Proceedings of the International Gas Turbine Congress 2003 Tokyo
November 2-7, 2003

IGTC2003Tokyo TS-112

Development of Large Scale Recuperator for Gas Turbine

Ryo AKIYOSHI[1], Kiwamu IMAI[2], Tatsuya SIODA[3], Katsunori ITO[1] and Masayoshi HORI[4]

[1] Thermal and Fluid Technology Dept., Research Laboratory
Ishikawajima-Harima Heavy Industries Co.,ltd.
1 Shin-Nakahara-cho. Isogo-ku, Yokohama 235-8501, JAPAN
Phone: +81-45-759-2869, FAX: +81-45-759-2210, E-mail: ryo_akiyoshi@ihi.co.jp
[2]Marine Application Gas Turbine Dept.
[3]Nuclear Power Project Dept.
[4]Research and Development Headquarters

ABSTRACT

Since 1997, a recuperated gas turbine engine for ship propulsion named SMGT (Super Marine Gas Turbine) is developed. SMGT engine has a recuperator to reduce fuel consumption. Authors developed the recuperator for SMGT.

High reliability, compactness, high heat transfer performance, and low pressure drops are required to the SMGT recuperator.

We designed and manufactured a recuperator for SMGT. Engine tests are performed from 2001 to 2003. Ignitions more than 400 times are tried, and run time exceeded 140 hours. A recuperator has been affected by thermal shock and soot, but it has worked while the engine tests.

NOMENCLATURE

A heat transfer area (m^2)
C constant
cm heat capacity (W/K)
E Young's modulus (Pa)
e maximum distance from axis (m)
H fin height (m)
h heat transfer coefficient (W/m^2K)
I moment of inertia if area (m^4)
K overall heat transfer coefficient (W/m^2K)
ΔL thermal expansion difference (m)
M bending moment (Nm)
Ntu number of transfer unit (-)
P fin pitch (m)
Δp pressure loss (Pa)
R capacity rate ratio (-)
r usual fouling factor (m^2K/W)
t thickness of separate plate (m)
ΔT temperature difference (K)
u velocity (m/s)
W load (N)
β thermal expansion coefficient (1/K)
δ thickness of soot (m)
ε strain (-)
λ thermal conductivity of soot (W/m·K)
ρ density (kg/m^3)
σ thermal stress (Pa)
ϕ exchanger effectiveness (-)
Subscript a air
g exhaust gas
0 absence of soot

Manuscript Received on September 24, 2003

Table 1 Specification of SMGT Recuperator

Exhaust Gas	Inlet Temperature	°C	673
	Inlet Pressure	MPaA	0.11
	Flow Rate	kg/s	9.6
	Pressure Loss	%	≦4
Air	Inlet Temperature	°C	290
	Inlet Pressure	MPaA	0.81
	Flow Rate	kg/s	8.6
	Effectiveness	%	≧83
	Pressure Loss	%	≦3
Size of Recuperator		m	1.4*1.2*1.2
Maximum Power of Engine		kW	2500
Fuel		-	Heavy oil
Lifetime		hour	10,000

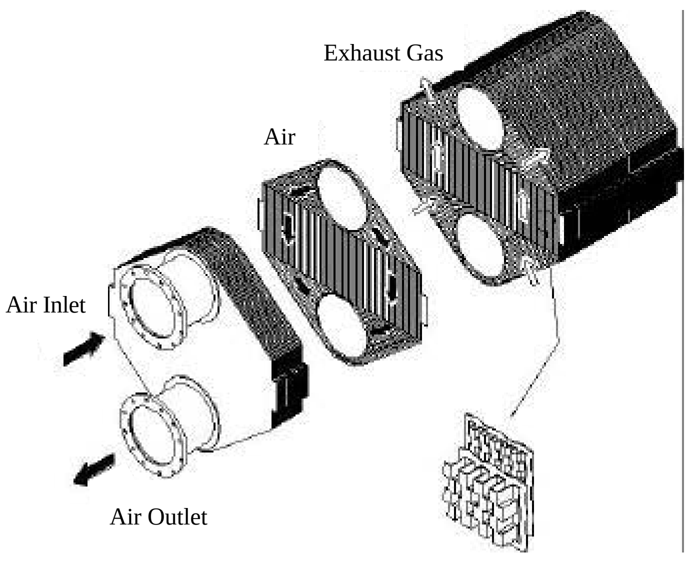


Fig.1 Cross Sections of SMGT Recuperator

INTRODUCTION

Recuperated gas turbine cycle isn't very popular. The first reason is that we can obtain high efficiencies by combined cycle or co-generations. The second reason is that it isn't easy to design compact and enduring recuperators. Recuperators for gas turbine are expected for transportations mainly.

