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Experimental Study of a 3-kW Stirling Engine

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A 3-kW Stirling engine, NS03T, which is a two-piston type with a unique V-shape cylinder arrangement, was developed for a household heat-pump system for air conditioning. By adopting this cylinder arrangement and a well-designed heat exchanger system, an approximate 50% indicated efficiency was obtained. In this paper the specifications and modifications of the 1984-year model NS03T engine are presented, which was the first engine and had the highest efficiency and performance among the NS03T engines. The indicated efficiency of the NS03T engine is presented by using experimental data and analyzed data by a mathematical model, Stirling Engine Thermodynamic and Mechanical Analysis. Also, the regenerator losses are analyzed by a method that is able to derive accurate regenerator loss from engine operating data. The results show that the design of the heat exchanger is important for high-efficiency engines.

Nomenclature

A = cross area
D, d = diameter
h = heat-transfer coefficient
L, l = length, distance, thickness
p = pitch
Q = heat quantity
S = surface area
T = temperature
W = power
η = efficiency, thermal efficiency
λ = thermal conductivity

Subscripts

air = air
b = burner
C = cooler, cooling water
c = compression
cc = compression cylinder
cond = conduction
e = expansion
ec = expansion cylinder
eff = effective
ex = exhaust
i = indicated
in = input
int = internal
mech = mechanical
pre = preheater
r = regenerator
rc = regenerator low-temperature side (cooler side)
rh = regenerator high-temperature side (heater side)
rloss = regenerator loss
s = output, shaft

Introduction

RELATED to recent substantial environmental and energy problems, Stirling cycle technology has attracted attention. With the aid of a heat-recovery mechanism between two isochoric processes of Stirling cycle, the cycle efficiency becomes higher. The regenerating mechanism, so-called regenerator, plays a very important role. If the regenerator works without any loss, the required heat input supplied at the heater and/or the expansion cylinder becomes less. For realization of Stirling engines with higher regenerator efficiency, it is essential to design and develop a suitable regenerator for each machine.

A 3-kW Stirling engine, NS03T, was developed for a household heat-pump system by Toshiba from 1982 to 1988.[1] The engine is a two-piston type with a unique V-shape cylinder arrangement. The 1984 engine achieved a 48% indicated efficiency and a 31% thermal efficiency. The NS03T was reassembled recently to study its high performance in detail at the National Defense Academy. The engine has being improved to develop practical small generator and cogeneration systems. It is interesting to analyze the engine performance based on reliable experimental data. Also, it is useful to arrange such technical information for design and development of new practical Stirling engines.

In this paper the indicated efficiency of the NS03T engine is analyzed by using experimental and analyzed data obtained at Toshiba and experimental data measured at the Mechanical Engineering Laboratory in Tsukuba. The data of the regenerator loss are calculated with a proposed method, which is able to derive the accurate regenerator loss from the engine operating data. In addition, the paper will try to introduce the specifications of the NS03T engine in detail.

Specifications of NS03T

Table 1 shows the design parameters of the 1984 engine. From 1982 to 1988 the NS03T engine project was funded by the New Energy Development Organization as one of national projects of the Moonlight program. The final model engine, which was installed in a heat-pump system, was constructed with lighter and compact components. Figure 1 represents the schematic view of the 1987-year model engine. The engine is a deformed in-line design with two pistons and cylinders mounted on a pressurized crankcase. The cylinders are arranged in unsymmetrical V form at an angle of 60 deg. The crank has two throws set at a 40-deg angle to give a phase difference the pistons of 100 deg.

The target engine of this study is a 1984 engine, which is shown in Fig. 2. The initial specifications and performance of the engine have been previously reported.[2] After several improvements the engine performance was improved dramatically.[3] Table 2 shows detailed information about the 1984 model engine, which is described next.

Combustion Systems

The optimized combustion chamber and a set of a fuel injector with eight injection holes, a swirler, and a burner-throat realized a

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Table 1 Design parameters of the 3-kW engine

Item	Design parameter
Main fuel	Natural gas
Working fluid	Helium
Mean pressure	3–6 MPa
Maximum expansion space temperature	975 ± 50 K
Compression space temperature	<323 K (water cooling)
Engine speed	500–1500 rpm
Maximum output power	>3 kW
Maximum thermal efficiency	32%
NOx	<150 ppm
Noise level	<60 dB(A)
Mass	<75 kg
Height	<800 mm

