Optimization of Thermal Components in a Stirling Engine

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Abstract: Stirling engines are known as high efficiency engines. In theory their efficiency is equal to Carnot efficiency but in practice due to some reasons their efficiency reduces. One of the most important sources of efficiency reductions is imperfect heat transfer. In this way, heat exchangers are key components in the Stirling engines. There may be three or four heat exchangers in the Stirling engine system, 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 as shown in Fig. 1, 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].

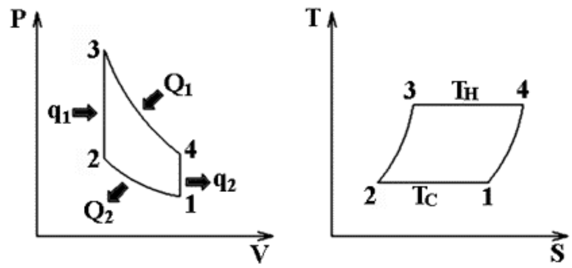


Fig. 1. Ideal Stirling Cycle

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. T_C	300 K	Heat source Temp. T_H	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. Stirling Engine Heat Exchangers

Heat exchangers have an important role in Stirling engine. There may be three or four heat exchangers in the Stirling engine system. These are illustrated in Fig.1 which includes a heater, cooler, regenerator and some auxiliary ones like, pre-heater. The heater transfers heat from external source to the working fluid contained within the engine working space. The cooler does just reverse, it absorbs heat from engine working fluid adjacent to compression space and rejects to atmosphere through coolant. The regenerator acts as a thermal sponge alternatively accepting heat from working fluid in one stage and rejecting heat back to working fluid in other stage. Three main heat exchangers will be explained more as follows[4].

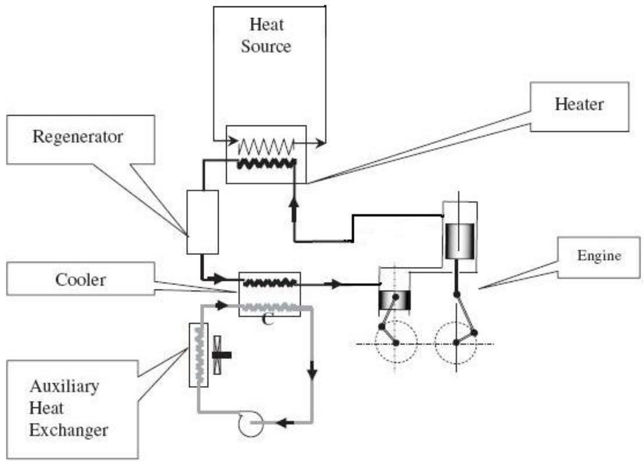


Fig.2. Heat exchangers in Stirling Engine

3.1 Heater

The heat transfer phenomenon in heater is as follows:

(a) Convective heat transfer from external heating source to walls of heater tubes or fins.

(b) Conductive heat transfer through outer tube wall surface to inner surface.

(c) Convective heat transfer from internal wall of tube to the working fluid.

In the relatively simple case of steady turbulent flow analytical techniques are not available. The heat transfer coefficient must then be determined using the well-known Reynolds analogy in its original or in a modified form. This analogy relates heat transfer to fluid friction, using standard non-dimensional parameters[4].

The design of heater is difficult because of the tube requirements. The design is also affected by the choice of heat source. The outer tube surface will usually experience a high temperature low-pressure steady flow environment. The inner tube surface will experience a high pressure, high temperature, very unsteady flow. The heat transfer coefficients will be significantly different at inside and outside and thus it will be almost evitable that surface area requirements will not be comparable with each other. There are two further constraints; one is the inner to outer diameter ratio will be determined by both pressure and thermal loading and second the optimum diameter ratio may not be in harmony with the surface area needs. All these factors may also be out of step with the frictional drag and dead space requirements[4].

