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ABOUT THE STUDY OF THE THERMAL STRESS FOR NAVAL SYSTEMS

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Abstract: *In this paper are presented and analyzed the effects of thermal expansion on gas evacuation piping from naval power plants and technical protection possibilities to prevent structures from deformations; also are analyzed the possibilities for the use of thermal expansion for tightening the main screws for power plant propulsion.*
Keywords: *compensating pipe, linear temperature expansion coefficient, overall heat exceanger temperature, thermal stress.*

INTRODUCTION

In machinery, mechanisms and naval installations, mass flows of fluids circulating at different temperatures so that metal structures subject to dimensional changes. These changes are in volume:

$V = V_0 \cdot (1 + \beta \cdot \Delta t)$ (1), where:

- V_0 [m^3] - the volume at the reference temperature;
- V [m^3] - the volume at the heating operation;
- β [(^{0}C)$^{-1}$] - the volume expansion coefficient;
- Δt[0C] - the temperature difference between the operating temperature and the reference temperature.

Depending on the operating temperature, the metal structures can compress or expand. Geometric reports of steel structure allow to take into account, sometimes, only by one dimension - that is the direction for the study of thermal stress.

Result:

$L = L_0 \cdot (1 + \alpha \cdot \Delta t)$ (2),

- L_0 [m] - length at the reference temperature;
- L [m] - length at mounting temperature or the operation temperature and the reference temperature;
- α[^{0}C^{-1}] - the specific linear expansion coefficient of the material from which is made the metal structure;
- Δt - the temperature difference between the mounting temperature or the operation temperature and the reference temperature.

Equation (2) becomes:

$\Delta L = L_0 \cdot \alpha \cdot \Delta t$ (3) or:

$\frac{\Delta L}{L_0} = \alpha \cdot \Delta t$ (4)

$\sigma = \frac{\Delta L}{L_0} \cdot E \quad \left[\frac{kN}{m^2}\right]$ (5),

- σ $\left[\frac{kN}{m^2}\right]$ - tension or unit loading;
- E $\left[\frac{kN}{m^2}\right]$ - the module of longitudinal elasticity.

Temperature change will cause stress in a fixed pipe. When temperature is changed - stress introduced in a fixed pipe:

$\sigma = \alpha \cdot E \cdot \Delta t \quad \left[\frac{kN}{m^2}\right]$ (6) where:

- σ - $\left[\frac{kN}{m^2}\right]$ stress;
- α - [^{o}C^{-1}] - linear temperature expansion coefficient
- E - $\left[\frac{kN}{m^2}\right]$ - modulus of elasticity of the piping material;

- Δt - difference temperature change from installation temperature [^{o}C];
- $\Delta t = t_i - t_{ref}$;
- t_{ref} – reference temperature;

The stress must not exceed maximum allowable stress for the chosen piping material. Be aware that with frequently temperature changes - the stress cycle (with stress well below the maximum allowable limit) may fatigue the pipe.

Ship classification societies attach great importance of this issue. As an example, Det Norske Veritas among others, issued the general piping design for standardization like:

Stress calculation:

- When a thermal stress analysis of a piping system between two or more anchor points is carried out, the system shall be treated as a whole. The significance of all parts of the line, of restraints such as solid hangers, sway braces and guides and of intermediate restraints built in for the purpose of reducing loads on equipment or small branch lines, shall be duly considered. The stress analysis shall be carried out on the assumption

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that the piping system expands from 20°C to the highest operating temperature. The modulus of elasticity to be used for the pipe material, is the value of same at 20°C.

- In carrying out a thermal stress analysis, stress concentration factors found to exist in components other than straight pipes, shall be taken into account. In cases where it is known that such components possess extra flexibility, this may be incorporated in the stress calculations. Stress concentration factors and flexibility factors given, will be accepted for use in the calculations when other substantiated factors may be lacking.

