POLISH MARITIME RESEARCH 1 (105) 2020 Vol. 27; pp. 96-106
10.2478/pomr-2020-0010

APPLICATION OF STIRLING ENGINE TYPE ALPHA POWERED BY THE RECOVERY ENERGY ON VESSELS

Jacek Kropiwnicki
Gdansk University of Technology, Faculty of Mechanical Engineering, Poland

ABSTRACT

The Stirling engine is a device in which thermal energy is transformed into mechanical energy without any contact between the heat carrier and the working gas enclosed in the engine. The mentioned feature makes this type of engine very attractive for the use of the recovery energy taken from other heat devices. One of the potential applications of Stirling engines is the use of thermal energy generated in the ship's engine room for producing electricity. The work presents the concept of the Stirling engine type alpha powered by the recovery energy. The model of Stirling engine developed in this work allows a quantitative assessment of the impact of the design features of the engine, primarily the heat exchange surfaces and the volume of control spaces, on the achieved efficiency and power of the engine. Using an iterative procedure, Stirling engine simulation tests were carried out taking into account the variable structural features of the system. The influence of the size of the heater and the cooler, as well as the effectiveness of the regenerator and the temperature of the heat source on the efficiency and power produced by the Stirling engine have been presented.

Keywords: Stirling engine,the recovery energy,efficiency,regenerator

NOMENCLATURE

A_g – heater area of the heat exchange [m^2],
A_w – cooler area of the heat exchange [m^2],
c_p – specific heat capacity of the working fluid at constant pressure [J/(kg·K)],
c_v – specific heat capacity of the working fluid at constant volume [J/(kg·K)],
$\dot{E}_x$ – rate of increase of the internal energy of the working gas in the space x [W],
HVM – heater (cooler) volume multiplication [-],
m – total mass of gas in the engine [kg],
m_x – mass of the working gas in the space x [kg],
$\dot{m}_{xy}$ – mass flow of the working gas between the spaces x and y [kg/s],
$\dot{m}_{in(out)}$ – mass flow of the working gas coming into (out of) the analysed space [kg/s],
p – pressure of the working gas [Pa],
Q_{in} – energy delivered to the Stirling engine [J],
Q_{out} – energy loss during cooling of the Stirling engine [J],
$\dot{Q}_x$ – rate of heat transferred into the space x [W],
$\bar{\dot{Q}}_h$ – average rate of heat delivered to the working gas in the heater [W],
$\bar{\dot{Q}}_k$ – average rate of heat received from the working gas in the cooler [W],
R – gas constant [J/(kg·K)],
T_x – temperature of the working gas in the space x [K],
T_{xy} – temperature of the working gas flowing between the spaces x and y [K],
$T_{in(out)}$ – temperature of the working gas coming into (out of) the analysed space [K],

W – net work produced by the Stirling engine [J],
$\dot{W}_x$ – rate of work done on the surroundings in the space x [W],
$V_{SC.c}$ – volume of the compression space swept capacity [m^3],
$V_{SC.e}$ – volume of the expansion space swept capacity [m^3],
V_x – volume of the space x [m^3],

GREEK SYMBOLS

α – angle of the phase shift between the spaces of expansion and compression,
DT_r – regenerator temperature difference,
ε – regenerator effectiveness,
η_{th} – thermal efficiency of the Stirling engine,
ϕ – angle of rotation of the crankshaft shaft with respect to the cylinder of the expansion space,
κ – isentropic exponent,
t_c – time of the cycle,

SUBSCRIPTS

c – compression space,
e – expansion space,
g – gas supplying the heat to the engine,
h – heater,
k – cooler
r – regenerator,
ref – reference value,
$test$ – tested value,
w – cooling water.

INTRODUCTION

The contemporary aspiration of modern economies to minimise energy consumption and the emission of toxic compounds is reflected in the development of energy-saving drives in land and sea transport [16]. New types of fuels are being used, which on the one hand enable the development of substances that are difficult to use in the energy sector, on the other hand giving the possibility of reducing the emission of toxic compounds [13, 14]. Modern marine propulsion systems mainly use reciprocating engines, which enable the production of mechanical energy that is used as the main drive and the drive for auxiliary devices, e.g. in hopper suction dredgers [2]. Ships, as autonomous energy objects, must also have facilities that produce electricity and heat. The complex production system of these three types of energy on the one hand increases the capital costs and complexity of the system, and on the other hand gives the opportunity to increase the overall energy efficiency of the ship by treating heat as an equivalent energy carrier. However, if the heat production exceeds the demand, it is treated as waste and returned to the atmosphere. However, the thermal energy leaving the combustion engine [11] can be transformed into mechanical or electric energy in specially designed recovery energy systems [18]. One of the most popular commercial solutions of this type is a system powered by exhaust gases from a reciprocating engine. The system uses a gas turbine, steam turbine or both [17, 24]. This solution makes it possible to reduce the specific fuel consumption by approx. 2% at nominal load. However, it is not very popular, mainly due to the high investment cost and the large space required for an additional installation, which is currently lacking in the modern engine room. For small boats like fishing or tourist vessels, the Stirling engine can be an effective solution for the recovery of energy and transforming it into electricity.

The Stirling engine [12] is a device in which thermal energy is transformed into mechanical energy without any contact between the heat carrier and the working gas enclosed in the engine. Stirling engines are characterised by relatively simple construction in comparison with internal combustion engines; the engine's moving parts do not have direct contact with the substance that supplies thermal energy to the device and operate at lower temperature differences. All this influences the durability of Stirling engines, and often eliminates servicing at all. Currently, as a result of the growing interest worldwide in the use of renewable energy sources to generate electricity [3, 15], a series of research works on Stirling engines are being conducted [3, 8, 10]. There are solutions that allow the direct use of the Stirling engine for propulsion, e.g. the system used by Saab (Kockums) in submarines [21]. In commercial applications, Stirling engines are not yet used as the main propulsion. However, research is being conducted in this area, for example in [9] the concept of a 20 MW Stirling engine using diesel as a heat source is presented. The authors admit that at the current stage of construction development, the efficiency achieved by a Stirling engine will be lower than that of traditional reciprocating engines for marine applications.

