Optimization of Stirling Engine Heat Exchangers

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Abstract: The heat exchangers are the most important parts of stirling engine. These heat exchangers are Heater, Regenerator and Cooler which exchange heat to and from the engine. The design and configuration of these heat exchangers effect on engine performance. The optimization of these parts for a 20 kw stirling engine is shown in this paper.

Key-words: Stirling Engine, Cooler, Heater, Regenerator, Heat Exchanger, Optimization, Performance.

1. Introduction

Recent critical environmental and energy challenges caused to concentrate on new energy sources and trying to find effective means of energy conservation by researchers. Stirling engine is One of the most effective means for producing power. This engine was invented by Sir Robert Stirling in 1816. It is an externally heated, clean and runs almost silently on any source of energy [1].

Stirling engine operate under stirling cycle processes, which consist of two isothermal and two isochoric process. The engine works between two temperatures T_H and T_C, theoretically the thermal efficiency of the engine is equal to Carnot. Alternative compression and expansion of working gas under isothermal process cause to produce work. Main parts of engine are crank mechanism, heater, cooler and a regenerator. There are about 280 configurations for this engine based on drive mechanism, type and location of heat exchangers and working fluid. Heat exchangers in a stirling engine play a main rule on performance parameters of the engine so improving the design and construction of heat exchangers cause to improvement in engine performance[2].

2. Design goals

A 20 kW engine has been designed and built as a research tool for the investigation of multi-kilowatt Stirling engine performance. Details of this engine are given in table 1. While the original design of the engine is quite satisfactory for research purposes, its power output and efficiency can be improved upon. Such improvements are of interest if the engine is considered as a pilot model, for example, as a stationary generator[3].

Table 1. Engine specification

Engine configuration	Alpha	Phase angle	90°
Max. cycle pressure	15 MPa	Working gas	Helium
Piston stroke	5 cm	Displacer stroke	5 cm
Piston bore	9 cm	Displacer bore	9 cm
Cold swept volume	318 cm^3	Hot swept volume	318 cm^3
Heat sink Temp.	300 K	Heat source Temp.	900 K
Design speed	3000 rpm	Indicated power	9.8 kw
Indicated efficiency	0.11	Regenerator load	118 kw
Heater load	91 kw	Cooler load	59 kw

Without going into details of the circumstances under which the generator is to be employed, let it be supposed that a comparison of running costs with capital costs has shown a gain of 5% in power to be of equal benefit to a gain of 0.01 in efficiency. Some of the changes to be considered will also affect the construction costs of the engine, and penalty functions will be attached to these changes as they are discussed. Optimization will be carried out for a design speed of 3000 rpm, assuming that helium is to be used as the working gas and that the cycle pressure is limited to a maximum value of 15 MPa.

The goal to be pursued is therefore an optimization of power output against efficiency and construction costs. The modifications to be considered in this paper will be restricted to the heat exchangers. These components can be replaced without drastic alteration to the remainder of the engine, though the structure of the remainder of the engine will impose constraints on certain of the heat exchanger design variables. The method to be employed involves simulating an initial design in detail, then assessing the benefits of small modifications using the second-order design model, OPTIMUM.

The initial design specification and its predicted performance are given in table1, which was generated using the third-order program STRENG. (The figures for power and efficiency quoted in this paper are always for indicated power and efficiency. A rough estimate of the corresponding brake power and efficiency can be obtained by multiplying these figures by a factor of 0.8). The greater realism of the third-order model leads to the absolute values of its performance predictions being rather more pessimistic than those obtained from the second-order program OPTIMUM. Nevertheless, there is good agreement on the position of the optimum design.

3. Re-design of the Heater

The first step in re-design is to study the initial design point.

Table 2. Heater properties and performance

No. of tubes*	52	Tube i.d.	2.28 mm
Tube length	50 cm	Tube o.d.	4.76 mm
Dead volume	110 cm^3	Wall thickness	1.24 mm
Heat transfer area	1860 cm^2	Cross sectional flow area	2 cm^2
ΔT between metal & gas	25.8 K	ΔT across metal	11.8 K
Power lost in fluid friction	kw	Von-mises stress	62.9 MPa

*Material is 321 S/S

The detailed heater design is given in table 2. Some comment should be made on the source of the performance figures quoted. The figures for power output and efficiency are obtained from the third order program, STRENG. In this program, flow frictional work is calculated as a separate term, but the effects of the two secondary variables, dead volume and fluid temperature in the heater, are inextricably linked. While this is the most accurate way of calculating the effect of these variables on performance it is useful for design purposes to try to separate their individual contributions.