The two parameters, which are of prime importance for the internal heater surfaces are the heat transfer coefficient and the friction factor. With knowledge of these two factors performance of heat exchanger can be assessed and the optimum dimensions of a proposed design can be formulated for given thermodynamic specifications. The rate at which the tube inner wall heat flux can be transferred to the gas depends upon the inner tube film coefficient of heat transfer, the mass flow rate and the specific heat of the gas[4].

3.2 Cooler

In principle, two different kinds of cooling methods there are in Stirling engines. So they may be air-cooled or water-cooled. To reduce temperature of working fluid the cooling system of engine required to handle cooling load almost twice than of cooling load of conventional engines. As the coolant temperature increases there is considerable fall in thermal efficiency, so it is desirable to have coolant temperature at minimum possible value. The flow conditions are as similar as heater but at the lower temperature. Almost all engine designers have adopted water-cooling and the outer cooler tubes experience the same flow conditions as in conventional engine. The semi-empirical heat transfer coefficients for this condition are better documented[4].

3.3 Regenerator

Some of the heat supplied by external source to working fluid is converted in to useful work and while flowing out hot expanded gases from expansion space to the cooler the rest of the heat is stored in a regenerator. After cooling in the cooler and compressing in compression space gases flows back to expansion space through regenerator. The stored heat in regenerator is given back to the working fluid during back-flow. This process is called as regeneration. And the efficiency of the Stirling cycle machine depends on the efficiency of the regeneration process or regenerator[4].

Ideal regeneration is achieved when the fluid entering and leaving the matrix does so one of the two constant temperatures, T_H at the expansion end and T_C at the compression end of the regenerator. This is possible only if operations are carried out infinitely slowly or the heat transfer coefficient or the area of heat transfer is infinite. This is also possible if the heat transfer capacity of fluid is zero or heat capacity of the matrix is infinite. The fundamental requirements of the regenerator are specified by the thermodynamic processes of the ideal cycle. In order to achieve maximum efficiency the heat rejected during the isochoric process 4–1 must be returned to the gas in the isochoric process 2–3. Ideally, the heat transfer is achieved reversibly in the regenerator. A linear temperature gradient from T_{max} to T_{min} is maintained along the length of the regenerator. The working fluid enters in the regenerator in thermodynamic state 4 and starts transferring its heat to the regenerator material and leaves at state 1. During this process temperature of each element of the regenerator is raised. After the compression process the working fluid now enters in the regenerator at state 2 (minimum cycle temperature) and flows back through regenerator receiving the heat stored in the regenerator by increasing its temperature from T_2 to T_3[4].

But in practical cases the regenerator of engine operates with conditions far from assumptions for the ideal case. The temperature of working fluid entering the regenerator is not constant because the pressure, density and velocity of working fluid vary over a wide range. The effectiveness of the regeneration process largely depends upon the thermal capacity of the material also[4].

3.3.1 Regenerator analysis

There are various kinds of materials which can be used for the regenerator matrix, such as steel wool, steel felt, wire mesh, fine pipes, spring mesh, stacked screen, packed balls, metal foils and parallel plates etc. To achieve high heat storage capacity the matrices of above materials are used in current regenerators. To improve heat transfer coefficient and to establish the minimum temperature difference between matrix and the fluid it is necessary to expose the maximum surface area of matrix, therefore matrix should be finely divided[4].

Thus following are the desirable characteristics of regenerator matrix:

1. For maximum heat capacity so, a large, solid matrix is required.
2. For minimum flow losses so, a small, highly porous matrix is required.
3. For minimum dead space so, a small, dense matrix is required.
4. For maximum heat transfer so, a large, finely divided matrix is required.

5. For minimum contamination so, a matrix with no obstruction is required.
Hence design of regenerator is a matter of optimization of regenerator volume for best values of above parameters[4].

3.3.2 Heat transfer and fluid friction in regenerator

For simplifying the analysis of regenerator following assumptions are made:
(a) The thermal conductivity of the matrix material of regenerator is constant.
(b) The specific heats of fluid and matrix do not change with temperature.
(c) The fluid flow and temperature that is constant over the flow section.
(d) The heat transfer coefficients and fluid velocities are constant with time and space.
(e) The rate of mass flow is constant.
(f) The pressure drop across the regenerator is negligible.
(g) The gas flow in duct is one-dimensional.
(h) The working gas is assumed to be a perfect gas.

