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# A superelement for analyzing the coupling between crankshaft torsional and axial vibration

SUMMARY

*Description of the coupling phenomenon of torsional and axial vibration appearing in the ship engine crankshaft is presented. The applied computational model suitable for analyzing the coupling phenomenon consists of special superelements which couple the vibrations, rigid finite elements and deformable beam elements.*

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The models in which no coupling phenomenon between torsional and axial vibration was accounted for, have been hitherto used for dynamic analysis of ship propulsion systems. Comparison of the calculation results of such systems with the measurement results clearly demonstrates that a strong relationship exists between torsional and axial vibration [4,5,6,7]. The closer natural frequencies of the vibrations the more pronounced phenomenon of their coupling; the investigations reveal a much stronger effect of the torsional vibration on axial vibration than in reverse (Fig.1).

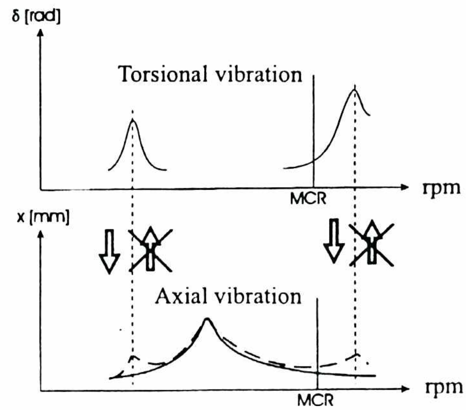


*Fig.1. Scheme of mutual interaction of torsional and axial vibration*
*—— calculated*
*- - - - measured*

The performed investigations demonstrate that the much stronger influence of the torsional vibration on axial vibration is caused by :

- much greater values of the dynamic tangent forces acting on cranks in respect to the propeller-induced axial forces
- a specific spatial configuration of the crank where the torsional moments induce the axial deformations and the axial forces do not cause any torsional deformations (Fig.2).

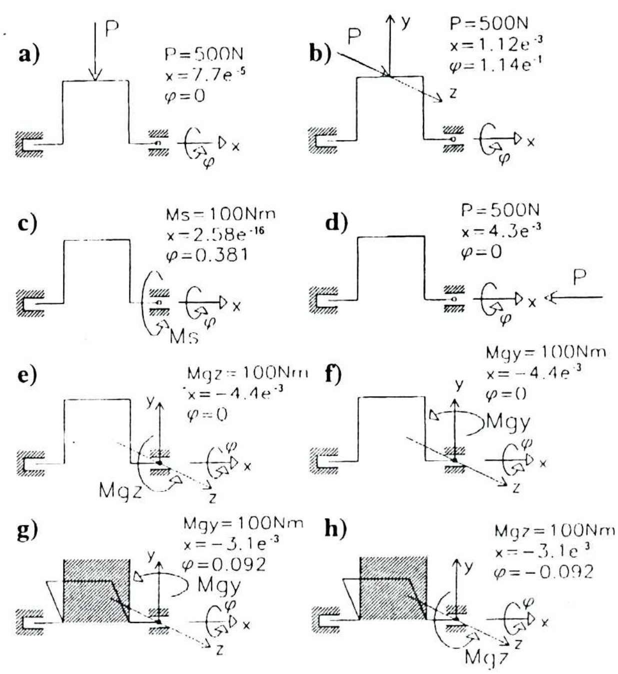


*Fig.2. Deformations of a single crank under different loading (displacement x - [m], angle φ - [rad])*

The combustion engine crankshaft is one of the elements which cause coupling the torsional and axial vibration . The coupling effect is especially pronounced in the modern propulsion systems with two-stroke diesel engines in which crankshaft geometry was changed due to increasing the piston stroke and maximum cylinder pressure. In result the mass and inertia moments of the crankshaft were increased, but its stiffness lowered simultaneously. The investigation results published in [7] prove that the torsional - axial vibration coupling depends on both above mentioned features of crankshafts.

It can be observed from the systems e) to h) shown in Fig.2 that the reaction forces acting on the main journals must be directed outside the crank plane to cause coupling the torsional and axial deformations of the crank. It can happen only when the vectors of the bending moments $M_{gy}$ and $M_{gz}$ are directed outside the crank plane. On the basis of the observations it can be stated that a correct interpretation of the phenomenon must be based on determination of torsional--axial - bending deformations of at least two neighbouring cranks mutually displaced by an angle, as coupling the torsional and axial vibration results from the bending moments action.

An analysis of the mechanical system which consists of two cranks mutually displaced by the angle 2δ, and is excited by the torsional moment M and axial force F acting in the external main bearings (Fig.3) makes it possible to determine the open form of the coupling coefficient of the torsional and axial vibrations as follows [4] :

$$C_{14} = \frac{C_{21} \cdot C_{34} \cdot \cos\delta \cdot \sin\delta}{C_{33} \cdot \sin^2\delta + C_{22} \cdot \cos^2\delta} \qquad (1)$$

where :

$C_{ik}$ - flexibility coefficient of the crank, defined as its displacement in the direction „i" due to the unit force acting in the direction „k".