The first point to be noted from table 2 is that the heater tube wall thickness is overdesigned for the expected operating temperature and pressure. The stress limit for the tube material is 74 MPa, while the Von-Mises stress actually experienced is only 60 MPa. The stress level is a function of the ratio d_o/d_i, and it is seen from Figure 1 that this ration may be reduced to 1.3 without exceeding the safety limit. It is not possible to set this ratio finally at this stage, as stress is also a function of tube number and length. However, it may be born in mind that do can be reduced, allowing the geometric limit on the number of tubes to be relaxed.

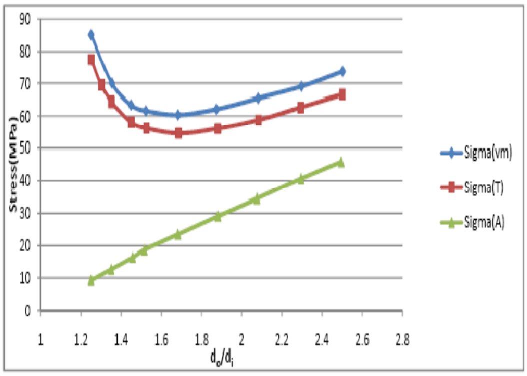


Fig. 1. Stress variation with diameter ratio

Table 2 gives a very high figure for fluid frictional work in the heater. This must be reduced, and a reduction can be accomplished in three ways. Firstly, heater tube length could be reduced. Frictional work would be reduced in proportion to tube length. Dead space in the heater would be reduced in the same ratio, but the temperature drop between source and working fluid would be increased in inverse ratio. The Von-Mises stress on the tubes would also be increased[4].

A second option is to increase the number of heater tubes, reducing pressure drop. It also has the consequence of increasing dead volume, of reducing the temperature drop across the tube metal, and, in much smaller degree of reducing the metal-to-gas temperature drop. The Von-Mises stress on the tube is reduced. With this option, unlike the others, a penalty in increased cost and difficulty of construction is attached to each additional tube. At some point, also, there will be a geometrical limit on the number of tubes that can be added. At the design point, this limit is 140 tubes.

The third option that of increasing the internal diameter of the tube, offers much the best way of achieving the very large reduction in pressure drop needed. It is seen that a 10% increase in tube diameter produces a drop of 50% in the frictional work. The dead volume is increased and the temperature drop between metal and working fluid is also slightly increased. If the outer diameter is varied to keep do/di constant, the Von-Mises stress will not be affected[4].

To select one of these methods of reducing frictional work, it is useful to know which of the other two secondary variables is the more critical. From table 3 it may be seen that a suitable strategy may be to reduce frictional losses by simultaneously shortening the tube length and increasing the tube diameter. This combination rapidly reduces friction without too great a penalty in increased dead volume. There is some reduction in heater temperature, but this can be accepted, as the temperature drop across the tubes at the design point is very small.

Table 3. Initial sensitivity analysis for heater

Change in Primary Variables		Effect on secondary variables			Effect on Power				Effect on efficiency				Total benefit
		ΔT	ΔP	V_{HD}	ΔT	ΔP	V_{HD}	NET	ΔT	ΔP	V_{HD}	NET	
L	-10%	+10%	-10%	-10%	-0.5%	+2%	+0.5%	+2%	-0.001	+0.006	--	+0.005	+0.009
d_i	+10%	+4%	-47%	+20%	-0.25%	+10%	-1%	+8.7%	--	+0.027	--	+0.027	+0.0447
n	+10%	-4%	-17%	+10%	+0.25%	+4%	-0.5%	+3.7%	--	+0.01	--	+0.01	+0.016
d_o/d_i	+10%	+2%	--	--	-0.1%	--	--	-0.1%	--	--	--	--	--
d_o/d_i	-10%	-2%	--	--	+0.1%	--	--	+0.1%	--	--	--	--	--

To find the optimum combination, the second-order program is used to investigate 40 combinations of tube length and diameter. The results of this simulation are shown in figure 2. It is seen that increasing the diameter of the tubes first lends to an increase in net power output, due to the reduction in fluid friction; further increase then leads to a fall in power, due to the increase in dead volume. The engine performance is less sensitive to changes in tube length. It is seen that shortening the tubes produces the greatest improvement for low values of di when frictional losses predominate. As tubes of wider bore are considered, the gains from reducing friction and dead volume are balanced by the loss in heat transfer area. Thus for a tube internal diameter of 3.25 mm, there is little to choose between tube lengths in the range 500-350 mm; the 450 mm tube is the best by a narrow margin.