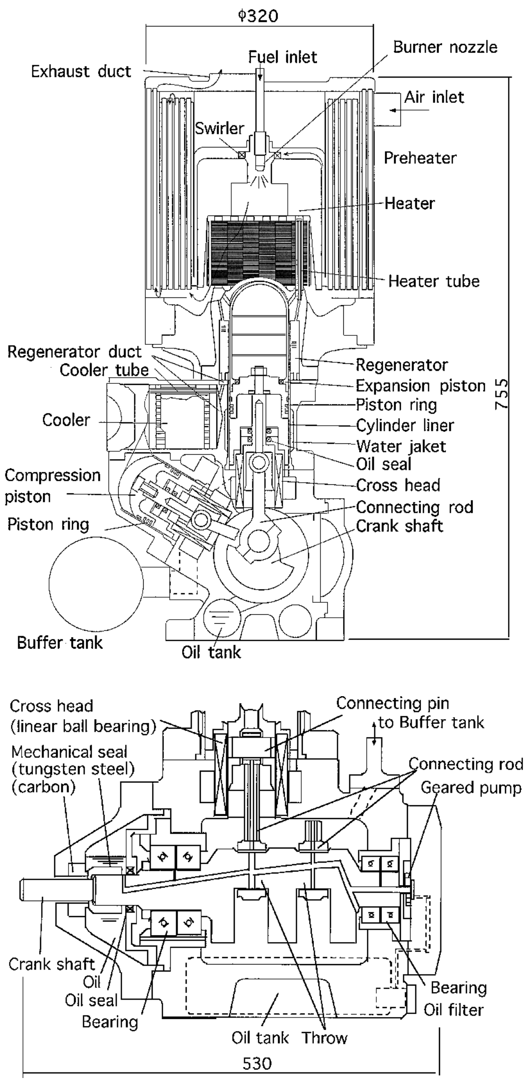


Fig. 1 1987-year model engine.

short flame length of about 150 mm. It has uniform temperature distribution of the combustion gas whose temperature goes up to 1773 K. The preheater is a counterflow-type fresh-air heater surrounding the combustor to reduce heat loss. The preheater matrix consists of 6 thin stainless cylinders and 180 tubes. The cylinders are arranged concentrically, and the tubes are set between the cylinders to enlarge the heating surface. The combustion gas flowing in the opposite direction in the tubes heats fresh air flowing through each annular duct between the cylinder walls. It has a heating surface of 1.3 m^2. The pressure drop of the combustion system is about 1.36 kPa, which includes 0.91 kPa of the flow-control valve.

The convective heat transfer is substantially improved by tangential and turbulent flow developed by the swirler of the combustor, which is located above the heater head. In addition, thermal radiation from high-temperature materials surrounding the heater increases the heat transfer. Also, uniformity of heater wall temperature and combustion gas velocity distribution, which passes between the fins of the heater, increases the heat transfer. The distance between the nozzle and the heater head was adjusted to obtain the minimum temperature difference between the top and the bottom of the heater of 140 K. Layers of one sheet of mesh size number 35 and wire size

Table 2 Engine specifications

Item	Data
1) Engine	
Type	Two piston
Swept volume	
Expansion	192 cm^3
Compression	173 cm^3
Volume phase angle	100 deg
Compression ratio	1.63
Crankcase	Pressurized
2) Combustor	
Type	Swirled diffusion burner
Combustion capacity	4.5–13 kW
Turndown ratio	1:3
Air excess ratio	0.8–2.0
Combustor volume	1,000 cm^3
Maximum load ration	13 MW/m^3
3) Preheater	
Type	Shell and tube
Heat-transfer area	
Combustion gas side	1.3 m^2
Fresh air side	1.3 m^2
4) Piston	
Bore	70 mm
Stroke	
Expansion	50 mm
Compression	45 mm
Unswept volume	
Expansion	40 cm^3
Compression	20 cm^3
Rod diameter	20 mm
Connecting rod length	108 mm
5) Heater	
Type	Bayonet tube with fin
Heater tube	
Outer diameter	16 mm
Length	152 mm
Number	12
Material	Hastelloy-X
Heat transfer area	
Working fluid side	0.09 m^2
Combustion gas side	0.63 m^2
Dead volume	100 cm^3
6) Regenerator	
Type	Canned
Number	1
Dead volume	150 cm^3
Matrix	
Material	Stainless steel 304
Mesh	150-46 + 200-47
Outer diameter	70 mm
Sheet number	210 + 250
Length	60 mm
Void ratio	0.65
7) Cooler	
Type	Double tube w/inner fin
Cooler tube	
Outer diameter	13 mm
Length	150.5 mm
Number	14
Material	Stainless steel 304

(Continued)

Table 2 Engine specifications (continued)

