A simulation model of energy distribution in ship combustion engine

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ABSTRACT

In the paper a model of energy flow and distribution in ship diesel engine cylinder was presented. This is a model of discrete parameters, being a continuation of the author's research on simulation of energy processing within ship main propulsion engine [2,3]. The model in question makes it possible to calculate energy flow values delivered and transferred during every working cycle of the engine cylinder. Results of application of the model for 6ZA40S Sulzer engine installed on B672 ship were also attached. The results were compared with results of test-stand measurements of the engine, performed at different load levels.

Keywords : computer simulation, ship diesel engines, dynamic processes, energy balance and distribution

INTRODUCTORY NOTE

Knowledge of energy flow and distribution within combustion engine is important for many reasons. It makes it possible to estimate and optimize engine's efficiency as well as to diagnostically model its technical state. In the case of analyses concerning ship propulsion system it enables to determine values of input energy flows to auxiliary systems and waste heat utilization devices.

PHYSICAL MODEL

Cylinder's space together with a working medium contained in it forms a thermodynamical system which can be described by means of the equation of the 1st principle of thermodynamics. The energy delivered to the cylinder in the form of fuel flow is processed, due to combustion process, into flows of internal work, thermal energy transferred by combustion chamber walls, as well as of engine exhaust gas energy.

The energy transferred by combustion chamber walls is absorbed by media contained in the systems of fresh water cooling cylinders, heads and injectors, as well as of lubricating and piston cooling oil. Whereas the exhaust gas energy is transferred to drive turbine of supercharging system, to produce steam in waste heat boiler, as well as to the atmosphere and the water cooling head and turbine. Internal work of cylinder's working cycle is transferred through the crankshaft-piston system to the propeller, to drive auxiliary devices of the engine, as well as consumed during overcoming friction drag within the engine. A small part of the combustion energy is transferred through the engine's elements to the environment.

The following assumptions have been made :

- ➲ Instantaneous temperature, pressure and gas constant of a medium take the same values in every point of the space over piston.
- ➲ Heat exchange surface area in the cylinder contains only three crucial ones : that of lower plate of cylinder head together with valves, of piston head, as well as changeable surface area of cylinder liner which is in contact working medium. Average thickness of walls of cylinder liner, lower plate of cylinder head, piston head and cylinder block, as well as material parameters of the walls, are assumed constant.
- ➲ Heat flows only in perpendicular direction to particular surfaces.

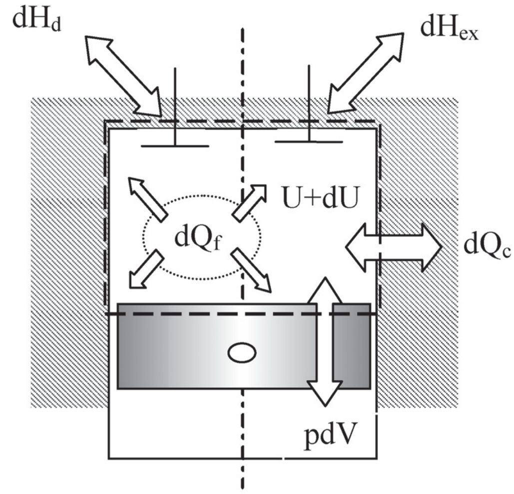

Fig. 1. *Schematic diagram of energy balance for the thermodynamical system "cylinder". **Notation** : dH_d - input enthalpy increment, dH_{ex} - output enhtalpy increment, dQ_f - heat emitted during fuel combustion, dQ_c - heat exchanged with walls, pdV - work done by working medium, U - working medium internal energy*

MATHEMATICAL MODEL

Energy balance in the cylinder can be written in the following form :

$$\frac{dU}{d\tau} = \frac{dQ_f}{d\tau} - \frac{dQ_c}{d\tau} - p\frac{dV}{d\tau} + \frac{dH_d}{d\tau} - \frac{dH_{ex}}{d\tau} \quad (1)$$

The derivatives appearing in (1) express rate of change of the following flows :

$\frac{dU}{d\tau} = \dot{U}(\tau)$ - of internal energy in the cylinder

$\frac{dQ_f}{d\tau} = \dot{Q}_f(\tau)$ - of heat of combustion of fuel injected to the cylinder

$\frac{dQ_c}{d\tau} = \dot{Q}_c(\tau)$ - of heat exchanged with the walls

$p(\tau)\frac{dV}{d\tau} = N_i(\tau)$ - of internal work performed by working medium

$\frac{dH_d}{d\tau} = \dot{H}_d(\tau)$ - of input enthalpy

$\frac{dH_{ex}}{d\tau} = \dot{H}_{ex}(\tau)$ - of output enthalpy.