- The thermal expansion resultant stress σ_r is defined as:

$$\sigma_r = \sqrt{\sigma_b^2 + 4\tau^2} \quad \left[\frac{N}{mm^2}\right] \qquad (7)$$

$$\sigma_b = \frac{\sqrt{(i_1 M_1)^2 + (i_0 M_0)^2}}{2} \quad \left[\frac{N}{mm^2}\right] \qquad (8)$$

, represent total bending stress;

$\tau = \frac{M_T}{2Z}$ $\left[\frac{N}{mm^2}\right]$, represented torsional stress,

M_T – torsion moment [Nm],

M_I – bending moment in plane of member [Nm];

M_0–bending moment traverse to plane of member [Nm];

i_I - stress concentration factor for in-plane bendingmoments;

i_0 - stress concentration factor for out-of-plane bending moments;

Z = section modulus in bending of member (mm^3).

When the member cross-section in non-uniform, the section modulus of the matching pipe shall be used. For branched systems, where the branch diameter is less than the header diameter, the branch section modulus may be taken as the smaller value of:

$$\pi r_b^2 t_h \text{ and } \pi r_b^2 i_{ib} t_b$$

r_b - mean cross-sectional radius of branch [mm]

t_h - thickness of pipe which matches header [mm]

t_b - thickness of pipe which matches branch [mm]

i_{ib} - in-plane stress concentration factor for branch.

The resultant stress σ_r is at no point of the piping system to exceed the corresponding stress range σ_{int}:

$$\sigma_{int} = 0.75\sigma_{tk} + 0.25\sigma_{tv} \qquad (9)$$

σ_{tk} - permissible pipe wall stress at 100°C or lower $\left[\frac{N}{mm^2}\right]$;

σ_{tv} - permissible pipe wall stress at maximum working temperature of system $\left[\frac{N}{mm^2}\right]$;

For low temperature piping σ_{int} shall be determined upon special consideration.

The sum of axial bending stress in the pipe wall due to static loading (pipe weight) and axial tensile stress
due to internal pressure, is at no point in the system to exceed the permissible stress σ_{tv}.

Table 1-Linear Temperature Expansion Coefficient [α]

Linear Temperature Expansion Coefficient [α]	[^{0} C^{-1}]
Brass	18,7 ·10^{-6}
Epoxy, cast resins & compounds, unfilled	(45-65) ·10^{-6}
Invar	1,5·10^{-6}
Nickel	13·10^{-6}
Steel	12·10^{-6}
Steel Stainless Austenitic (304)	17,3·10^{-6}
Steel Stainless Austenitic (310)	14,4·10^{-6}
Steel Stainless Austenitic (316)	16·10^{-6}
Steel Stainless Ferritic (410)	9,9·10^{-6}
Carbon steel pipes	11,6·10^{-6}
Austenitic steel pipes	13,5·10^{-6}

Table 2- Tensile Modulus

Tensile Modulus [E]	[kN/m^2]
Brass	1,02 ·10^8
Copper	1,17 ·10^8
Epoxy, cast resins & compounds, unfilled	(0,02-0,03) ·10^8
Nickel	1,7 · 10^8
Steel, stainless ANSI 302	1,8 ·10^8
Steel, Structural ASTM-A36	2 ·10^8

Table 3 - International standards for ambient reference conditions

25 - charge air coolant temperature;*

*10** - cooling water temperature (minimum for lubricating oil cooler).*

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A. AXIAL FORCE THAT EMERGED FROM TEMPERATURE DIFFERENCE

Axial force developed into a pipe due to thermal expansion or thermal compression:

$$F = \sigma_t \cdot E \cdot \Delta t \cdot S_u \quad (10),$$

S_u – surface of the pipe section [m^2];

$$S_u = \frac{\pi}{4}(d_e^2 - d_i^2) \quad [\text{m}^2], \text{ where:}$$

d_e - the outer diameter of the pipe [m];
d_i – the inner diameter of the pipe [m];

$$s = \frac{d_e - d_i}{2} \quad [\text{m}] \quad (11), \text{ the pipe thickness;}$$

$$d_m = \frac{d_i - d_e}{2} \quad [\text{m}] \quad (12), \text{ the average diameter of the piupe.}$$

Table 4 are determined the forces at different values of pipe heating. It is noted that, although the temperature differences are small, flat expansion pipes are required to avoid altering the geometry of the metal structure.

Table 4 - The axial force developed in the pipe by the thermal expansion.