Item	Data
Heat transfer area	
Working fluid side	0.10 m^2
Combustion gas side	0.14 m^2
Dead volume	45 cm^3
8) Seal	
Piston seal	
Type	3-piece-type piston ring
Ring height	2.0 mm
Stage number	4
Main material	Polyimide18/Polyimide W
Rod seal	
Type	Scraper type
Stage number	2
Main material	VS83
Shaft seal	
Type	Mechanical seal
Balance ratio	0.60
Seal surface wide	2.0 mm
Material	Carbon/super-hard steel
Helium storage volume	
Crank room	7,700 cm^3
Buffer tank	10,000 cm^3
9) Output mechanism	
Type	60-deg V-crank
Lubrication system	Integrated geared pump
Fly wheel inertia	0.9 kg m

20 (35-20; wire pitch p is 0.725 mm; wire diameter d is 0.274 mm) and seven sheets of 35-50 ($p = 0.725$ mm; $d = 0.193$ mm) stainless-steel wire mesh screens and a 1-mm thickness radiation plate were located around the backside of the heater tube to enhance both the radiation and convectional heat transfer.

Heat-Exchanger System

The heater is constructed of 12 heater tubes, which are connected to the expansion cylinder concentrically. It was designed to achieve uniformity of the temperature in the tube wall and the flow of the working fluid. The heater is a double tube composed of an outer tube with round-shaped external fins and an inner tube with an inner core. The outer surfaces of the inner tube and core have four longitudinal fins contacting the outer tube and the inner tube, respectively. The outer tube is made of a heat-resistant alloy, Hastelloy-X and the others are made of copper. The temperature difference of each tube is less than 10 K at average heater wall temperature of 1070 K. The amount of heat transfer achieved is 7760 W at which point the net shaft power is 3 kW.

The cooler is a conventional shell and tube-type heat exchanger. The cooler has 14 tubes, and each of them has an inner core with four longitudinal fins similar to those of the heater inner tubes. Each tube has longitudinal grooves in the outer surface to discard heat to water with higher heat-transfer ratio. Helium flows in the 0.75-mm wide passage formed by the tube and core.

The regenerator is a can type with 60 mm length. Stacked stainless-steel 304 wire mesh is used as a regenerator matrix. The matrix is made up of 560 screens of 150-46 ($p = 0.169$ mm; $d = 0.061$ mm) and 200-47 ($p = 0.127$ mm; $d = 0.050$ mm) wire mesh. Each screen has a 3-mm-diam hole in its center, and a rod is passed through the hole to compose a cylindrical matrix. The diameter of the matrix is the same as the piston diameter of 70-mm. Three wheel-shaped plates with 1-mm thickness were inserted in the matrix, and two were put at both ends to decrease side leakage loss.

For the regenerator design concept the regenerator temperature efficiencies η_{rth}, η_{rtc} are useful.

$$\eta_{\mathrm{rth}} = \frac{T_{\mathrm{rh}} - T_c}{T_h - T_c} \tag{1}$$

$$\eta_{\mathrm{rtc}} = \frac{T_h - T_{\mathrm{rc}}}{T_h - T_c} \tag{2}$$

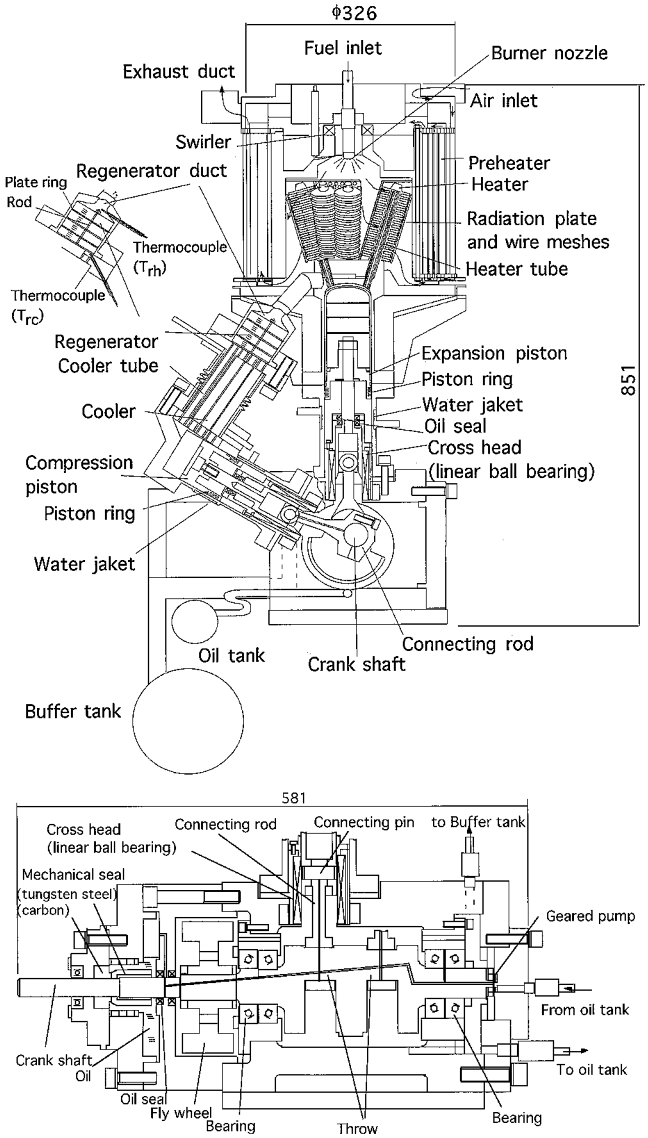


Fig. 2 1984-year model engine.