To conclude, the Stirling engine is not yet considered to be the main propulsion of the ship, due to the low degree of technological development and lower efficiency than modern low-speed internal combustion engines. However, it can be an effective solution to reduce fuel consumption on ships [9, 26, 27], because it uses energy otherwise emitted to the atmosphere. For example, in [21], the authors presented the concept of generating electricity using a Stirling engine powered by exhaust gases from a 578 kW main engine. Assuming the exhaust gas stream and the system efficiency, the authors calculated the potential fuel savings at 3%.

Such systems do not have to be installed in the immediate vicinity of the source of the heat, which seems particularly attractive for the limited space of the engine room. The thermal energy is supplied to the Stirling engine through the walls of the heater, which has some thermal inertia. That enables uninterrupted work for several minutes without a heat supply from external devices. Such a solution can also be adopted as a support for the backup power system for a few minutes.

Fig. 1 shows the energy system diagram of a fishing vessel equipped with a main engine, an oil-fired boiler and an electric power generator driven from the crankshaft of the engine. The use of the Stirling engine in such a system makes it possible to recover energy leaving the main engine in the form of exhaust gases. However, in the case of switching off the main drive and starting the oil-fired boiler only for heat production, the system gives the possibility of producing additional electricity using the exhaust gas leaving the boiler.

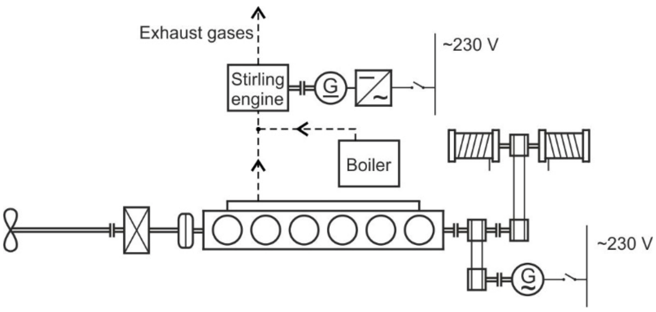

Fig. 1. The energy system diagram of a fishing vessel equipped with a main engine, an oil-fired boiler and the Stirling engine producing electricity

The work presents the concept of the Stirling engine type alpha powered by the recovery energy. The model of the Stirling engine in the form of a set of equations is presented. The model developed in this work allows a quantitative assessment of the impact of the design features of the Stirling engine, primarily the heat exchange surfaces and the volume of control spaces, on the achieved efficiency and power of the engine. Prospectively, application of this model will enable optimisation of the energy recovery system, which uses a Stirling engine powered by low-temperature energy sources. Using the iterative procedure, Stirling engine simulation tests were carried out taking into account the variable structural features of the system. The influence of the size of the heater and the cooler, as well as the effectiveness of the regenerator and the temperature of the heat source on the efficiency and power produced by the Stirling engine are presented.

MODEL OF THE STIRLING ENGINE

The principle of operation of the engine without internal combustion with a fixed mass of gas (the working medium) is shown in Fig. 2. The gas enclosed in the elastic sphere is cyclically heated and cooled. Due to changes of the gas temperature, the sphere expands or shrinks cyclically. The point is to synchronise the processes of heating (expanding) and the mechanical energy being received by the user. Similarly, the cooling process (compression) must be synchronised with the supply of energy, e.g. using the kinetic energy accumulated in the crank mechanism. The diagram presented does not include the regeneration process, which is very important in the Stirling engine, dramatically increasing the efficiency.

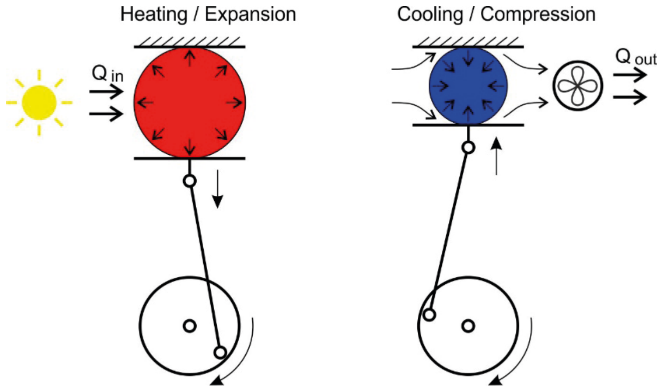

Fig. 2. The principle of operation of the engine without internal combustion with a fixed mass of working gas

The principle of operation of the engine operating according to the Stirling theoretical cycle is shown in Fig. 3. The Stirling theoretical cycle consists of four transformations (Fig. 3):
1-2 intensive cooling of the cylinder during compression,
2-3 stopping the piston at top dead centre (TDC) and providing heat from the regeneration process,
3-4 intensive heating of the cylinder during expansion,
4-1 stopping the piston at bottom dead centre (BDC) and heat recovery from the regeneration process.

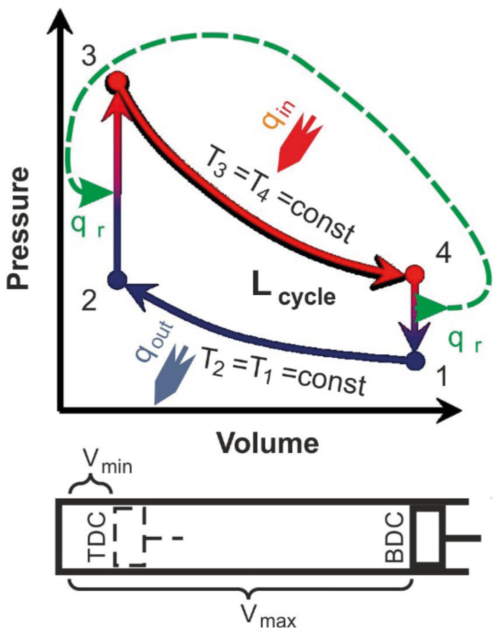

Fig. 3. The Stirling engine theoretical cycle

In the Stirling engine's theoretical cycle, it is assumed that regeneration takes place in a perfect way. The practical design of the Stirling engine can be derived by solving three major technical problems:
- continuity of movement of the displacement element,
- transfer of the total mass of gas from low to high temperature space and back without volume change,
- completing the process of heat regeneration in one cycle.