The entire heat transfer process is carried out reversibly. The temperature differential between working fluid and the regenerator must be infinitesimal. To fulfill this requirement certain conditions have to be satisfied. First reversible process can assure if process is in thermodynamic equilibrium, i.e. regenerator system must pass through a series of equilibrium states. But this can never be achieved in practice hence every attempt must be made to satisfy the remaining condition, which can be readily identified by fundamental equation for the dynamic behavior of the regenerator[4].

4. 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 2 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 d_o can be reduced, allowing the geometric limit on the number of tubes to be relaxed.

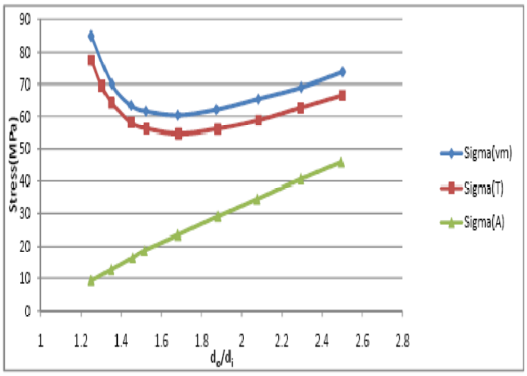


Fig.3. Stress variation with diameter ratio

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[5].

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 d_o/d_i constant, the Von-Mises stress will not be affected[5].

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 4. 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 d_i 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.

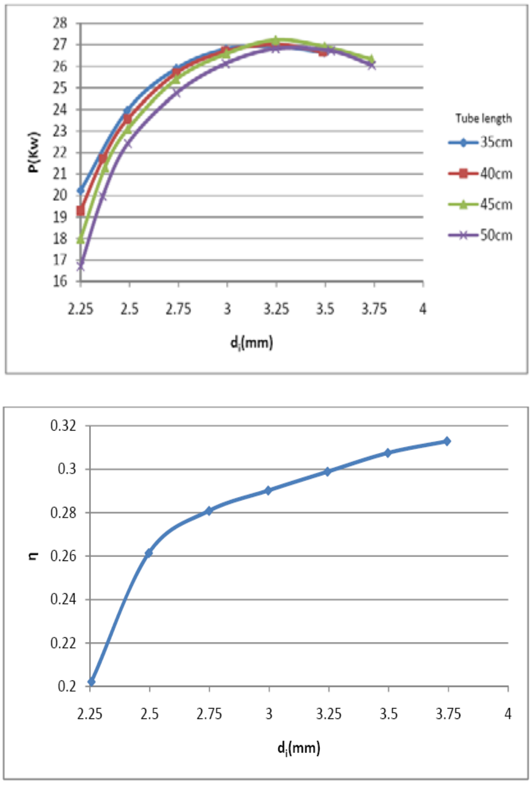
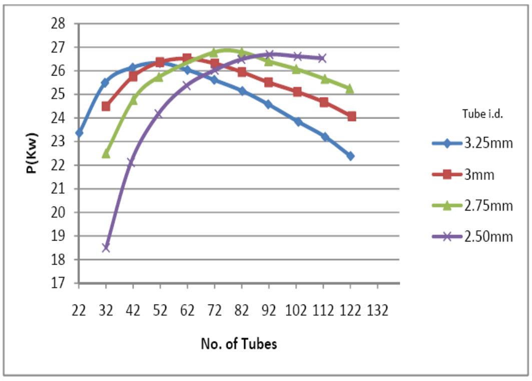


Fig.4. Heater sensitivity (No. of tubes=52)

Note: The variation in efficiency with tube length is slight and reflects the variation in power

In Fig.5, 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.

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.