where T_h, T_c, T_{rh}, T_{rc} represent heater wall temperature, cooler wall temperature, gas temperature at the high-temperature side (heater side) of the regenerator, and gas temperature at the low-temperature side (cooler side) of the regenerator, respectively. It is difficult to measure the actual temperature swing. Although the temperatures were measured by using thermocouples sealed in a stainless-steel sheath (shown in Fig. 2) and a slow digital voltmeter, the defined efficiencies show the discrepancy from the ideal situation. If the regenerator works ideally, T_{rh} is close to T_h, and T_{rc} becomes T_c. The measured thermal efficiency was around 86% for the high-temperature side η_{rth} and 94% for low temperature side η_{rtc}. The pressure drop is approximately 0.3 MPa at 6 MPa mean engine pressure and 1000 rpm engine speed.

Driving System

Four-stage low-friction loss and low-wear piston rings are used for the NS03T engine. Three-piece type, which consists of two plane rings, an inner ring, and a steel tension ring, is used for each piston. The material of the plane and inner rings is polyimide in which carbon fiber is added to reduce wear and increase heat resistance. The friction coefficient of the assorted rings is less than 0.1. The mechanical seal is set at the end of the crankcase where lubricant is supplied by a forced lubrication system. The friction loss of the mechanical seal is about 140 W at 6 MPa and 1000 rpm.

A crank mechanism with cross head is adopted. The cross head plays the role of supporting the side force loading to the piston rod.

The cross heads slide in linear ball bearings. Lubricant is transferred by a geared pump whose swept volume is 1 cm^3. A phase difference between expansion and compression pistons is selected at an angle of 100 deg. The crankshaft is supported by ball-bearing sets. A flywheel, made of heavy alloy, is attached to the crankshaft.

Engine Performance

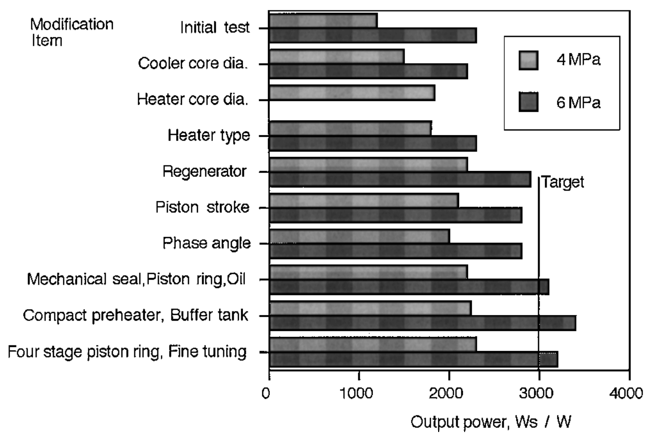
The engine performance was measured at the official midterm evaluation test under New Energy Development Organization in Japan. It was held at the Mechanical Engineering Laboratory in Tsukuba in 1984. At that time detailed performance data for other Moonlight-program engines by Mitsubishi Electric (NS03M), Aishin Seiki (NS30A), and Sanyo (NS30S) were also measured. Major testing results of the NS03T engine were already reported in the literature.[2] Table 3 shows the more detailed results of the NS03T engine at the maximum power and efficiency conditions. The definitions of the powers, heats, losses, and efficiencies are shown in Table 4. Even though the 1984 model engine was designed after only two years from the start of the project and was developed for only six months, it had satisfactory performance as a practical engine. It is noteworthy that the indicated efficiency of 48.8% is very high. The associated targets, like engine mass, dimension, exhaust gas characteristics, were not realized until the final evaluation test after another three years.[1]