Among the many practical constructions of Stirling engines, three main types can be distinguished: alpha, beta and gamma [12]. In the presented work only the alpha type will be considered. In Fig. 4 the simplified layout of the Stirling engine type alpha is presented. The layout is presented in 4 versions corresponding to each process detailed in the theoretical cycle above (Fig. 3).

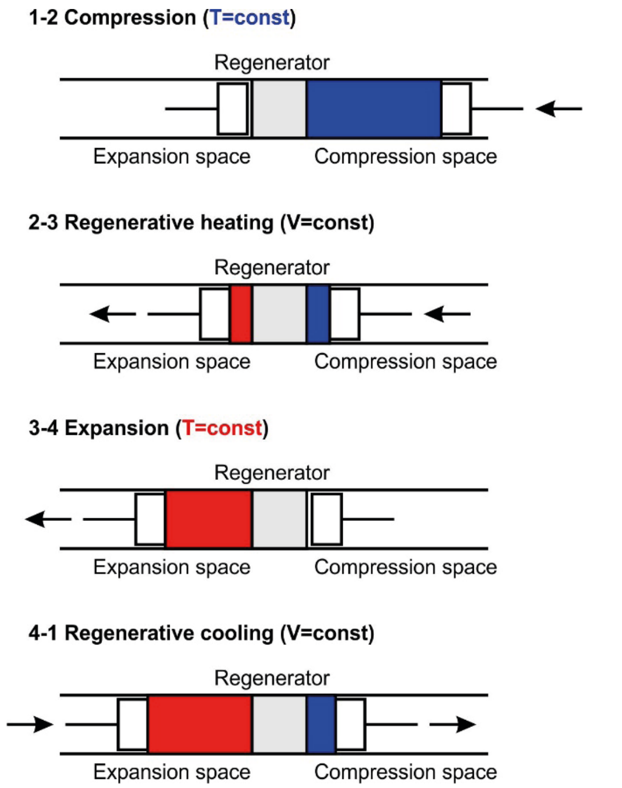

Fig. 4. Simplified layout of the Stirling engine type alpha with the four phases of the cycle

The theoretical Stirling cycle is a highly idealised thermodynamic cycle [1, 17], so its use is not recommended for quantitative analysis. For practical analysis, the isothermal or adiabatic model [4, 22, 23, 25] is most often used, which allows the functional separation of engine working spaces into: compression space, cooler, regenerator, heater and expansion space. Unfortunately, based on the fundamental assumptions for the isothermal model, the temperature of the gas in compression and expansion space is constant (as well as in each distinguished space). This implies that these exchangers are useless and the whole heat exchange process take place in the compression and expansion space only. Obviously this cannot be correct, since the cylinder walls are not able to cover the expected heat demand. Additionally, this approach prevents calculation of the size of the heat exchangers (cooler and heater). In real machines the compression and expansion spaces will tend to be adiabatic rather than isothermal. For this reason, in this work, the analysis of Stirling engine operation will be carried out using the adiabatic model.

Fig. 5 presents the diagram of the analysed Stirling engine type alpha. The analysis was conducted with the assumption that there is no heat exchange in the compression space, the regenerator, the expansion space or in adjacent pipelines (the grey band indicates perfect insulation).

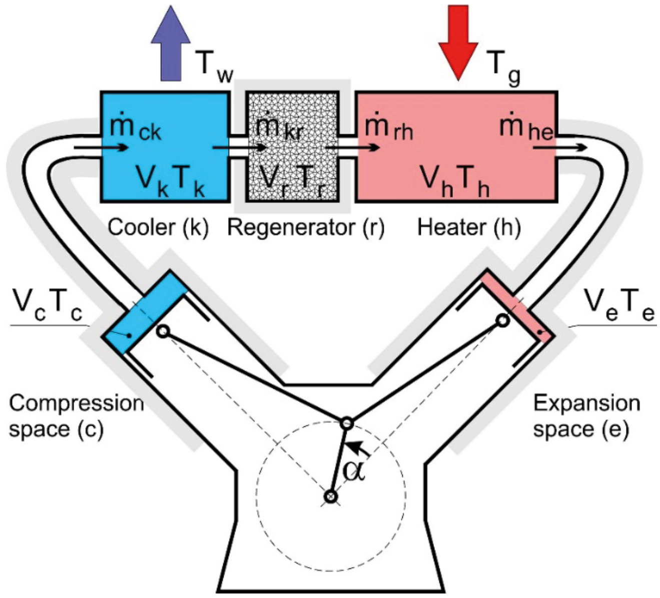

Fig. 5. Adiabatic arrangement of spaces of the Stirling engine type alpha

The starting point of the analysis is that the instantaneous pressure in each element of the Stirling engine is the same for the indicated crankshaft position. The total mass of gas in the machine is constant, thus:

$$m = m_c + m_k + m_r + m_h + m_e = \text{const} \quad (1)$$

Hence, Eq. (1) can be transformed to the following form:

$$\dot{m}_c + \dot{m}_k + \dot{m}_r + \dot{m}_h + \dot{m}_e = 0 \quad (2)$$

For the compression space the energy equation can be given in the form:

$$\dot{Q}_c + \left(c_p \cdot \dot{m}_{in} \cdot T_{in} - c_p \cdot \dot{m}_{out} \cdot T_{out}\right) = \dot{W}_c + \dot{E}_c \quad (3)$$

where:

$$\dot{W}_c = p \cdot \dot{V}_c \quad (4)$$

$$E_c = c_v \cdot m_c \cdot T_c \quad (5)$$

using the equation of state we obtain:

$$E_c = \frac{c_v}{R} \cdot p \cdot V_c \quad (6)$$

hence:

$$\dot{E}_c = \frac{c_v}{R} \cdot \left(\dot{p} \cdot V_c + p \cdot \dot{V}_c\right) \tag{7}$$

