2009-08-31

Preliminary Modelling Results for an Otto Cycle/Stirling Cycle Hybrid-engine-based Power Generation System

Barry Cullen
Dublin Institute of Technology, barry.cullen@dit.ie

Jim McGovern
Dublin Institute of Technology, jim.mcgovern@dit.i

Michel Feidt
University Henri Poincare of Nancy, michel.feidt@ensem.inpl-nancy.fr

Stoian Petrescu
University Politehnica of Bucharest, stoian.petrescu@yahoo.com

Proceedings of ECOS 2009

22nd International Conference on Efficiency, Cost, Optimization
Simulation and Environmental Impact of Energy Systems
August 31 – September 3, 2009, Foz do Iguaçu, Paraná, Brazil

PRELIMINARY MODELLING RESULTS FOR AN OTTO CYCLE / STIRLING CYCLE HYBRID-ENGINE-BASED POWER GENERATION SYSTEM

Barry Cullen, barry.cullen@dit.it

Jim McGovern, jim.mcgovern@dit.ie

Traditional and Sustainable Energy Research Group, Dublin Institute of Technology, Bolton Street, Dublin 1, Ireland

Stoian Petrescu, stoian.petrescu@yahoo.com

Department of Engineering Thermodynamics, University Politehnica of Bucharest, Splaiul Independentei, 313, 060042 Bucharest, Romania

Michel Feidt, michel.feidt@ensem.inpl-nancy.fr

L.E.M.T.A., U.R.A.C.N.R.S.7563, University "Henri Poincare" of Nancy1, Avenue de la Foret de Haye, 54504, Vandoeuvre-les-Nancy, France

Abstract

This paper presents preliminary data and results for a system mathematical model for a proposed Otto Cycle / Stirling Cycle hybrid-engine-based power generation system. The system is a combined cycle system with the Stirling cycle machine operating as a bottoming cycle on the Otto cycle exhaust.

The application considered is that of a stationary power generation scenario wherein the Stirling cycle engine operates as a waste heat recovery device on the exhaust stream of the Otto cycle engine.

This paper is primarily concerned with the development of a model for a suitable Stirling cycle engine capable of running on the high grade thermal energy present in the exhaust stream of the industrial Otto cycle engine under consideration. The Otto cycle engine is not modelled, with the relevant engine exhaust and performance parameters taken from published data. This was deemed a suitable step as the Otto cycle engine is an established technology and the engines on which the model is to be based are commercially available. Therefore use of the published data in this manner was favourable as it limited the extent of the model required and allowed focus on the Stirling cycle engine requirements.

The modelling procedure for the Stirling engine follows the traditional sequence of Zero Order, First Order, Second Order as suggested in the classical literature. Zero Order analysis is completed in the form of the Beale analysis, First Order analysis utilises a Schmidt style method while the Second Order analysis utilises the Direct Method model.

Modelling is based on data available for an industrial Otto engine system operating on natural gas, with the input variables being specified as the exhaust recoverable energy content and temperature at a constant speed of 1500rpm.

1. INTRODUCTION

The concept of heat recovery from a generation plant for the purpose of additional power generation, as opposed to space or process heating, is well established. Combined cycle gas turbines were first installed in the United States in 1949 (Cengel and Boles 2002). Since then a wealth of different systems have been developed and installed. The typical form for such systems utilises a Brayton cycle gas turbine coupled to a Rankine cycle steam turbine operating on steam raised through heat recovery from the exhaust system of the gas turbine. Such a coupling offers significant increases in total plant efficiency. Maximum net efficiency of such systems can reach 60% (Franco and Casarosa 2002; Engelbert, Fuller et al. 2004). Typical single cycle power plant generally operate at efficiencies under 40%.