E [kN/m^2]	2·10^8	2·10^8	2·10^8	2·10^8	2·10^8	2·10^8
[α] [(^{0}C)$^{-1}$]	12·10^{-6}	12·10^{-6}	12·10^{-6}	12·10^{-6}	12·10^{-6}	12·10^{-6}
t_0 [^{0}C] ISO	25	25	25	25	25	25
t_i [^{0}C]	30	35	40	45	50	55
dt [^{0}C]	5	10	15	20	25	30
[σ] [kN/m^2]	120·10^2	240·10^2	360·10^2	480·10^2	600·10^2	720·10^2
[σ] [bar]	120	240	360	480	600	720
Su [m^2]	10^{-2}	10^{-2}	10^{-2}	10^{-2}	10^{-2}	10^{-2}
F []kN]	120	240	360	480	600	720
F [t_f]	12,18	24,36	36,55	48,73	61	73

Obs. Pipe d_m=250 [mm]; s=12.73 [mm]; S_u=10^{-2}[m^2]

B. TIGHTENING OF THE MAIN SCREWS

Main screws are considered: tie-rods for assembling the heat engine, engine mounting bolts on the base or supporting structure of the ship, connecting rod big end bolts to assemble the head and the cylinder head bolts.
Determine the clamping force:

$$F_{st} = (2 \div 3) F_{dim} \quad [\text{kN}] \quad (13),$$

F_{dim} [kN] –sizing force.
The clamping force can be obtained by the following methods:

International standards for ambient reference conditions	t_{air} [^{0} C]	p [bar]	Humidiy [%]	t_{sw} [^{0}C]
STP - Standard Temperature and Pressure IUPAC (International Union of Pure and Applied Chemistry)	0	1		
STP - Standard Temperature and Pressure USA	15,6	1,0132		
NTP - Normal Temperature and Pressure	20	1,0132		
SATP - Standard Ambient Temperature and Pressure	25	1,0132		
ISA - International Standard Atmosphere	15	1,0132	0	
ICAO Standard Atmosphere	15	1,0132		
ISO 3046-1:2002(E);ISO 15550:2002(E)	25	1	30	25*
IACS M28(1978):Tropical ambient reference conditions	45	1	60	32
Winter ambient reference conditions	10	1	60	10**

- hydraulic devices for bolt elongation<
- elongation of the screw by turning the nut with torque wrench;
- bolt elongation with star key with beating;
- bolt elongation by thermal expansion, electrical resistance being mounted in the main bolt .

For check assembly- tightening of main screw , the following methods are used:
- measuring the pressure in the hydraulic device;
- bolt elongation measurement;
- measuring the angle of rotation of the nut;
- measuring the temperature of the heating screw thereof.

$$F_{st} = \sigma_{st} \cdot A_s \quad [\text{kN}] \quad (14);$$

$$\sigma_{st} = \xi_{st} \cdot E \quad \left[\frac{N}{mm^2}\right] \quad (15), \text{ where:}$$

$$\xi_{st} = \frac{\Delta L_{st}}{L_0} \quad [-] \quad \text{- the relative elongation of the bolt;}$$

ΔL_{st} [mm] - the actual elongation of the bolt;
L_0 [mm] - length of the screw at the reference temperature;

$$F_{st} = \frac{\Delta L_{st}}{L_0} \cdot E \cdot A_s \quad [\text{kN}] \quad (16);$$

$$\sigma_{st} = \frac{\Delta L_{st}}{L_0} \cdot E \quad \left[\frac{N}{mm^2}\right] \quad (17)$$

$$\sigma_{st} = \alpha \cdot E \cdot \Delta t \quad \left[\frac{N}{mm^2}\right]; \quad (18),$$

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$\Delta t = \frac{1}{\alpha} \cdot \frac{\Delta L_{st}}{L}$, [^{0}C] (17), determined in Table 5.

Table 5- Tightening the main screws, by thermal expansion.

