# Hybrid Stirling Solar Engine Micro-Cogeneration Unit simulation for given localities of the Czech Republic

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**Abstrakt**

Využití Stirlingova solárního motoru, tepelného motoru s vnějším přívodem tepla v podobě zkoncentrovaného slunečního záření, není pro severní zeměpisné šířky příliš přínosné. Rešerše předchozích výzkumů Stirlingova solárního motoru a mikrokogeneračních jednotek vyústila v novou koncepci tzv. Hybridního Stirlingova solárního motoru (HSSM)- mikrokogenerační jednotky založené na technologii Dish/Stirling se Stirlingovým motorem s volnými písty. HSSM využívá jak sluneční energii, tak i doplňkový zdroj tepla, plynový hořák, který je začleněn v kolektoru HSSM, jehož hlavní funkcí je zajištění min. elektrického a tudíž i tepelného výkonu (10 $kW_e$, resp. 14,6 kW) požadované doby provozu (24h a 8h provoz). Článek prezentuje výpočtový model HSSM a jeho simulaci v programu MATLAB pro zadané parametry a lokalitu České republiky - Prahu. Výsledky simulace poukazují na významné ušetření konvenčního zdroje energie, zemního plynu.

**Abstract**

Use of solar powered Stirling engine, the external combustion heat engine utilizing concentrated solar radiation, is rare for higher latitudes. The review of previous research on Stirling solar engine and micro-cogeneration units resulted in new concept of solar powered micro-cogeneration unit: Hybrid Stirling Solar Engine (HSSE) - micro-cogeneration unit based on Dish/Stirling technology with Free-piston Stirling engine (FPSE). HSSM uses both concentrated solar power and additional source of heat, the gas burner integrated in HSSM receiver with main purpose to keep minimum electric power (10 kW) and consequently thermal power (10 $kW_e$ and 14,6 kW, respectively), during required time operation (24h and 8h of operation). The paper presents computational model of HSSM and its simulation in program MATLAB for given locality of the Czech Republic – Prague . The results of simulation has revealed substantial savings of conventional source of energy, the natural gas.

## 1. Introduction

The first solar-powered Stirling engine (SSE) was invented in 1870 when the engine was adapted by Ericcson to operate with solar energy [1]. It was a open-cycle hot-air engine using a spherical mirror concentrator to heat steam and which used steam to drive the engine. There were many attempts to construct effective SSE until 80's when Dish/Stirling (DS) technology was developed by *Advanco Corporation*, *United Stirling AB*, *McDonnell Douglas Aerospace Corporation (MDA)*, the *US Department of Energy (DOE)*, and *NASA's Jet Propulsion Laboratory* [2]. Nowadays *Stirling Energy Systems* company introduced a matured SSE 25 $kW_e$ unit using dish/Stirling technology and its first attempt of commercialization - 20-year power purchase agreement to build 500 $MW_e$ (20,000 units, each of 25 $kW_e$) solar plant using this technology in California [3]. It would be the world's largest solar facility.

## 2. Introduction to new concept of Hybrid Stirling Solar Engine Micro-Cogeneration Unit (HSSE)

### 2.1. The overall system

As indicated in Fig. 1, the dish/engine systems use a parabolic mirror (for that reason called as Dish/Stirling systems) to reflect and concentrate incoming direct normal insolation to focal point where the power conversion unit (PCU) is situated. The PCU is consisting of cavity into which the sun rays are concentrated, receiver absorbing the heat from concentrated radiation and transferring to Stirling engine by means of heater tubes or heat pipe.. Stirling engine converts the energy into mechanical energy that subsequently drives alternator.

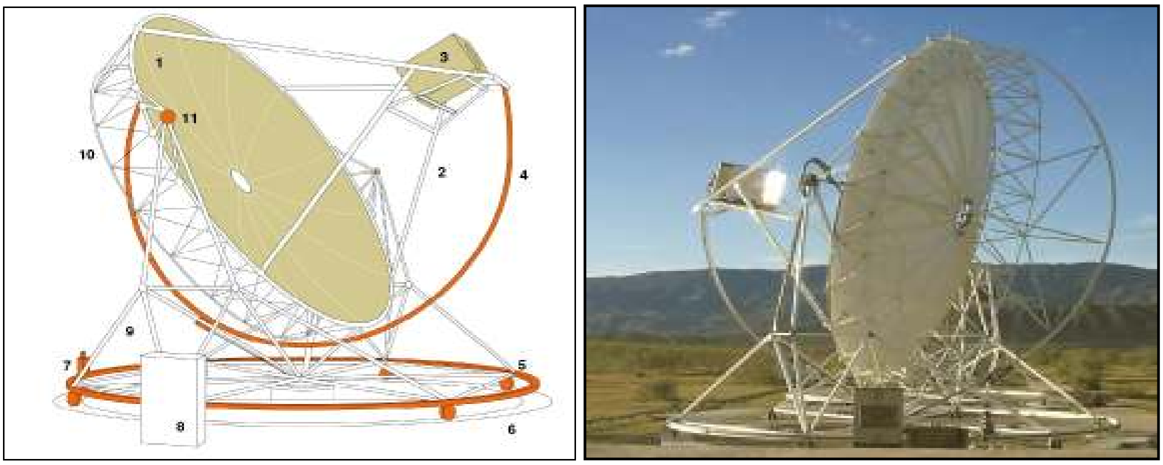


*Figure 1: The scheme of Dish/Stirling system (left) - the Envirodish – and its current prototype (right) developed by German-Spain-France consortium [4] : 1) concentrator, 2) PCU support structure, 3) PCU, 4) elevation drive arc, 5) azimuth drive arc, 6) foundation, 7) azimuth drive, 8) switch case, 9) elevation bearing, 10) concentrator ring truss, 11) elevation bearing.*

Dish/engine systems are characterized by high efficiency, modularity, autonomous operation, and an inherent hybrid capability (the ability to operate on either solar energy or fossil fuel, or both). Of all solar technologies, dish/engine systems have demonstrated the highest solar-to-electric conversion efficiency (29.4%) and annual efficiency (25%) [4]. Therefore they have the potential to become one of the least expensive sources of renewable energy. The modularity of dish/engine systems allows them to be deployed individually for remote applications, or grouped together for small-grid (village power) or end-of-line utility applications.