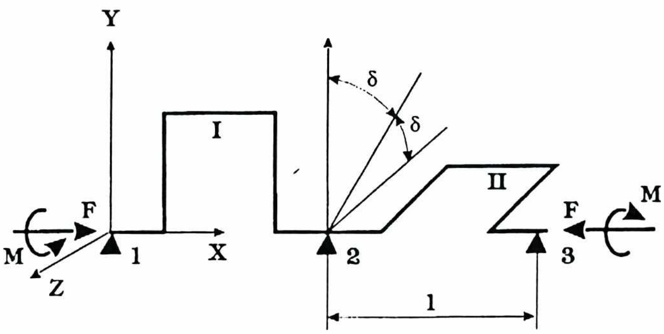


*Fig.3. System of two cranks under the axial force F and torsional moment M*

In the investigations carried out by the Chair of Mechanics and Strength of Materials, Technical University of Gdańsk, the calculation model of the crankshaft was used, which consisted of the superelements (SE) formed on the basis of the deformable beam elements, and of the rigid finite elements (Fig.4).

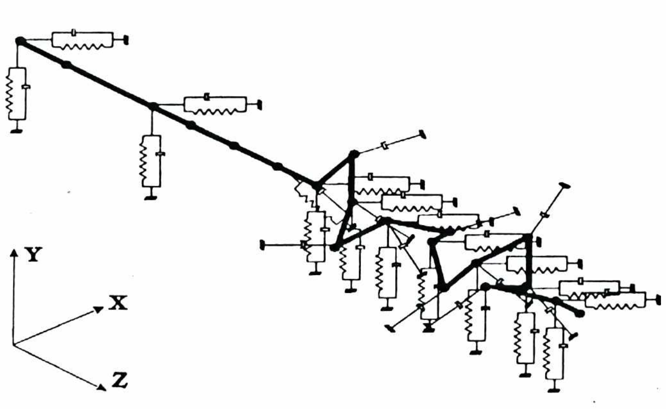


*Fig.4. Simplified calculation model of the ship propulsion system in which the crankshaft is idealized by the superelements which couple the torsional and axial vibration*

The SE of 12 degrees of freedom (Fig.5), especially elaborated for such investigations, models a single crank.

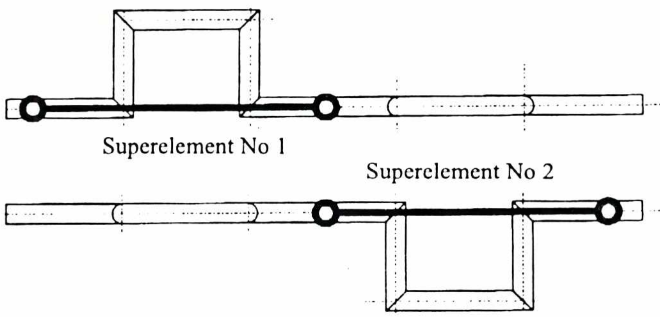


*Fig.5. A crankshaft fragment modelled by two superelements of 12 d.o.f. each*

The coupling of the torsional and axial vibration is just accounted for by introducing the additional coefficient $C_{14}$ to the superelement flexibility matrix **C** (Fig.6) :

$$\mathbf{C} = \mathbf{K}_{sz}^{-1} = \begin{bmatrix} C_{11} & C_{12} & C_{13} & C_{14} & C_{15} & C_{16} \\ & C_{22} & C_{23} & C_{24} & C_{25} & C_{26} \\ & & C_{33} & C_{34} & C_{35} & C_{36} \\ & & & C_{44} & C_{45} & C_{46} \\ & & & & C_{55} & C_{56} \\ \text{sym.} & & & & & C_{66} \end{bmatrix} \qquad (2)$$

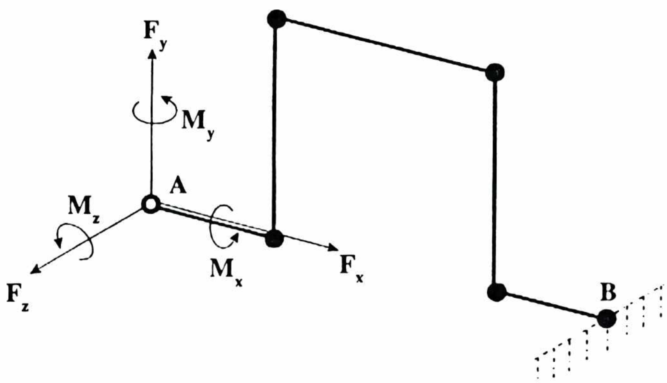


*Fig.6. Model of the one-end-fixed crank used to determine the crank flexibility coefficients*

The SE dynamic flexibility matrix formed on the basis of the global dynamic stiffness matrix of the crank sub-system, can be presented in the following form [9] :

$$\overline{\mathbf{B}}_{AA}(s) = \mathbf{B}_{AA}(s) - \mathbf{B}_{AB}(s) \cdot \mathbf{B}_{BB}^{-1}(s) \cdot \mathbf{B}_{BA}(s)$$
$$s = j \cdot \omega \qquad j = \sqrt{-1} \qquad (3)$$

where :

$\mathbf{B} = (K + j\omega C - M\omega^2)$ - global dynamic stiffness matrix of the crank

$\mathbf{B}_{mn}$ - a relevant block of the global dynamic stiffness matrix of the crank

$\omega$ - forcing load frequency.