Since 1997, advanced development of a recuperated gas turbine engine for ship propulsion named Super Marine Gas Turbine, SMGT has been progressing under Japanese MLIT. SMGT engine has a recuperator to reduce fuel consumption. Authors developed the recuperator for SMGT. High reliability, compactness, high heat transfer performance, and low pressure drops are required to the recuperator.

Reliable compact heat exchanger for gas turbine isn't easy to design, because a thermal shock by start up and shut down of gas turbine are very strong. We selected symmetrical shape to flow direction and width direction of recuperator to reduce thermal stress (Ito et al, 1996). Temperature distribution in the neighborhoods of air outlet channel causes thermal stress around air outlet, so we set heat transfer fins all around air outlet channels to reduce thermal stress (Hori et al., 1994)(Nagamori et al., 1995). We have confirmed reliability more than 1500 cycle by cyclic test (Yoshimura et al., 1998).

Recuperator is relatively big to gas turbine engine. So usual recuperators for gas turbines are compact heat exchangers to consist with high heat transfer performance and low pressure drops. There aren't many samples of large scale compact heat exchangers.

We tried to develop a large scale recuperator against to these problems.

DESIGN AND COMPONENT TESTS

Specifications of a recuperator are shown in table 1. Plate fin type heat exchanger is selected to satisfy both of high performance and compactness. Cross sections of a recuperator are shown in figure 1. An usual fouling factor is accounted for design. Sizes of fins and a recuperator are decided as follows. There are a number of combinations of those to satisfy exchanger effectiveness and pressure losses. Lines in figure 2 shows minimum size to satisfy exchanger effectiveness and gas pressure loss. A design point was selected to minimize volume of recuperator as shown in figure 2. Design results are shown in table 2. Shapes of offset fins are rectangle as shown in figure 3.

Exchanger effectiveness and pressure losses are confirmed by subscale heat exchange tests with a heat exchanger whose plate size is the same to a recuperator and number of pairs of exhaust gas and air is 8.

Material for fins and separating plates are selected by corrosion tests to expose the test pieces to heavy oil exhaust gas of whitch maximum temperature is 680 °C. Corrosion test results in 1,000 hours are extrapolated to lifetime 10,000 hours linearly, and a material and thickness are defined.

MANUFACTURING OF RECUPERATOR

A Recuperator consists of 134 pairs of fins and separating plates. They were joined to 6 blocks by brazing. They were welded as shown in figure 4. A recuperator was set in an exhaust gas duct as shown in figure 5, and put onto an exhaust duct as shown in figure 6. And then, air headers are connected to air inlet duct from a compressor and to air outlet duct to combustion chambers.

ENGINE TESTS RESULTS

Engine tests were carried out from November 2001 to February 2003. Figure 7 shows changes of exchanger effectiveness and pressure losses. Horizontal axis shows the total run time. Effectiveness tends to decrease and exhaust gas pressure loss tend to increase with ignition trial times. They are due to soot deposition more than estimation to gas channels. Before a

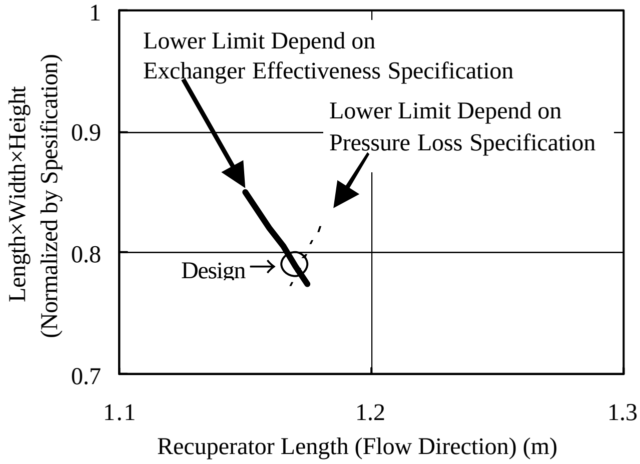


Fig.2 Optimization of Size of Fin and Recuperator

Table 2 Design of SMGT Recuperator

Fin Type	Counter Flow Part		Offset Fin
	Cross Flow Part		Strait Fin
Fin Height	Air	mm	2.0
	Exhaust Gas	mm	6.0
	Width	m	1.22
Recuperator	Length (Flow Direction)	m	1.17
	Height (Pile Direction)	m	1.29
	Pile Number	Pair	134

