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Conceptual Design of Recuperator for Ultramicro Gas Turbine

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ABSTRACT

Conceptual design of recuperator for palm-top size (nearly 3kW output) and button size (nearly 15W output) micro turbines has been made. In the case of a palm-top gas turbine an offset-strip-fin (OSF) recuperator was employed. The basic performance of OSF recuperator was evaluated from the viewpoint of recuperator volume for a specified effectiveness and pressure loss. In addition an optimization was made for the recuperator size to maximize the turbine efficiency. Based on these analyses a conceptual design of annularly arranged box-type recuperators was presented. In the case of a button-sized turbine a micro-channel recuperator was considered, and an optimum size was determined taking into account the deterioration of effectiveness due to the longitudinal heat conduction of the wall.

NOMENCLATURE

A	frontal area
Cp	specific heat
D	hydraulic diameter of channel
f	friction factor
G	mass flow rate
H	height
h	heat transfer coefficient
j	j factor ($= Nu / RePr^{1/3}$)
L	length
N	number of transfer unit
Nu	Nusselt number ($= Dh / \lambda$)
p	pressure
Pr	Prandtl number ($= C_p \mu / \lambda$)
Re	Reynolds number ($= \rho u D / \mu$)
S	heat transfer area
T	temperature
T^*	dimensionless temperature ($= (T - T_{a,in})/(T_{b,in} - T_{a,in})$)
u	velocity ($= G/(\rho A)$)
V	volume
W	width
δ	thickness
ε	effectiveness of recuperator
ζ	pressure loss ratio (defined by Eq.(23))
η	thermal efficiency
λ	thermal conductivity
μ	viscosity
ρ	density

Subscripts

a	compressed air
b	exhaust gas
c	compressor
f	fin
HX	recuperator
in	inlet
o	atmospheric condition
out	outlet
tot	total
w	wall

INTRODUCTION

In order to prevent global warming due to the emission of carbon dioxide and reduce the consumption of fossil energy resources the effective use of energy is all the more important. From this viewpoint the distributed energy system, i.e. cogeneration system, is attractive, because the waste thermal energy exhausted from the onsite electric power generation can be utilized for heating and cooling at the demand site. Micro gas turbines play important roles in such distributed energy systems because of their small size and weight, low emission, vibration-free operation, and low maintenance cost. Micro turbines as small as 25-75 kW range have been currently developed which meet the energy requirement in office building, restaurant, hospital, apartment house, etc. Further reduction of turbine size, for example 3kW power output, could be applied to cogeneration systems for an individual house. Further possibility of reducing turbine size is a button-sized gas turbine as was proposed by Epstein et al. (1997). Such an ultramicro gas turbine will be useful for power sources of mobile electronic equipments.

As mentioned above the downsizing of gas turbines will find various applications as long as their thermal efficiency is sufficiently high. As is the case of current micro turbines recuperated cycle, as shown in Fig.1, is mandatory because the

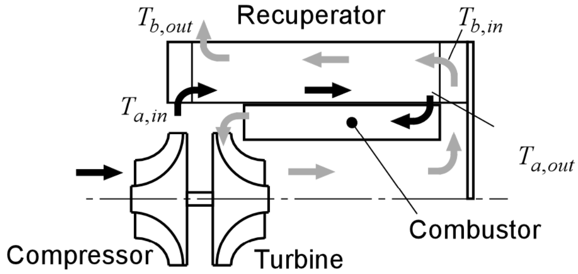

Fig.1 Recuperated gas turbine

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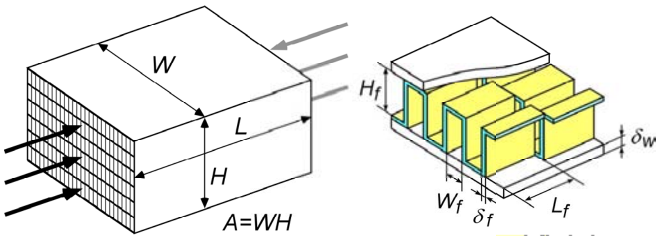


Fig.2 Definition of recuperator and offset strip fin size

compression ratio and turbine inlet temperature of micro turbines are small, and the efficiency of simple cycle is too low. It should be noted that the volume of turbine generally varies in proportion to $G^{3/2}$, where G is gas flow rate. On the other hand the volume of recuperator is roughly proportional to G as shown in later, which brings the increase of recuperator size relative to turbine size with the decrease of power. Therefore precise optimization of recuperator becomes more and more important with the reduction of turbine size. Although there have been excellent papers which discuss the performance of recuperator (McDonald and Wilson, 1996, Utriainen and Sunden, 2002), optimization considering the relationship between the thermal efficiency and recuperator size has not been well documented. Generally speaking larger recuperator has higher effectiveness and lower pressure loss, therefore, the optimization is a problem to minimize the recuperator volume or weight to achieve desired cycle efficiency. Such an optimization will depend on specific turbine conditions such as pressure ratio, temperature ratio, compressor efficiency and turbine efficiency. In the first part of this paper an optimization and design of recuperator for a postulated 3kW turbine system are discussed. In the second part the performance of ultramicro recuperator using micro channel for a button-sized gas turbine is analyzed where the deterioration of recuperator effectiveness due to the axial heat conduction of the wall is considered.