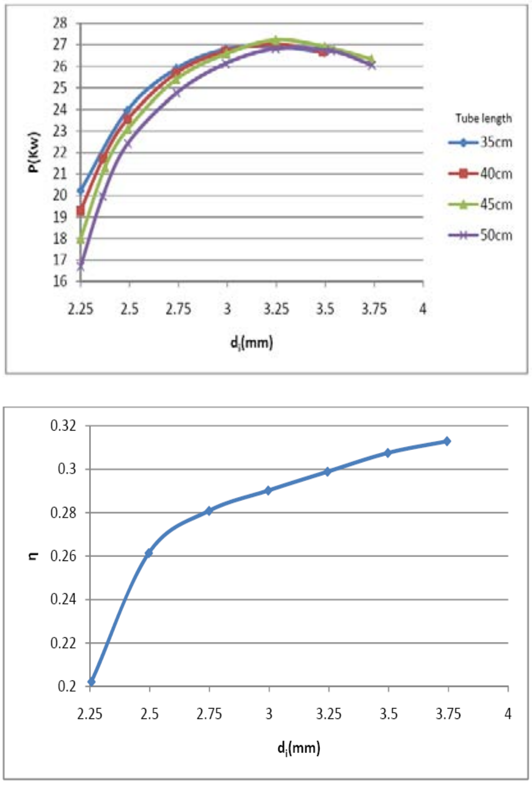
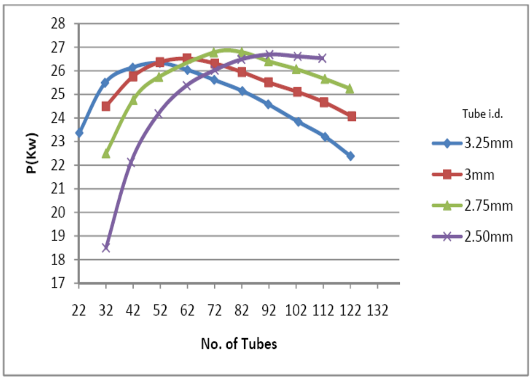


Fig. 2. Heater sensitivity (No. of tubes=52)

<u>Note: The variation in efficiency with tube length is slight and reflects the variation in power</u>

In fig. 3, the effects of varying the number of tubes and their internal diameter are investigated, keeping the tube length fixed at 450 mm. As the number of tubes is increased, performance is improved by the reduction in fluid friction until a maximum is reached, after which the increased dead volume becomes the predominant factor and performance falls away. The number of heater tubes needed to give optimum performance increases as the tube diameter is reduced; if these two modifications are made together, moving towards a design of very many fine tubes performance shows a slight, very slow improvement. It must be borne in mind that each additional tube brings an increase in construction costs. As the number of heater tubes must be doubled to produce an increase in power of 500 watts from the design point, it may well be argued that this modification is not justified.



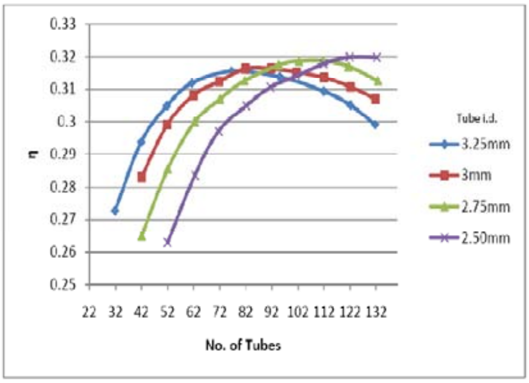


Fig. 3. Heater sensitivity (Tube length=45 cm)

It is concluded that the optimized heater will have 52 tubes, each 450mm long with an internal bore of 3.25mm. Stress calculations show that the corresponding tube external diameter should be 4.3 mm. STRENG is run to give a more accurate estimate of the new design's performance; an improvement in power of 8.5 kW and in efficiency of 0.10 is predicted as shown in table 4.