Assuming that heat transferred into the compression space is zero as well as the mass flow of the working gas coming into the analysed space, Eq. (3) can be simplified:

$$-c_p \cdot \dot{m}_{ck} \cdot T_{ck} = p \cdot \dot{V}_c + \frac{c_v}{R} \cdot \left(\dot{p} \cdot V_c + p \cdot \dot{V}_c\right) \tag{8}$$

And assuming that:

$$\dot{m}_{ck} = -\dot{m}_c \tag{9}$$

we obtain:

$$\dot{m}_c = \frac{1}{R \cdot T_{ck}} \cdot p \cdot \dot{V}_c + \frac{1}{\kappa \cdot R \cdot T_{ck}} \cdot \dot{p} \cdot V_c \tag{10}$$

Analogically, for the expansion space the rate of mass can be calculated:

$$\dot{m}_e = \frac{1}{R \cdot T_{he}} \cdot p \cdot \dot{V}_e + \frac{1}{\kappa \cdot R \cdot T_{he}} \cdot \dot{p} \cdot V_e \tag{11}$$

For the cooler, the energy equation can be given in the form:

$$\dot{Q}_k + \left(c_p \cdot \dot{m}_{ck} \cdot T_{ck} - c_p \cdot \dot{m}_{kr} \cdot T_{kr}\right) = \dot{W}_k + \dot{E}_k \tag{12}$$

where:

$$\dot{E}_k = \frac{c_v}{R} \cdot \left(\dot{p} \cdot V_k + p \cdot \dot{V}_k\right) \tag{13}$$

Assuming that for the heat exchanger the following conditions are met:

$$\dot{W}_k = 0 \text{ and } \dot{V}_k = 0$$

we obtain:

$$\dot{Q}_k = \frac{c_v}{R} \cdot \dot{p} \cdot V_k - \left(c_p \cdot \dot{m}_{ck} \cdot T_{ck} - c_p \cdot \dot{m}_{kr} \cdot T_{kr}\right) \tag{14}$$

Analogically, for the regenerator and the heater we obtain respectively:

$$\dot{Q}_r = \frac{c_v}{R} \cdot \dot{p} \cdot V_r - \left(c_p \cdot \dot{m}_{kr} \cdot T_{kr} - c_p \cdot \dot{m}_{rh} \cdot T_{rh}\right) \tag{15}$$

$$\dot{Q}_k = \frac{c_v}{R} \cdot \dot{p} \cdot V_h - \left(c_p \cdot \dot{m}_{rh} \cdot T_{rh} - c_p \cdot \dot{m}_{he} \cdot T_{he}\right) \tag{16}$$

For the cooler, where the volume and the temperature are constant the equation of state can be transformed into the following form:

$$\dot{p} \cdot V_k = \dot{m}_k \cdot R \cdot T_k \tag{17}$$

hence, we obtain the definition of the derivative of the mass concentration in the cooler:

$$\dot{m}_k = \dot{p} \cdot \frac{V_k}{R \cdot T_k} \tag{18}$$

Analogically, for the regenerator and the heater we obtain:

$$\dot{m}_r = \dot{p} \cdot \frac{V_r}{R \cdot T_r} \tag{19}$$

$$\dot{m}_h = \dot{p} \cdot \frac{V_h}{R \cdot T_h} \tag{20}$$

Inserting Eqs. (10), (11), (18), (19) and (20) into (2) we obtain:

$$\dot{p} = -p \cdot \kappa \cdot \frac{\left(\frac{\dot{V}_c}{T_{ck}} + \frac{\dot{V}_e}{T_{he}}\right)}{\frac{V_c}{T_{ck}} + \frac{V_e}{T_{he}} + \kappa \cdot \left(\frac{V_k}{T_k} + \frac{V_r}{T_r} + \frac{V_h}{T_h}\right)} \tag{21}$$

For the compression space the equation of state takes the form:

$$T_c = \frac{p \cdot V_c}{m_c \cdot R} \tag{22}$$

and can be transformed to the following form:

$$\dot{T}_c = \frac{V_c}{m_c \cdot R} \cdot \dot{p} + \frac{p}{m_c \cdot R} \cdot \dot{V}_c - \frac{p \cdot V_c}{R} \cdot \frac{1}{m_c^2} \cdot \dot{m}_c \tag{23}$$

hence:

$$\dot{T}_c = \frac{T_c}{p} \cdot \dot{p} + \frac{T_c}{V_c} \cdot \dot{V}_c - m_c \cdot T_c \cdot \frac{1}{m_c^2} \cdot \dot{m}_c \tag{24}$$

and:

$$\dot{T}_c = T_c \cdot \left(\frac{\dot{p}}{p} + \frac{\dot{V}_c}{V_c} - \frac{\dot{m}_c}{m_c}\right) \tag{25}$$

where:

$$m_c = \frac{p \cdot V_c}{T_c \cdot R} \tag{26}$$

Analogically, for the expansion space we obtain:

$$\dot{T}_e = T_e \cdot \left(\frac{\dot{p}}{p} + \frac{\dot{V}_e}{V_e} - \frac{\dot{m}_e}{m_e}\right) \tag{27}$$

where:

$$m_e = \frac{p \cdot V_e}{T_e \cdot R} \tag{28}$$

The mass flows used in earlier equations can be defined using the following relations:

$$\begin{cases} \dot{m}_{ck} = -\dot{m}_c \\ \dot{m}_{kr} = \dot{m}_{ck} - \dot{m}_k \\ \dot{m}_{rh} = \dot{m}_e + \dot{m}_h \\ \dot{m}_{he} = \dot{m}_e \end{cases} \tag{29}$$

In the regenerator the gas flows cyclically from the cooler to the heater and in the reverse direction. The gas is respectively heated and cooled by the metal mesh placed in the regenerator. The net heat transfer per cycle is zero. The regenerator quality is defined by the regenerator effectiveness (ε), which parameter allows us to calculate the regenerator temperature difference:

$$\Delta T_r = \frac{(T_h - T_k)}{2} \cdot (1 - \varepsilon) \tag{30}$$

Hence the temperature of the gas coming into the cooler from the regenerator will be higher than the temperature of gas in the cooler by 2xΔT_r; analogically, the gas coming into the heater from the regenerator will be lower than the temperature of gas in the heater by 2xΔT_r.