By taking into account that the internal energy of working medium is a function of the temperature T and its chemical content F

$$\frac{du}{d\tau} = \frac{\partial u}{\partial T}\frac{dT}{d\tau} + \frac{\partial u}{\partial F}\frac{dF}{d\tau} \qquad (2)$$

where :

F - ratio of the combusted fuel mass m_{fb} and the mass, m_a, of the air contained in the mass of working medium m :

$$F = \frac{m_{fb}}{m_a} \;,\; m = m_{fb} + m_a \qquad (3)$$

and

$$c_v = \frac{\partial u}{\partial T}$$

the equation (1) can be transformed to the form :

$$\frac{dT}{d\tau} = \frac{1}{mc_v}\left(\begin{array}{l} \frac{dQ_f}{d\tau} - \frac{dQ_c}{d\tau} - p\frac{dV}{d\tau} + h_d\frac{dm_d}{d\tau} + \\ - h_{ex}\frac{dm_{ex}}{d\tau} - u\frac{dm}{d\tau} - m\frac{\partial u}{dF}\frac{dF}{d\tau} \end{array}\right) \qquad (4)$$

On its basis the instantaneous temperature of working medium in the cylinder after the calculation step $\Delta\tau$, can be determined :

$$T(\tau + \Delta\tau) = T(\tau) + \frac{dT}{d\tau}\Delta\tau \qquad (5)$$

The equation (5) together with the state equation of working medium make it possible to determine course of temperature and pressure in the cylinder in function of time or crank angle [3]. To this end is also necessary to determine rates of the above mentioned energy flows as well as energy parameters of working medium in every calculation step of cycle. The working medium flows through inlet and outlet valves in normal and reverse direction depending on pressure values in the cylinder and manifolds as well as instantaneous state of opening of the valves, were determined. Different cases of the equation (1) regarding cycle phases realized in the cylinder, were taken into account.

For instance, in the charging phase only inlet valves through which fresh air flows in the cylinder, are opened. Depending on difference of the pressure p in the cylinder and p_d in inlet manifold, the following flows through inlet valves take place :

- at $p_d > p$ - **normal flow** - in which the mass flow $(\frac{dm_d}{d\tau})$ enters the cylinder - of the specific enthalpy of air, h_d, in the manifold, resulting in changing the chemical content of working medium F and its internal energy u in the cylinder :

$$\frac{dQ_f}{d\tau} = 0 \;\;;\;\; \frac{dm_{ex}}{d\tau} = 0 \;\;;\;\; \frac{dF}{d\tau} = -\frac{(F+1)F}{m}\frac{dm}{d\tau}$$

The equation (4) is of the following form :

$$\frac{dT}{d\tau} = \frac{1}{mc_v}\cdot$$

$$\cdot\left(-\frac{dQ_c}{d\tau} - p\frac{dV}{d\tau} + h_d\frac{dm_d}{d\tau} - u\frac{dm}{d\tau} - m\frac{\partial u}{dF}\frac{dF}{d\tau}\right)$$

- at $p_d < p$ - **reverse flow** - during which the mass flow $(-\frac{dm_d}{d\tau})$ flows out - of the specific enthalpy of working medium, h, in the cylinder. The chemical content of working medium, F, and the internal energy u of the medium in the cylinder remain unchanged :

$$\frac{dQ_f}{d\tau} = 0 \;\;;\;\; \frac{dm_{ex}}{d\tau} = 0 \;\;;\;\; \frac{dF}{d\tau} = 0$$

The equation (4) takes the following form :

$$\frac{dT}{d\tau} = \frac{1}{mc_v}\left(-\frac{dQ_c}{d\tau} - p\frac{dV}{d\tau} - h\frac{dm_d}{d\tau} - u\frac{dm}{d\tau}\right)$$

When an engine is in a steady working state, it is possible to determine an approximate state of working medium after each of the working cycle in a given cycle point. Energy balance in the engine can be expressed as follows :

$$\dot{Q}_d = N_i + \dot{Q}_c + \dot{Q}_{ex} + \dot{U} \qquad (6)$$

where :

$\dot{Q}_d$ - rate of the energy delivered to cylinder

N_i - internal work power (indicated power) of cylinder

$\dot{Q}_c$ - flow rate of heat exchanged with walls

$\dot{Q}_{ex}$ - energy rate of gas exhausted from cylinder

$\dot{U}$ - internal energy rate of working medium in cylinder.