Table 4. The optimized heater and its predicted performance

No. of tubes*	52	Tube i.d.	3.25 mm
Tube length	45 cm	Tube o.d.	4.3 mm
Dead volume	110 cm^3	Wall thickness	0.525 mm
Heat transfer area	2400 cm^2	Cross sectional flow area	4 cm^2
ΔT between metal & gas	35 K	ΔT across metal	5 K
Power lost in fluid friction	kw	Von-Mises stress	74 MPa
Engine power improvement	8.5 kw	Engine efficiency improvement	0.10

4. Cooler Re-Design

Study of the cooler properties as given in table 5 shows that the chief limitation on performance is the poor heat transfer between the inner surface of the cooler tubes and the working gas, with a consequently high value for the mean temperature of gas in the cooler. The flow friction in the cooler is extremely low, even at 3000 rpm. This suggests that an improved performance may be obtained by reducing the tube bore and increasing their length and number. These options are compared in table 6.

Table 5. Cooler properties and performance

No. of tubes*	150	Tube i.d.	3.25 mm
Tube length	10 cm	Tube o.d.	4.00 mm
Dead volume	124 cm^3	Wall thickness	o.375 mm
Heat transfer area	1530 cm^2	Cross sectional flow area	12 cm^2
ΔT between metal & coolant	50.4 K	ΔT across metal	5.5 K
Power lost in fluid friction	w	ΔT tube to gas	31.4 K
Engine power **	18.3 kw	Engine efficiency **	0.21

* Material is 321 S/S

**Assuming heater design as given in table 4

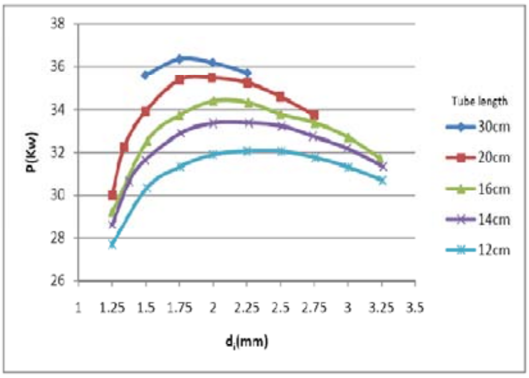
Table 6 suggests a reduction of bore is the best policy, perhaps. in combination with an increase in tube length.

Table 6. Initial sensitivity analysis for cooler

Change in Primary Variables		Effect on secondary variables			Effect on Power				Effect on efficiency				Total benefit
		ΔT	ΔP	V_{KD}	ΔT	ΔP	V_{KD}	NET	ΔT	ΔP	V_{KD}	NET	
L	+10%	-10%	+10%	+10%	+1.9%	Negle.	-1.36%	+0.54%	+0.0059	--		+0.0059	+0.007
d_i	-10%	-8%	+47%	-20%	+1.5%	Negle.	+3.01%	+4.51%	+0.0047	--		+0.0047	+0.014
n	+10%	-2%	-17%	+10%	+0.4%	Negle.	-1.36%	-0.96%	+0.0012	--		+0.0012	-0.0007

Optimization will proceed slightly differently if the flow rate of the coolant is regarded as fixed. In most applications, it should be possible to increase the coolant flow rate as necessary to keep the temperature drop between coolant and tube exterior constant. In a few cases, the supply of coolant may be limited, or the power required to pump the coolant may have to be deducted from the engine's power output. It will be assumed that neither of these limitations applies here; the optimization program can readily be modified to take them into account[5].

The effect on engine power and efficiency of reducing the bore of the cooler tubes while increasing their length is shown in fig.4. It is noted that reduction of the bore produces an increase in power due to the improved heat transfer and reduced dead volume; beyond a certain maximum, the marginal increase in fluid friction outweighs these gains. The same factors produce a steady fall in efficiency as bore is reduced. (It will be remembered that adiabatic analysis shows that reduction of dead volume produces slight fall in engine efficiency)[5].