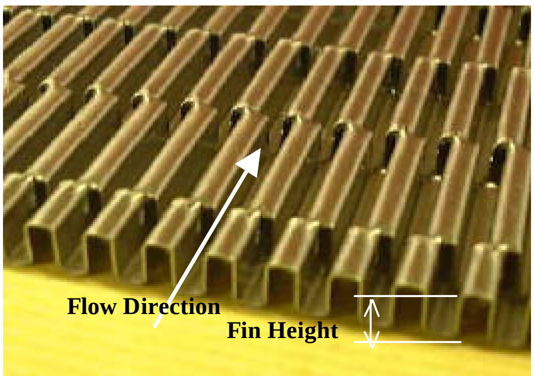


Fig.3. Offset Fin for Exhaust Gas Channel

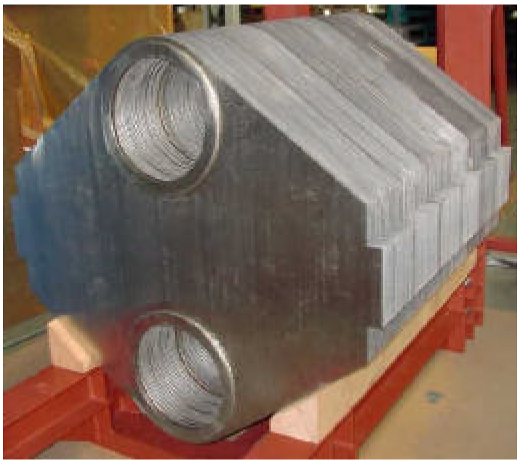


Fig.4 A Recuperator

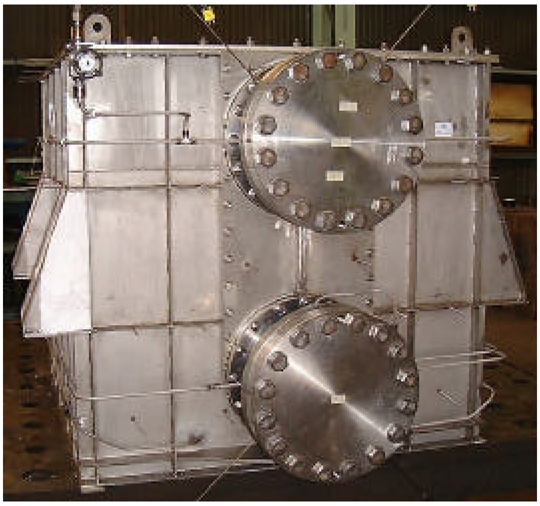


Fig.5 A Recuperator Set in an Exhaust Gas Duct

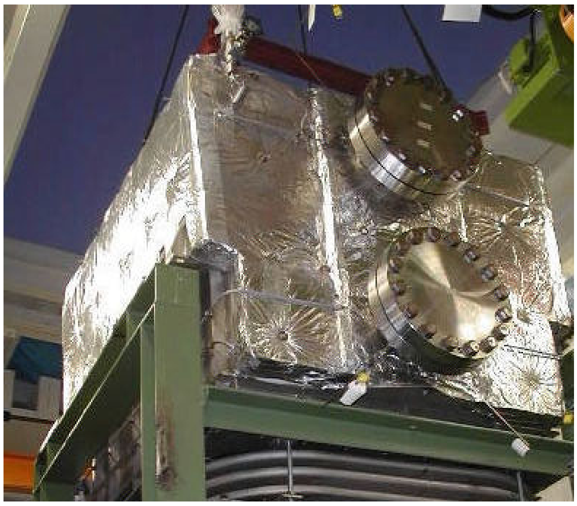


Fig.6 A Recuperator Put on an Exhaust Gas Duct

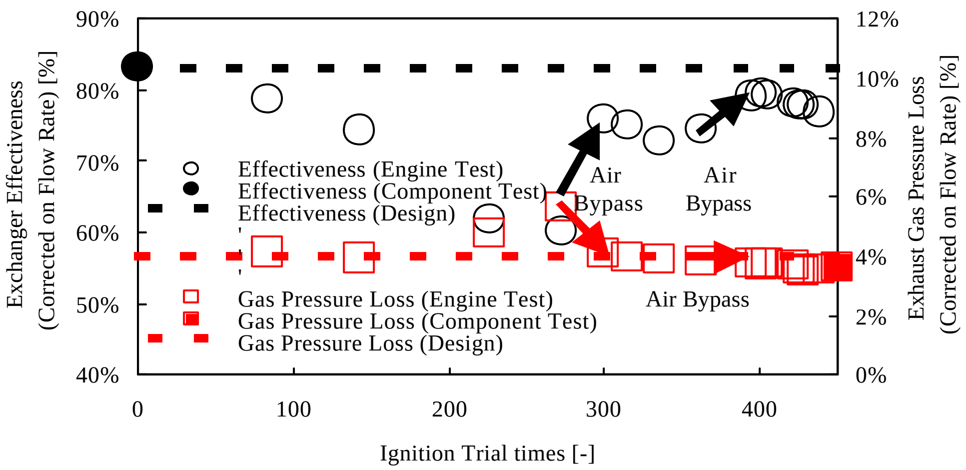
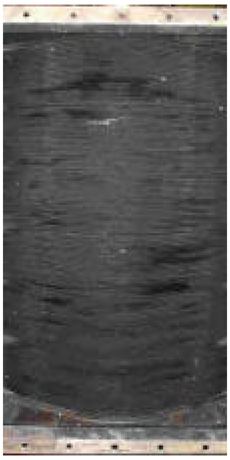
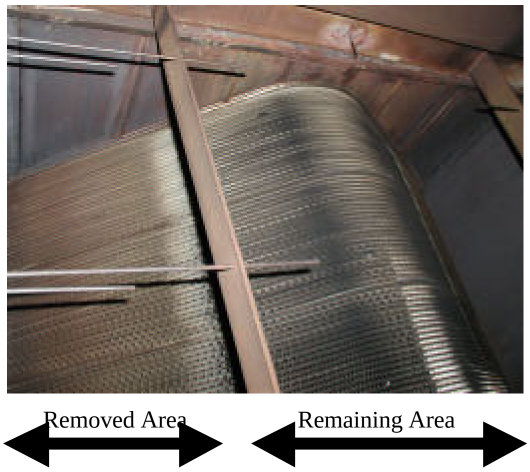


Fig.7 Change of Exchanger Effectiveness and Gas Pressure Loss



(a) Deposited Soot



(b) After Removal of Soot by Air Bypass

Fig.8 Soot Adhesion in Exhaust Gas Outlet

measurement in which effectiveness reaches to minimum, gas channel of the recuperators were checked. There are deposited soot on outlet of exhaust gas as shown in figure 8. There was no soot on gas inlet side due to high temperature.

For removal of soot, heating the recuperator by air bypass was tried. A part of soot removed by air bypass as shown in figure 8, and effectiveness increased. But it didn't achieve to a specification by the remains of soot. Gas pressure loss decreased by air bypass, and it achieved to a specification. Air pressure loss was low enough to a specification. Test results after second air bypass are shown in table 3.

At a check of recuperator after engine tests, there is a little leakage in gas inlet.

INFLUENCES OF SOOT DEPOSITION