In the steady state the internal energy rate $\dot{U}$ should be equal to zero in an ideal model which is practically unavailable. However a model in which the value is relatively small, can be deemed correct.

The instantaneous flow rate of heat transferred through walls, $\dot{Q}_c$, as well as temperature distribution in walls can be determined by means of a model of heat exchange process in the cylinder – environment system [3]. Heat exchange – through walls – between working medium and cooling medium is an unsteady process. All the parameters : heat exchange surface area, temperature of walls, as well as the convective heat-transfer coefficient α are changeable in function of crank angle. The heat exchange with cylinder walls was modelled by means of the finite element method. The cylinder liner was divided into n finite elements in which concentrated temperatures of cylinder liner wall and of cooling water, are determined.

Changeable temperature in the direction of wall thickness *x* are derived from the Fourier equation :

$$\frac{\partial T_s}{\partial \tau} = a_s \frac{\partial^2 T_s}{\partial x^2} \tag{7}$$

with the 3rd kind boundary conditions on the side of working medium and cooling water :

$$\alpha(T - T_{s1}) = -\lambda_s \left(\frac{\partial T_s}{\partial x}\right)_{x=0}$$
$$\alpha_w (T_{s2} - T_w) = -\lambda_s \left(\frac{\partial T_s}{\partial x}\right)_{x=\delta_s} \tag{8}$$

where :

- α , α_w – convective heat-transfer coefficient on the side of working medium and on the side of cooling water, respectively
- a_s , λ_s – temperature equalization coefficient and thermal conductance of wall, respectively
- T_{s1} , T_{s2} – internal and external cylinder liner wall temperature, respectively
- δ_s – wall thickness.

Due to significant temperature and velocity changes of working medium contained in cylinder in different phases of working cycle, a value of the coefficient α is calculated in function of crank angle, in accordance with the empirical formula [5] :

$$\alpha(\varphi) = 130\, d^{-0.2} p(\varphi)^{0.8} T(\varphi)^{-0.53} \cdot \left[C_1 C_m + C_2 \frac{V_s T_1}{p_1 V_1} (p(\varphi) - p_{ob}) \right]^{0.8} \tag{9}$$

where :

- φ [deg] - crank angle
- d [m] - cylinder diameter
- p [bar] - working medium pressure in cylinder
- C_m [m/s] - mean piston speed
- $C_1 = 6.18$ - for compression, combustion and expansion phase, 2.28 - for charge exchange phase
- $C_2 = 0.00324$ [m/s K] - for integral combustion chamber, 0.00622 [m/s K] - for divided combustion chamber
- V_s [m^3] - piston displacement
- T_1 , p_1 , V_1 - working medium parameters at the beginning of compression
- p_{ob} [bar] - pressure in cylinder in case of external drive.

The delivered energy flow is determined as the total energy contained in the flow of fuel oil delivered to the cylinder, and in the flow of charging air :

$$\dot{Q}_d = \dot{m}_f W_f + \dot{m}_d h_d \tag{10}$$

The cylinder internal work flow N_i is transferred through crankshaft – piston system to drive engine shaft . Its magnitude depends on working medium pressure in the cylinder and on engine rotation speed. The latter is determined by means of the ship motion model [2] considered as an energy consumer model.