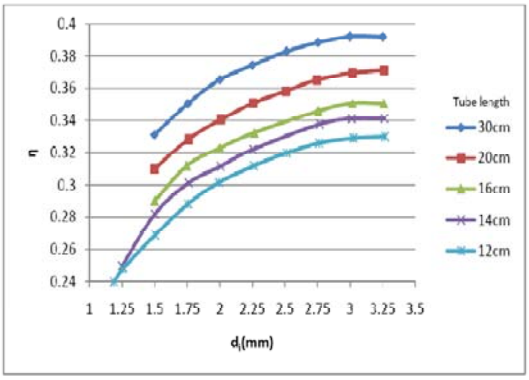


Fig. 4. Cooler sensitivity

The peak in the curve relating engine power to cooler tube internal diameter becomes both sharper and higher as the tube length is increased-sharper, because the two limiting factors of flow friction and dead volume are both increased in the longer tubes- higher, because the dominating factor between these two extremes is heat transfer, which is improved as the heat transfer area is increased. Efficiency increases with tube length, except for the narrowest tubes.

As in the case of the heater, it is found that heat transfer, and hence overall performance, is improved only very slowly by increasing the number of tubes. This parameter will therefore be left unmodified.

It is seen from fig.4 that the maximum efficiency and maximum power occur at different points. It is therefore necessary to make use of the design guide mentioned above, that a 5% improvement in power will be considered equivalent to an increase of 0.01 in efficiency. Applying this rule gives the optimum design as having tubes 300 mm long with a bore of 2.25 mm. This implies an increase in cooler tube length of 200 mm, however, which may necessitate an overall re-design and lead to an unacceptably bulky engine. For a penalty of 1 kW in power and 0.02 in efficiency, the cooler tubes can be made 200 mm long with a bore of 2.5 mm. Which of these options is preferred will depend on the details of the application being considered.The optimized cooler design is given in table 7.

Table 7. The optimized cooler and its predicted performance

No. of tubes*	150	Tube i.d.	2.25 mm
Tube length	30 cm	Tube o.d.	2.8 mm
Dead volume	179 cm^3	Wall thickness	0.275 mm
Heat transfer area	3800 cm^2	Cross sectional flow area	6 cm^2
ΔT between metal & coolant	8.5 K	ΔT across metal	2 K
Power lost in fluid friction	2.4kw	ΔT tube to gas	13 K
Engine power improvement	9.3 kw	Engine efficiency improvement	0.109

5. Regenerator Re-Design

The initial design point for the regenerator and properties of regenerator material are specified in tables 8 and 9. the sensitivity of regenerator performance to small departures from this design point is analyzed in table 10[6],[7].

Table 8. Regenerator specification and performance

Wire diameter d_w (Stacked screen)*	100 μm	Wall thickness	1 cm
Bore	9 cm	Length	5 cm
Porosity (ε)	0.6	Dead volume	115 cm^3
Cross sectional flow area	38 cm^2	Heat transfer area	3.05 m^2
Engine power**	27.5 kw	Efficiency	0.319
Thermal mass ratio	31	Flow ratio	1.7
Reduced length	97	Net enthalpy flux	3 kw
Heat pump power less	900 w	Power lost in friction	10 kw

*Material 321 S/S

**Assumes heater and cooler design as given in tables 4 & 7

Table 9. Material properties of 321 S/S

Conductivity	23 w/m.K	Specific heat	500 j/Kg.K
Young modulus	2×10^{11} N/m^2	Poisson ratio	0.283
Expansion coefficient	16×10^{-6} K^{-1}	Density	7800 Kg/m^3

Table 10 shows that the most serious weakness of the initial regenerator design is the high value for fluid friction. This may be reduced by increasing the matrix porosity, increasing the matrix fiber diameter, reducing the regenerator length or increasing the regenerator bore. The first two of these parameters can be varied without necessitating any changes in the overall engine design, by changing the matrix to be placed in the regenerator housing. It seems sensible to investigate this option first.