Table 3 shows differences between engine test and component test on exchanger effectiveness and exhaust gas pressure loss. We calculated a soot thickness, and predicted effectiveness and pressure loss with uniform soot on exhaust gas channel as follows (Kays and London, 1964).

Exchanger effectiveness is calculated approximately as follows.

$$\phi = (1-E)/(1-RE) \quad (1)$$

$$E = \exp[-(1-R)Ntu] \quad (2)$$

$$R = (cm)_a/(cm)_g \quad (3)$$

$$Ntu = KA/(cm)_a \quad (4)$$

$$1/(KA) = 1/(h_g A_g) + r/A_g + \delta/(\lambda A_g) + 1/(h_a A_a) \quad (5)$$

$$h_g = C_g (u_g/u_{g0})^{0.8} \quad (6)$$

Table 3 Tests Results

		Exchanger Effectiveness	Pressure Loss	
			Air	Gas
Specification	%	≧83.0	≦3.0	≦4.0
Engine Test Result	%	79.1	1.8	3.8
Component Test Result	%	83.2	2.2	3.6

$$u_g/u_{g0} = HP/\{(H-2\delta)(P-2\delta)\} \qquad (7)$$

Pressure loss of exhaust gas is expressed as follow by results of engine tests.

$$\Delta P = C_p \rho (u_g/u_{g0})^{1.8} \qquad (8)$$

Calculated exchanger effectiveness and pressure loss are shown in figure 9 as black lines. A blue arrow shows difference of pressure loss between engine test and component test. A cross point shows that a thickness of remaining soot is about 0.1 mm, and decrease of exchanger effectiveness by soot is 4 points. Therefore, if soot deposition is usual, exchanger effectiveness of this recuperator achieves to a specification.