The flow of energy contained in cylinder exhaust gas, $\dot{Q}_{ex}$, is determined as the enthalpy of cylinder exhaust gas flows :

$$\dot{Q}_{ex} = h_{ex} \dot{m}_{ex} \tag{11}$$

The flow of heat exchanged with cylinder walls by working medium can be considered as a source of the particular heat flows absorbed by cooling media: that of cylinder liner cooling water, $\dot{Q}_{wc}$, of cylinder head cooling water, $\dot{Q}_g$, of piston cooling water, $\dot{Q}_{wt}$, of lubricating oil, $\dot{Q}_{ol}$, as well as of the environment, $\dot{Q}_{ot}$:

$$\dot{Q}_c = \dot{Q}_{wc} + \dot{Q}_g + \dot{Q}_{wt} + \dot{Q}_{ol} + \dot{Q}_{ot} \tag{12}$$

If water temperature distribution within cylinder cooling jacket is known [3], the flow of heat absorbed by cylinder cooling water, $\dot{Q}_{wc}$, can be determined from the equation :

$$\dot{Q}_{wc} = \dot{m}_w c_w \left[T_{w,1} - T_{w,n} \right] \tag{13}$$

where :

- $T_{w,1}$, $T_{w,n}$ – cooling water temperature at inlet to and outlet from cylinder jacket, respectively.

The flow of heat transferred through cylinder head walls in steady state can be determined by using the following equation :

$$\dot{Q}_g = \alpha A_g (T - T_{og}) \tag{14}$$

where :

- A_g , T_{og} – area and temperature of head's lower plate surface, repectively.

The flow of heat absorbed by cylinder head cooling water, $\dot{Q}_{wg}$, is determined as the sum of the heat transferred by head's lower plate, $\dot{Q}_g$, and the heat flow given up by exhaust gas to valve and exhaust channel walls. The latter is estimated to be 10% of the total heat transferred to cooling medium [1,4].

Similarly the flow of heat transferred by piston head walls in steady state can be determined by using the equation :

$$\dot{Q}_{wt} = \alpha A_t (T - T_t) \tag{15}$$

where :

- α - working medium convective-heat coefficient
- A_t , T_t - area and temperature of piston head surface, respectively.

The flow of heat given up to the environment is determined as the heat absorbed by the engine outer surface having the temperature equal to that of cylinder block. This is expressed by the equation :

$$\dot{Q}_{ot} = \alpha_{ot} A_b (T_b - T_{ot}) \tag{16}$$

where :

- α_{ot} - ambient air convective-heat coefficient
- A_b , T_b - area and temperature of outer surface of cylinder block, respectively.

The flow of heat absorbed by lubricating oil, $\dot{Q}_{ol}$, can be determined as the rest of thermal energy balance (12) :

$$\dot{Q}_{ol} = \dot{Q}_c - \dot{Q}_{wc} - \dot{Q}_{wt} - \dot{Q}_g - \dot{Q}_{ot} \tag{17}$$

Fig.2 and 3, show the information on distribution of the energy flowing through 6ZA40S engine, given by its manufacturer [6]. In Fig.2. is presented the percentage distribution of the energy of fuel oil delivered to the engine, into : mechanical energy, heat transferred to cooling media and environment, as well as energy contained in exhaust gases. The balance was elaborated for nominal conditions of the engine's operation in the tropical zone and it did not account for the heat recovered in waste heat boiler. However in Fig.3 shares of the enumerated flows are shown in function of the engine's load, related to its rated load. Information on the conditions in which the characteristics have been obtained, is lacking.

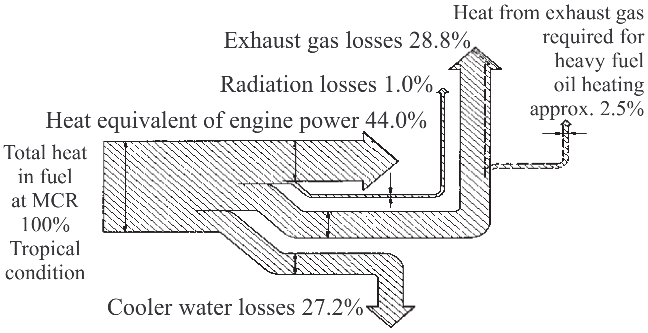

Fig. 2. *Sankey's energy flow diagram for 6ZA40S engine in nominal conditions acc. [6]*