EVALUATION OF PLATE-FIN RECUPERATOR WITH OFFSET STRIP FINS

There exist several kinds of heat transfer surfaces used for recuperator such as plate-fin and primary surface (Utriainen and Sunden, 2002). In this study a plate-fin recuperator with offset strip fins, as shown in Fig.2, is employed because the manufacturing technology and reliability are generally well established. In this section the basic performance of offset-strip-fin channel is identified from the viewpoint of compactness compared with a simple rectangular channel (i.e. straight fin).

The heat-transfer surface area to achieve a given recuperator effectiveness, ε, is determined by means of the number of transfer unit, N. In the case of a counter-flow recuperator, the relationship between N and ε is given by

$$N_{tot} = \frac{\varepsilon}{1-\varepsilon} \tag{1}$$

where

$$\varepsilon = \frac{T_{a,out} - T_{a,in}}{T_{b,in} - T_{a,in}} \tag{2}$$

$$\frac{1}{N_{tot}} = \frac{1}{N_a} + \frac{1}{N_b} \tag{3}$$

$$N = \frac{hS}{C_p\, G} \quad \text{for each fluid.} \tag{4}$$

As shown by the above equations, it is obvious that the heat transfer surface area, S, to achieve a desired effectiveness is reduced by increasing the heat transfer coefficient, h. The heat transfer coefficient is increased by the increase of flow velocity, however, it brings the increase of pressure loss. Therefore the pressure loss must be specified to determine the surface area. The pressure loss for one side of fluid is given by the definition of friction factor f:

$$\Delta p = \frac{4L}{D}\frac{1}{2}\rho u^2 f = \frac{2L}{D}\frac{G^2}{\rho A^2} f \tag{5}$$

where hydraulic diameter, D, and frontal area, A, of one side are related to volume, V, and length, L, as follows:

$$D = \frac{4V}{S} \tag{6}$$

$$A = \frac{V}{L} \tag{7}$$

In addition the heat transfer coefficient is customarily expressed in terms of j factor, because the ratio j/f is independent or a weak function of Re number:

$$j = \frac{Nu}{RePr^{1/3}} = \frac{h\,A\,Pr^{2/3}}{C_p\, G} \tag{8}$$

Equation (8) is rewritten by using Eqs.(4), (6) and (7):

$$L = \frac{DPr^{2/3}N}{4j} \tag{9}$$

From Eqs.(5) and (9) the frontal area and volume of one side is expressed as follows:

$$\frac{A}{G} = \left(\frac{Pr^{2/3}N}{2\rho\Delta p}\frac{f}{j}\right)^{1/2} \tag{10}$$

$$\frac{V}{G} = \frac{AL}{G} = \frac{Pr\,N^{3/2}}{2\,(2\rho\Delta p)^{1/2}}\frac{D}{j}\left(\frac{f}{j}\right)^{1/2} \tag{11}$$

Equations (9)-(11) are the basic equations to determine the size of one fluid side in a recuperator. It is noted that the length does not depend on mass flow rate, G, and the frontal area is proportional to G. Because j and f are functions of Re number, it is convenient to rewrite Eqs.(10) and (11) in dimensionless forms as follows:

$$Re = D^*\left(\frac{j}{f}\right)^{1/2} \tag{12}$$

$$V^* = \frac{D^*}{j}\left(\frac{f}{j}\right)^{1/2} = \frac{D^{*2}}{j\,Re} \tag{13}$$

where

$$Re = \frac{D}{\mu}\frac{G}{A} \tag{14}$$

$$D^* = \left(\frac{2\rho\Delta p}{Pr^{2/3}N}\right)^{1/2}\frac{D}{\mu} \tag{15}$$

$$V^* = \frac{4\rho\Delta p}{\mu Pr^{4/3}N^2}\frac{V}{G} \tag{16}$$

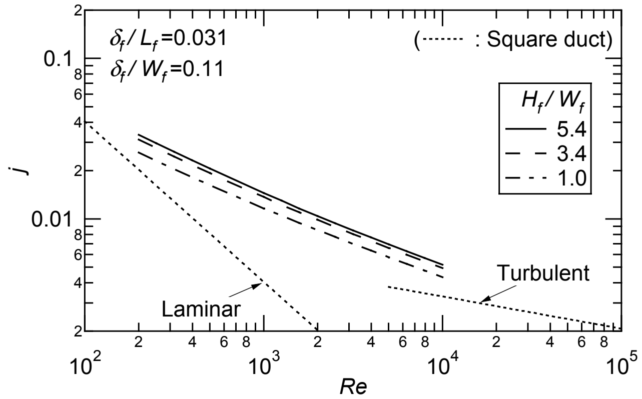
Because the non-dimensional diameter, D^*, is determined by operating conditions (N and Δp) and hydraulic diameter of channel, Re number is determined by Eq.(12) for a given value of D^* by using correlations for j and f versus Re number. Then the non-dimensional volume, V^*, is determined by Eq.(13). Therefore V^* is only a function of D^* for a specific type of recuperator, and the relationship between V^* and D^* can definitely compare the compactness of different types of heat transfer surface under the same operating condition and hydraulic diameter.