Simultaneously, there are efforts by *ENATEC* company from Netherlands to commercialize a new micro-cogeneration unit using Free-piston Stirling engine (FPSE) with gas burner for households. Gas burner is situated close to heater head of FPSE and can be modulated in various output. Part of the heat from the Stirling burner is released to the linear alternator of FPSE and part to the central heating system by means of cooling water of FPSE. The Stirling micro CHP indicated that Dish/engine system equipped with FPSE and gas burner can also be used as cogeneration unit buildings. Thus, both systems review resulted in new concept – Hybrid Stirling Solar Engine Micro-Cogeneration Unit, showed in Fig. 2 [5].

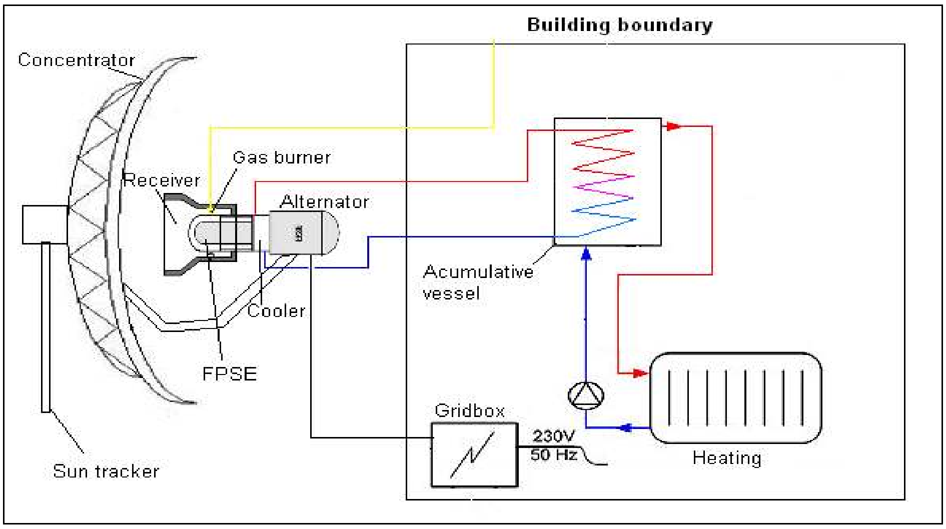


*Figure 2: Scheme of HSSE system and its application for building. The yellow line represents distribution of gas to Stirling gas burner, the red and blue lines represent heat transfer fluid for heating system of building and for cooling FPSE respectively, thus utilizing the waste heat of engine for building [5].*

HSSM uses both concentrated solar power and additional source of heat - the gas burner incorporated in HSSM receiver of PCU with main purpose to keep minimum electric and consequently thermal power (for instance 10 $kW_e$ and 14,6 kW, respectively) during the absence of solar radiation and during required time operation simultaneously. The thermal power is produced by means of FPSE cooler and stored in accumulative vessel.

The magnitude and parameters of the HSSE system are designed according location and energy demand of building. The higher latitudes the larger concentrator is required in order to capture favorable amount of solar energy. For instance, the location of 25 kW SSE (from *SES*) in Los Angeles, state California, represents the 55,000 kWh of electricity. It corresponds to 14,526 $m^3$ of natural gas burned in gas turbine power plant [3]. This is only the conversion of solar energy into electrical. The possibility of using waste heat occurred during Stirling thermodynamic cycle is obvious. Thus overall efficiency of the system and thermal efficiency of FPSE is enhanced.

### 2.2. Principle of Free-piston Stirling engine (FPSE)

Stirling engine used in SSE systems is high-temperature, high-pressure externally heated engine that uses a hydrogen or helium, the hermetically sealed working gas. Due to concentrated solar energy the working gas reaches temperatures as high as 800 °C (depending on concentration ratio, defined as the ratio of concentrator area and receiver area). In the Stirling cycle, the working gas is alternately heated and cooled by constant-temperature and constant-volume processes. Stirling engines incorporate an efficiency-enhancing regenerator that captures heat during constant-volume cooling and replaces it when the gas is heated at constant volume. The higher efficiency of regenerator is the higher efficiency of Stirling engine [6]. Fig. 3 shows the four basic processes of a Stirling engine incorporated in Dish/Stirling system, so-called α-modification, which consists of a V engine, with both pistons coupled to a common crankshaft. The spaces above the pistons constitute the compression and expansion volumes coupled by a duct containing the regenerator and additional heat-exchangers (heater tubes of receiver and cooler).