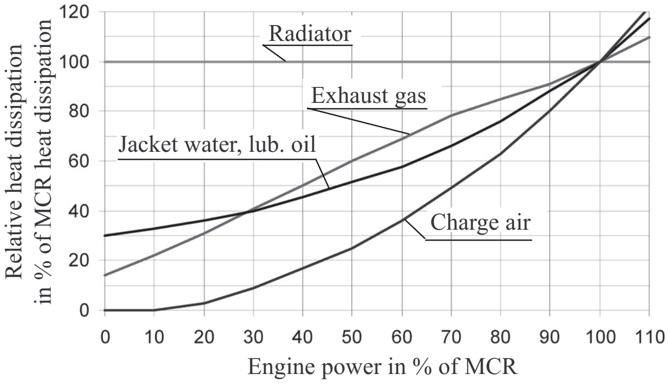

Fig. 3. *Percentage shares of thermal energy flows through 6ZA40S engine in function of its load acc. [6]*

CALCULATION RESULTS

The model of energy flow and distribution, together with that of heat exchange process in the cylinder-environment system, was programmed in Borland Pascal code of Delphi package. Calculations of the model were carried out by applying changeable integration step with the use of a PC fitted with Pentium II processor [2].

In Table the results of calculation of the model of energy distribution within cylinder are presented for different engine load levels. The following flows and their percentage shares in the balance : mechanical energy (cylinder indicated power) (item. 3), heat transferred from working medium through combustion chamber walls (item. 4), heat contained in gas exhausted from cylinder (item. 5), heat transferred through cylinder liner and head walls (item. 7 and 8), heat absorbed by lubricating oil (item. 10) , as well as heat given up to the environment (item. 11).

The difference between percentage share of the energy delivered to and that transferred from cylinder in steady state was shown in item 13 as the relative error of calculations. In reality the value has not necessarily to be equal to zero; it equals internal energy increment of working medium. This is due to a non-linearity of cyclic work of the engine.

The calculated share of mechanical energy appeared higher by abt. 2% than that presented in the Sankey's diagram (Fig. 2) for 100% engine load. Hence it results that the value given in the diagram is the percentage share of the effective power measured at the engine shaft.

From Fig. 2. it results that the value given by the engine's manufacturer as the share in the energy balance of the heat absorbed by cooling water, is understood as the heat absorbed by all cooling systems (that of fresh water, lubricating oil, and charging air); and the share of the exhaust gas energy is calculated for exhaust gas behind the turbine.

On the basis of the calculation results the energy balance and its diagram for the modelled engine was elaborated, as shown in Fig.4.

Calculation results of energy distribution in 6ZA40S engine

No	Item			Desired load factor			
1	Engine load		%	100	90	75	50
2	Flow of energy contained in fuel oil	Q_f	kW	1187	1061	915	634.6
			%	100	100	100	100
3	Cylinder indicated power	N_i	kW	555	494	421	284
			%	46.8	46.6	46.0	44.8
4	Flow of heat transferred through walls from working medium	Q_c	kW	207	188	166	121
			%	17.4	17.7	18.1	19.1
5	Flow of heat contained in cylinder exhaust gas	Q_{ex}	kW	412	359	312	215
			%	34.7	33.8	34.1	33.9
6	Flow of heat contained in exhaust gas behind turbine	Q_{ex}	kW	362.0	309.8	269.9	190.4
			%	30.5	29.2	29.5	30
7	Flow of heat absorbed by cylinder liner cooling water	Q_{wc}	kW	76.5	70.8	61.2	42.5
			%	6.4	6.7	6.7	6.7
8	Flow of heat transferred through walls of cylinder head's lower plate	Q_g	kW	46.2	43.4	43	41.9
			%	3.9	4.1	4.7	6.6
9	Flow of heat absorbed by cylinder head cooling water	Q_{wg}	kW	66.9	62.2	59.6	54
			%	5.6	5.9	6.5	8.5
10	Flow of heat transferred to lubricating oil	Q_{ol}	kW	79.4	69.1	57.6	32.8
			%	6.7	6.5	6.3	5.2
11	Flow of heat given up to environment	Q_{ot}	kW	4.9	4.7	4.2	3.8
			%	0.4	0.4	0.5	0.6
12	Cylinder energy balance (3 + 4 + 5)		%	98.9	98.1	98.3	97.7
13	Relative error		%	-1.09	-1.86	-1.74	-2.30