Table 3 Engine performance

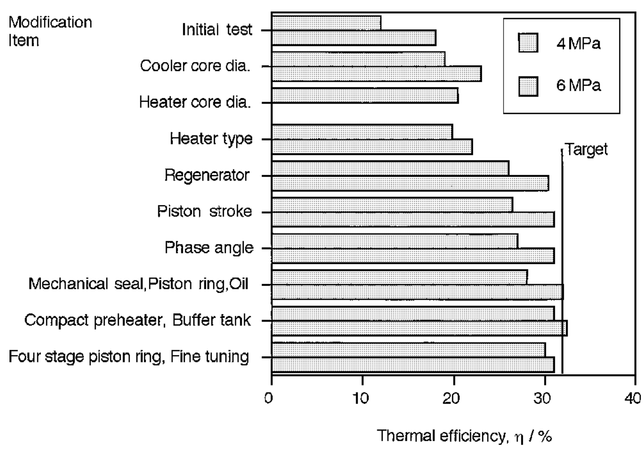
Measuring item	Engine data
1) Maximum output conditions	
Heater tube average temperature	1062 K
Gas temperature in exp. space	959 K
Mean cycle pressure	6.1 MPa
Engine speed	1250 rpm
Cooling water temperature	298 K
Cooling water flow rate	20×10^3 cm^3/min
Output power	3.46 kW
Thermal efficiency η	27.3%
Burner efficiency η_b	81.4%
Preheater efficiency η_{pre}	57.0%
Interconversion efficiency η_{int}	49.9%
Indicated efficiency η_i	46.5%
Mechanical efficiency η_{mech}	72.2%
2) Maximum efficiency conditions	
Heater tube average temperature	1071 K
Gas temperature in exp. space	971 K
Mean cycle pressure	6.1 MPa
Engine speed	900 rpm
Cooling water temperature	298 K
Cooling water flow rate	20×10^3 cm^3/min
Output power	3.00 kW
Thermal efficiency η	31.1%
Burner efficiency η_b	80.7%
Preheater efficiency η_{pre}	57.1%
Interconversion efficiency η_{int}	52.7%
Indicated efficiency η_i	48.8%
Mechanical efficiency η_{mech}	78.9%
3) Partial load characteristics	
Pressure range	3–6.1 MPa
Speed range	500–1400 rpm
Power variable ratio	1:4.13
Turndown ratio	1:3.69
4) Cooling temperature dependence	
Cooling inlet water temperature	283–323 K
Decrement of shaft power	12%
Decrement of thermal efficiency	9%
Exhaust gas	
NOx	290 ppm
HC	0.9 ppm
CO	185 ppm
Mass	270 kg
Noise level	Approximately 72 dB(A)
Temperature distribution on heater tube (at 6 measuring points)	±10 K

Table 4 Definitions of powers and efficiencies

Definition	Symbol, Equation
1) Power	
Expansion work (power)	$W_e = n \oint P_e \, dV_e$
Compression work (power)	$W_c = n \oint P_c \, dV_c$
Indicated work (power)	$W_i = W_e - W_c$
Output power	W_s
Mechanical loss	$W_{mech} = W_i - W_s$
2) Quantity of heat	
Heat input	Q_{in}
Exhaust gas heat at preheater inlet (quantity)	$Q_{ex,in}$
Preheater recovered heat (quantity)	Q_{pre}
Cooling heat in cooler (quantity)	Q_c
Cooling heat in exp. cyl. water jacket (quantity)	Q_{ec}
Cooling heat in comp. cyl. water jacket (quantity)	Q_{cc}
Regenerator loss	$Q_{rloss} = Q_c + Q_{ec} - W_c$
Effective heat input	$Q_{eff} = Q_{rloss} + W_e$
3) Efficiency	
Thermal efficiency	$\eta = W_s / Q_{in} = \eta_b \eta_i \eta_{mech}$
Burner efficiency	$\eta_b = Q_h / Q_{in}$
Preheater efficiency	$\eta_{pre} = Q_{pre} / Q_{ex.in}$
Internal conversion efficiency	$\eta_{int} = W_i / W_e$
Indicated efficiency	$\eta_i = W_i / Q_{eff}$
Mechanical efficiency	$\eta_{mech} = W_s / W_i$



Output power



Thermal efficiency

Fig. 3 Improved engine performance and modification items.

The initial performance of the 1984 model engine was not completely as desired. As shown in Fig. 3, the performance was improved by many experimental procedures with an aid of mathematical analysis. All of the modifications were done during six months. The modifications of the heat-exchanger system influenced the engine performance significantly, especially the optimization of the cooler and the regenerator. In this paper a method for optimizing the regenerator is presented.

Regenerator Loss of 3-kW Stirling Engine

To optimize the regenerator, regenerator losses that are intricately connected each other must be clarified. To regenerate heat during the short period of time when the working gas passes through it, a good regenerator must possess large heat-transfer surface, and the regenerator temperature efficiencies are close to unity. However, a regenerator with good heat-transfer characteristics and high heat capacity is likely to cause too much pressure drop of the working gas flow. The desirable matrix has porosity, high heat capacity, and low flow impedance.