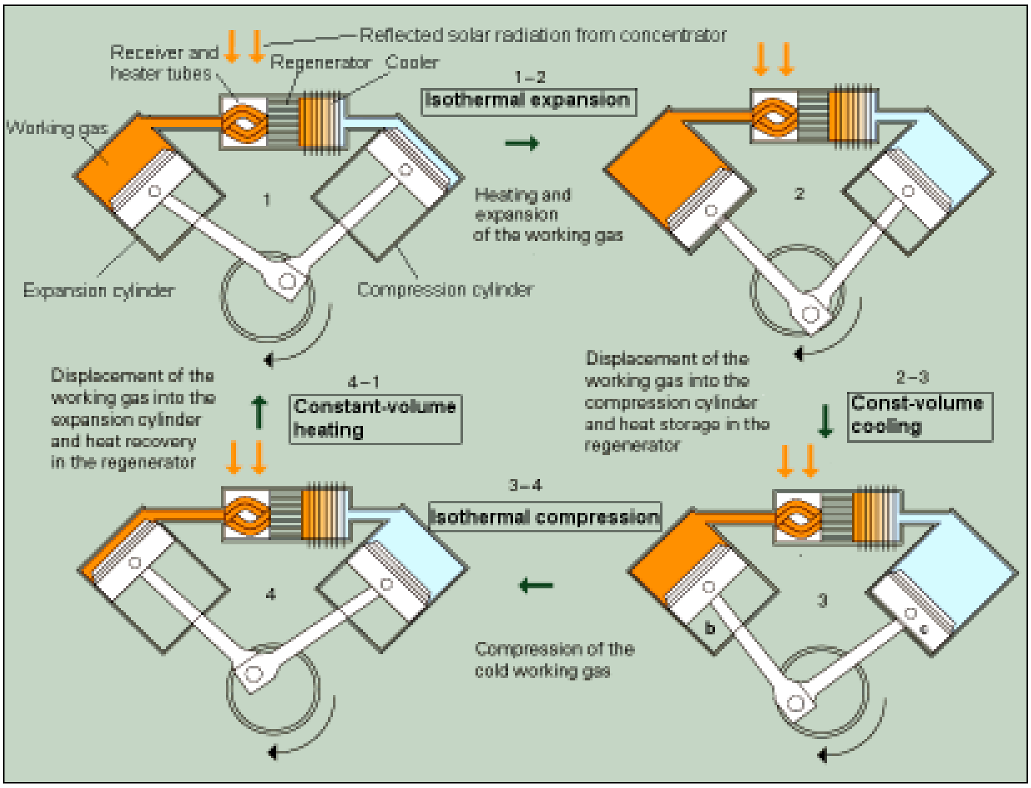


*Figure 3: Description of four thermodynamic processes occurring in Stirling cycle during solar irradiation.*

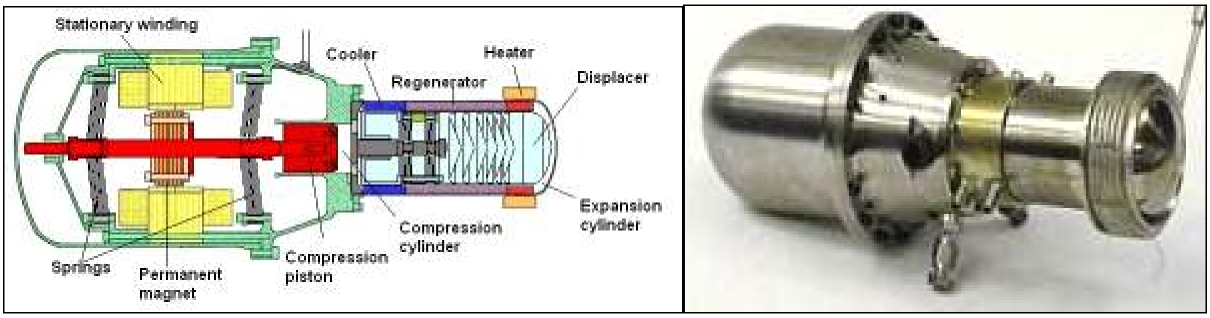


*Figure 4: Scheme (left) and photo (right) of current Free-piston Stirling engine manufactured by Infinia - Stirling Technology Copany [7].*

There are a number of mechanical configurations that implement these constant-temperature and constant-volume processes. Most involve the use of pistons and cylinders. Some use a displacer (a piston that displaces the working gas without changing its volume) to shuttle the working gas back and forth from the hot region to the cold region of the engine. For most engine designs, power is extracted kinematically by a rotating crankshaft.

An exception is the free-piston configuration initially designed by W. Beale in the late 1960's [6]. This configuration is the most up-and-going. Free piston Stirling engines (FPSE) have no kinematic mechanism coupling the reciprocating elements to each other or to a common rotating shaft. Instead, the elements move entirely in response to the gas or other spring forces acting upon them as shown in Figure 5.

The main advantages of FPSE relative to kinematic Stirling engines are: the simpler mechanical design, the possibility to avoid any side force on the piston and also, by using flexure bearings, the possibility to eliminate all rubbing parts, reducing engine wear, the high energy conversion (thermal to mechanical) efficiency, easy starting, high performance, very long life, and low cost [8]. For all that the FPSE haven't been utilized in solar energy. In case of HSSE micro-cogeneration unit it's a crucial component.

## 3. Methods

The simulation of complete HSSM micro-cogeneration unit in different operational modes is performed in MATLAB and based on its computational model demonstrated in the Fig. 5. The main outputs are electrical and thermal power during the solar operational mode ($P_{el.}$ and $P_{heat}$, respectively) and corresponding amount of el. and thermal energy ($Q_{el.}$, $Q_{heat}$, respectively). Additionally is considered so called hybrid mode which means that the gas burner is in operation if the $P_{el}$ is lower than 10 kW$_e$ (24h hybrid mode) or burner is in operation if the $P_{el}$ is lower than 10 kW$_e$ during the period from 8:00 until 16:00, only (8h hybrid mode). Thus another power, gas burner power $P_{gas,hyb}$, is computed via FPSE efficiency obtained from FPSE computational thermodynamic model. Corresponding amount of electrical and thermal energy produced in hybrid mode is $Q_{el,hyb.}$ and $Q_{.heat,hyb}$, respectively.