Combined cycle systems involving reciprocating engines such as Otto cycle or Diesel cycle engines are less common but exist. Review of the literature reveals a number of patents and other publications concerning waste heat recovery from the exhaust system of Otto engines for the express purpose of additional mechanical power generation. This is done using a Rankine cycle turbine (Clawson 2006; Badami, Mura et al. 2008), or in other cases a Stirling cycle engine (Hiroshi Yaguchi and Daisaku Sawada 2003; Mori 2004; Sawada 2007). Similar systems for Diesel cycle engines also appear (Johansson 2003). Otto cycle engines are routinely used as prime movers for small scale (<4MW) power systems such as light industrial cogeneration systems and stand-by power generation. The issues pertaining to efficiency gains in such engines have been analysed in previous work(Cullen and McGovern 2007). Electrical power efficiencies for such systems generally increase with rated output, with the largest units typically capable of efficiencies above 40%.

The present study was undertaken to investigate the possible gains that may be made through the use of a Stirling cycle engine as a bottoming cycle on a natural gas fired Otto cycle engine exhaust waste heat. The exhaust heat is considered due to its high grade (763K). The Stirling engine is considered due to the relative simplicity of the plant and its high theoretical efficiency. It may therefore warrant consideration for inclusion in a combined cycle plant over a Rankine cycle engine. Comparison of the two systems is not included in the current work though.

2. METHODOLOGY

Previous work on the combination of the two engines in the manner proposed has been completed by the first author. A rudimentary thermodynamic analysis yielded expressions for the total work and total efficiency of the combined plant from the ideal cycles(Cullen and McGovern 2008). Also, an energy system study of the concept for automotive use was completed(Cullen and McGovern 2008). The methodology of the present study follows closely that developed in the latter work, with thermodynamic data for the Otto engine being taken from manufacturer specifications. These were the engine brake power, exhaust gas temperature and recoverable energy content, and the engine cooling circuit temperature. These parameters were then used as source and sink constraints for the modelling of the Stirling engine.

3. THE OTTO CYCLE INDUSTRIAL ENGINE

The Otto cycle industrial engine selected for the simulation was a Liebherr TBG 9408 natural gas fired cogeneration unit. Its rated brake power at 1500 rpm is 255 kW. A summary of its full specification is offered in Tab. 1(Oberdorfer 2005). The exhaust temperature and sensible energy is utilised as the thermal source for the Stirling engine, with the cooling water system used as the thermal sink. Arrangement of the Stirling engine water cooling circuit is treated as being in series with that of the Otto engine.

Table 1. – Technical Data Liebherr TBG 9408 Natural Gas Spark Ignition Engine, 1500rpm

Technical Data – Liebherr TBG 9408				
Load	%	100	75	50
Mechanical Power	kW	255.5	192	128
Exhaust Thermal Power (cooled to 393K)	kW	160	125	88
Exhaust Gas Temperature	K	763	763	763
Engine Cooling Water Jacket Temperature	K	351	351	351
Brake Thermal Efficiency, η	%	35.6	34.5	33.1

4. THE STIRLING ENGINE

The central concern of the modelling process was the selection of a Stirling cycle engine suitably designed to operate with the exhaust stream of the Otto cycle engine as the thermal source, and the water circuit of the Otto engine as the thermal sink. Rather than a “ground-up” design of the engine, the method employed was to conduct a survey of existing engine designs currently or recently available and to attempt to match such a design to the specified Otto engine. Such a method is desirable due to time and cost constraints on design and testing of a bespoke engine.

4.1 Modelling of the Stirling Engine

Modelling and simulation of the Stirling engine was conducted in accordance with the traditional sequence of Zero, First and Second Order steps, each successive step offering a more rigorous approximation to real engine operation.

4.1.1 Zero Order

The term Zero order modelling was first used to describe the rudimentary "back of the envelope" type calculations that were first popularised by William Beale of Sunpower Inc in Athens, Ohio (Walker, Fauvel et al. 1994). Its genesis lies in empirical observation and experience more than mathematical and scientific principles. As a result, its use is traditionally considered difficult to justify(Walker 1980; Walker, Fauvel et al. 1994). Analyses based on the ideal cycle are typically described as Zero order as well (Walker, Fauvel et al. 1994). Various studies of this type have been developed. Erbay and Yavuz (Erbay and Yavuz 1997) analyse the cycle for the case of polytropic processes in the power and displacer volumes. Feidt et al (Feidt 1995) develop a model based on the ideal cycle and a generalized form of the heat transfer law at the source and sink. Account is also taken of the effect of various irreversibilities in the system Tlili et al (Tlili, Timoumi et al. 2008) develop a model based on the ideal cycle but include for irreversibility in the engine and imperfect regeneration. The equation originally derived by Beale has been expressed in a number of forms. The form adopted for this study was that developed by Walker, West and Senft as indicated by Kontragool and Wongwises(Kongtragool and Wongwises 2005):

