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ABOUT THE STUDY OF REQUESTS BUCKLING VERIFICATION OF THE ELEMENTS OF POWER PLANT PROPULSION

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***Abstract**: On study based of requests from the power propulsion plant, the authors develop in this paper a study of buckling heat engine piston rod at the head of the cross, the strut rod and shaft intermediaries for power propulsion plant.*

***Key words**: buckling, strut rod, propulsion power plant*

Introduction

For parts-metallic structures of naval propulsion plants, compression can be applied to situations of exploitation when axial forces grow, the issue of verification of buckling.

This consists of:

a. determining the critical buckling force
b. determining the critical buckling pressure
c. determination the coefficient of safety buckling
d. depending on the results obtained can take precautions to avoid operational achieving critical force buckling
e. resizing piece to increase the safety coefficient buckling

Buckling phenomenon is dangerous because it is irreversible.

Critical buckling tension may fall to limit proportionalities (elasticity) of the material from which it is made corresponding to the piece, is called elastic buckling.

If the tension is above the critical elastic buckling, buckling is called the elastic-plastic.

Either: F_{crf} [kN] - critical buckling force
F_{ef} [kN] - effective buckling force

The safety coefficient of buckling is determined with:

$$c_f = \frac{F_{crf}}{F_{ef}} \quad [1]$$

with the condition like $c_f > 1$

Depending on the functional importance of the piece of naval propulsion power plant, buckling safety coefficient will be accepted in a particular reference domain.

Critical buckling force

$$F_{crf} = \frac{\pi^2 \cdot E \cdot I_{min}}{L_f^2} \text{ [KN]} \quad [2]$$

E [kN/m²] - elastic modulus specific for the material it is made subjected to compressive load the piece. For the manufacture of the materials used in naval propulsion power plant, determination of the elastic modulus should be performed in the material strength laboratories recognized by classification societies and for materials recognize by them.

I_{min}[m⁴] - moment of inertia calculated for minimal section of the piece put under compression. This is calculated based on the geometric size and shape of the respective section.

L_f^2 [m²] - buckling length is determined depending on the length and attachment, assembling piece in the system.

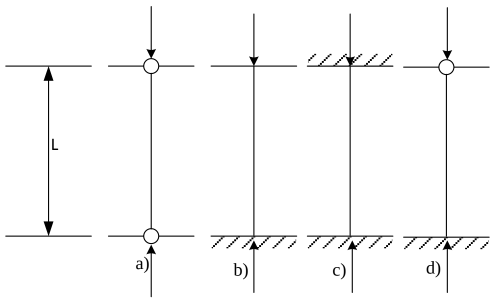

Fig. 1. Modes of assembling the piece

L – length of piece, a) double articulated bar, b) embedded bar,
c) double embedded bar, d) embedded and articulated bar

For this cases results:

a) $L_f = L$
b) $L_f = 2xL$
c) $L_f = L/2$
d) $L_f = 0.707xL$

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For fixing arrangement are taken into account two perpendicular planes of section fixing joints, otherwise for articulated plan L_f = L, but for the normally plan be considered embedded, so L_f = L/2 what will lead to the determination of two critical forces buckling.

$F_{crf\ (PO)}$ [kN] - critical buckling force for articulated plan – called oscillation plan

$F_{crf\ (PI)}$ [kN] - critical buckling force for normal plan on oscillation plan – called embedded plan

To equalize the two critical buckling forces, proceed to mass redistribution of minimum calculation section so:

$$I_{min(PO)} = 4 \times I_{min\ (PI)} \qquad [3]$$

By equalizing critical safety buckling forces is obtained equalization of critical safety buckling coefficients for both plans assembling piece