	Studs fixing pedestal	Connecting rod bolt head	Cylinder head bolt	Coupling bar
L [mm]	500	1500	1200	9000
dl [mm]	0,25-0,35	0,65-1,05	0,6-0,84	4,5-6,3
dl/L	$(5\text{-}7)\cdot 10^{-4}$	$(4{,}3\text{-}7)\cdot 10^{-4}$	$(5\text{-}7)\cdot 10^{-4}$	$(5\text{-}7)\cdot 10^{-4}$
E [kN/m^2]	$2\cdot 10^8$	$2\cdot 10^8$	$2\cdot 10^8$	$2\cdot 10^8$
[α] [(^{0}C)$^{-1}$]	$13\cdot 10^{-6}$	$13\cdot 10^{-6}$	$13\cdot 10^{-6}$	$13\cdot 10^{-6}$
dt [^{0}C]	39-54	33-54	39-54	39-54
t_0 [^{0}C] ISO	25	25	25	25
t_i [^{0}C]	64-79	58-79	64-79	64-79

t_i [^{0}C]- Heating temperature; $t_0 = t_{ref}$ [^{0}C]=25^{0}C
ISO 3046-1:2002(E);ISO 15550:2002(E)

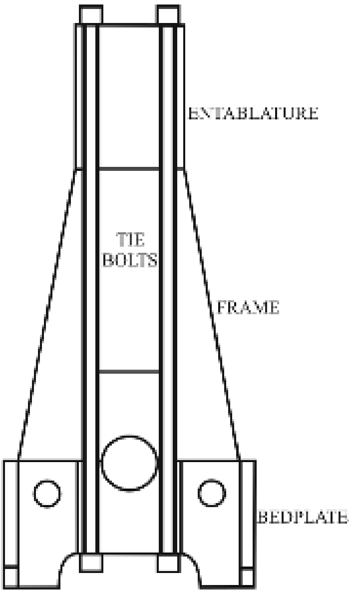

Figure1a – Coupling Bars [2.pg. 42]

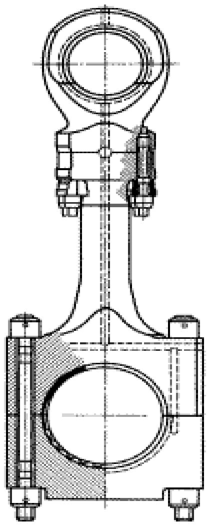

Figure 1b – Join pin for crank end of connecting-rod
[2, pg. 84]

Figure 1c- Cylinder head bolts [3]

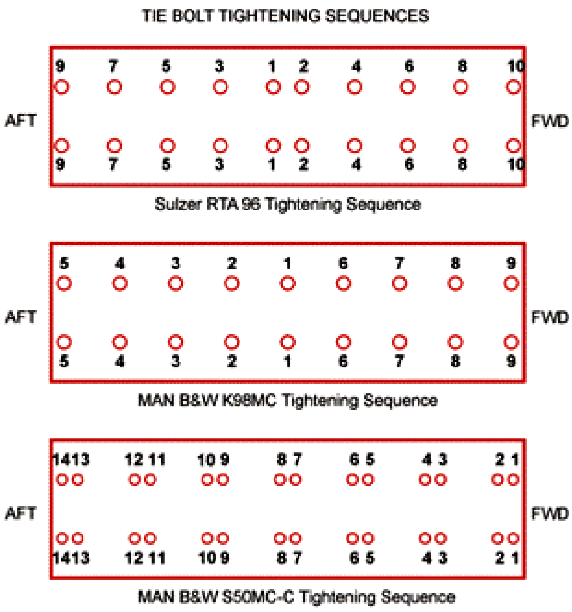

Figure 2 – The bolt tightening sequences [2, pg.44]

C. FIXING THE MAIN ENGINE ON THE PEDESTAL

At the coupling of the engine with shaft line it is mounts thrust bearing, which requires that the first

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studs in the aft, which fix the engine on the pedestal, be calibrated on the holes in the engine flange and in the holes in the pedestal, and the holes from the forward of engine not be clearance so as not to allow the engine, expansion, In view of the temperatures of the motor flange, respectively the flange of the pedestal, in Table 6 are determined clearances required for assembly, to avoid blocking the engine longitudinal expansion and its cracking in the area of engine frame and block cylinders.