Table 10. Initial sensitivity analysis for the regenerator

Change in Primary Variables		Effect on secondary variables						Effect on Power			Effect on efficiency					Net change		
		V_{RD}	W_F	W_{HP}	H_∞	H_{HP}	Q_C	V_{RD}	W_F	W_{HP}	W_F	W_{HP}	H_∞	H_{HP}	Q_C	power	η	benefit
ε	+10%	+10%	-10%	-10%	+32%	+20%	--	-1.6%	+7.5%	-0.4%	+0.4%	-.002	-0.006	+0.001	--	+5.5%	+0.032	+0.042
d_W	-10%	+4%	-47%	+20%	-15%	--	--	--	-1.6%	--	-0.01%	--	+0.003	--	--	-1.6%	-0.007	-0.01
L	+10%	-4%	-17%	+10%	-10%	--	-10%	-1.6%	-1.5%	--	-0.01%	--	+0.002	--	+0.0004	-3.1%	-0.008	-0.014
D	+10%	+2%	--	--	-10%	--	+20%	-3.2%	+6%	--	+0.03%	--	+0.002	--	-0.0008	+2.8%	+0.031	+0.037

Changing the geometry of the matrix, from stacked screens to spheres, say, is likely to worsen rather than improve performance. There is also little scope for improvement by substituting another matrix material for stain less steel; the conductivity of the matrix has no significant effect on performance, and the thermal densities of feasible alternative materials differ little from that of stainless steel. So the only matrix properties of interest are porosity and wire diameter. The effects of varying these are shown in fig. 5[8].

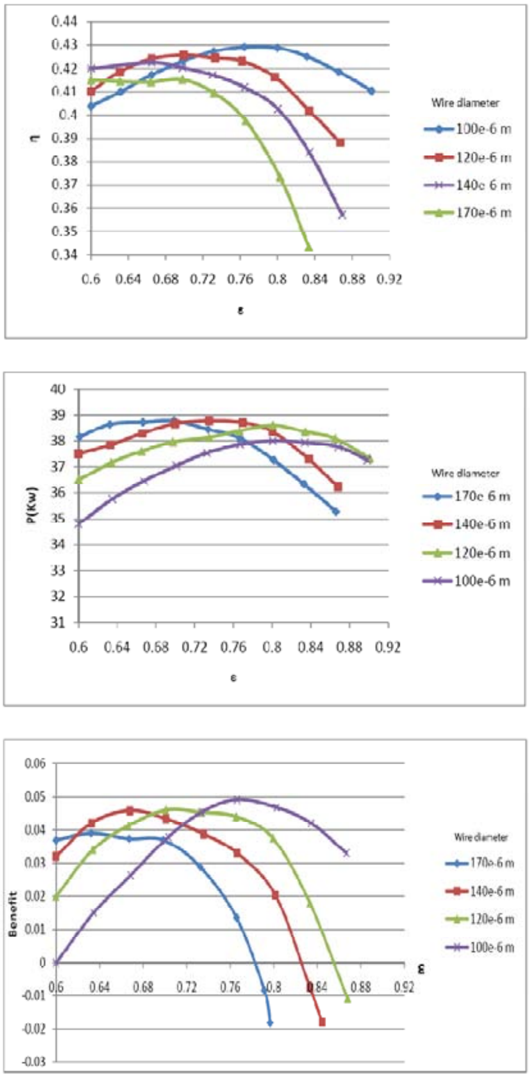


Fig. 5. Optimization of ε and d_w

Fig. 5 shows that regenerator performance is very sensitive to small variations in porosity. The curves presented show the effect of porosity varying from 0.60 to 0.76. Increasing porosity initially produces an increase in power and efficiency, due to the reduction in fluid friction. As porosity increases further, the reduction in effectiveness leads to a net reduction in engine efficiency. Next, the increase in the "heat pump" power loss and in dead volume bring about a net reduction in power. The optimum balance between these effects is determined by the wire diameter; the coarser the wires of which the matrix is made up, the denser the optimum porosity.

For the values of ε and d_w investigated, there are a range of combinations giving equal net benefit, though different balances between power and efficiency. A fine, porous matrix will increase efficiency at some cost in power, while a coarse, dense matrix gives higher power and lower efficiency.

The regenerator's performance can only be further improved by modifying its aspect ratio. Because this implies either a modification of the entire engine design, altering the cylinder bore to match that of the regenerator, or the introduction of abrupt changes in flow path cross-section, leading to increased dead volume and flow losses, a full investigation of this option cannot be made here. However, some idea of the influence of aspect ratio on performance can be gained from figs. 6 and 7, in which L and D are varied for regenerator matrices having wire diameters of 100 µm and 50 µm respectively[9].