B.1.Checking the buckling of piston rod

D [m]	0.5	0.72
S_p[m^2]	0.196	0.407
d_{TP}[m]	0.15	0.216
S [m]	2	3
L_{TP} [m]	3	4.5
E [kN/m^2]	2.1×10^8	2.1×10^8
I_{min} [m^4]	2.48×10^{-5}	1.068×10^{-4}
p_{maxaid} [bar]	135	140
p_{maxaid} [kN/m^2]	135×10^2	140×10^2
p_{maxdss} [bar]	150	155
p_{maxdss} [kN/m^2]	150×10^2	155×10^2
F_{pmax} [kN]	29.4×10^2	63×10^2
F_{crf} [kN]	2.28×10^4	4.373×10^4
c_f[-]	7.75	6.941

B.2. Checking the buckling of piston rod

D [m]	0.84	1.08
S_p[m^2]	0.554	0.855
d_{TP}[m]	0.25	0.32
S [m]	3.2	2.8
L_{TP} [m]	5	5
E [kN/m^2]	2.1×10^8	2.1×10^8
I_{min} [m^4]	1.91×10^{-4}	5.044×10^{-4}
p_{maxaid} [bar]	140	140
p_{maxaid} [kN/m^2]	140×10^2	140×10^2
p_{maxdss} [bar]	155	155
p_{maxdss} [kN/m^2]	155×10^2	155×10^2
F_{pmax} [kN]	85.87×10^2	132×10^2
F_{crf} [kN]	6.33×10^4	16.73×10^4
c_f[-]	7.38	12.6

D [m] - cylinder diameter; S_p [m^2] – piston area; d_{TP} [m] – rod of piston diameter; S [m] - piston stroke; L_{TP} [m] – rod of piston length; I_{min}[m^4] – inertia moment; p_{maxard} [kN/m^2] –maximum combustion pressure; p_{maxdss} [kN/m^2] – maximum pressure for safety valve; F_{pmax}[kN] – maximum forte of pressure; σ_{efmax} [kN/m^2] - maximum effective tension; F_{crf} [kN] – critical force buckling; σ_{crf} [kN/m^2] - critical tension buckling; c_f [-] - buckling safety factor;

$$S_p = \frac{\pi \cdot D^2}{4}\ [m^2]; \qquad [4]$$

$$I_{min} = \frac{\pi \cdot d_{TP}^2}{64}\ [m^4]; \qquad [5]$$

$$F_{pmax} = p_{maxdss} \cdot \frac{\pi \cdot D^2}{4}\ [kN]; \qquad [6]$$

$$\sigma_{efmax} = \frac{F_{pmax}}{\frac{\pi \cdot d_{TP}^2}{4}} [\frac{kN}{m^2}]; \qquad [7]$$

$$F_{crf} = \frac{\pi^2 \cdot E \cdot I_{min}}{\left(\frac{L_{TP}}{2}\right)^2}\ [kN] \qquad [8]$$

$$\sigma_{crf} = \frac{F_{crf}}{\frac{\pi \cdot d_{TP}^2}{4}} [\frac{kN}{m^2}]; \qquad [9]$$

$$c_f = \frac{F_{crf}}{F_{pmax}} = \frac{\sigma_{crf}}{\sigma_{efmax}}; \qquad [10]$$

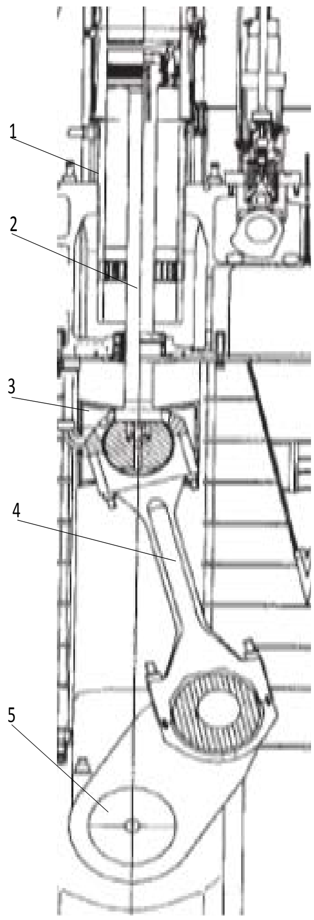

Fig.2 piston rod and connecting rod for MAN S46 [B6]