In the applied calculation model the excitation exerted by the engine in the form of the tangent and radial forces acting on the cranks, is determined on the basis of the indicator diagram of cylinder pressure changes with the account taken of inertia forces induced by masses in translational and rotational motion. Reducing the forces to the external nodes is performed in compliance with the following relationship [9] :

$$\bar{\mathbf{h}}_A(s) = \mathbf{h}_A(s) - \mathbf{B}_{AB}(s) \cdot \mathbf{B}^{-1}_{BB}(s) \cdot \mathbf{h}_B(s) \tag{4}$$

By making use of the notation applied in (3) and (4) the global dynamic stiffness matrix of the crankshaft can be described as follows :

$$\mathbf{B}_w = diag(\mathbf{B}_1, \mathbf{B}_2, \mathbf{B}_3 \dots \mathbf{B}_n) \tag{5}$$

where :

n - number of cranks

$\mathbf{B}_i = \bar{\mathbf{B}}_{AA\,i} + \bar{\mathbf{B}}_{sprzegi}$ , and

$\bar{\mathbf{B}}_{AA\,i}$ - i-th superelement dynamic stiffness matrix

$\bar{\mathbf{B}}_{sprzegi}$ - coupling matrix of the crankshaft torsional and axial vibration (determined for i-th crank).

The vector of the generalized excitation in the nodes of the model can be presented as :

$$\mathbf{h}_w(j\omega) = vec(\bar{\mathbf{h}}_{A1}, \bar{\mathbf{h}}_{A2} \dots \bar{\mathbf{h}}_{An}) \tag{6}$$

where :

$\bar{\mathbf{h}}_{Ai}$ - generalized excitation vector of i-th superelement.

The following equation system should be solved to determine the vibration amplitudes of the crankshaft :

$$\mathbf{B}_w(j\omega) \cdot \mathbf{q}_w(j\omega) = \mathbf{h}_w(j\omega) \tag{7}$$

where :

$\mathbf{q}_w$ - vector of generalized displacement amplitudes of the crankshaft for the harmonic excitation components taken into account.

Response of the system in function of time was calculated by means of the method described in [2].

## CONCLUSIONS

The proposed superelement makes it possible to account for the coupling phenomenon of the torsional and axial vibration in the dynamic analysis of marine combustion engines. As the elements of the superelement flexibility matrix describing the coupling are of the open form, an analysis can be performed of the influence of particular crankshaft design parameters on the vibration coupling. Due to this reason it will be possible, at a further investigation stage, to introduce optimization methods to the ship propulsion systems design in order to minimize the detrimental effect of coupling the torsional and axial vibration. It will make modifying the system's design already at the design stage possible.

Moreover, decision variables and an optimization objective function can be easily determined by making use of the elaborated computational model of the propulsion system.

### NOMENCLATURE

h - generalized excitation vector applied to crank
$\bar{h}$ - superelement generalized excitation vector
$h_w$ - crankshaft generalized excitation vector
K - global crank stiffness matrix
$K_w$ - superelement stiffness
MCR - maximum continuous rating of engine
$\delta$ - half of the angle between cranks

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## Conferences

# SHIP PROPULSION PLANTS

On 16÷17 October the Maritime University of Szczecin hosted XIX International Symposium on Ship Propulsion Plants. It was organized by Institute of Technical Exploitation of Ship Power Plants, Mechanical Faculty of the University, within the celebration programme of 50th anniversary of maritime education in Western Pommerania.

The symposium was the next of the series of such meetings devoted to problems of designing and exploitation of ship power plants, successively organized by 5 largest scientific and educational centres of the specialty in Poland.

Scientific Committee of the symposium selected 51 papers to be presented at 6 topical sessions. The papers were generally concerned with the following problems :

- Design elements of ship power plants, selection and co-operation of their components.
- Exploitation processes of ship power plant machines and systems.
- Ship power plant operation and marine environment protection.
- Optimization of energy processes.
- Diagnostics of technical state of marine engine components and of performance of energy transformation cycles.
- Composite materials for repair of ship machines and installations.
- Laboratory and service investigations of characteristics of power and propulsion devices.
- Education of engineers in the area of designing.

The scope of themes of the symposium was found very wide and many papers interesting. Most of the papers was prepared by Polish scientists from :

- Gdynia Maritime Academy
- Maritime University of Szczecin
- Polish Naval Academy, Gdynia
- Ship Design & Research Centre, Gdańsk
- Technical University of Gdańsk
- Technical University of Szczecin.

Foreign guests prepared 14 papers. They represented :

- Far Eastern Technical University, Vladivostok, Russia
- Technical University, Murmansk, Russia
- Ukrainian Maritime Technical University, Nikolayev.