a pressure difference between machines sharing the working gas by an open port, the work difference during such a gas transfer is lost. From that reason, the pressure traces of connected volumes are drawn twice in ***Figure 9*** (see, e.g., 1 and 2 or 4 and 5 during compression/ expansion), once for bigger, once for smaller volume. It presents the work dissipation by pressure losses in connection pipes. Beside this pressure difference caused by flow resistances of ports and pipes, another loss may occur in positive-displacement machines: the pressure unbalance and a loss of the work due to process unsteadiness. If the flow expands to a connected volume with temporarily decreased pressure, the irreversible change takes place and work potential is lost (see, e.g., well-known Joule experiment used for finding internal energy features). During these changes, temperature increase occurs because the lost work is transformed into the internal (thermal) energy, which could be dangerous for engines of this type, moreover. To avoid unsteady pressure differences, the pressure tuning should be used before two volumes are connected. Smooth reversible pre-compression or pre-expansion must be arranged to avoid these losses. This old knowledge from compound steam engines has been almost forgotten. The lost work due to pre-compression contributes to higher efficiency but creates a work capacity loss, of course. Like in the case of the compression in an ICE, this loss cannot be avoided by simple removal of compression, however, because it would change the shape of the complete indicator diagram with resulting poor efficiency.

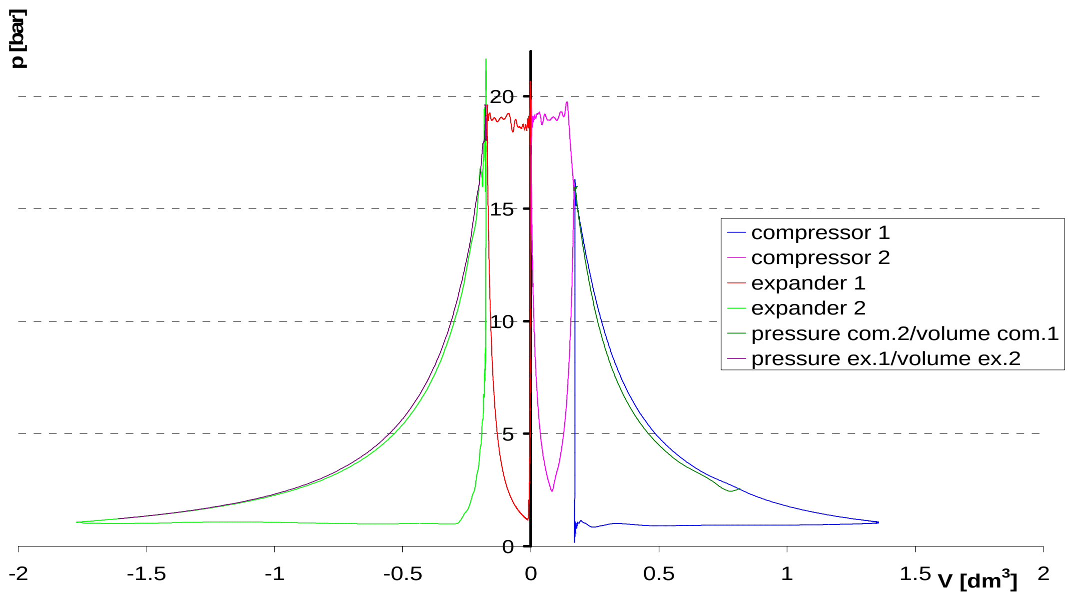

Figure 9 *Pressure diagrams of all positive displacement machines drawn against the cumulative volumes of compressing or expanding machines. The useful work is produced mainly in the last machine 5 with slight contribution of 4.*

Pre-expansion was used in the stage 2 including connecting pipe before transfer port from 1 is opened at the angle of 200°. The pre-compression at the right side of expansion stages is achieved using connection between 5 and opposite side of piston in 4 (“second” 4 – 4“s“ in ***Figure 7***). If the connected volume is small (a piston at TDC without any connecting pipe), pressure is balanced almost immediately without significant loss and necessity of pre-compression.

The optimization may involve even different position of the cranks of individual machines. Up until now, this possibility was not assessed as it could lead to unbalancing of the engine. This question is currently solved by multi-parametric optimization including final setting of engine component swept volumes and gas exchange timing. The results will be published soon together with experiment results.