1 – piston, 2 – piston rod,
3 - cross head, 4 – connecting rod,
5 - crankshaft

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C.1. Checking the buckling of connectingstrut rod

D [m]	0.5	0.72
S [m]	2	3
L_{TB} [m]	2.63	3.95
$F_{pmaxdss}$ [kN]	$29.4x10^2$	$63x10^2$
E [kN/m^2]	$2.1x10^8$	$2.1x10^8$
d_{TB}[m]	0.18	0.22
I_{min} [m^4]	$5.15x10^{-5}$	$1.27x10^{-4}$
$F_{crf\ PO}$ [kN]	$15.43x10^3$	$1.688x10^4$
$c_{f\ PO}$ [-]	5.25	2.679
$F_{crf\ PI}$ [kN]	$6172x10^3$	$6.72x10^4$
$c_{f\ PI}$ [-]	21	10.716

C.2. Checking the buckling of connecting strut rod

D [m]	0.84	1.08
S [m]	3.2	2.8
L_{TB} [m]	4.21	4
$F_{pmaxdss}$ [kN]	$85.87x10^2$	$132x10^2$
E [kN/m^2]	$2.1x10^8$	$2.1x10^8$
d_{TB}[m]	0.28	0.35
I_{min} [m^4]	$3.01x10^{-4}$	$7.37x10^{-4}$
$F_{crf\ PO}$ [kN]	$3.52x10^4$	$9.55x10^4$
$c_{f\ PO}$ [-]	4.099	7.23
$F_{crf\ PI}$ [kN]	$14.08x10^4$	$38.2x10^4$
$c_{f\ PI}$ [-]	16.39	28.92

L_{TB} [m] – connecting rod length; $\lambda = 0.25 \div 0.38$; $\lambda = \frac{S}{2 \cdot L_{TB}}$; d_{TB} – connecting rod diameter;
$F_{crf\ (PO)}$ [kN] - critical buckling force for connecting rod oscillation plan;
$F_{crf\ (PI)}$ [kN] - critical buckling force for connecting rod embedded plan;
$\sigma_{crf(PO)}$ [kN/m^2] - critical tension buckling for connecting rod oscillation plan;
$\sigma_{crf(PI)}$ [kN/m^2] - critical tension buckling for connecting rod embedded plan;
$c_{f(PO)}$ [-] - buckling safety factor for connecting rod oscillation plan;
$c_{f(PI)}$ [-] - buckling safety factor for connecting rod embedded plan;

$$F_{crf(PO)} = \frac{\pi^2 \cdot E \cdot I_{min}}{L_{TB}^2}\ [kN]; \quad [11]$$

$$F_{crf(PI)} = \frac{\pi^2 \cdot E \cdot I_{min}}{\left(\frac{L_{TB}}{2}\right)^2}\ [kN]; \quad [12]$$

$$I_{min} = \frac{\pi \cdot d_{TB}^2}{64}\ [m^4]; \quad [13]$$

$$\sigma_{efmax} = \frac{F_{pmax}}{\frac{\pi \cdot d_{TB}^2}{4}} [\frac{kN}{m^2}]; \quad [14]$$

$$\sigma_{crf(PO)} = \frac{F_{crf(PO)}}{\frac{\pi \cdot d_{TB}^2}{4}} [\frac{kN}{m^2}]; \quad [15]$$

$$\sigma_{crf(PI)} = \frac{F_{crf(PI)}}{\frac{\pi \cdot d_{TB}^2}{4}} [\frac{kN}{m^2}]; \quad [16]$$

$$c_{f(PO)} = \frac{F_{crf(PO)}}{F_{pmax}} = \frac{\sigma_{crf(PO)}}{\sigma_{efmax}}; \quad [17]$$

$$c_{f(PI)} = \frac{F_{crf(PI)}}{F_{pmax}} = \frac{\sigma_{crf(PI)}}{\sigma_{efmax}}; \quad [18]$$

Material limitations [B4]

Where shafts may experience vibratory stresses close to the permissible stresses for transient operation, the materials are to have a specified minimum ultimate tensile strength (σ_B) of 500 N/mm^2. Otherwise materials having a specified minimum ultimate tensile strength () of 400 N/mm^2 may be used.