INFLUENCES OF RAPID GAS TEMPERATURE CHANGE

Exhaust gas temperature distributions are measured on inlet and outlet section by 16 thermocouples arranged in square pattern. Figure 10 shows usual temperature changes of exhaust gas inlet and air outlet at start up. Exhaust gas inlet temperature rises in 1 or 2 minutes immediately after an ignition. Immediately after a shutting down of engine, exhaust gas inlet temperature fall down rapidly, too. This recuperator was designed to bear these usual thermal stresses.

Figure 11 shows maximum and minimum change of exhaust gas inlet in each run. Maximum changes occur at start up, and minimum changes occur at shut down. Rapid temperature change cause strong thermal stress. When exhaust gas temperature rises up very rapidly, almost separating plates expand downward as red arrow in figure 12 very rapidly. Both of terminal plates of a recuperator shown as the right end plate in figure 12 are thicker to reinforce recuperator, so they and their neighbor plates expand lately. The difference of expands causes strong thermal stress at neighbor to terminal plates shown by blue arrow in figure 12. They can be calculated as follows.

$$\sigma = Me/I \qquad (9)$$

$$M = WH \qquad (10)$$

$$W = 3EI\Delta L/H^3 \qquad (11)$$

$$I = Pt^3/12 \qquad (12)$$

$$\Delta L = \varepsilon L \qquad (13)$$

$$\varepsilon = \beta \Delta T \qquad (14)$$

$$e = t/2 \qquad (15)$$

It is possible that thermal stress reaches 700MPa, and it excesses a tensile strength of separate plate. Damages seem to occur by one or a couple of rapid gas temperature changes. The cause isn't low cycle fatigue.

CONCLUSION

A recuperator for 2500kW gas turbine has developed. Following results are obtained.

(1) One of the largest compact type heat exchanger for gas turbine recuperator is designed to minimize a volume of recuperator by optimization of size of fins and a recuperator (1.4m*1.2m*1.2m).

(2) Exchanger effectiveness decreased by soot more than estimation. Heating by air bypass is effective to soot removing Exchanger effectiveness increased, and gas pressure loss decreased by soot removing.

(3) In engine tests, ignitions more than 400 times are tried, and run time exceeds 150 hours. After all tests, a little leakage of air recognized in gas inlet side. Exhaust gas temperature have changed very rapidly by unusual burning condition. They cause big thermal stresses those exceed a tensile strength of material and leakage.

ACKNOWLEDGEMENT

This development has done as a part of development of SMGT (Super Marine Gas Turbine), and has supported by MLIT (Ministry of Land, Infrastructure and Transport), CATT

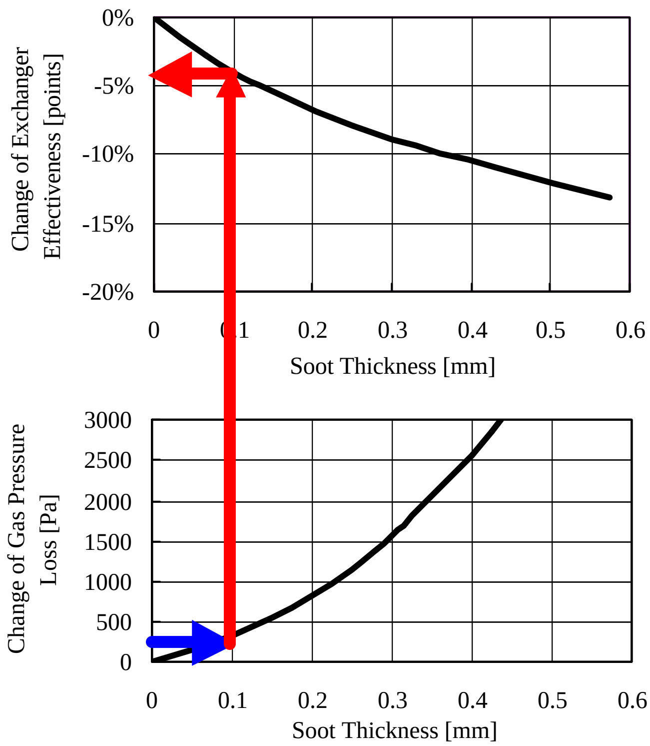


Fig.9 Influences of Soot Thickness to Exchanger Effectiveness and Gas Pressure Loss