There are always considerable losses that cannot be avoided: heat transfer, pressure drop, heat conduction, side leakage, matrix friction, and duct shape losses. Their interrelation becomes more complicated by the fact that the flow in the regenerator is not uniform and is reciprocating with a phase angle of 90–100 deg. For the purpose of providing appropriate design information for regenerator, many papers have been published and reported empirical information about regenerator and matrix.[4–6] However, such information was normally arranged based on experimental data, which were obtained by using an element test stand not equipped with the whole heat-exchanger system and cylinder set. With only such information it is difficult to evaluate the actual performance of a regenerator installed in actual engines.[7]

To predict engine and cooling machine performance, a Stirling Engine Thermodynamic and Mechanical Analysis (SETMA) has been developed. SETMA divides the working space into five controlled volumes and can analyze engine and cooler performance with enough accuracy for industrial uses.[3,8] It is a useful analysis to design Stirling cycle machines and to determine their dimensions. Figure 4 shows the SETMA analysis results for indicated efficiency of the NS03T engine. It is a function of the regenerator length and the diameter at 4.1 MPa, 1000 rpm with 1070 K heater-wall temperature and 298 K cooler-wall temperature. The result shows that the highest efficiency is obtained with 60 mm length and 70 mm diam. The calculated regenerator loss (reheat loss) is 480 W.

Such a mathematical model is very useful to design and develop an engine; however, experimental procedure is required to determine the actual dimension. Figures 5–8 show measured results about one of the parameter tests of the 1984 engine to determine the final dimension and the matrix of the regenerator at 5.9 MPa with 1130 K heater-wall temperature and 290 K cooler-wall temperature. Four matrix sets shown in Table 5 were prepared for the test. Each data set of the matrix is fitted with a curve in Figs. 4–8. Figure 5 shows the indicated power curves when installing each matrix. It indicates that the matrix of 100-42 with a short body and 150-46 and 200-47 with a longer body promise to play a good role as the matrix of the 1984 engine.

As shown in Fig. 6, 100-42 is suitable for lower engine speed operation relative to indicated efficiency. The mixed type gets the better performance at the higher engine speed region. In Figs. 7

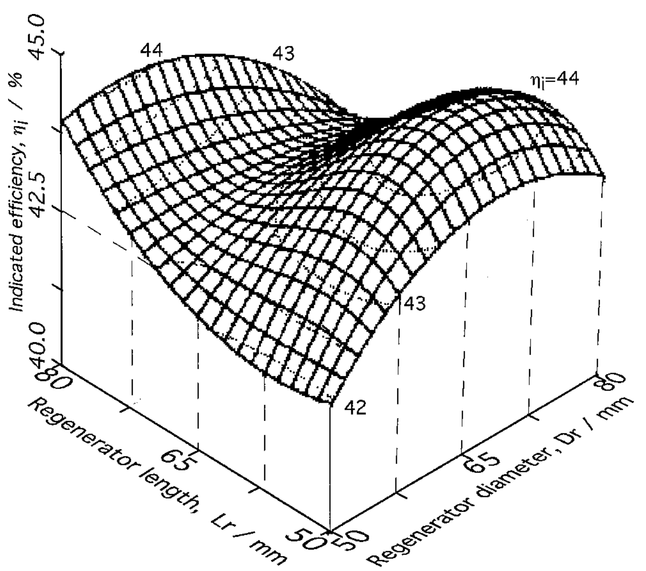


Fig. 4 Calculated indicated performance.

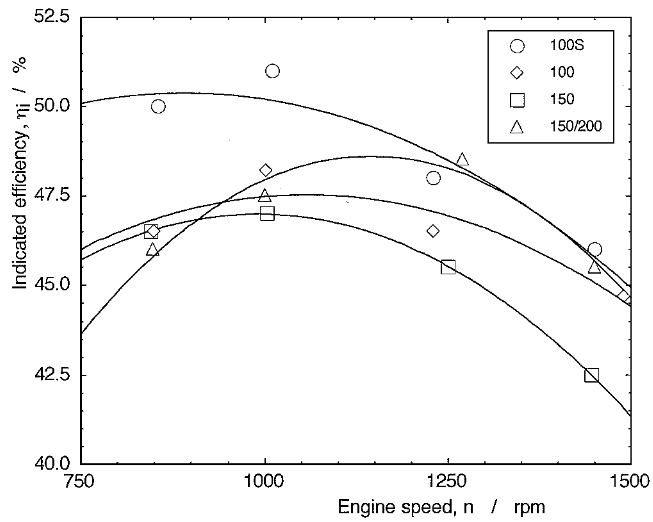


Fig. 6 Indicated efficiency of matrix sets.