Both cases show engine power increasing steadily as regenerator length is reduced. This is to be expected; power would be maximized by doing away with the regenerator altogether, provided enough heat could be supplied to make up for the resulting inefficiency. At a given length, power initially increases with increasing bore, this is a result of the reduction in friction. Beyond a certain point, this increase is outweighed by the increase in "heat pump" power loss and dead volume.

Efficiency is at first improved by shortening the regenerator, as a result of the reduction in fluid friction. Further shortening increases conduction losses and reduces regenerator effectiveness. Increase of the regenerator bore increases effectiveness, but also increases conduction losses[9].

Comparing figs. 6 and 7, it is seen that the finer the wires making up a regenerator, the shorter and wider its optimum dimensions will be. Fig.7, for a matrix made up of 50 µm wires, shows an optimum design having a length of between 1 and 2 cm and a bore of about 13 cm. In the construction of these sensitivity curves, the only factor limiting length reduction and increase of bore has been conductivity loss. In practice, difficulties in designing an engine around a very short, wide regenerator might impose a stricter limit.

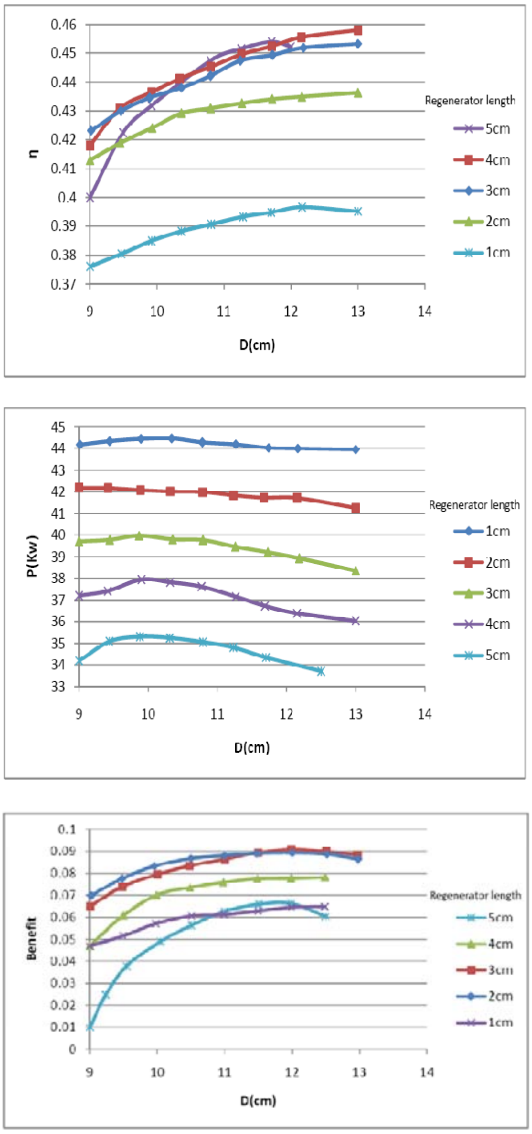


Fig. 6. Optimization of L and D (d_w=0.1 mm)

The data presented in figs. 5, 6 and 7 does not suffice to define the optimum regenerator design if all four of the primary variables are allowed to vary freely, though it does indicate the area in which the optimum may be expected to lie. If the regenerator bore is taken as fixed, a further improvement on the initial design point may be obtained by simply shortening the regenerator, which can be done without necessitating major changes in the rest of the engine. This option was investigated by varying ε against dw for a range of matrix lengths between 2.0 and 5.0 cm.

It was found that the optimum matrix porosity and wire diameter depended on matrix length; the shorter the matrix, the denser the optimum packing and the finer the wires. The height of the optimum did not vary as greatly as its position; at a length of 2.5 cm, the net benefit obtained from a matrix having wire diameter of 60 μm and a porosity of 0.72 was almost equal to that of a matrix with wire diameter 70 μm and porosity 0.74, though the balance between improvement in power and in efficiency was different in each case. The optimum design point finally selected had a regenerator length of 2.5 cm, a wire diameter of 60 μm and a porosity of 0.72. The full specification and performance of this design are given in table 11.