$$P = p_m V_p f F\left(\frac{1-\tau}{1+\tau}\right) \tag{1}$$

Equation (1) offers an expression for the brake power output of the Stirling engine P, as a function of the mean pressure within the engine, p_m, the swept volume of the power piston, V_p the frequency at which it operates, f, and the ratio of the source and sinks temperatures, τ. The term F is an empirical factor. For the ideal cycle, F is 2. A value in the range 0.25 – 0.35 is offered as being representative of real engines. The equation can be easily manipulated to specify Vp as the subject:

$$V_p = \left(\frac{P}{p_m f F\left[\frac{(1-\tau)}{(1+\tau)}\right]} \right) \tag{2}$$

The first step of the model was to approximate the power that might be recovered from the exhaust stream of the Otto engine through a rudimentary energy balance. A typical thermal efficiency for a Stirling engine can be specified as circa 30%. Therefore, multiplying this by the available energy in the exhaust stream will allow us an approximate insight into the power output of a Stirling cycle engine running on this thermal source. For the Otto engine under investigation, the thermal output of the exhaust is 160kW. Therefore, a 30% efficient Stirling engine might convert 48kW of this to recoverable mechanical energy. We proceed with this as the design rating for the Stirling output.

We wish to determine the required swept volume for the Stirling engine to produce 48kW at the nominal 1500rpm to match the Otto engine speed. We therefore require more information in the form of the source and sink temperatures, the mean engine cycle pressure, and finally we specify the factor F.

The source temperature is 763 K, the temperature of the Otto exhaust, the sink is 351 K, the Otto cooling water temperature. The mean pressure is arbitrarily specified as 20MPa and the F value is selected as 0.3.

Under these conditions, equation (2) yields a power piston swept volume of 865cc.

In order to briefly analyse the effect of source temperature variation on the expected power output and size requirement of the engine, a short sensitivity analysis was performed.

Arbitrarily increasing the source temperature to 1000 K for all other parameters maintained the same indicates a required power piston swept volume of 666 cc, a reduction of 23% on that required for the lower temperature.

For the swept volume of 865cc, an increase in temperature to 1000K indicates a power output of 63kW, an increase of 31% on that possible at the lower temperature.

The calculated swept volume is 865cc and is used as a first estimate for the following modelling step, the first order model for source temperature of 763 K.

4.1.2 First Order

First Order modelling is a term that describes a level of analytical study of the ideal Stirling cycle that was originally developed in the nineteenth century by the German mathematician Gustav Schmidt (Urieli and Berchowitz 1984). The term "First Order" has been generally adopted to signify its priority over other "Zero Order" methods in terms of mathematical rigour and approach. Although it still relies on highly idealised assumptions and therefore generates profoundly optimistic performance predictions, it nonetheless is a useful tool for analysis and understanding of the cycle. The assumptions made in the model are described at length elsewhere (Urieli and Berchowitz 1984; Walker, Fauvel et al. 1994).