For use in the following formulae in this UR, σ_B is limited as follows:

- For carbon and carbon manganese steels, a minimum specified tensile strength not exceeding 600 N/mm^2 for use in M68.5 and not exceeding 760 N/mm^2 in M68.4.
- For alloy steels, a minimum specified tensile strength not exceeding 800 N/mm^2.
- For propeller shafts in general a minimum specified tensile strength not exceeding 600 N/mm^2 (for carbon, carbon manganese and alloy steels).

Where materials with greater specified or actual tensile strengths than the limitations given above are used, reduced shaft dimensions or higher permissible vibration stresses are not acceptable when derived from the formulae in this UR.

Shaft diameters [B4]

Shaft diameters are not to be less than that determined from the following formula:

$$d = F \cdot k \cdot \sqrt[3]{\frac{p}{n_0} \cdot \frac{1}{1 - \frac{d_i^4}{d_0^4}} \cdot \frac{560}{\sigma_B + 160}}\ [mm] \quad [19]$$

where:
d = minimum required diameter in [mm]
d_i = actual diameter in [mm] of shaft bore
d_0 = outside diameter in [mm] of shaft. If the bore of the shaft is $\leq 0.40 x d_0$, the expression
$1 - \frac{d_i^4}{d_0^4}$ may be taken as 1.0
F = factor for type of propulsion installation
F_1= 95 for intermediate shafts in turbine installation, diesel installations with hydraulic (slip type) couplings, electric propulsion installations
F_2= 100 for all other diesel installations and all propeller shafts
k = factor for the particular shaft design features, see M68.6
n_0 = speed in revolutions per minute of shaft at rated power
p = rated power in kW transmitted through the shaft (losses in gearboxes and bearings are to be disregarded)
σ_B = specified minimum tensile strength in N/mm$_2$ of the shaft material, see M68.3

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The diameter of the propeller shaft located forward of the inboard stern tube seal may be gradually reduced to the corresponding diameter required for the intermediate shaft using the minimum specified tensile strength of the propeller shaft in the formula and recognizing any limitations given in M68.3. [4]

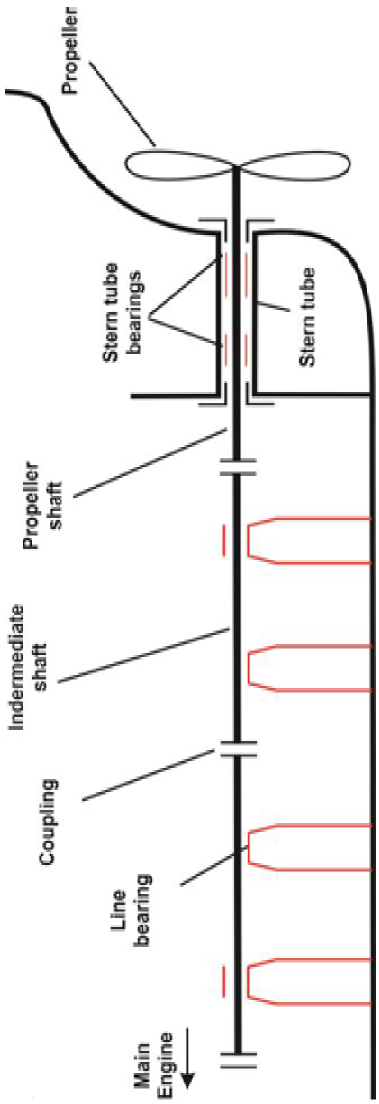

Fig. 3 The arrangement of ship shafting system [B5]