Firstly it is computed the receiver (heater) temperature $T_h$ according to intensity of solar radiation $I_{b,n}$ incident on concentrator every hour during the reference year (taken from the database created in simulation program TRNSYS), according to concentrator efficiency $\eta_{con.}$ and receiver efficiency $\eta_{re}$. The efficiencies mentioned above as well as concentrator area, $S_{con}$, namely 53 m$^2$, are specified in accordance with measured values performed on Envirodish system (electrical power $P_{el}$ = 10 kW$_e$) in Odeillo, France, on 9 January 2007 [9].

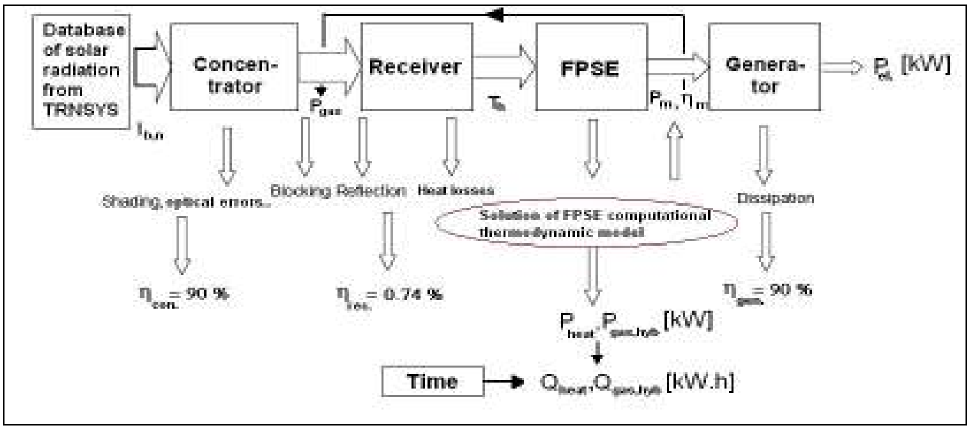


*Figure 5: Schematic diagram of HSSM micro-cogeneration unit computational model solved in MATLAB.*

Simultaneously parameters of FPSE are defined in order to complete the inputs for computational thermodynamic model. In Stirling engine during the compression and expansion processes and operational reasonable speed (say, 1000 rev/min), it is likely the processes are nearer adiabatic (no heat-transfer) than isothermal (infinite heat-transfer) which implies that the net heat transferred over the cycle must be provided by the heat exchangers. Thus, the computational thermodynamic model is based on "Ideal Adiabatic Model" (IAM) introduced by Urieli [10].

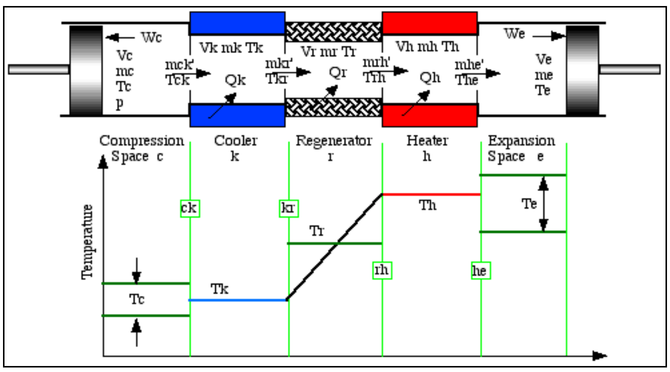


*Figure 6: Scheme of FPSE Ideal Adiabatic Model (IAM) with temperature diagram [10].*

Engine is considered as a five component serially connected model (Fig. 6), consisting respectively of a compression space *c*, cooler *k*, regenerator *r*, heater *h* and expansion space *e*. Each component is considered as a homogeneous entity or cell, the gas therein being represented by its instantaneous mass *m*, absolute temperature *T*, volume *V* and pressure *p*, with the suffix *c, k*, *r*, *h*, and *e* identifying the specific cell. The compression and expansion spaces are adiabatic, in which no heat is transferred to the surroundings. Double suffix (*ck*, *kr*, *rh*, *he*) representing the four interfaces between the cells. Enthalpy is transported across the interfaces in terms of a mass flow rate $\dot{m}$ and an upstream temperature *T*. The arrows on the interfaces represent the positive direction of flow, arbitrarily defined from the compression space to the expansion space. Notice from the temperature distribution diagram that the temperature in the compression and expansion spaces ($T_c$ and $T_e$) are not constant, but vary over the cycle in accordance with the adiabatic compression and expansion occurring in the working spaces. Thus the enthalpies flowing across the interfaces *ck* and carry the respective adjacent upstream cell temperatures, hence temperatures $T_{ck}$ and *The* are conditional on the direction of flow and are defined algorithmically as follows:

$$\text{if } \dot{m}_{ck} > 0 \text{ then } T_{ck} = T_c \text{, else } T_{ck} = T_k \text{,} \tag{1}$$

$$\text{if } \dot{m}_{he} > 0 \text{ then } T_{he} = T_h \text{, else } T_{he} = T_e \text{,} \tag{2}$$

In the ideal model, there is no gas leakage, the total mass of gas M in the system is constant, and there is no pressure drop, hence p is not suffixed and represents the instantaneous pressure throughout the system. Consider first the energy equation applied to a generalized cell (Fig. 7) which may either be reduced to a working space cell or a heat exchanger cell enthalpy is transported into the cell by means of mass flow $\dot{m}_i$ and temperature $T_i$, and out of the cell by means of mass flow $\dot{m}_o$ and temperature $T_o$.