A number of treatments similar to the type developed initially by Schmidt exist. Organ presents a summary of the numerous different models available, along with useful information on the merits and limits of each(Organ 1992). Urieli and Berchowitz (Urieli and Berchowitz 1984), Thombare (Thombare and Verma 2006), Reader (Reader and Hooper 1983) and Hargreaves(Hargreaves 1991) among others all provide isothermal models similar to Schmidt's original work. Carlson et al (Carlson, Commisso et al. 1990) develop an ideal model with non isothermal heat exchange. Models of this type suggest an improvement on the isothermal model as they eliminate the necessity for infinite heat transfer and impractically slow engine speed associated with isothermal working spaces. Finkelstein(Finkelstein 1998) presents an alternative isothermal model that aims to avoid some of the pitfalls of Schmidt style analysis. Kontragool and Wongwises (Kongtragool and Wongwises 2006) offer an isothermal model with emphasis on system dead volume and imperfect regeneration. Walker (Walker 1980) develops a similar model in terms of four dimensionless ratios pertaining to the engine geometry. The advantage of this method is the removing of geometric design of engine components to a secondary design phase, however its usefulness is disputed by some (Urieli and Berchowitz 1984). In all cases, Schmidt style models are highly idealised and therefore unsuitable for use as sole design aids. The engine efficiency predicted by the Schmidt model for instance is equivalent to the Carnot efficiency for the given source and sink temperatures, an impossibility in any real engine. Considerable work has been done though on correlating the actual performance of an engine to the Schmidt style analysis prediction for that engine. It is therefore possible to use a correction factor to align the ideal model with realistic performance expectations. This treatment is advocated in the literature as an acceptable initial design step. Walker suggests a correction factor of between 0.3 and 0.6 as suitable to align with real operating limits, with 0.5 being suggested as representative of a reasonably well designed engine.

The First Order modelling step taken in the present study utilised the Schmidt type approach offered by Walker. This approach was adopted due to its use of the dimensionless parameters and the body of work available for optimisation of these parameters. It was therefore possible to model a Stirling engine based on the principle of scaling from the specific work parameter. Reference to the consolidated design charts compiled by Walker (Walker 1980) allows selection of the following ratios: τ, the ratio of the cold end temperature to the hot end temperature T_c/T_h; κ, the ratio of the swept volumes in the system, V_{comp}/V_{exp}; α, the phase angle by which volume variations in the expansion space *lead* those in the compression space; χ, the ratio of the dead volume in the system to the expansion space volume. The method employed for the model was to calculate the brake power and efficiency using the modelling equations, then applying a correction factor of 0.5 to align the model results with realistic operating limits, as outlined above.

The following values were selected: $\tau = 0.46$; $\kappa = 0.85$; $\alpha = 0.55\pi$ rads; $\chi = 1$. Application of the simulation equations for the power piston swept volume derived from the Beale analysis yielded the following results after correction:

Table 2. – Results of First Order Model, First Iteration. Stirling Engine at 1500rpm

First Order Analysis - Results Iteration 1		
Mean Pressure	MPa	20
Mechanical Power	kW	62
Expansion Space Swept Volume, V_{exp}	cc	865
Total Swept Volume, V_T	cc	1600
Brake Thermal Efficiency, η	%	26.9

The results in Tab. 2 indicate that a second iteration of the model is required. The power output and thermal efficiency of the engine indicate that insufficient energy is available in the exhaust stream to allow the power output calculated. The second iteration therefore involved alteration of the expansion space swept volume, V_{exp} until the required heat admission to the system matches that available from the Otto exhaust. The results for this second iteration are given in Tab. 3.

Table 3. – Results of First Order Model, Second Iteration. Stirling Engine at 1500rpm

First Order Analysis - Results Iteration 2		
Mean Pressure	MPa	20
Mechanical Power	kW	43
Expansion Space Swept Volume, V_{exp}	cc	604
Total Swept Volume, V_T	cc	1117
Brake Thermal Efficiency, η	%	26.9

The results presented in Tab. 3 are considerably more favourable. By implication the heat required from the source for the engine, the Otto engine exhaust gases, is 159.85kW, an almost exact match for that known to be available.

4.1.3 Second Order

Second order analysis is a further step in complexity for modelling of the engine. Whilst the first order method is an improvement in terms of mathematical complexity and rigour over the zero order methods, it is nonetheless a highly idealised representation of the engine and its performance. Use of empirical factors such as the correction factor used above is a way of overcoming the optimism of the model outputs; however it is not necessary to elaborate on the shortcomings of this method. Implicit in the correction factor used are losses associated with real engine operation. Second order analysis attempts to quantify these losses. The term generally refers to the de-coupled methods of analysis that offer a refinement of the first order method by accounting for the various loss mechanisms that occur in the engine. These can be generally defined as Heat Transfer losses and Flow Power losses (Dyson, Wilson et al. 2004). Inherent in this method is engine component design. A wealth of analyses exist that offer design schemes for components such as heat exchangers, regenerators, seals, pistons, displacers and all associated components. Readers are referred to (Organ 1992; Organ 1997; Feidt 2000; Feidt 2002; Senft 2002; Kuosa, Kaikko et al. 2007) for more detailed descriptions. Other methods exist. Petrescu et al (Petrescu, Stanescu et al. 1992; Petrescu and Harman 1994; Petrescu, Costea et al. 2002) have applied the Direct Method for processes with Finite Speed to the Stirling cycle to account for the irreversibilities present in the non ideal situation. Domingo details a solution method for a Second order method applied to Stirling heat pumps (Domingo 1985).