By using the above method the compactness of offset-strip-fin channel is compared with a simple straight channel with square cross section (i.e. straight-fin channel). Correlations for j and f factors are given as follows (Manglik and Bergles, 1995):

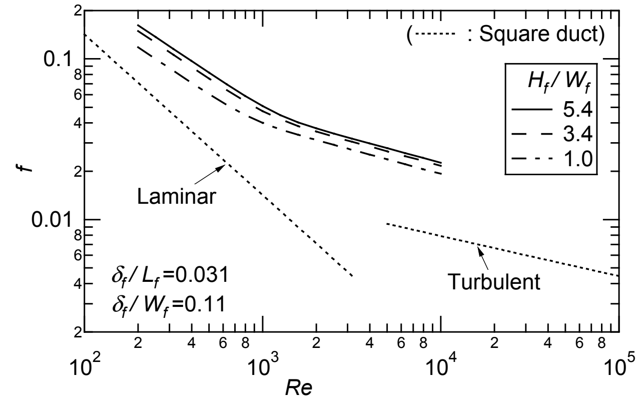
Offset strip fin

$$j = 0.6522Re^{-0.5403}\left(\frac{W_f}{H_f}\right)^{-0.1541}\left(\frac{\delta_f}{L_f}\right)^{0.1499}\left(\frac{\delta_f}{W_f}\right)^{-0.0678}$$

$$\times\left[1 + 5.269\times10^{-5}Re^{1.340}\left(\frac{W_f}{H_f}\right)^{0.504}\left(\frac{\delta_f}{L_f}\right)^{0.456}\left(\frac{\delta_f}{W_f}\right)^{-1.055}\right]^{0.1}$$

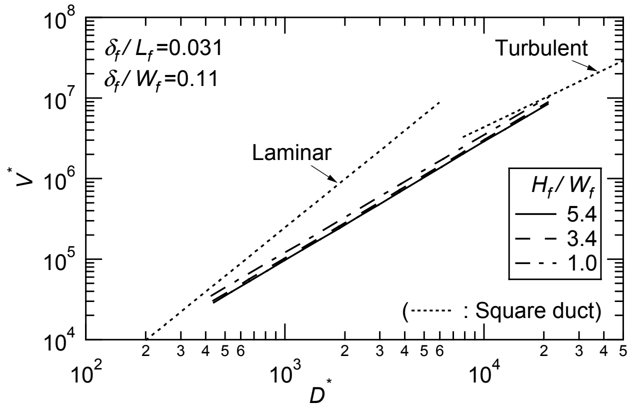


(a) j factor

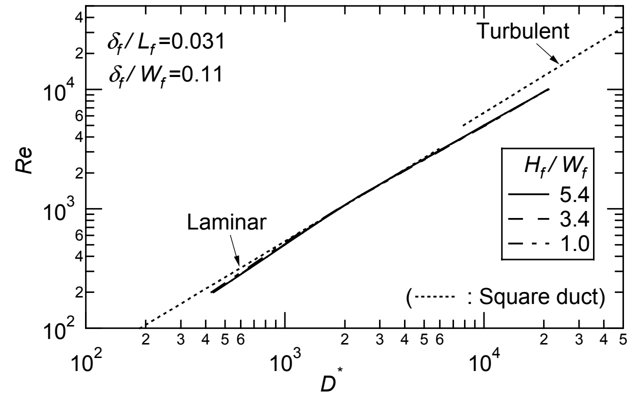


(b) friction factor

Fig.3 Characteristics of offset strip fin channel



(a) Non-dimensional volume



(b) Relationship between Re number and D^*

Fig.4 Comparison of recuperator volume between offset strip fin channel and square duct

(17)

$$f = 9.6243 Re^{-0.7422} \left(\frac{W_f}{H_f}\right)^{-0.1856} \left(\frac{\delta_f}{L_f}\right)^{0.3053} \left(\frac{\delta_f}{W_f}\right)^{-0.2659} \times \left[1 + 7.699\times10^{-8} Re^{4.429} \left(\frac{W_f}{H_f}\right)^{0.920} \left(\frac{\delta_f}{L_f}\right)^{3.767} \left(\frac{\delta_f}{W_f}\right)^{0.236}\right]^{0.1} \tag{18}$$

Square duct (fully developed):

Laminar flow

$$Nu = 3.6 \tag{19a}$$

$$j = 4.91 / Re \qquad \text{(for } Pr = 0.7\text{)} \tag{19b}$$

$$f = 14.2 / Re \tag{20}$$

Turbulent flow

$$Nu = 0.022 Re^{0.8} Pr^{0.5} \tag{21a}$$

$$j = 0.021 Re^{-0.2} \qquad \text{(for } Pr = 0.7\text{)} \tag{21b}$$

$$f = 0.0791 Re^{-0.25} \tag{22}$$

Figure 3 shows the j and f characteristics versus Re number. Both j and f factors of offset-strip-fin channel are larger than those of square duct. The relationships between V^* and D^* are compared in Fig.4(a). Figure 4(b) is the auxiliary relationship between Re number and D^*. It is apparent that the reduction of hydraulic diameter brings the decrease of channel volume. Further it is shown that the volume of offset-strip-fin channel is smaller than that of square duct especially in laminar flow region with higher Re number.

The recuperator size is determined by the above method when effectiveness and pressure loss are specified, however, the method to determine optimum pressure loss is unknown. The design of recuperator is to minimize the recuperator volume to achieve a desired thermal efficiency of gas turbine cycle. In the next section the relationship between cycle thermal efficiency and recuperator volume is examined.