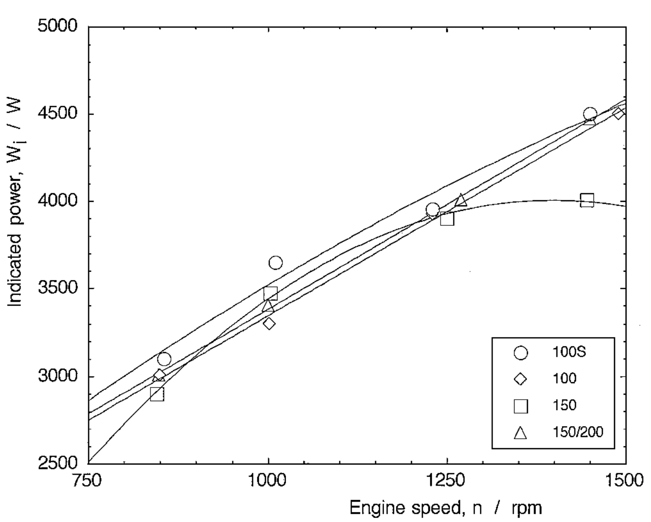


Fig. 5 Indicated power of matrix sets.

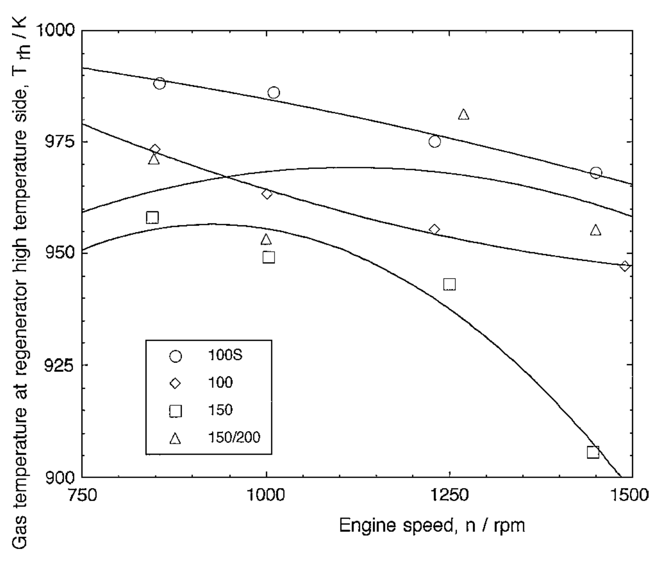


Fig. 7 Gas temperature at high-temperature side.

Table 5 Specifications of tested regenerator

Engine speed	100S	100	150	150/200
Dimension $D_r \times L_r$, mm	70×56	70×60	70×60	70×60
Mesh number	100-42	100-42	150-46	150-46 + 200-47
Sheet number	227	240	428	210 + 250
Wire diameter, mm	0.101	0.101	0.061	0.061, 0.050
Pitch, mm	0.254	0.254	0.169	0.169, 0.127
Matrix surface area, mm^3	2.56×10^9	2.76×10^9	4.18×10^9	5.52×10^9
Equiv. flow area, mm^2	1.41×10^6	1.41×10^6	1.32×10^6	1.41×10^6
Dead volume, mm^3	1.29×10^5	1.38×10^5	1.35×10^5	1.38×10^5

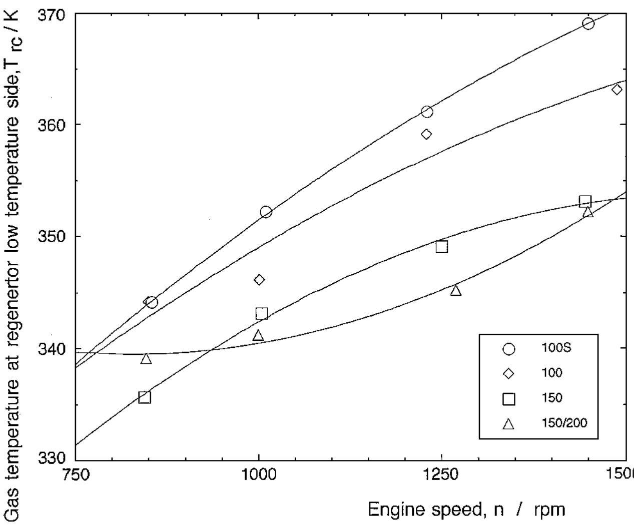


Fig. 8 Gas temperature at low-temperature side.