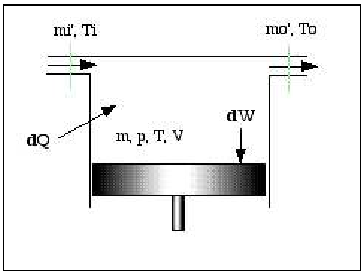


*Figure 7: Scheme of generalized cell representing either a working space cell or heat exchanger cell [10].*

The mathematical statement of first energy equation according to generalized cell is:

$$\mathrm{d}Q + \left(c_p T_i \dot{m}_i - c_p T_o \dot{m}_o\right) = \mathrm{d}W + c_v \mathrm{d}\left(mT\right), \tag{3}$$

where $c_p$ and $c_v$ are the specific heat capacities of the gas at constant pressure and constant volume respectively. This equation is the well known classical form of the energy equation for non steady flow in which kinetic and potential energy terms have been neglected. It's assumed that the working gas is ideal. This is a reasonable assumption for Stirling engines since the working gas processes are far removed from the gas critical point. Furthermore, the temperature profile of regenerator is linear, the pressure in each cell is homogenous and the heat transfer and friction coefficients for each exchanger are calculated using correlation. Applying the state, mass and energy equation for each control volume (cell) resulting in a differential equation system:

$$\mathrm{d}T_c = T_c\left(\frac{\mathrm{d}p}{p}+\frac{\mathrm{d}V_c}{V_c}+\frac{\mathrm{d}m_c}{m_c}\right), \tag{4}$$

$$\mathrm{d}T_e = T_e\left(\frac{\mathrm{d}p}{p}+\frac{\mathrm{d}V_e}{V_e}+\frac{\mathrm{d}m_e}{m_e}\right), \tag{5}$$

for temperatures in compression and expansion space, respectively, for three heat exchangers

$$\mathrm{d}Q_k = \frac{V_k c_v \mathrm{d}p}{r} - c_p\left(T_{ck}\dot{m}_{ck} - T_{kr}\dot{m}_{kr}\right), \tag{6}$$

$$\mathrm{d}Q_r = \frac{V_r c_v \mathrm{d}p}{r} - c_p\left(T_{kr}\dot{m}_{kr} - T_{rh}\dot{m}_{rh}\right), \tag{7}$$

$$\mathrm{d}Q_h = \frac{V_h c_v \mathrm{d}p}{r} - c_p\left(T_{rh}\dot{m}_{rh} - T_{he}\dot{m}_{he}\right). \tag{8}$$

and finally for the work done in the compression and expansion cells

$$\mathrm{d}W = \mathrm{d}W_c + \mathrm{d}W_e = p\mathrm{dV_c} + p\mathrm{dV_e}, \tag{9 -10}$$

The specific engine configuration and geometry defines $V_c$, $V_e$, d$V_c$, and d$V_e$ as analytic functions of cycle angle θ (θ = [0, 2π]), and the heat exchanger geometry defines the void volumes $V_k$, $V_r$, $V_h$. The chosen working gas - hydrogen - specifies $R$ (gas constant ), $c_p$, $c_v$, and $\kappa$ (specific heat ratio, $c_p/c$, ). The operating conditions are cooler temperature, $T_k$ = 313 K, heater (receiver) temperature $T_h$, and thus the mean effective temperature $T_r$. Specifying the total mass of working gas $m$ is a problem, since this is not normally a known parameter. The approach used was to specify the mean operating pressure, $p_{\mathrm{mean}}$ = 7,5 MPa, and then use the Schmidt analysis to evaluate $m$. Even though the IAM is independent of operating frequency we nevertheless specify it, $f$ = 25 Hz, in order to evaluate power and other time related effects (such as thermal conduction loss in the regenerator housing.). The values of $p_{\mathrm{mean}}$ and $f$ where specified accordance with experiences on current FPSE prototype.

The IAM is treated as "quasi steady-flow" system, thus over each integration interval the four mass flow variables $\dot{m}_{ck}$, $\dot{m}_{kr}$, $\dot{m}_{rh}$, *and* $\dot{m}_{he}$ remain constant and there are no acceleration effects. Thus we consider the problem as that of solving a set of seven simultaneous ordinary differential equations (eq. 4-10) by means of the classical fourth-order Runge-Kutta method for every value of $T_h$. System is formed as an initial value problem by assigning arbitrary initial conditions, and integrating the equations through several complete cycles until a cyclic steady state has been attained. The compression and expansion space temperatures are thus initially specified at $T_k$ and $T_h$ respectively. Work $W$ is done on the surroundings by virtue of the varying volumes of the working spaces $V_c$ and $V_e$, and heat $Q_h$ is transferred from the receiver at $T_h$ and heat $Q_k$ is rejected at the $T_k$ to the cooler , hence to the accumulative vessel The regenerator is externally adiabatic, heat $Q_r$ is transferred internally from the regenerator matrix to the gas flowing through the regenerator void volume $V_r$.

## 4. Results and Discussion

In the following charts results are presented obtained from solution of computational HSSM model in MATLAB, presented above. The main parameters are: $S_{con}$ = 53 m$^2$, $\eta_{con}$ = 0.9, $\eta_{rec}$ = 0.7, $T_k$ = 313 K, $f$ = 25 Hz, $p_{\mathrm{mean}}$ = 7,5 MPa. The first Fig. 8 presents the electric