9. Results achieved

The engine cycle was simulated including heat losses due to cooling to keep reasonable wall temperature below 180°C. The adjusted Woschni's formula was used for a heat transfer coefficient estimate. The heat flux to cooled walls in expander stages 4 and 5 was evaluated to 19% of heat supplied in a heat exchanger. The simulation takes into consideration all major sources of losses in this way.

The achieved traces of engine's indicated power and efficiency are shown in Figure 10 and Figure 11. Despite the low temperature of heat supply the efficiency is quite good – it is comparable to the competitive concept of a more complicated SOLO Stirling engine with the heat supply temperature 650°C, pressures between 15 and 20 MPa (10 times higher!) and helium used as a working gas, where the engine efficiency between 18-23% was reached.

On the other side, the specific power at 6 000 r.p.m. of crank (3 000 r.p.m. at double-acting piston) is not very high. The reference swept volume at inlet is 1.2 dm^3, the other ones are between 1.6 and 0.2 dm^3 in the case under consideration. The low specific power would not be a major problem for a stationary unit but it is associated with low mechanical efficiency, which could hamper the brake parameters of engine – as well-known from any low-power engine. However, these low values of specific power acquired through measurements on our prototype are mainly due to limitations of the testing environment. The results of preliminary measurements at cold machines gave mechanical efficiency estimate of 75-80%, which would give still acceptable results with temperatures of 600 – 650°C.

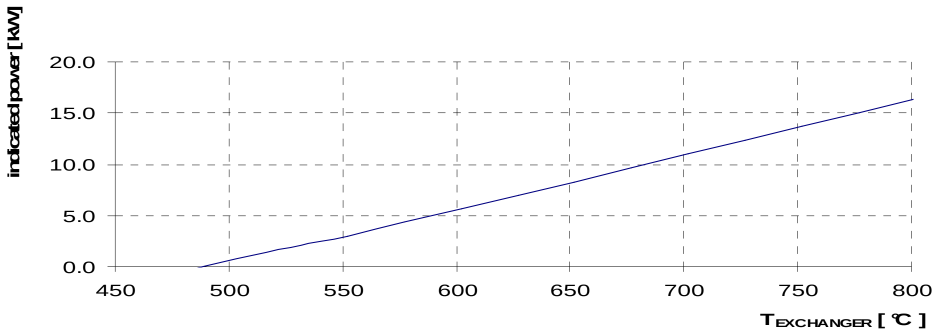

Figure 10 *Indicated power after optimization in dependence on heat supply temperature*

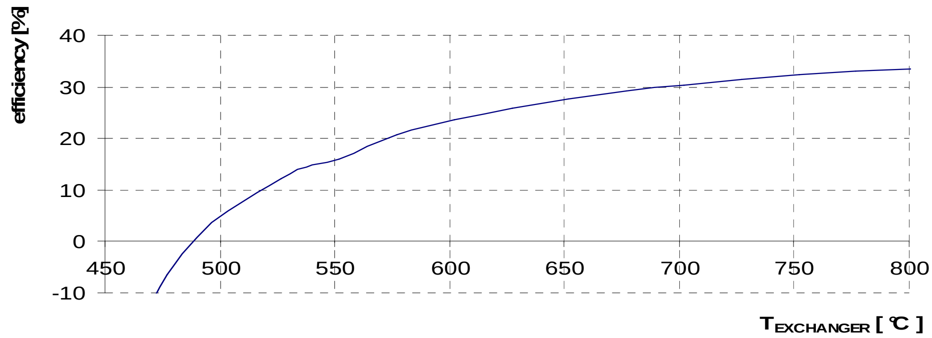

Figure 11 *Indicated efficiency in dependence on heat supply temperature*

10 Conclusions

The paper presents the preliminary results concerning the estimation of potential offered by a novel engine concept, using the procedures according to [7].

The current results show the reasonable level of efficiency achieved at low inlet temperatures but the limits may be posed by comparatively small power density. The experiments planned for the next year will elucidate the potential more. There are still other problems in design to be solved but the current experience offers realistic hope to overcome them.

The components – positive displacement machines – can be used, moreover, as compressors or expanders individually, which seems to be very important immediately useful side-effect of the research performed today. The advantages of those machines are high speed and fully balanced machine with very small minimum volumes, suitable for high volumetric efficiency.

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List of symbols

BDC	bottom dead center
E	expander
ICE	internal combustion engine
IMEP	indicated mean effective pressure
K	compressor
s	inlet
SK	combustion chamber
TDC	top dead center
v	outlet

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