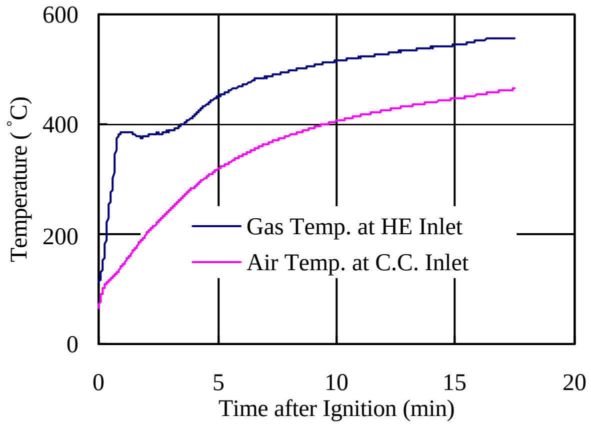


Fig.10 Temperature Change at Start Up of Engine

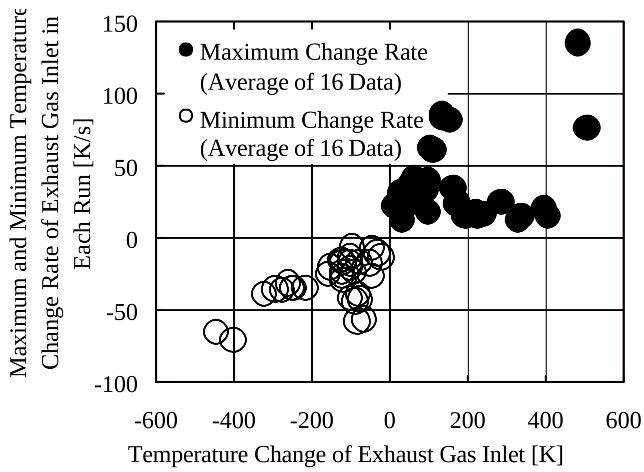


Fig.11 Maximum Temperature Change Rate at Gas Inlet

(Corporation for Advanced Transport and Technology), and the Nippon Foundation.

References

Yoshimura, Y., et al., 1998, "Investigation on Recuperator for Gas Turbine", IHI Technical Report, Vol. 38, pp. 308-312.

Hori, M., et al., 1994, "Investigation on Unsteady State Heating Characteristics of Plate Fin Heat Exchanger for Gas Turbine, 1st. Unsteady State Temperature Characteristics", *Proceedings, 9th lecture meeting by Gas Turbine Society of Japan*, pp. 283-289.

Nagamori, K., et al., 1995, "Investigation on Unsteady State Heating Characteristics of Plate Fin Heat Exchanger for Gas Turbine, 2nd. Simulation Test Results", *Proceedings, 10th lecture meeting by Gas Turbine Society of Japan*, pp. 157-162.

Ito, K., et al., 1996, "Investigation on Unsteady State Heating Characteristics of Plate Fin Heat Exchanger for Gas Turbine, 3rd. Test Results by Large Scale Heat Exchanger", *Proceedings, 11th lecture meeting by Gas Turbine Society of Japan*, pp. 107-111.

Kays, W. M., and London, A. L., 1964, "Compact Heat Exchangers", McGraw-Hill Inc., NY., 2nd ed., pp.10-18.

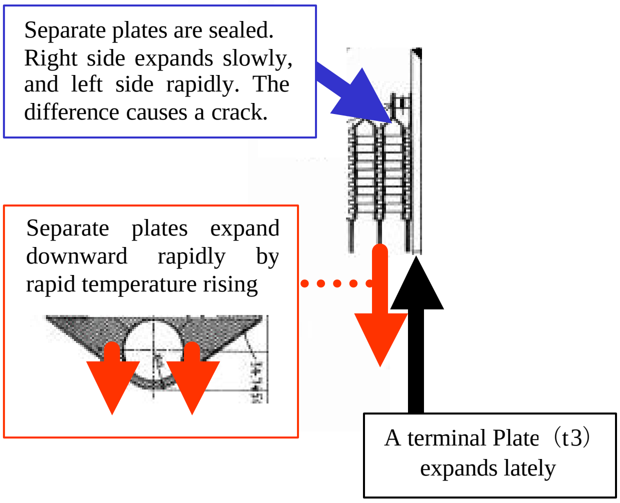


Fig.12 Thermal Stress by Rapid Exhaust Gas Temperature Change