OPTIMUM OPERATING CONDITION OF RECUPERATOR AND DESIGN FOR 3kW TURBINE

Optimum condition to maximize thermal efficiency

In order to examine the optimum size of practical recuperator a 3kW class micro turbine as shown in Table 1 is assumed. Fin dimensions employed in this study are $W_f = 1.02\text{mm}$, $H_f = 3.2\text{mm}$, $L_f = 3.2\text{mm}$, $\delta_f = 0.1\text{mm}$, $\delta_w = 0.3\text{mm}$. These dimensions correspond to the finest fin currently applicable to the recuperator. Variations of specific volume V/G, and pressure loss ratio ζ_{HX} with the change of frontal area A/G at constant effectiveness are shown in Fig.5. The pressure loss ratio ζ_{HX} is defined by

$$\zeta_{HX} = \frac{\Delta p_b}{p_o} + \frac{\Delta p_a}{p_{c,out}} \tag{23}$$

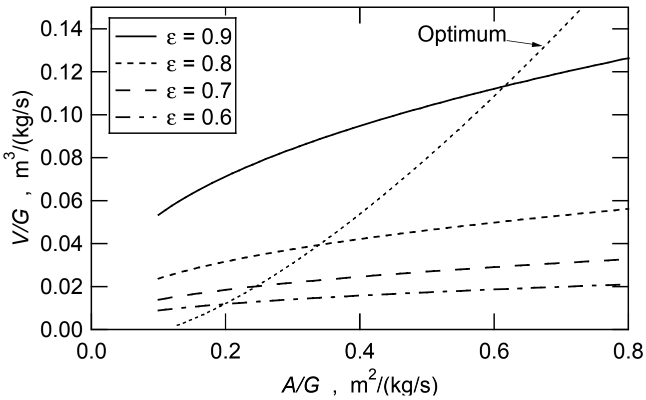
where Δp_a and Δp_b denote pressure loss of compressed air side and exhaust gas side, respectively. In addition the fin efficiency η_f was taken into account in the analysis to give effective heat transfer area:

$$\eta_f = \frac{\tanh(mH_f/2)}{mH_f/2}, \qquad m = \sqrt{\frac{2h}{\lambda_w \delta_f}} \left(1 + \frac{\delta_f}{L_f}\right) \tag{24}$$

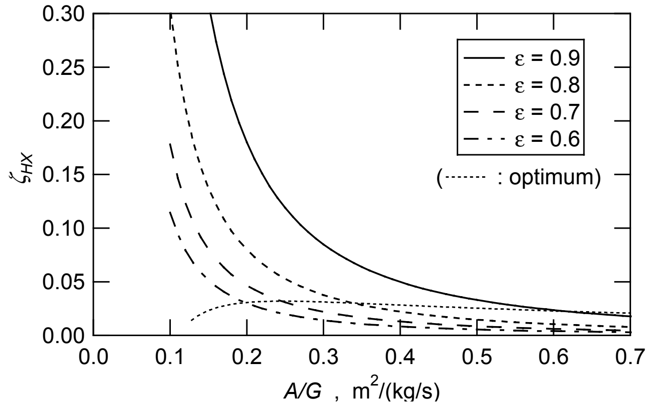
As shown in Fig.5(a) the recuperator volume decreases with the decrease of A/G because the heat transfer coefficient increases due to the increase of flow velocity, which brings the increase of

Table 1 Specification and performance of postulated 3kW class micro gas turbine

Gas flow rate	0.03 kg/s
Turbine inlet temp.	1223 K
Pressure reatio	3
Turbine efficiency	0.8
Compressor efficiency	0.75
Combustor pressure loss	3 %
Without recuperator	
Power output	3.66 kW
Efficiency	14.4 %
With the present recuperator	
Effectiveness of recuperator	0.8
Pressure loss of recuperator	6.3 %
Power output	3.24 kW
Efficiency	28.5 %



(a) Specific volume



(b) Pressure loss ratio

Fig.5 Variation of recuperator size and pressure loss with specific frontal area at constant effectiveness

pressure loss as shown in Fig.5(b).

By using these results the cycle thermal efficiency is obtained based on the turbine operating conditions listed in Table 1. Figure 6 shows relationships between thermal efficiency, η_{GT} , and recuperator volume at constant recuperator effectiveness. The thermal efficiency increases with the increase of recuperator volume due to the decrease of pressure loss and approaches an asymptotic value. It is noted that an optimum effectiveness exists which maximize the thermal efficiency at a specific recuperator volume. At a given recuperator volume the increase of effectiveness beyond the optimum value brings the decrease of thermal efficiency due to the increase of pressure loss. On the other hand the decrease of effectiveness below the optimum value reduces the thermal efficiency due to the decrease of effectiveness by itself. The optimum conditions for several kinds of recuperator

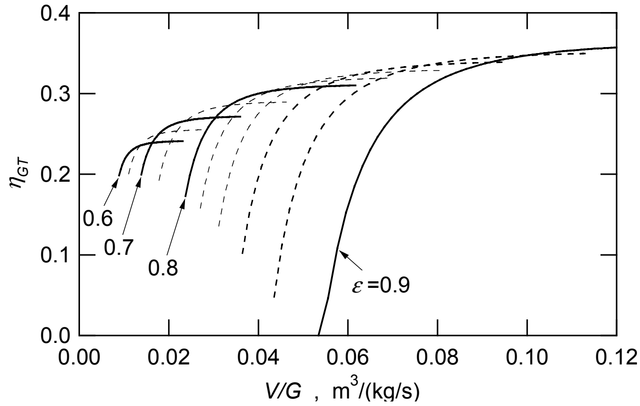
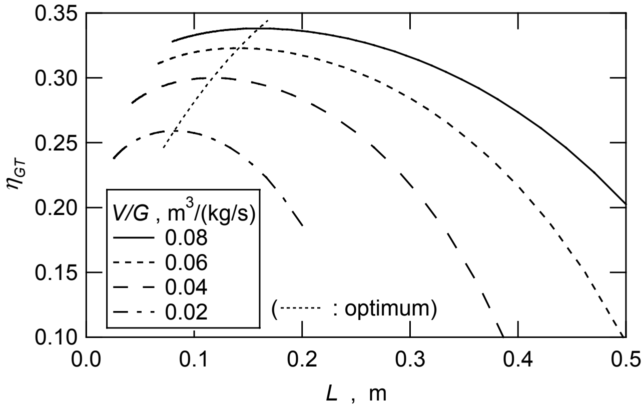
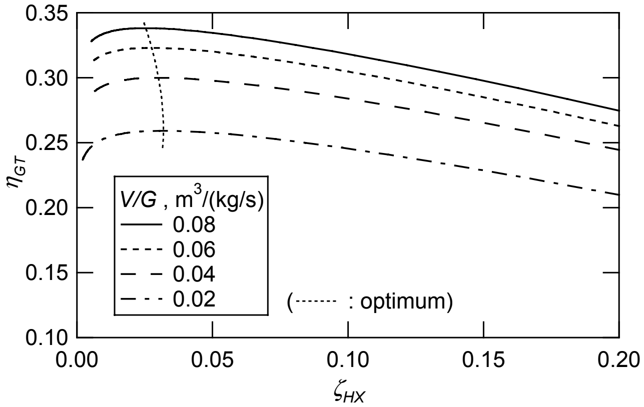