The net work produced by the Stirling engine in the expansion and compression spaces for the whole cycle equals:

$$W = \int_0^{\tau_c} p \cdot \left(\dot{V}_c + \dot{V}_e\right) \cdot dt \tag{31}$$

The energy delivered to the engine is given by:

$$Q_{in} = \int_0^{\tau_c} \dot{Q}_h \cdot dt \tag{32}$$

The energy loss during cooling is given by:

$$Q_{out} = \int_0^{\tau_c} \dot{Q}_k \cdot dt \tag{33}$$

The thermal efficiency of the Stirling engine can be calculated using the following formula:

$$\eta_{th} = \frac{W}{Q_{in}} \tag{34}$$

For more realistic analysis it is convenient to define the temperature of the gas delivering heat to the Stirling engine (the waste energy) and the temperature of the water receiving heat from the device instead of the temperature of the working gas respectively in the heater and the cooler [5, 6]. Hence, assuming the energy is transferred with 100% efficiency through the heat exchanger, we can define the average rate of heat delivered to the working gas in the heater:

$$\bar{\dot{Q}}_h = \frac{Q_{in}}{\tau_c} = k_g \cdot A_g \cdot \left(T_g - T_h\right) \tag{35}$$

and the average rate of heat received from the working gas in the cooler:

$$\bar{\dot{Q}}_k = \frac{Q_{out}}{\tau_c} = k_w \cdot A_w \cdot (T_w - T_k) \tag{36}$$

The equations presented in this section are used to determine the desired parameters of the working gas, e.g., the pressure, temperature in the compression space, expansion, engine efficiency, etc. Some of the variables contained in the equations we can assume, for example, the volume of the compression space:

$$V_c = 0.5 \cdot V_{SC.c} \cdot \left(1 + cos(\varphi - \alpha - \pi)\right) \tag{37}$$

and for the expansion space:

$$V_e = 0.5 \cdot V_{SC.e} \cdot \left(1 + cos(\varphi - \pi)\right) \tag{38}$$

Some others of the variables are conditional on the direction of mass flow. Hence, the temperature of the working gas coming into the cooler from the compression space can be defined using the following formula:

$$T_{ck} = \begin{cases} T_c \; if \; \dot{m}_{ck} \geq 0 \\ T_k \; if \; \dot{m}_{ck} < 0 \end{cases} \tag{39}$$

Analogically, for the temperature of the working gas coming into the regenerator from the cooler we obtain:

$$T_{kr} = \begin{cases} T_k \; if \; \dot{m}_{kr} \geq 0 \\ T_r \; if \; \dot{m}_{kr} < 0 \end{cases} \tag{40}$$

for the temperature of the working gas coming into the heater from the regenerator we obtain:

$$T_{rh} = \begin{cases} T_r \; if \; \dot{m}_{rh} \geq 0 \\ T_h \; if \; \dot{m}_{rh} < 0 \end{cases} \tag{41}$$

for the temperature of the working gas coming into the expansion space from the heater we obtain:

$$T_{he} = \begin{cases} T_h \; if \; \dot{m}_{he} \geq 0 \\ T_e \; if \; \dot{m}_{he} < 0 \end{cases} \tag{42}$$

When the initial values of the variables are not known, e.g., pressure, the temperature in the compression and expansion space, etc., we need to use an iterative procedure for calculations. In such a procedure, we assume that the cycle is of a repetitive nature, which means that the calculations can be completed when the respective values at the beginning and the end of the cycle are the same, e.g., the temperature of the working gas in the compression space. Hence, the results of the calculations of one cycle of work are used as the initial conditions in the next iteration. In the iterative procedure, the initial pressure and temperature values in all control volumes are assumed, as well as the temperature of the exhaust gas supplying heat to the engine and the temperature of the cooling water. In the first calculation step, the direction of the working gas flow in the Stirling engine is determined on the basis of the volume change balance in the expansion and compression space. Hence the initial values of the gas stream and its temperature between control volumes are calculated. In the next steps, only the results from the previous calculation steps are taken into account. After each calculation step, the indicated work is calculated along with the amount of heat supplied, removed and accumulated by the regenerator. The convergence was controlled by the conformity of the compression and expansion temperatures at the end of one cycle and the beginning of the next one, as well as by the energy balance released and absorbed in the regenerator, which should equal zero.

INFLUENCE OF THE DESIGN FEATURES ON THE OPERATING PARAMETERS

The calculations were made for the system: heat source - Stirling engine powered with the waste heat (Fig. 1). It was assumed that the stream of exhaust gases flowing to the Stirling engine heater can be delivered by an internal combustion engine and an oil-fired boiler. It was assumed that the temperature of the exhaust gases heating the Stirling engine can be changed in the range of 450–750 K. The Stirling engine cooler is fed with 323 K cooling water. The adiabatic model of the Stirling engine described in the previous section was applied to test the influence of the size of the heater and the cooler, as well as the effectiveness of the regenerator and the temperature of the heat source on the efficiency and power produced by the Stirling engine. The calculations were carried out using the author's iterative procedure described in the previous section. Table 1 shows the basic specification of the analysed Stirling engine.

Tab. 1. Main parameters and proprieties of the analysed Stirling engine type alpha

Parameter	Value
Working gas	air
Charge pressure	1.0 MPa
Compression (cold) volume: swept capacity	730 cc
Pipeline (cold part)	427 cc
Cooler volume (reference value)	304 cc
Regenerator volume	289 cc
Heater volume (reference value)	1140 cc
Pipeline (hot part)	427 cc
Expansion (hot) volume: swept capacity	730 cc

Using the constitutive equations given in the previous section, the temperatures (Fig. 6) in the analysed spaces were calculated for the selected temperature of the heat supply gas and cooling water. Also given was the condition of convergence in the iterative procedure that the mean pressure of the working gas equals 1.0 MPa. It can be observed that, due to the assumed non- isothermal transformation taking place in the compression and expansion space only, there the temperature follows the change of pressure and total volume of the working gas (left part of Fig. 6). In Fig. 6, right part, the conditional temperatures are presented. The levels of these conditional temperatures change rapidly due to the change of direction of the working gas flow.