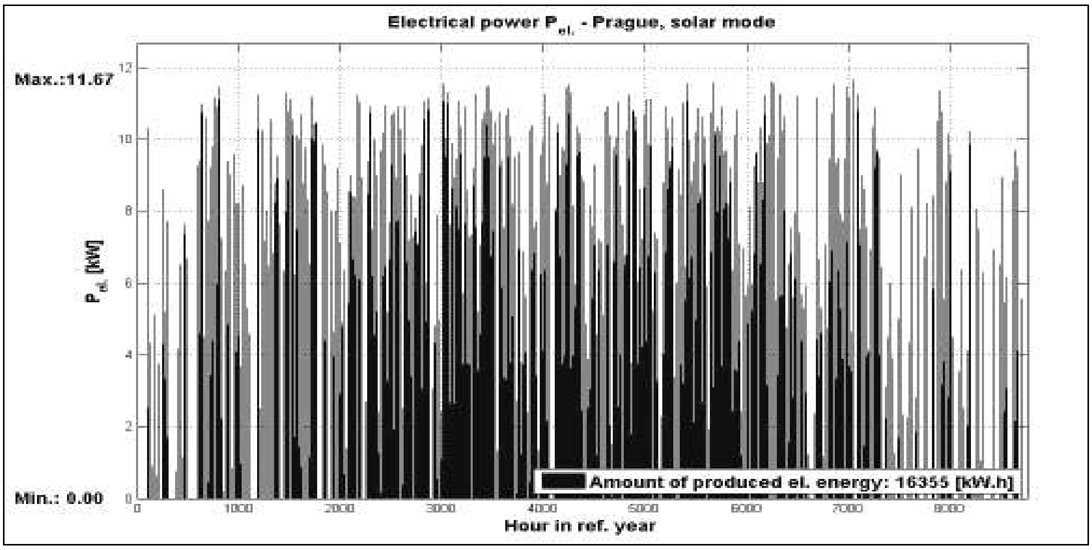


*Figure 8: Electric power $P_{el.}$ as a function of solar radiation during the ref. year in Prague (solar mode). The maximum $P_{el.}$ = 11.67 kW and the amount of produced el. energy $Q_{el.}$ = 16355 kW.h*

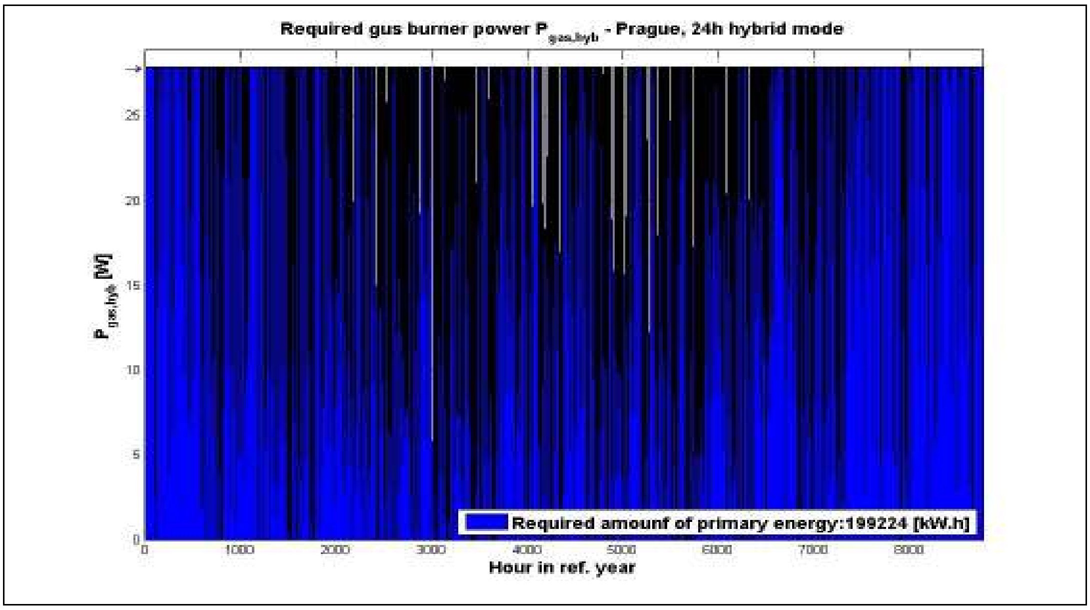


*Figure 9: Gas burner power $P_{gas,hyb.}$ as a function of solar radiation during the ref. year in Prague, 24h hybrid mode (gas burner is in operation if the $P_{el.}$ of the HSSM is lower than 10 $kW_e$). The maximum $P_{gas,hyb}$ = 27.87 kW (marked by arrow) and the required amount of primary energy $Q_{el,hyb.}$ = 199224 kW.h.*

power $P_{el.}$ of HSSM in Prague during the reference year (simulated in TRNSYS) in solar mode. It means that the el. power produced within the range 0 ÷ 11.67 kW and produced total amount of el. energy $Q_{el..}$ = 16.35 MW.h is obtained only during solar radiation. The production of thermal power as well as amount of heat is notable in solar mode: $P_{heat}$ is within the range of 7.33 ÷ 16.56 kW and $Q_{heat}$ = 34 MWh, respectiely (Fig. 10).

In the Fig. 9 is presented so called 24 hourly hybrid mode which means that additional gas burner is in operation if the $P_{el.}$ is lower than 10 $kW_e$. Thus, the min. $P_{el.}$ = 10 $kW_e$ and min. cooler thermal power $P_{heat}$ = 14.6 $kW_e$ (Fig. 11). The required energy amount of primary energy for gas burner during the 24h hybrid mode is enormous: $Q_{el,hyb.}$ = 199 MW.h. However, the great amount of "wasted heat" from cooler, $Q_{el,hyb.}$ = 128 MW.h can be utilized for building. Hence, another hybrid mode - 8 hourly - was performed on the same computational model in MATLAB, in order to simulate save mode. The results are presented