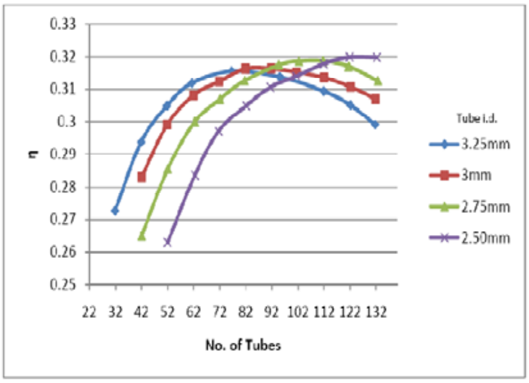


Fig.5. Heater sensitivity (Tube length=45 cm)

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

5. 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.

The effect on engine power and efficiency of reducing the bore of the cooler tubes while increasing their length is shown in fig.5. 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).

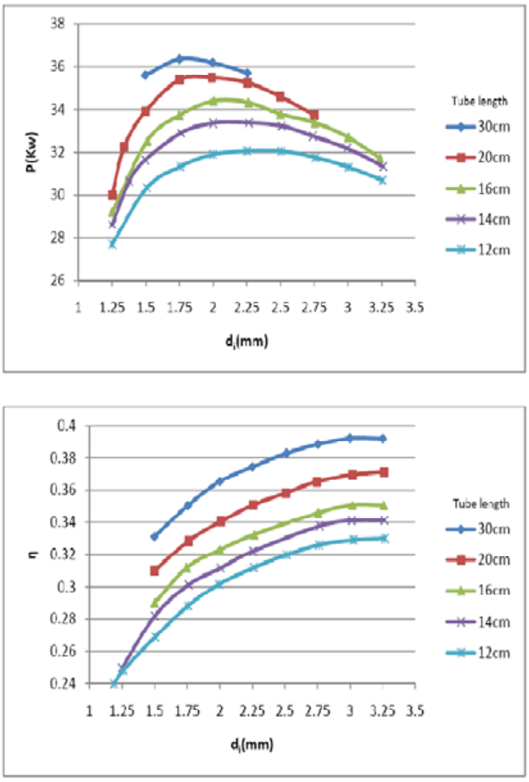


Fig.6. Cooler sensitivity

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.5 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

6. 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.7[8].

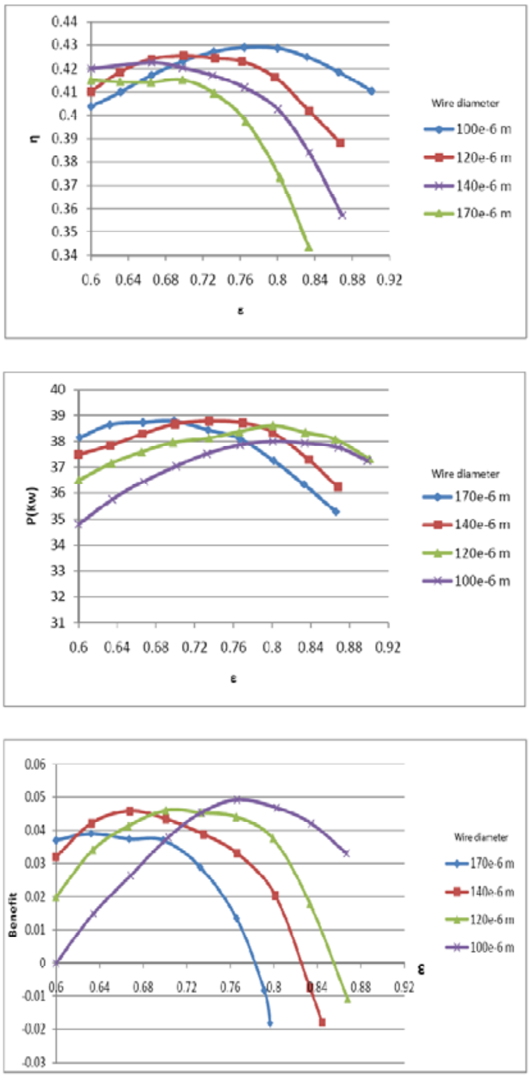


Fig.7. Optimization of ε and d_w

Fig.7 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 brings 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 100μm and 50μm 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 finer wires making up a regenerator, the shorter and wider its optimum dimensions will be. Fig.7, for a matrix made up of 50μm 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.