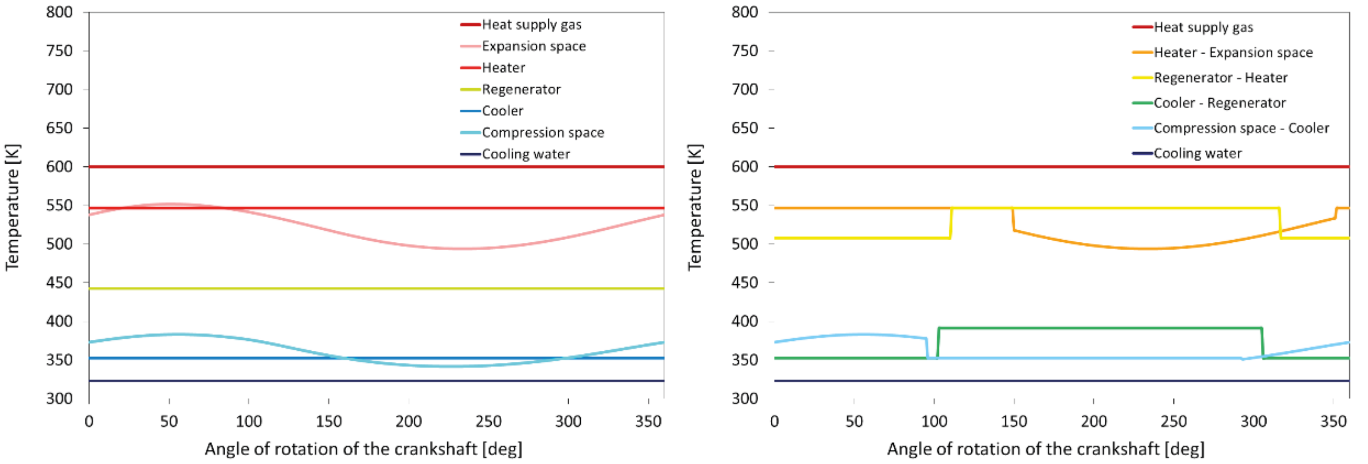

Fig. 6. Temperatures distribution for one cycle; regenerator effectiveness 80%

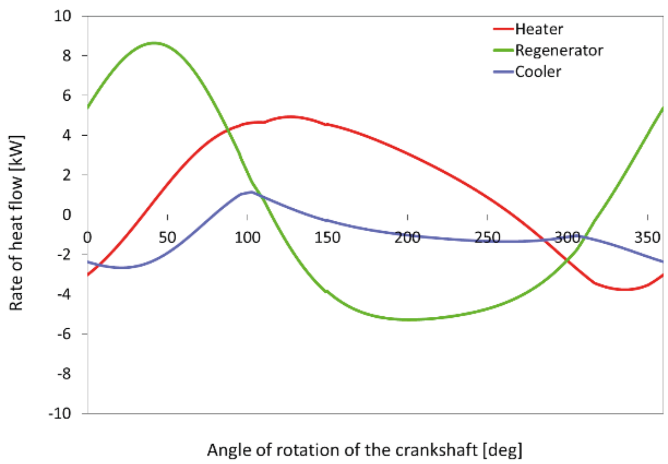

Fig. 7. Rate of the heat flow in: heater, regenerator and cooler; regenerator effectiveness 80%; temp. of the heat supply gas 600 K; temp. of the cooling water 323 K

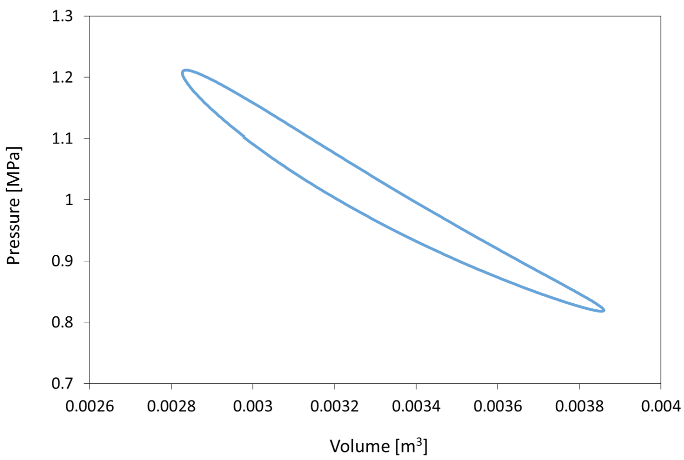

Fig. 8. Volume–pressure diagram; regenerator effectiveness 80%; temp. of the heat supply gas 600 K; temp. of the cooling water 323 K

Fig. 7 presents the rate of the heat flow in the heater, regenerator and cooler. It can be observed that the energy balance is positive for the heater and negative for the cooler, which respectively corresponds to energy delivery and reception. The energy balance for the regenerator equals zero, which proves correct convergence of the calculation. Fig. 8 presents the volume–pressure diagram of the tested Stirling engine. It can be observed that the mean pressure is 1.0 MPa, which is in line with the assumed calculation condition.

Fig. 9 presents the influence of the regenerator effectiveness on the pressure change of the working gas in the tested Stirling engine. For the assumed test conditions the regenerator effectiveness, in the range 80–100%, has an explicitly strong influence on the efficiency; for reduced regenerator effectiveness the influence is weak (Fig. 10). That result leads to the important conclusion that the effort put into the regenerator development pays back effectively if we operate in the high value range. On the other hand, a low-quality regenerator that gives off high hydraulic resistance brings more costs than benefits and may not be needed at all.