The Direct Method was utilised in the present study. This method involves the direct integration of equations based on the first law for processes with finite speed. The technique was adopted due to its good agreement with actual engine performance data . Its inclusion of detailed geometric parameters concerning the engines and its predictive capability render it a suitable second order model.

Initially a United Stirling 4-95 Mark II engine was selected as a likely candidate for the combined scenario. This engine is established and has been used in the Vanguard solar thermal system in the past (Stine and Diver 1994). This model was selected as its rated output appeared congruent with that which was expected from the First order analysis.

The main geometric and performance data for the 4-95 Mk II engine are given in Tab. 4(Stine and Diver 1994).

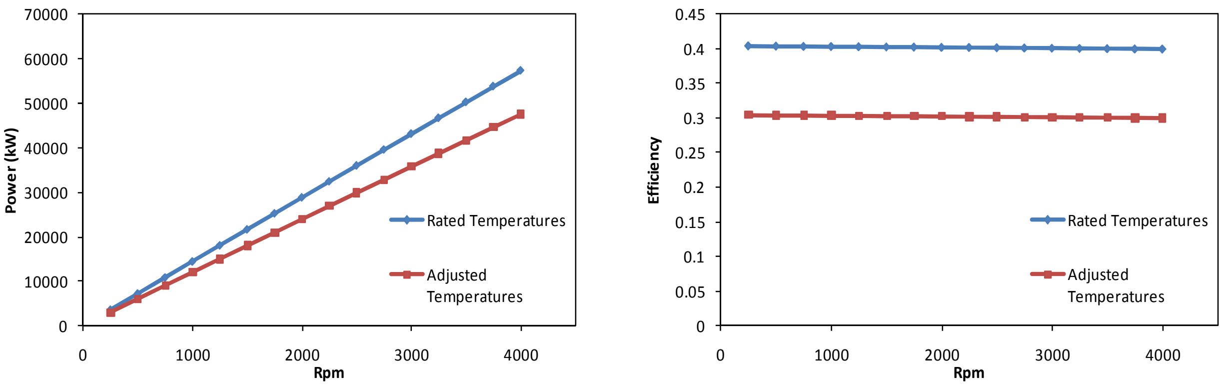
As in the first order model, an iterative procedure was used for the second order model. The 4-95 Mk II was modelled using the Direct Method. As the Otto engine exhaust gas was to act as the thermal source for the engine and the cooling water as the sink, the system model was altered to reflect these changes. The source temperature was adjusted to the 763K of the exhaust gases. The sink temperature was adjusted to 351K of the jacket cooling for the Otto engine. The Figure 1 shows the power and efficiency for the 4-95MkII engine under both rated conditions and the modified temperature conditions.

It can be seen from the figure that the engine is capable of producing the required 48kW at approximately 4000 rpm. Therefore in the combined engine scenario, the two engines would be loaded at different speeds, as the Otto engine operates at 1500rpm.

Alternatively, to assess if geometric changes to the engine might reduce the required engine speed, it was decided to perform a sensitivity analysis on several of the parameters of the engine to investigate what affect there would be on the engine power output. The parameters investigated were 1) the cylinder diameter, 2) the regenerator diameter. Details of the altered parameters for the analysis are presented in Tab. 5. The simulation results of the analysis are presented in Figure 2.