Fig.6 Relationship between cycle thermal efficiency and recuperator specific volume at constant effectiveness



(a) Efficiency vs. length



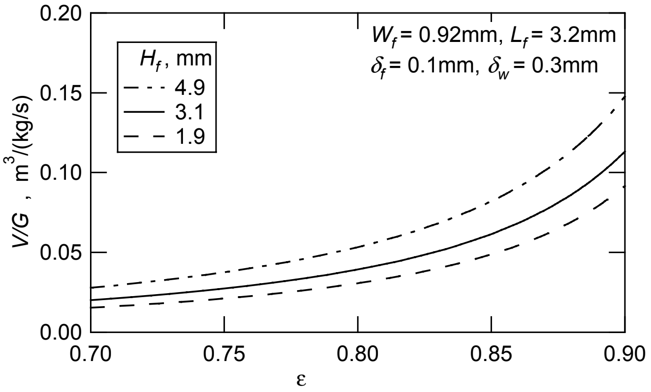
(b) Efficiency vs. pressure loss ratio

Fig.7 Variation of cycle thermal efficiency with recuperator length and pressure loss at constant volume

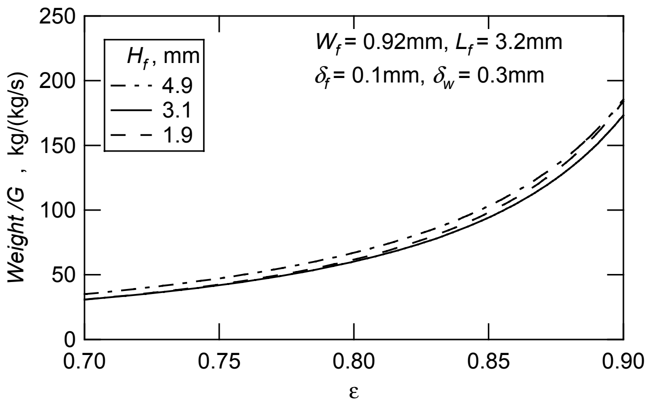
volume are shown in Fig.7(a) as the relationship between thermal efficiency and recuperator length. The dotted line denotes the optimum point. The optimum point is also plotted in Fig.5 by a dotted line. Figure 7(b) shows the relationship between thermal efficiency and pressure loss. It should be noted that the change of thermal efficiency with pressure loss near the optimum point is small. Although it is not discussed in this paper, an additional pressure loss exists due to the distributor section of recuperator, and the additional pressure loss generally increases with the recuperator width. When this additional pressure loss is taken into account, larger length, i.e. larger pressure loss, than the optimum value may be preferable practically.

Effect of fin height

In this section the effect of fin dimensions on the recuperator performance is examined. Although it was suggested by a preliminary study that the decrease of fin pitch W_f and thickness



(a) Specific volume



(b) Specific weight

Fig.8 Relationship between optimum recuperator size and effectiveness for various fin height

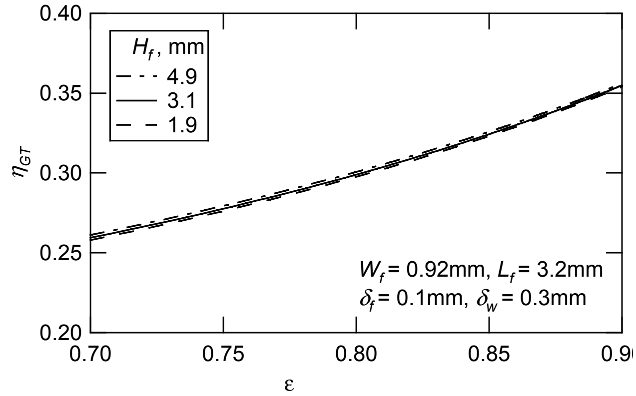


Fig.9 Relationship between cycle thermal efficiency and recuperator effectiveness at optimum (minimum volume) point