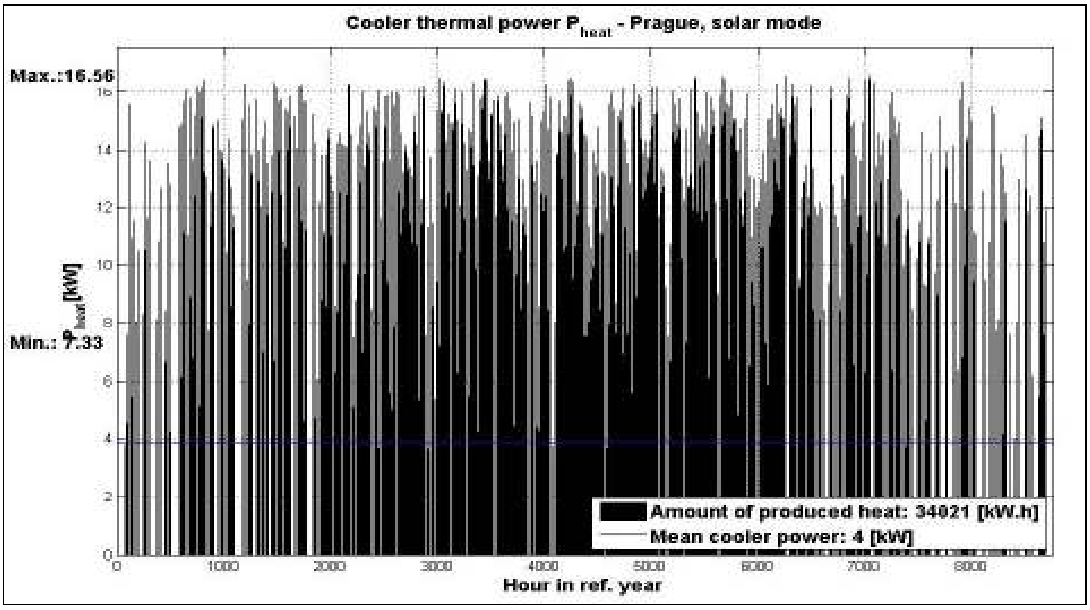


*Figure 10: Cooler power $P_{heat.}$ as a function of solar radiation, within the range of 7.33÷16.56 kW, during the ref. year in Prague (solar mode). The mean value of maximum $P_{heat.}$ = 4 kW and the amount of produced heat $Q_{heat.}$ = 34021 kW.h.*

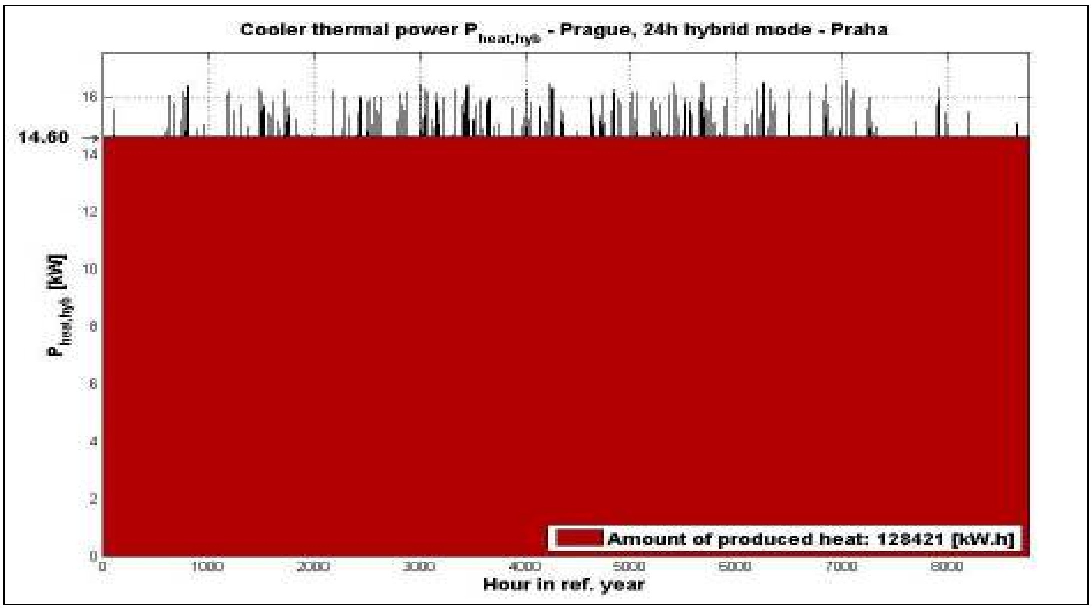


*Figure 11: Cooler thermal power $P_{heat,,hyb.}$ as a function of solar radiation during the ref. year in Prague, 24h hybrid mode (gas burner is in operation if the $P_{el.}$ of the HSSM is lower than 10 $kW_e$). The minimum $P_{gas,hyb}$ = 14.6 kW (marked by arrow) and the amount of produced heat $Q_{heat,,hyb.}$ = 128421 kW.h.*

in the Fig. 12 and 13. The amount of required primary energy decreased to $Q_{el,hyb.}$ = 49 MW.h. (Fig. 12) and simultaneously the amount of produced heat decreased to $Q_{el,hyb.}$ = 43 MW.h. The total amount of obtained energy (electrical and thermal) in 8h hybrid mode is 72,3 MW.h.

If we consider the consumed energy by gas burner in the 8h hybrid mode , $Q_{el,hyb.}$ = 49 MW.h, the "Coefficient Of Performance" (*COP*) is equal to 1,4 which corresponds to average heat pump (if the overall *COP* is considered - mainly the production of electricity in power plant). Finally, the total amount of saved energy performed by solar mode are noticeable: 50,38 MW.h ($Q_{el.} + Q_{heat,}$ Fig. 8 and 10, respectively) which corresponds to 4800 $m^3$ of natural gas considering the conventional gas furnace.