Table 6 - Dilation of the engine compared to the base structure of the ship

$[\alpha_{FM}]$ $[(^0C)^{-1}]$	$13\cdot10^{-6}$	$13\cdot10^{-6}$	$13\cdot10^{-6}$	$13\cdot10^{-6}$	$13\cdot10^{-6}$
$[\alpha_{FP}]$ $[(^0C)^{-1}]$	$12\cdot10^{-6}$	$12\cdot10^{-6}$	$12\cdot10^{-6}$	$12\cdot10^{-6}$	$12\cdot10^{-6}$
L_0 [mm]	10^4	10^4	10^4	10^4	10^4
t_0 [^{0}C] ISO	25	25	25	25	25
t_{FM} [^{0}C]	10	20	25	30	35
Δt_{FM} [^{0}C]	-15	-5	0	+5	+10
t_{FP} [^{0}C]	10	20	20	20	20
Δt_{FP} [^{0}C]	-15	-5	-5	-5	-5
ΔL_{FM} [mm]	-1,95	-0,65	0	+0,65	+1,3
ΔL_{FP} [mm]	-1,80	-0,6	-0,6	-0,6	-0,6
$\sum\Delta L$ [mm]	+0,15	+0,05	-0,6	-1,25	-1,9

D. STEAM PIPING EXPANSION

Table 7 - Saturated Steam Piping and Superheated Steam Piping

t_0 [^{0}C] ISO	25	25	25	25	25	25	25
L_0 [mm]	10^4	10^4	10^4	10^4	10^4	10^4	10^4
$[\alpha_1]$ $[(^0C)^{-1}]$	$11,6\cdot10^{-6}$	$11,6\cdot10^{-6}$	$11,6\cdot10^{-6}$	$11,6\cdot10^{-6}$	$11,6\cdot10^{-6}$	$11,6\cdot10^{-6}$	$11,6\cdot10^{-6}$
p [bar]	5	7	10	18	20	50	100
t_{VAP} [^{0}C]	150	163	178	206	212	266	316
t_{SI} [^{0}C]	250	250	275	300	325	350	400
Δt_{VAP} [^{0}C]	125	138	153	181	187	241	291
ΔL_{VAP}[mm]	14,50	16,00	17,70	21,00	22,00	28,00	33,08
Δt_{SI} [^{0}C]	175	225	250	275	300	325	375
ΔL_{SI}[mm]	20,30	26,10	29,00	32,00	34,80	37,70	43,60

E. The Gas Piping Expansion, To Internal Combustion Engines

Table 8 - The gas piping of internal combustion engines

t_0 [^{0}C] ISO	25	25	25	25	25	25
t_{EV} [^{0}C]	350	400	450	500	550	600
$[\alpha_2]$ $[(^0C)^{-1}]$	$13,5\cdot10^{-6}$	$13,5\cdot10^{-6}$	$13,5\cdot10^{-6}$	$13,5\cdot10^{-6}$	$13,5\cdot10^{-6}$	$13,5\cdot10^{-6}$
Δt [^{0}C]	325	375	425	475	525	575
L_0 [mm]	10^4	10^4	10^4	10^4	10^4	10^4
ΔL[mm]	43,88	50,63	57,38	64,13	71,00	77,63

F. THE EXPANSION OF GAS PIPING FROM STEAM BOILERS

Table 9 - The gas piping - burner steam boiler

t_0 [^{0}C] ISO	25	25	25	25	25	25
t_{EV} [^{0}C]	125	150	175	200	225	250
Δt [^{0}C]	100	125	150	175	200	225
L_0 [mm]	10^4	10^4	10^4	10^4	10^4	10^4
ΔL [mm]	13,50	16.88	20,25	24,00	27,00	30,38

G. THE EXPANSION OF GAS PIPING FROM GAS TURBINES

Table 10 - The gas piping from gas turbines

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t_0 [^{0}C] ISO	25	25	25	25	25	25
t_{EV} [^{0}C]	525	550	600	625	700	750
Δt [^{0}C]	500	525	575	600	675	725
L_0 [mm]	10^4	10^4	10^4	10^4	10^4	10^4
ΔL [mm]	67,50	71,00	77,63	81,00	91,13	98,0

Observation – Compensating pipe shown in figure 3a,b,c.

a)
Compensating pipe for flue gas exhaust ducts [4]

a) Compensating pipe for The gas piping - burner steam boiler [4]

b) *axial compensating pipe for linear expansion* [5]

Figure 3 – Dilatation compensating pipe

CONCLUSIONS

Particular attention should be given to fixing metal structures to enable the processing of dimensional changes. In design and installation must be determined sense of expansion to maintain safety in operation. In operational the procedures to preheating, startup and keep the operating temperatures must be respected. Attention should be given to the heating main bolts fortheir dismanting!.

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