Table 4. – Technical Data United Stirling 4-95 Mk II Engine, at 1800rpm

Technical Data - United Stirling 4-95 Mk II		
Power (rated)@1800rpm	kW	25
Swept Volume	cc	540
Bore	mm	55
Stroke	mm	40
Gas Temperature (high)	K	993
Coolant Temperature	K	323
Mean Gas Pressure	MPa	20
Brake Thermal Efficiency, η	%	41
Configuration	double - acting alpha	
Working Gas	-	Hydrogen
No. of Cylinders	-	4

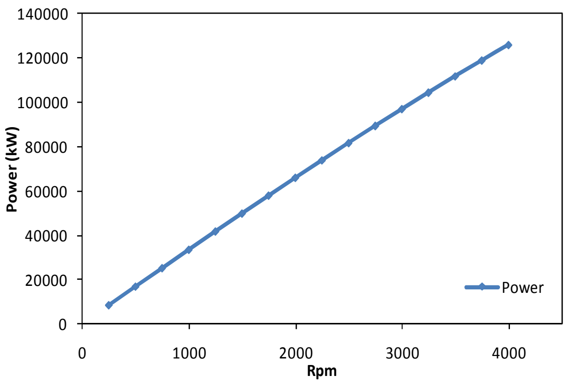


(a) (b)

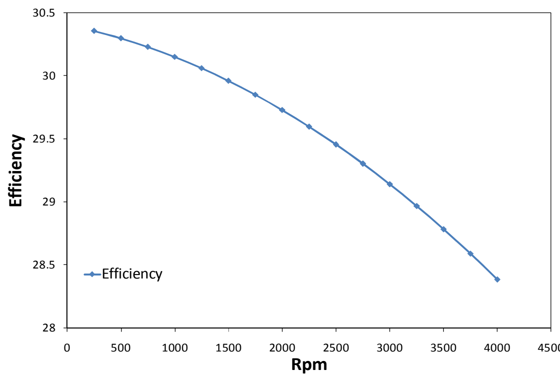
Figure 1. Simulation Performance Data – United Stirling 4-95 Mk II engine with rated and altered source and sink temperatures (a) Power (b) Efficiency

Table 5. – Sensitivity analysis parameters

Sensitivity Analysis - Parameters		
	Iteration 1	Iteration 2
Cylinder Diameter (m)	0.055	0.095
Regenerator Diamter (m)	0.057	0.067



(a)



(b)

Figure 2. Simulation Performance Data – United Stirling 4-95 Mk II engine with altered system parameters (a) Power (b) Efficiency

The sensitivity analysis demonstrates that with the increased dimensions presented in Table 5 the engine is capable of producing 49.8kW at 1500rpm. Interestingly, the working volumes generated through the model are a close match for the sister engine, the United Stirling 4-275. As complete geometric data was not available for the 4-275 engine, though, it was not possible to fully investigate this.

5. THE COMBINED GENERATOR

The combined system involves the use of the Stirling cycle engine as a waste heat recovery device on the exhaust stream of the Otto engine.

The combined engine is therefore simulated by combining the outputs of the Otto cycle natural gas engine and the Stirling cycle engine as in the following expression:

$$P_{hybrid} = P_{Otto} + P_{Stirling} \tag{3}$$

Where P_{hybrid} is the total power of the hybrid engine, P_{Otto} is the brake power output of the Otto cycle engine and $P_{Stirling}$ is the brake power output of the Stirling cycle engine. Similarly, the total first law efficiency of the unit is calculated with reference to the total energy input from the fuel supplied to the Otto cycle Natural Gas Engine:

$$\eta_{hybrid} = \frac{P_{hybrid}}{P_{fuel}} \tag{4}$$

Where η_{hybrid} is the brake efficiency of the combined unit and P_{fuel} is the energy input per second of the fuel supplied to the Otto cycle engine.

The results of the simulation of the hybrid are presented in Tab. 6. Although the sensitivity analysis indicates a possible power output of 49.8kW for the Stirling at 1500 rpm, the final result is taken as 48kW to allow for the energy balance as mentioned previously.