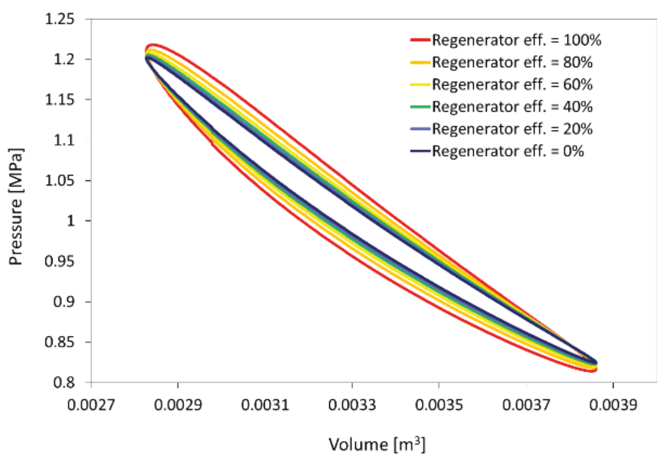

Fig. 9. Volume–pressure diagram for different values of regenerator effectiveness; temp. of the heat supply gas 600 K; temp. of the cooling water 323 K

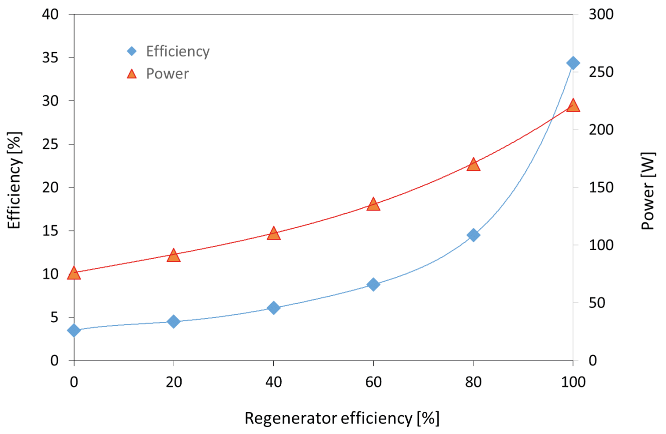

Fig. 10. Influence of the regenerator effectiveness on the efficiency and power of the Stirling engine; temp. of the heat supply gas 600 K; temp. of the cooling water 323 K

Fig. 11 presents the influence of the temperature of the heat supply gas on the efficiency and power of the Stirling engine. The poor heat exchange conditions between the working gas (air) and the gas supplying the heat (exhaust gases) cause a rise of the heat exchange surface to cover the heat demand. That results in a relatively large volume of the heater, which means a relatively low efficiency of the Stirling engine, even for a higher temperature range. Nevertheless, the Stirling engine power increases almost proportionally to the temperature.

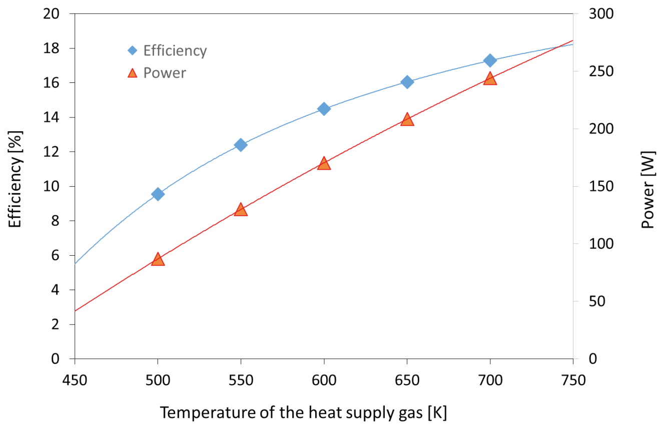

Fig. 11. Influence of the temp. of the heat supply gas on the efficiency and power of the Stirling engine; temp. of the cooling water 323 K; the regenerator effectiveness 80%

The analysis of the influence of the heater/cooler size on the efficiency and power of the Stirling engine was performed assuming that the relation between the gas volume and the heat exchange surface is constant. Additionally, it was assumed that the relation of the gas volume in the heater to the gas volume in the cooler is also constant. The heater and cooler size were defined using one parameter – the heater volume multiplication:

$$HVM = \frac{V_{h\ test}}{V_{h\ ref}} = \frac{A_{g\ test}}{A_{g\ ref}} = \frac{V_{k\ test}}{V_{k\ ref}} = \frac{A_{w\ test}}{A_{w\ ref}} \tag{43}$$

The influence of the heater/cooler size (HVM) on the efficiency and power of the Stirling engine is presented in Fig. 12. For the initial HVM range, the rise of the heater volume causes a significant increase in the amount of thermal energy supplied to the engine, which results in a sudden increase of efficiency and power. In the vicinity of HVM = 1, the maximum of the efficiency and the power is reached, and then, as a result of the significant impact of dead space, the efficiency and the power of the Stirling engine decreases.

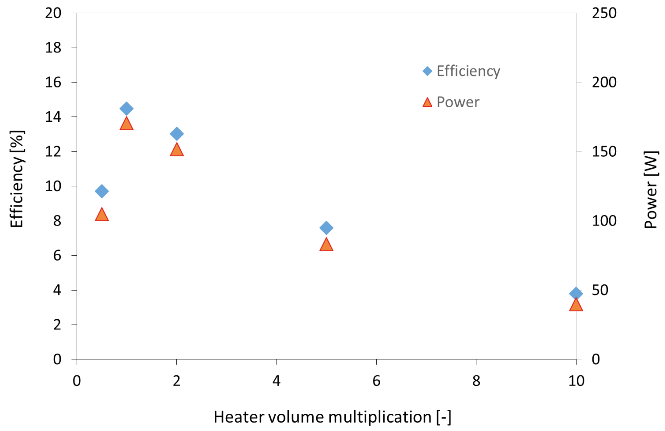

Fig. 12. Influence of heater/cooler size (HVM) on the efficiency and power of the Stirling engine; the regenerator effectiveness 80%; temp. of the heat supply gas 600 K; temp. of the cooling water 323 K

CONCLUSIONS

For small boats like fishing or tourist vessels, the Stirling engine can be an effective solution for recovering energy and transforming it into electricity. Such systems do not have to be installed in the immediate vicinity of the source of the heat, which seems particularly attractive for the limited space of the engine room. The adiabatic model of the Stirling engine presented in this paper has been evaluated for application in the iterative procedure, which in practice means that not all the initial conditions need to be known and that they will be determined as an effect of calculations in preceding calculation steps.