E.1.Checking of buckling intermediary shaft

	DE 4T	DE 2T
P_e [kW]	10000	20000
n [rpm]	450	90
n_0 [rpm]	90	90
F	100	100
σ_B[N/mm^2]	650	650
K	1	1
d [mm]	425	535
d [m]	0.425	0.535
L [m]	8.5	10.7
M_m [kNm]	1061	2122
W_{n1} [Nd]	20	20
W_{n1} [m/s]	10.29	10.29
F_{A1} [kN]	972	1944
F_{crf} [kN]	$1.84x10^5$	$2.91x10^5$
c_{f1} [-]	189	150
W_{n2} [m/s]	2	2
F_{A2} [kN]	5000	9487
c_{f2} [-]	37	31
W_{n3} [m/s]	0.5144	0.5144
F_{A3} [kN]	$1.944x10^4$	$3.69x10^4$
c_{f3} [-]	9.46	7.88

E.2. Checking of buckling intermediary shaft

	GT	ST
P_e [kW]	30000	60000
n [rpm]	5000	3000
n_0 [rpm]	100	100
F	95	95
σ_B[N/mm^2]	650	650
K	1	1
d [mm]	592	746
d [m]	0.592	0.746
L [m]	11.84	14.92
M_m [kNm]	2865	5730
W_{n1} [Nd]	20	20
W_{n1} [m/s]	10.29	10.29
F_{A1} [kN]	2916	5831
F_{crf} [kN]	$3.56x10^5$	$5.68x10^5$
c_{f1} [-]	122	96.55
W_{n2} [m/s]	2	2
F_{A2} [kN]	15002	30000
c_{f2} [-]	23	19
W_{n3} [m/s]	0.5144	0.5144
F_{A3} [kN]	$5.832x10^4$	$1.166\ x10^5$
c_{f3} [-]	6.10	4.867

P_e [kW] – effective power; n [rpm] –RPM of thermal machinery; n_0 [rpm] – RPM of intermediary shaft; F - factor for type of propulsion installation; σ_B [N/mm2] - specified minimum tensile strength in [N/mm^2] of the shaft material; K - factor for the particular shaft design features; d [m] – designed diameter of intermediary shaft;

$d = F \cdot K \cdot \left[\frac{P_e}{n_0} \cdot \frac{560}{\sigma_B + 160}\right]^{\frac{1}{3}} \cdot 10^{-3}\ [m]$; [20]

L [m] – length of intermediary shaft;

Mm [kNm] – torque;

$P = M \cdot \frac{\pi \cdot n_0}{30}\ \ [kW]$; [21]

W_n [Nd] – ship speed; F_{A1} [kN] – axial force in shaft;

$P = F_{A1} \cdot W_{N1}\,[kW]$; [22]

F_{crf} [kN] – critical force buckling;

$F_{crf} = \frac{\pi^2 \cdot E \cdot I_{min}}{\left(\frac{L}{2}\right)^2}\ \ [kN]$; [23]

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$I_{min} = \frac{\pi \cdot d^4}{64}$ $[m^4]$; [24]

c_{f1} [-] – buckling safety factor; W_{n2} [m/s], W_{n3} [m/s] – maneuvering speed vessel; F_{A2} [kN], F_{A3} [kN] – maneuvering axial force in shaft; c_{f2} [-], c_{f3} [-] – maneuvering buckling safety factor;

CONCLUSIONS

1. Rods pistons and connecting rods are cylindrical bar;
2. The diameter of the piston rod of B and the diameter of connecting rod of C may be increased by 15÷30%;
3. In the calculation of buckling hasn't were taken into account channels for oil circulation of piston rod used for piston head cooling; respective circulation channels of connecting rod used for lubricating oil from the camps crosshead to camp crankpin.
4. To avoid increased pressure engine cylinder must be adjusted injection system in accordance with procedures.
5. Periodic the safety valve mounted on the cylinder head must be checking and regulation;
6. Engine start be made with open purges;
7. Intermediate shafts are supported by radial bearings - buckling calculation must consider the distance between bearings;
8. Shaft calculation must be completed with torsional and axial vibration testing for propulsion power plant.

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