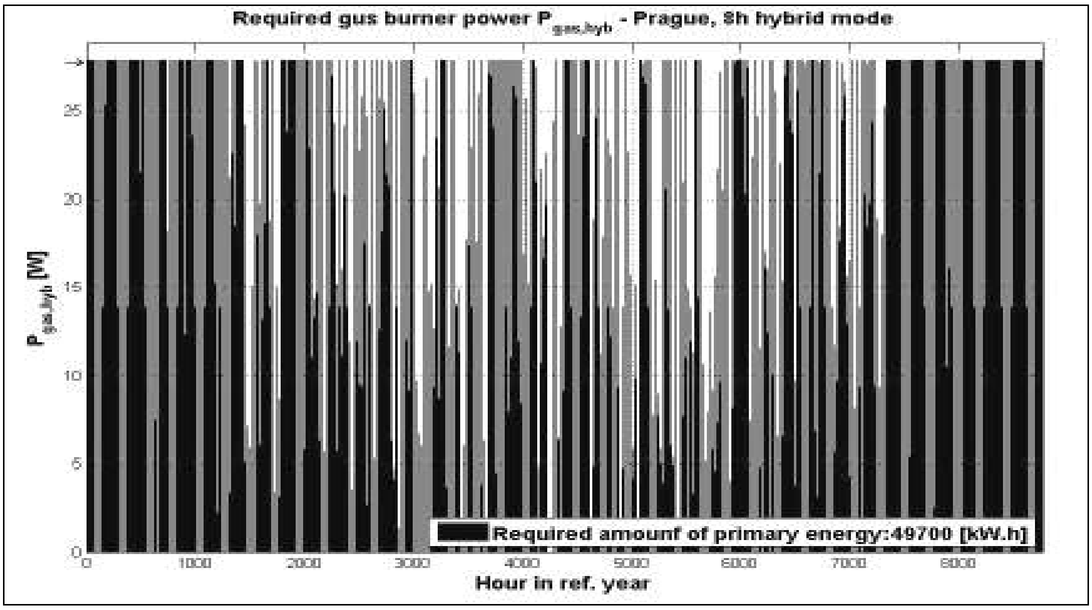


*Figure 12: Gas burner power $P_{gas,hyb.}$ as a function of solar radiation during the ref. year in Prague, 8h hybrid mode (gas burner is in operation if the $P_{el.}$ of the HSSM is lower than 10 $kW_e$ during the period from 8:00 until 16:00, only). The maximum $P_{gas,hyb}$ = 27.87 kW (marked by arrow) and the required amount of primary energy $Q_{el,hyb.}$ = 49700 kW.h.*

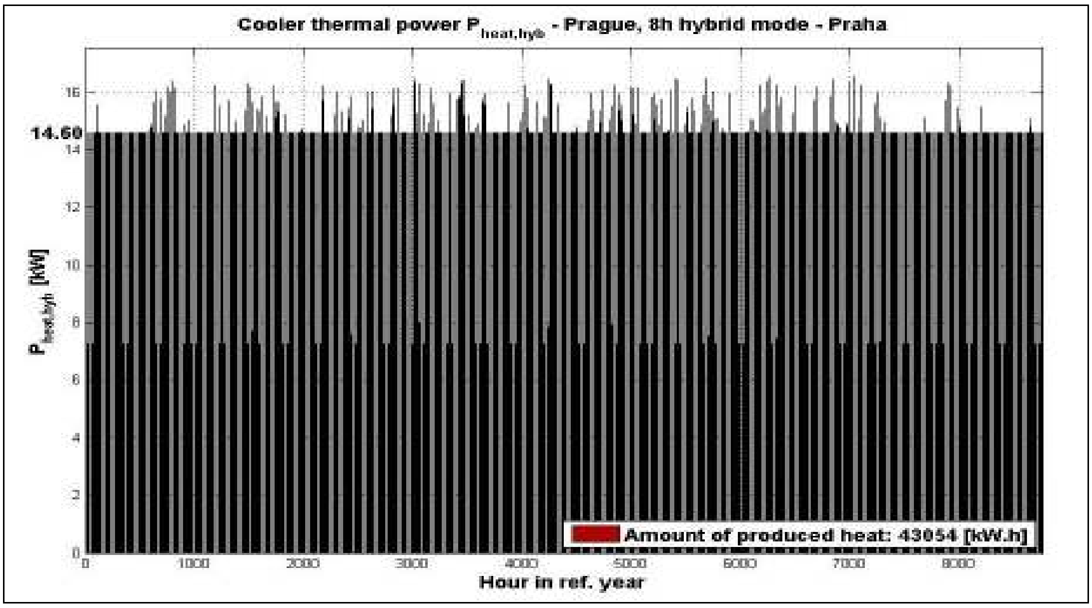


*Figure 13: Cooler thermal power $P_{heat,,hyb.}$ as a function of solar radiation during the ref. year in Prague, 8h hybrid mode (gas burner is in operation if the $P_{el.}$ of the HSSM is lower than 10 $kW_e$ during the period from 8:00 until 16:00, only). The minimum $P_{gas,hyb}$ = 14.6 kW (marked by arrow) and the amount of produced heat $Q_{heat,,hyb.}$ = 43054 kW.h.*

## 5. Conclusion

The computational model of new concept of micro-cogeneration unit based on Hybrid Stirling solar engine has been developed and simulated during solar and hybrid mode in program MATLAB. The crucial parameters ($S_{con}$, $\eta_{con}$, $\eta_{rec}$, $T_k$, $f$ and $p_{\mathrm{mean}}$) were obtained from measurements performed on Envirodish system in Odeillo, France, on 9 January 2007, and successfully applied on computational model. The results revealed the noticeable amount of saved conventional energy, namely 50,38 MW.h in solar mode. Furthermore, low noise and minimum maintenance are another advances acknowledge the HSSM micro-cogeneration unit subsequent development.

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