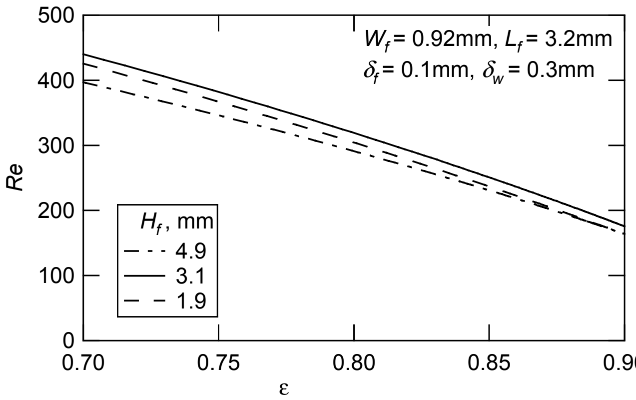


Fig.10 Relationship between *Re* number and recuperator effectiveness at optimum point

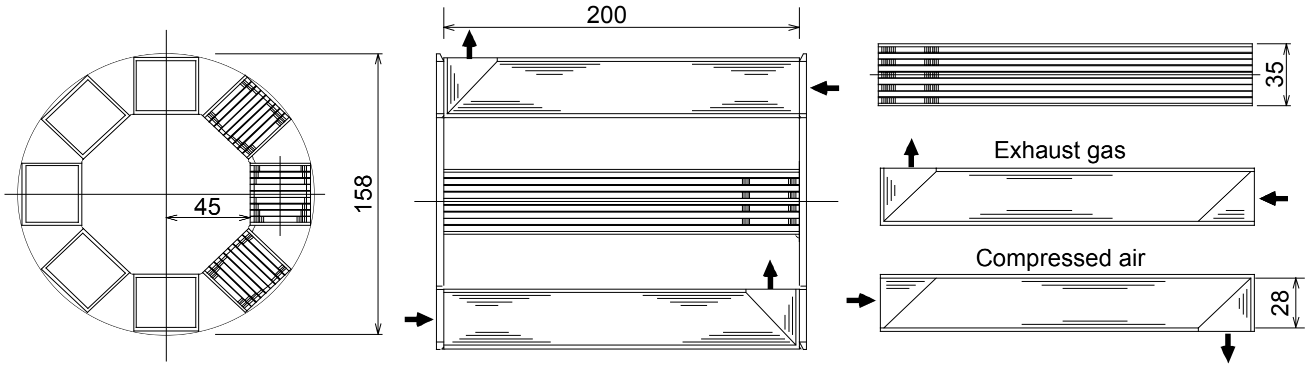


Fig.11 Designed recuperator for 3kW GT (ε=0.8)

δ_f reduces the recuperator size, the fin pitch and thickness employed in the previous section seem to be minimum values from practical points of view such as manufacturing, strength or resistance to corrosion. Therefore only the effect of fin height is discussed. Figure 8 shows the plots of recuperator volume and weight versus effectiveness at optimum condition for three kinds of fin height. The smallest fin height (H_f = 1.9mm) is preferable for the reduction of volume as shown in Fig.8(a), however, the intermediate fin height (H_f = 3.1mm) is preferable for the reduction of weight, as shown in Fig.8(b), due to the decrease of porosity with the decrease of fin height. In this study fin height of 3.1mm is adopted with emphasis on the material cost.

Figure 9 shows the relationship between thermal efficiency and effectiveness for different fin height. It is seen that the thermal efficiency is not affected by the fin height very much, which indicates that the pressure loss at the optimum point is nearly constant for different fin heights. Figure 10 shows the relationship between *Re* number and effectiveness. It is confirmed that the operating *Re* number is in the range where offset-strip-fin recuperator is effective as was show in Fig.4.

Design of recuperator for 3kW micro turbine

Based on the above-mentioned examination a practical design of recuperator for 3kW class micro turbine has been made. Although the cycle thermal efficiency increases with the recuperator effectiveness as shown in Fig.9, the volume and weight increases more steeply with effectiveness as seen in Fig.8. The tradeoff between thermal efficiency and recuperator size may depend on specific application. In the present study effectiveness 0.8 is adopted with emphasis on compactness. Figure 11 shows a result of proposed design. It consists of annularly arranged 8 box-type recuperators. Such an annular configuration surrounding the turbine as shown in Fig.1 can eliminate the external duct, reduce the

Table 2 Specification of postulated button-sized gas turbine

Overall outer diametr	12mm
Overall outer height	3mm
Gas flow rate	0.15 g/s
Turbine inlet temp.	1600 K
Pressure reatio	4
Turbine efficiency	0.65
Compressor efficiency	0.61
Combustor pressure loss	3 %
Without recuperator	
Power output	16.6 W
Efficiency	9.4 %
With the present recuperator	
Effectiveness of recuperator	0.8
Pressure loss of recuperator	7.8 %
Power output	13.0 W
Efficiency	17.6 %

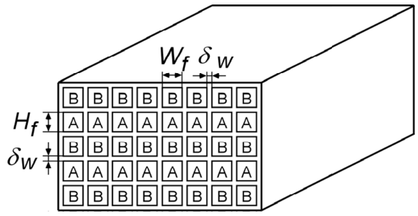


Fig.12 Micro-channel recuperator

heat loss from combustor and minimize the overall size of turbogenerator package. The total volume of matrix core (excluding distributor section, side plate and side bar) is 1186 cm^3, that is, V/G=0.040 m^3/(kg/s). The pressure loss and estimated cycle thermal efficiency are listed in Table 1. Although the thermal efficiency is slightly less than the optimum value shown in Fig.7 due to the additional pressure loss of distributor section, the estimated thermal efficiency, 28.5%, seems to be satisfactory for practical applications.