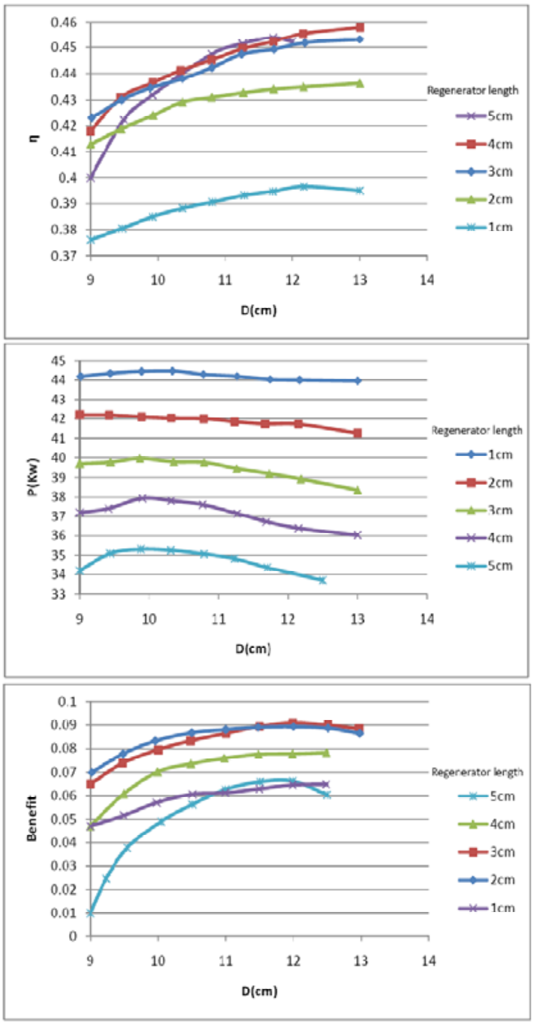


Fig.8. Optimization of L and D (d_w=0.1 mm)

The data presented in figs. 6, 7 and 8 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 d_w 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.

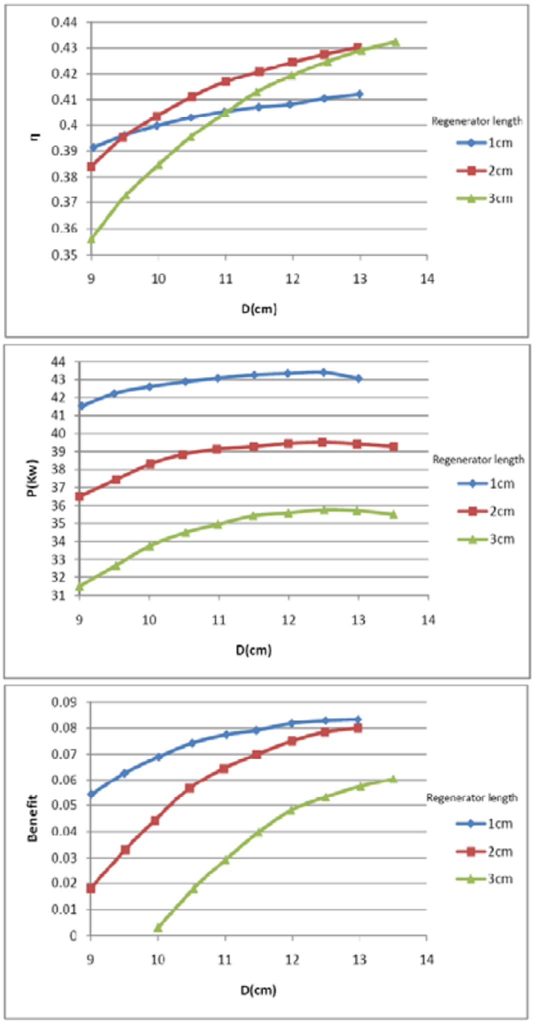


Fig. 9. Optimization of L and D (d_w=0.05 mm)

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

7. Conclusion

The use of the design techniques developed in this paper 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.

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