Table 6. Simulation Results – Hybrid engine based stationary power generation module, 1500rpm

Simulation Results - Hybrid Engine		
P_{fuel}	kW	717
Potto	kW	255
Pstirling	kW	48
Phybrid	kW	303
η_{Otto}	%	35.5
$\eta_{Stirling}$	%	29.9
η_{Hybrid}	%	42.2

6. CONCLUSIONS AND COMMENTS

Simulation results for an Otto cycle / Stirling cycle hybrid engine based stationary power generation prime mover are presented. A power gain of 48kW is predicted over the Otto engine system operating alone. This represents a brake thermal efficiency gain of 6.7% on the Otto cycle system operating alone. The additional power may be realised by using the United Stirling 4-95 MkII engine loaded at 4000 rpm or by altering certain geometric parameters to reduce to speed to approximately 1500 rpm to match that of the Otto engine.

The central work that is presented is the modelling of a suitable Stirling cycle engine that would be capable of operating on the high grade thermal energy available in the Otto engine exhaust stream. For this, an iterative method was utilised whereby an existing engine design was modelled using the Direct Method for processes with Finite Speed, with system parameters being altered until the work output of the Stirling engine is adjusted for the given inputs.

The system is a basic combined cycle; however the engine technologies considered are not those that are conventionally used, for instance the Brayton cycle and the Rankine cycle systems. The Otto cycle natural gas engine that is the central generator considered is typically used in small to medium scale industrial and commercial sector applications as standby power generators and as combined heat and power generators. The present analysis considers only the base case of mechanical power generation alone and does not take account of heat recovery issues that might affect application as a Combined Heat and Power generator. It is anticipated that such analyses will be presented at a later date.

Capital costs incurred in the addition of the Stirling engine to the system have not been analysed. A full appraisal of the system would necessarily include a cost-benefit analysis, which is planned for future publication.

7.0 RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper

References

Badami, M., M. Mura, P. Campanile and F. Anzioso (2008). "Design and performance evaluation of an innovative small scale combined cycle cogeneration system." Energy **33**(8): 1264-1276.

Carlson, H., M. B. Commisso and B. Lorentzen (1990). Maximum Obtainable Efficiency For Engines And Refrigerators Based On The Stirling Cycle. Energy Conversion Engineering Conference, 1990. IECEC-90. Proceedings of the 25th Intersociety.

Cengel, Y. A. and M. A. Boles (2002). Thermodynamics An Engineering Approach, McGraw Hill.

Clawson, L. G. (2006). High Efficiency Otto Cycle Engine with Power Generating Expander. U. S. P. Office. United States.

Cullen, B. and J. McGovern (2007). The Quest for More Efficient Industrial Engines: A Review of Current Industrial Engine Development and Applications. ASME Power, San Antonio, TX, ASME.

Cullen, B. and J. McGovern (2008). Energy System Feasibility Study of an Otto Cycle / Stirling Cycle Hybrid Automotive Engine. ECOS 2008. Krakow, Poland, AGH University of Science and Technology.

Cullen, B. and J. McGovern (2008). Proposed Otto Cycle / Stirling Cycle Hybrid Engine Based Power Generation System. ASME Power 2008. Orlando, Florida, ASME

Domingo, N. (1985). Comparative Analysis of a Stirling Heat Pump with Second and Third Order Computer Models, Oak Ridge National Laboratory.

Dyson, R. W., S. D. Wilson and R. C. Tew (2004). Review of Computational Stirling Analysis Methods, NASA.

Engelbert, C., R. A. Fuller, J. J. Fadok and B. Lueneburg (2004). Introducing the 1S.W501G Single-Shaft Combined-Cycle Reference Power Plant. ASME Power 2004. Baltimore, Maryland, ASME.

Erbay, L. B. and H. Yavuz (1997). "Analysis of the stirling heat engine at maximum power conditions." Energy **22**(7): 645-650.

Feidt, M., Costea, M., Popescu, G. (1995). Optimal Design of a Stirling Engine using a Generalized Heat Transfer Law Model. ROMA '95 CIRCUS Centro Internazionale di Ricerca e Calcolo Universitario Scientifico Rome, Italy.

Feidt, M., Costea,M., Petrescu,S. (2000). Heat Transfer Conductance or Area Minimization in a Stirling Cycle Engine. ECOS 2000. Universiteit Twente, Nederlands.

Feidt, M., Lesaos,K.,Costea,M., Petrescu,S. (2002). "Optimal allocation of HEX inventory associated with fixed power output or fixed heat transfer rate input." International Journal of Applied Thermodynamics **5**(1): 25-36.