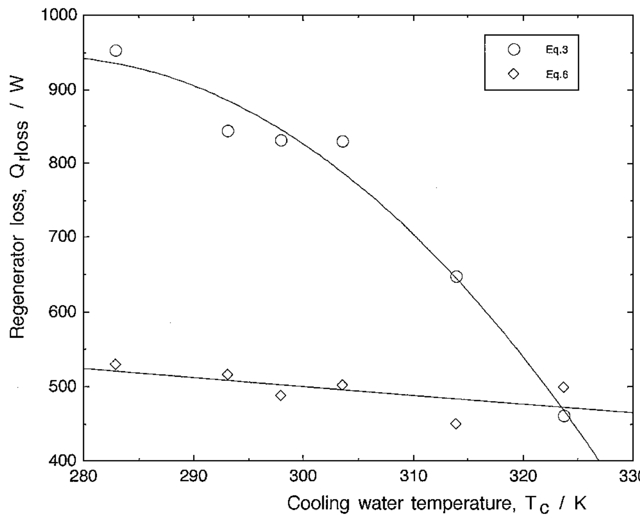


Fig. 9 Regenerator loss vs cooling temperature.

and 8, which show average gas temperatures at each end of the regenerator, it is clear that the mixed type has a higher efficiency. From the experimental and the SETMA analysis results the mixed regenerator was selected for the 1984 model engine. As show in Fig. 3, the engine has a higher performance with this regenerator and matrix.

Evaluation of Regenerator Loss

Regenerator performance is difficult to evaluate in an actual engine. Equation (3) is commonly referred to as regenerator loss:

$$Q_{\text{rloss}} = Q_c + Q_{\text{ec}} - |W_c| \tag{3}$$

However, the regenerator loss calculated from Eq. (3) easily shifted with operating conditions. For example, Fig. 9 shows the temperature dependence of the regenerator loss during the midterm evaluation test of the 1984 model engine at 6.1 MPa, 1000 rpm, 1070 K of the heater wall temperature. Figure 10 shows the regenerator loss with changing engine speed and charged with helium at 1070 K heater-wall temperature. The lines in the figures are curve fits to show the data behavior. Even though the temperature efficiency of the regenerator was kept at 86 and 94% by changing the cooling water temperatures, the regenerator loss in Fig. 9 is changed considerably. The reason seems to be that the heat flows in the engine were influenced by temperatures of the cooling water, the cylinders, and crankcase, and the air around the engine. Proposing a model of the heat flows is defined by using the following heat-transfer fundamentals.[9]

Heat conduction from element in engine:

$$Q_{\text{conduction}} = \lambda A(\Delta T/l) \tag{4}$$

Free convection from engine to air:

$$Q_{\text{heat transfer}} = hS\Delta T \tag{5}$$

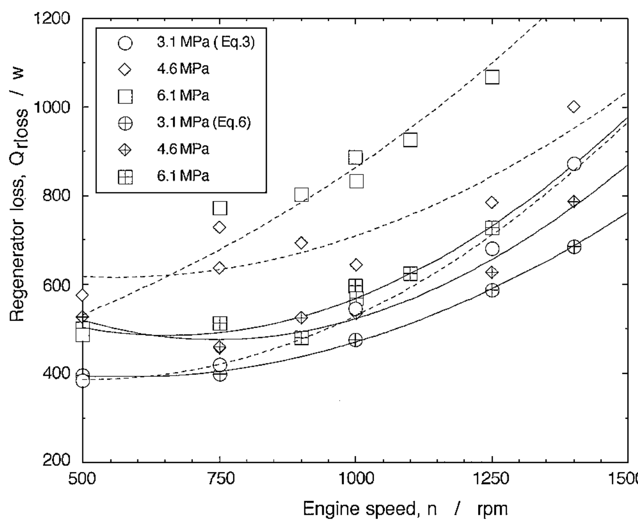


Fig. 10 Regenerator loss of NS03T.

A proposed method to evaluate regenerator loss from actual engine data is

$$Q_{\text{rloss}} = Q_c + Q_{\text{cc}} + Q_{\text{air}} - Q_{\text{cond}} - |W_c| \tag{6}$$

where Q_{air} is heat transfer from the cooler to air and Q_{cond} is heat leakage from the compression cylinder to the cooler.

Figures 9 and 10 include the regenerator loss of the engine based on Eq. (6). It shows more reasonable behavior, and it is similar to that of the SETMA analysis results. The calculated reheat loss is 500 W at 6 MPa and 1000 rpm. The calculated thermal conduction loss along the regenerator wall is 130 W. Thermal conduction loss through the matrix is 18 W. At 500 rpm the heat transfer of the matrix is decreased by the lower flow rate. Furthermore, the reheat energy becomes less so that the thermal conduction losses are dominant in the regenerator loss. The measured regenerator temperature efficiencies also decreased at lower engine speed.

Conclusions

In this paper the specifications of the 1984 NS03T engine were presented. The higher indicated efficiency of the NS03T engine was analyzed by using reliable experimental data and the mathematical model SETMA. The regenerator loss was analyzed with a proposed method that was able to derive the regenerator loss from engine operating data. The results show that the design of the heat exchanger is important for high-efficiency engines.

The NS03T was reassembled to study its performance in detail at National Defense Academy. The engine performance and the regenerator loss are currently being measured to improve them. In this study the developed engine will be used to develop a cogeneration system.

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