EVALUATION OF MICROCHANNEL RECUPERATOR FOR BUTTON-SIZED ULTRAMICRO TURBINE

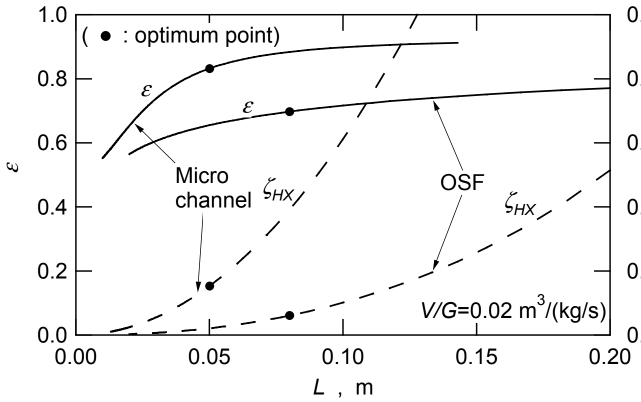
As a challenge to micro power sources, Epstein et al. (1997) has proposed a button-sized gas turbine as shown in Table 2 (note that efficiencies are estimated ones). In this section the possibility of recuperator compatible to such an ultramicro turbine is examined. For such a small-scale turbine a micro-channel recuperator based on MEMS (Micro-Electro-Mechanical Systems) technology, that is, a manufacturing technology based on semiconductor fabrication methods seems to be appropriate. Therefore, a recuperator with simple square channels ($H_f = W_f = 0.5$mm , $\delta_f = \delta w = 0.1$mm) is assumed as shown in Fig.12. Because the deterioration of effectiveness due to axial heat conduction of the wall will become significant with the decrease of recuperator size, an analytical solution considering the effect of wall conduction is used (Kroeger, 1967):

$$\varepsilon = 1 - \frac{1}{1 + N_{tot}(1+\lambda^*\Phi)/(1+\lambda^* N_{tot})} \tag{25}$$

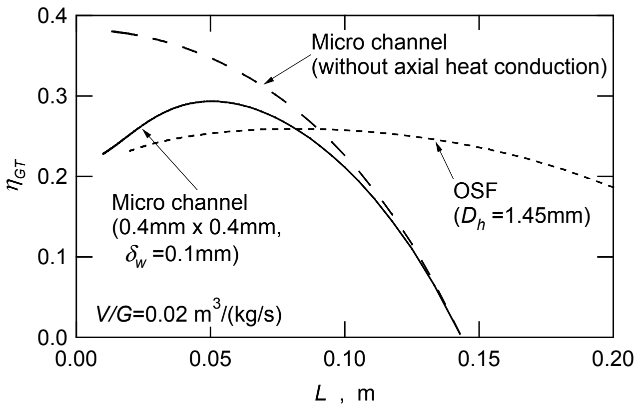
where

$$\Phi = \frac{\lambda^* N_{tot}}{1+\lambda^* N_{tot}} \sqrt{1 + \frac{4}{\lambda^*(N_a + N_b)}} \left(\coth(\beta) - \frac{\cosh(\alpha)}{\sinh(\beta)} \right) \tag{26}$$

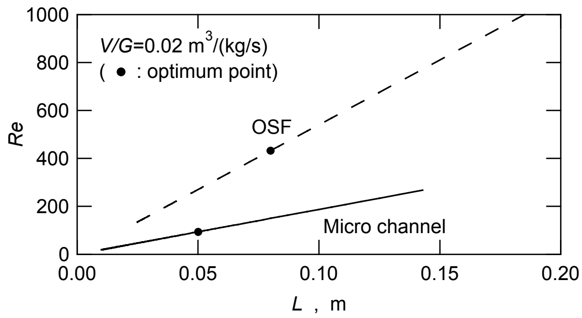
$$\alpha = N_b - N_a \quad , \quad \beta = \sqrt{(N_a + N_b)^2/4 + (N_a + N_b)/\lambda^*} \tag{27}$$



(a) Recuperator effectiveness and pressure loss



(b) Gas turbine efficiency



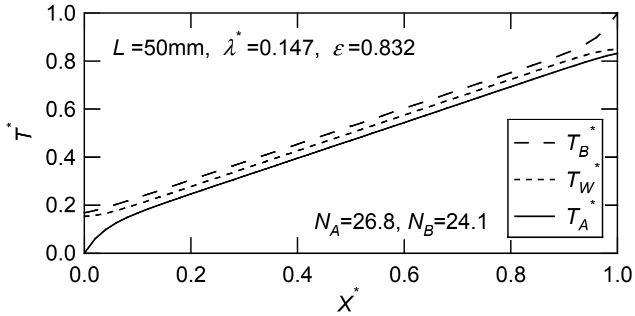
(c) Re number

Fig.13 Characteristics of micro-channel recuperator in comparison with offset strip fin recuperator (G=0.15 g/s)

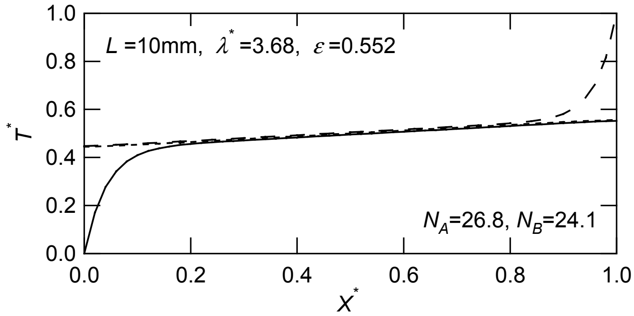
$$\lambda^* = \frac{\lambda_w A_w}{L C_p G} \tag{28}$$

It should be noted that Eq.(25) reduces to Eq.(1) when the wall thermal conductivity λ_w equals 0.