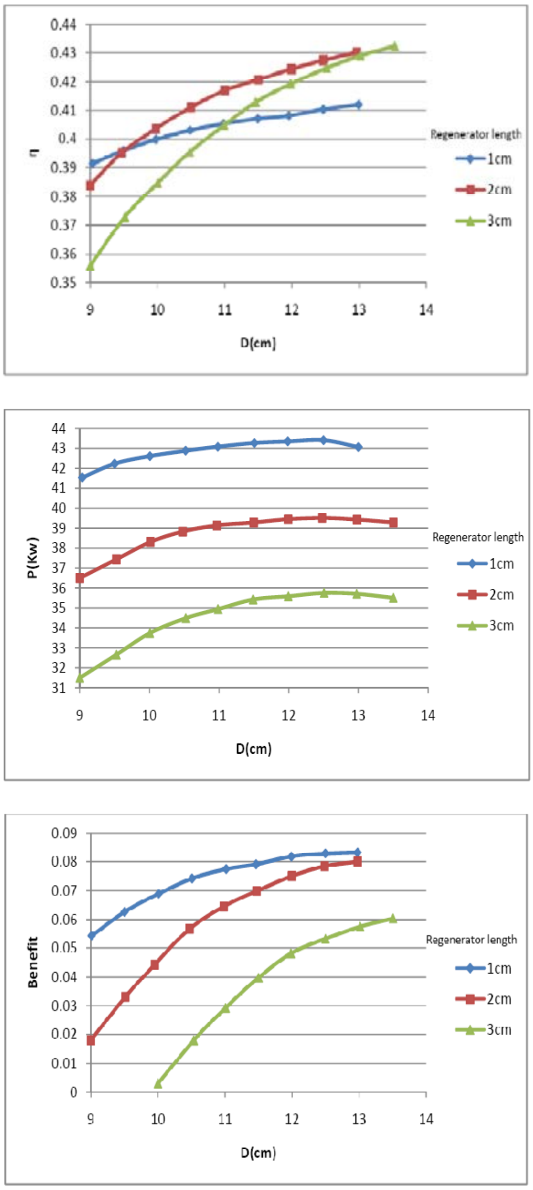


Fig. 7. Optimization of L and D (d_w=0.05 mm)

Table 11. The optimized regenerator and its predicted performance

Wire diameter d_w (Stacked screen)	60 μm	Wall thickness	1 cm
Bore	9 cm	Length	2.5 cm
Porosity (ε)	0.72	Dead volume	115 cm^3
Cross sectional flow area	45 cm^2	Heat transfer area	2.97 m^2
Engine power	32.9 kw	Efficiency	0.353
Thermal mass ratio	14.9	Flow ratio	1.7
Reduced length	85	Net enthalpy flux	6.3 kw
Heat pump power less	1.6 kw	Power lost in friction	3.7 kw

6. Conclusions

The use of the design techniques developed in Chapter Five has been demonstrated. It has been shown that these techniques can give an insight into the detailed interaction of factors determining performance, and can be used to produce a significant improvement in the predicted performances of a particular engine.

References:

[1] Zarinchang, J. (1982). “The Stirling Engine”. Shiraz University, Shiraz, Iran.

[2] Reader, G. T. & Hooper, C. (1983). “Stirling Engines”. London: Cambridge University Press, U.K.

[3] Dunn, P. D. & Rice, G. (1982). “20 kw UK Consortium Stirling Engine Specification and Manufacture” IMech E1982, Reading University, U.K.

[4] Senft, J. R. (1993). “Ringbom Stirling Engines”. New York: Oxford University Press.

[5] Thombare, D. G. & Verma, S. K. (2006). “Technological Development in the Stirling Cycle Engines”. *Renewable and Sustainable Energy Reviews*, Vol. 12, pp. 1-38

[6] Sonntag, R. E, Borgnakke, C. & Van Wylen, G. J. (2002). “Fundamentales of Thermodynamics”. New York: John Wiley & Sons, Inc. Sixth Edition.

[7] Shigley, J. E. & Mischke, Ch. R. (2001). “Mechanical aengineering Design”. New York: McGraw Hill, Inc. Sixth Edition.

[8] Organ, A. J. (1997). “The Regenerator and The Stirling Engine”. London: Mechanical Engineering Publication, U.K.

[9] Zarinchang, J. (1972). “Some Theoretical and Experimental Aspects of The Stirling Engine”. University of Reading, Reading, Berkshire, U.K.