The efficiency as well as the power of the tested Stirling engine are affected very intensely by the temperature of the heat supply gas and the regenerator effectiveness; the higher the values of those parameters, the higher the efficiency and the power. A particularly intensive influence of the regenerator effectiveness on the engine efficiency is observed for the range 80–100%, while for reduced regenerator effectiveness the influence is weak. That result leads to the important conclusion that the effort put into the regenerator development pays back effectively if we operate in the high value range. On the other hand, a low-quality regenerator that gives off high hydraulic resistance brings more costs than benefits and may not be needed at all.

The influence of the size of the cooler and the heater is not so obvious. On the one hand, increasing the size of heat exchangers causes more energy to be supplied to the engine, but on the other hand, it increases the dead space, which reduces the efficiency of the engine. The influence of those factors means that increasing the volume (surface) of the heat exchangers has a positive effect only for the small range (up to HVM=1), while a further volume increase results in a gradual decrease in the efficiency and power of the device.

REFERENCES

1. Bataineh K. M. (2018): *Numerical thermodynamic model of alpha-type Stirling engine*. Case Studies in Thermal Engineering, 12, 104-116.

2. Bocheński D. (2018): *Selection of main engines for hopper suction dredgers with the use of probability models*. Polish Maritime Research, 1(97), Vol. 25, 70-76.

3. Cheng C. H., et al. (2013): *Theoretical and experimental study of a 300-W beta-type Stirling engine*. Energy, 59, 590-599.

4. Chmielewski A., Gumiński R., Mączak J. (2016): *Adiabatic analysis of thermodynamic processes in the Stirling engine*. Proceedings of the Institute of Vehicles, 2(106), 13-20.

5. Cichy M., Kneba Z., Kropiwnicki J. (2017): *Causality in models of thermal processes in ship engine rooms with the use of Bond Graph (BG) method*. Polish Maritime Research, S1(93), Vol. 24, 32-37.

6. Cichy M., Kropiwnicki J., Kneba Z. (2015): *A model of thermal energy storage according to the convention of Bond Graphs (BG) and State Equations (SE)*. Polish Maritime Research, 4 (88), Vol. 22, 41-47.

7. Cieśliński J. T., et al. (2012): *Investigation of a Stirling engine as a micro-CHP system*. 3rd International Conference, Low Temperature and Waste Heat Use in Energy Supply Systems, Theory and Practice, Bremen, 33-38.

8. Gheith R., Aloui F., Ben Nasrallah S. (2013): *Experimental investigation of a Gamma Stirling engine*. Int. J. Energy Res., 37, 1519-1528.

9. Hirata K., Kawada M. (2005): *Discussion of Marine Stirling Engine Systems*. Proceedings of the 7th International Symposium on Marine Engineering. Tokyo, 1-5.

10. Karabulut H., et al. (2009): *An experimental study on the development of a b-type Stirling engine for low and moderate temperature heat sources*. Applied Energy, 86, 68-73.

11. Korczewski Z. (2015): *Exhaust gas temperature measurements in diagnostics of turbocharged marine internal combustion engines. Part I. Standard measurements*. Polish Maritime Research, 1(85) Vol. 22, 47-54.

12. Kropiwnicki J. (2013): *Design and applications of modern Stirling engines*. Combustion Engines, 3, 243-249.

13. Kropiwnicki J., et al. (2017): *Analysis of the possibilities of using of DME fuel in motor boat drive systems*. Combustion Engines, 4, 74-80.

14. Labeckas G., et al. (2018): *The effect of oxygenated diesel-n-butanol fuel blends on combustion, performance, and exhaust emissions of a turbocharged CRDI diesel engine*. Polish Maritime Research, 1(97), Vol. 25, 108-120.

15. Lane N. W., Beale W. T. (1999): *A biomass-fired 1 kWe Stirling engine generator and its applications in South Africa*. 9th International Stirling Engine Conference, South Africa, June 2-4.

16. Litwin W., Leśniewski W., Kowalski J. (2017): *Energy efficient and environmentally friendly hybrid conversion of inland passenger vessel*. Polish Maritime Research, 4(96), Vol. 24, 77-84.

17. MAN Energy Solutions (visited: 30.09.2019): https://turbocharger.man-es.com

18. Olszewski W., Dzida M. (2018): *Selected combined power systems consisted of self-ignition engine and steam turbine*. Polish Maritime Research Special Issue, S1(97), Vol. 25, 198-203.

19. Paul C. J., Engeda A. (2015): *Modeling a complete Stirling engine*. Energy, 80, 85-97.

20. Ramesh U. S., Kalyani T. (2012): *Improving the Efficiency of Marine Power Plant Using Stirling Engine in Waste Heat Recovery Systems*. International Journal of Innovative Research & Development, 1(10), 449-466.

21. Saab (visited: 30.09.2019): https://saab.com/naval/submarines-and-surface-ships/submarines/submarines/

22. Thombare D. G., Verma S. K. (2008): *Technological development in the Stirling cycle engines*. Renewable and Sustainable Energy Reviews, 12, 1-38.

23. Urieli I. (visited: 12.05.2019): *Stirling Cycle Machine Analysis*. https://www.ohio.edu/mechanical/stirling/

24. Wärtsilä (visited: 30.09.2019): https://www.wartsila.com/encyclopedia/term/waste-heat-recovery-(whr)

25. Wrona J., Prymon M. (2016): *Mathematical modeling of the Stirling engine*. Procedia Engineering, 157, 349- 356.

26. Yutuc W. E. (2016): *A Study on the Use of Stirling Engine Generator to Reduce Fuel Oil Consumption Onboard a Tanker Ship*. Journal of Engineering and Applied Sciences, 11(9), 2044-2049.

27. Zmuda A. (2010): *Estimation of the possibility of Stirling engine applications in LNG carrier power systems*. Scientific Journals of Maritime University of Szczecin, 21(93), 98-104.

CONTACT WITH THE AUTHOR

Jacek Kropiwnicki
e-mail: jkropiwn@pg.gda.pl
Gdansk University of Technology,
Faculty of Mechanical Engineering,
Ul. Narutowicza 11/12, 80-233 Gdańsk,
Poland