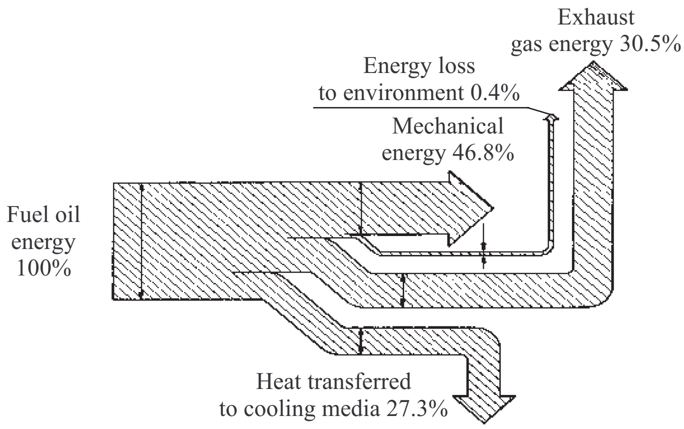

Fig. 4. *Sankey's energy flow diagram for 6ZA40S engine model at 100% load in nominal conditions*

On the basis of comparison of the results presented in Fig.3 and the results of calculation of the model it can be stated that the first has been elaborated for constant rated speed of the engine.

The research on dynamics of the processes was presented in the author's paper [3].

CONCLUSIONS

- In the paper the physical and mathematical model of energy flow and distribution within the engine were presented, which makes it possible to determine instantaneous values of particular energy flows in function of engine load.
- The model was exemplified by means of 6Z40S engine (installed as the main engine on B672 ship) and the research results were compared with the test-bed measurements of the engine.

- ❍ The condition of formal correctness of the modelled process, i.e. energy balance of the engine, was satisfied in every step of calculations at an acceptably low error.
- ❍ The model can be used as a tool for investigation of the engine operation process, influence of engine parameters on the process, as well as for diagnosing the engine technical state.
- ❍ Investigating the model makes it possible to provide new recommendations important for laboratory experiments and conditions of carrying out empirical tests.
- ❍ The model was used for programming a ship propulsion system simulator.

NOMENCLATURE

a_s - temperature equalization coefficient [m^2/s]
A - surface area [m^2],
c_v - specific heat [J/kgK]
F - ratio of combusted fuel mass and charging air mass
h - specific enthalpy [J/kg]
H - enthalpy [J]
m - mass [kg],
$\dot{m}$ - mass flow [kg/s]
N - power [W]
p - pressure [Pa]
Q - heat, energy [J]
Q_c - heat exchanged with walls of combustion chamber [J]
$\dot{Q}$ - heat flow, power [W]
T - temperature [K]
u - specific internal energy [J/kg]
U - internal energy [J]
V - volume [m^3]
W - calorific value [J/kg]

α - convective heat-transfer coefficient [W/m^2 K]
δ - thickness [m]
φ - crank angle [deg]
λ - thermal conductance [W/m K]
τ - time [s]

Indices

a - of air
b - of cylinder block
d - of inlet
ex - of exhaust gas
f - of fuel
fb - of combusted fuel oil
i - indicated
ot - of environment
s - of wall
w - of cooling water

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Conference

Jubilee Symposium

On 18÷19 November 2004

25th Symposium on Ship Power Plants

was held in Gdańsk.

As it can be observed the scientific conference gathering experts working in the domain of ship powering systems has had long tradition. It has been born from the idea of yearly meetings to exchange knowledge and experience in this area among scientific workers of five Polish Baltic Sea coast universities, namely :

- Gdańsk University of Technology
- Technical University of Szczecin
- Gdynia Maritime University
- Maritime University of Szczecin
- Naval University of Gdynia.

Every 5th year each of them serves as the organizer of a sucessive symposium in question. The 25th Symposium was organized by the Department of Ship Power Plants, Faculty of Ocean Engineering and Ship Technology, Gdańsk University of Technology. Its scope was very wide and covered the following topics :

Design, manufacturing and operation processes of ship power plants and their equipment with accounting for reliability, safety and environment protection.

It was reflected in 43 presented papers in preparation of which the representatives of Gdynia Maritime University with 11 papers, and Naval University with 10 papers contributed to the largest extent. The remaining universities took part in the Symposium: Gdańsk University of Technology - with 9 papers, Maritime University of Szczecin - with 8 papers, and Technical University of Szczecin - with 5 papers.

The Symposium program was enriched by presentation of products of several firms :

- ENAMOR - measurement instruments
- UNITEST - educational simulators
- MAN - B&W - ship engines
- Alfa Laval - fuel oil purifying devices
- INTERMASZ - filtration membrane devices.