Finkelstein, T. (1998). "A new isothermal theory for Stirling machine analysis and a volume optimization using the concept of 'ancillary' and 'tidal' domains." Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science **212**(3): 225-236.

Franco, A. and C. Casarosa (2002). "On some perspectives for increasing the efficiency of combined cycle power plants." Applied Thermal Engineering **22**(13): 1501-1518.

Hargreaves, C. M. (1991). The Philips Stirling Engine. Amsterdam, Elsevier Science Publishers B.V.

Hiroshi Yaguchi and Daisaku Sawada (2003). Exhaust Heat Recovery Apparatus. United States Patent Office. United States, Toyota Jidosha Kabushiki Kaisha.

Johansson, L. (2003). Exhaust Gas Alternator System. U. S. P. Office. United States of America, STM Power Inc.

Kongtragool, B. and S. Wongwises (2005). "Investigation on power output of the gamma-configuration low temperature differential Stirling engines." Renewable Energy **30**(3): 465-476.

Kongtragool, B. and S. Wongwises (2006). "Thermodynamic analysis of a Stirling engine including dead volumes of hot space, cold space and regenerator." Renewable Energy **31**(3): 345-359.

Kuosa, M., J. Kaikko and L. Koskelainen (2007). "The impact of heat exchanger fouling on the optimum operation and maintenance of the Stirling engine." Applied Thermal Engineering **27**(10): 1671-1676.

Mori, M. (2004). Power Device Equipped with Combustion Engine and Stirling Engine. E. P. Office, Honda Motor Co. Ltd.

Oberdorfer (2005). Technical Data OD250 Natural Gas Oberdorfer Kraft-Wärme-Kopplung G.m.b.H.

Organ, A. J. (1992). Thermodynamics and Gas Dynamics of the Stirling Cycle Machine, Cambridge University Press.

Organ, A. J. (1997). The Regenerator and the Stirling Engine, London Mechanical Engineering Publications.

Petrescu, S., M. Costea, C. Harman and T. Florea (2002). "Application of the Direct Method to irreversible Stirling cycles with finite speed." International Journal of Energy Research **26**(7): 589-609.

Petrescu, S. and C. Harman (1994). "The Connection Between the First Law and Second Law of Thermodynamics for Processes with Finite Speed - A Direct Method for Approaching and Optimization of Irreversible Processes " Journal of the Heat Transfer Society of Japan **33**(128): 60.

Petrescu, S., G. Stanescu, R. Constantinescu and A. Dobrovicescu (1992). The First Law of Thermodynamics for Closed Systems, Considering the Irreversibilities Generated by the Friction Piston - Cylinder, the Throttling of the Working Medium and the Finite Speed of the Mechanical Interaction. ECOS '92. Zaragoza, Spain, ASME.

Reader, G. T. and C. Hooper (1983). Stirling Engines. London, E. & F.N. Spon.

Sawada, D. Y., Hiroshi; Mitani, Shinichi (2007). Exhaust Heat Recovery Apparatus. World Intellectual Property Organisation, Toyota Jidosha Kabushiki Kaisha.

Senft, J. R. (2002). "Optimum Stirling engine geometry." International Journal of Energy Research **26**(12): 1087-1101.

Stine, W. B. and R. B. Diver (1994). A Compendium of Solar Dish/Stirling Technology, Sandia Laboratories.

Thombare, D. G. and S. K. Verma (2006). "Technological Development in the Stirling Cycle Engines." Renewable and Sustainable Energy Reviews(doi:10.1016/j.rser.2006.07.001).

Tlili, I., Y. Timoumi and S. B. Nasrallah (2008). "Thermodynamic analysis of the Stirling heat engine with regenerative losses and internal irreversibilities." International Journal of Engine Research **9**(1): 45-56.

Urieli, I. and D. M. Berchowitz (1984). Stirling Cycle Engine Analysis. Bristol, Hilger.

Walker, G. (1980). Stirling Engines. Oxford, Clarendon Press.

Walker, G., O. R. Fauvel, G. Reader and E. R. Bingham (1994). The Stirling Alternative, Gordon and Breach Science Publishers.