Figure 13 shows the characteristics of the micro-channel recuperator for G=0.15 g/s in comparison with those of offset-strip-fin recuperator discussed in the previous section. The recuperator volume is assumed to be 3 cm^3 (V/G=0.02 cm^3/(kg/s)) which results in effectiveness about 0.8 for the micro-channel recuperator. The wall material of the micro channel is assumed to be silicon. Although the applicability of silicon to high-temperature recuperator is suspicious, thermal conductivity of silicon was assumed because it is a representative material in MEMS technology. Figure 13(a) shows the variation effectiveness and pressure loss with the recuperator length. In the case of micro channel the heat transfer coefficient is constant as indicated by Eq.(19a), and N_{tot} is also constant under a constant volume (i.e. constant heat transfer area). However the effectiveness decreases with the deceases of recuperator length due to the wall axial conduction. In the case of offset strip fin (OSF) the variation of



(a) L = 50mm



(b) L = 10mm

Fig.14 Nondimensional axial temperature profile in micro-channel recuperator (V/G=0.02 m^3/(kg/s))

effectiveness is caused by the dependency of heat transfer coefficient on *Re* number. Figure 14 shows examples of axial temperature profiles in micro-channel recuperator. In the case of L=50mm, shown in Fig.14(a), the temperature profile is similar to that of usual counter-flow recuperator. However in the case of L=10mm, shown in Fig.14(b), the wall temperature is nearly uniform which indicates the deterioration of effectiveness due to wall conduction. Figure 13(b) shows the variation of cycle thermal efficiency with recuperator length, where turbine conditions listed in Table 1 (except gas flow rate) are assumed for the comparison with offset-strip-fin recuperator discussed in the previous section. The broken line in Fig.13(b) is the result when the axial heat conduction of wall is ignored, in which case the thermal efficiency increases monotonically with the decrease of recuperator length due to the decrease of pressure loss with constant effectiveness. On the other hand, when the axial wall conduction is considered, the thermal efficiency has a peak at a certain length due to the decrease of effectiveness by the axial wall conduction. Still the maximum thermal efficiency for the micro-channel recuperator is larger than that for offset-strip-fin due to the small hydraulic diameter of the micro channel. Fig.13(c) shows the variation *Re* number. In the case of micro channel *Re* number is nearly 100 at the optimum operating condition. It should be noted that, in the case of such a ultramicro recuperator, a simple rectangular channel is appropriate because offset-strip-fin channel is not effective in such a low *Re* number region as was shown in Fig.4.

Figure 15 shows a recuperator design for the postulated button-sized turbine. Because the flow length, about 60mm, is too long compared with the turbine size, three-path configuration may be the best choice for the present condition. The estimated performance of the button-sized turbine with the proposed recuperator is included in Table 2. The power output is reduced due to the pressure loss of recuperator, however the thermal efficiency increases significantly. Although the recuperator size seems to be rather too large compared with the button-sized turbine, the size of fuel tank may be reduced by the addition recuperator. Such a total compactness of recuperated turbine system should be judged depending on specific practical application.

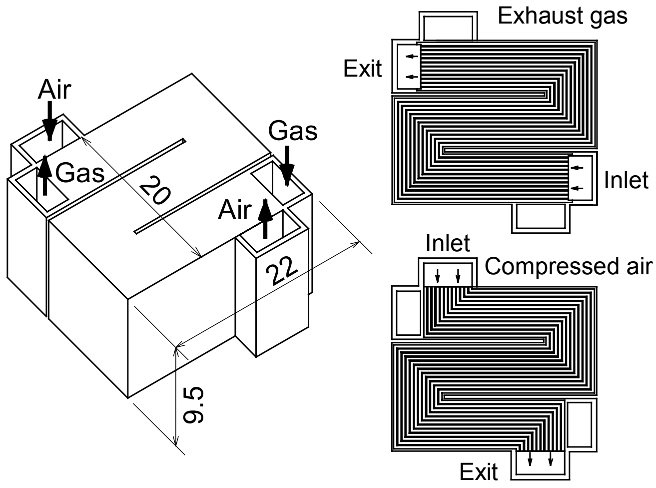


Fig.15 Micro recuperator for ultramicro 15W turbine (exhaust gas: 8 layers, air: 7 layers, ε=0.8)

CONCLUSIONS

Conceptual design of recuperator for palm-top size (nearly 3kW output) and button size (nearly 15W output) micro turbines has been made. In the case of 3kW class turbine an offset-strip-fin recuperator was employed. Dimensionless parameters were derived which enables absolute comparison of recuperator compactness. It was shown that the present offset-strip-fin recuperator with the finest fins currently available is effective especially in high *Re* number laminar region, which meets the operating condition of the designed turbine system. The optimum design of recuperator was presented by considering the relationship between cycle thermal efficiency and recuperator size, and an annular layout of multiple box-type recuperators was proposed which eliminates extra duct and reduces heat loss. In the case of button-sized turbine a micro channel recuperator was designed. Although the effectiveness of micro-channel recuperator is larger than that of offset-strip-fin recuperator, it suffers from deterioration of effectiveness due to the axial heat conduction of the wall. Because the optimum length of recuperator is significantly larger than the turbine size, a three-path cube-shaped recuperator was proposed. Although the volume of recuperator is much larger than that of turbine, the reduction of fuel consumption might compensate the recuperator volume by reducing the fuel tank size.

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