ZBIGNIEW KORCZEWSKI, Assoc.Prof.,D.Sc.,M.E.
Technical Institute of Ship's Maintenance
Polish Naval University, Gdynia

An approach to modelling of gasdynamic processes in turbocharging system of marine four-stroke diesel engine

Contemporary naval ships are usually propelled by four-stroke, high - or medium - speed diesel engines charged by means of turbo-blowers. Excess of the air delivered by the turbo-blower should be maintained within the limits which guarantee complete combustion of fuel in engine cylinders [3,5]. However during unsteady operation processes of the engine (such as starting, changes of rotational speed or load) the limiting values of the air excess number can be exceeded due to time-lag occurring within the engine charging system (considered jointly with the automatic control system). In the result of the excessive enrichment (or depletion) of the air-fuel mixture the combustion process is disturbed which substantially influences values of the mean indicated pressure, specific fuel oil consumption, mechanical and thermal loads of engine structural elements, exhaust gas toxicity as well as engine noisiness. In extreme cases misfiring can happen in combustion chambers even causing spontaneous stopping of the engine. Such events are well known for operators of ship propulsion systems [4].

MODELLING OF GASDYNAMIC PROCESSES

Modelling of dynamic energy processes occurring in the engine turbocharging systems of constant inlet pressure to the turbine, usually applied to two-stroke engines, can be realized in a generally well-known manner [4,5,6].

However the pulsatory systems applicable to the four-stroke engines, in which the turbo-blower is supplied with the exhaust gas of the peak outlet pressure (Büchi systems) introduce some singularities into calculation algorithms of elements of a considered flow system and their mutual connections. A phenomenon which should be specially carefully considered when analyzing dynamic behaviour of such systems is a character of changes of the quantities triggering the unsteady processes within the turbo-blower, namely flow of energy (enthalpy) due to wave pressure impulses of the exhaust gas from the engine.

Change courses of the delivered enthalpy flow $\dot{H}_{x-x}$ of the same amount of the exhaust gas at the outlet section of the radial-flow turbine guide vanes of the turbine-blower are schematically shown in Fig.1 for the two compared turbocharging systems.

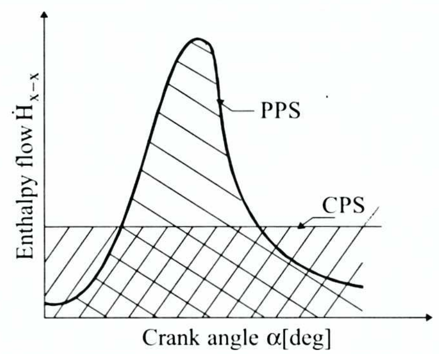

Fig.1. Schematic diagram of change courses of the delivered enthalpy flow $\dot{H}_{x-x}$ (at the same exhaust gas amount in the outlet section of the radial-flow turbine guide vanes of the turbine-blower) for the two turbocharging systems : ***PPS*** *– pulsatory pressure system ;* ***CPS*** *– constant pressure system*

Due to pulsation of the exhaust gas mass flow before the turbine its efficiency is lower than that of the constant pressure system, however kinetic energy of the exhaust gas from the engine is utilized in a greater degree. Moreover, the cylinder scavenging process is more effective due to an appropriate course of the pressure waves in the exhaust gas piping. Another factor which substantially influences time-lag of the unsteady processes within the turbocharging system, is the applied solution of the engine speed governor and its gasdynamic coupling with the air-charging system [3,5].

OBJECT OF INVESIGATION

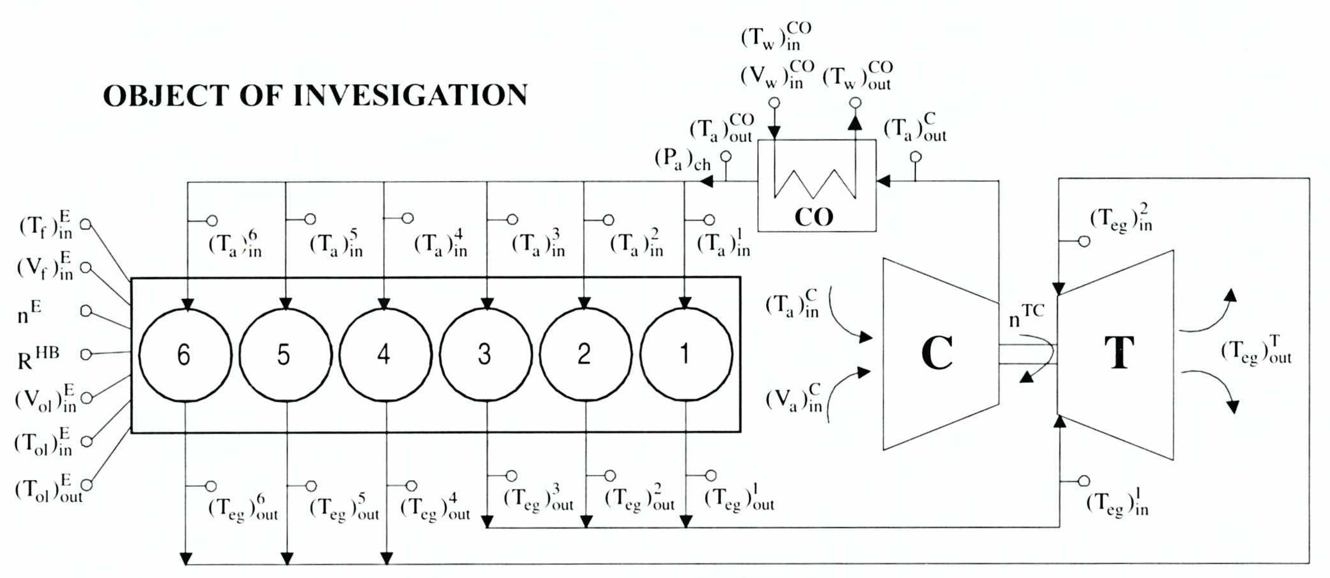

Fig.2. Scheme of the turbocharging system of 6AL20/24 SULZER diesel engine

The turbocharging system of 6AL20/24 SULZER, four-stroke diesel engine, with the pulsatory admission system of the turbine and with cooling of the charging air (Fig.2) was selected to practically model the above described gasdynamic processes.

The system in question is composed of the following units :

- ⇨ the turbo-blower rotor unit with the turbine T and radial compressor C
- ⇨ the air – and exhaust gas passages
- ⇨ the charging air cooler CO.

In the attached scheme the points of measurements of the engine control parameters, carried out by means of a computerized measurement recording system elaborated by the Technical Institute of Ship's Maintenence, Polish Naval University, are also indicated. The measurements were carried out to parametrically identify the gasdynamic processes occurring within the engine turbocharging system in question.

The measured engine parameters, whose values have been recorded at 10 Hz sampling frequency, as well as characteristics of the applied measuring gauges are presented in Table.

The single charging system of the engine of the 1-4-2-6-3-5 fuel injection sequence is equipped with a radial turbine of double-passage pulsatory admission of exhaust gas.

Course of exhaust gas changeable pressure in piping before the turbine is similar to that presented in Fig.3. This solution in which outlet valve opening lasts over the crank angle $\alpha_{out} = 340^o$ makes favourable overlapping of the pressure wave impulses from two cylinders over the crank angle $\alpha_{out} = 100^o$ possible.

Due to that effect the lowest exhaust gas pressure in outlet passages is higher than the outlet counterpressure. This way the turbine's efficiency is improved, but at the expense of a greater amount of the work to remove the exhaust gas.

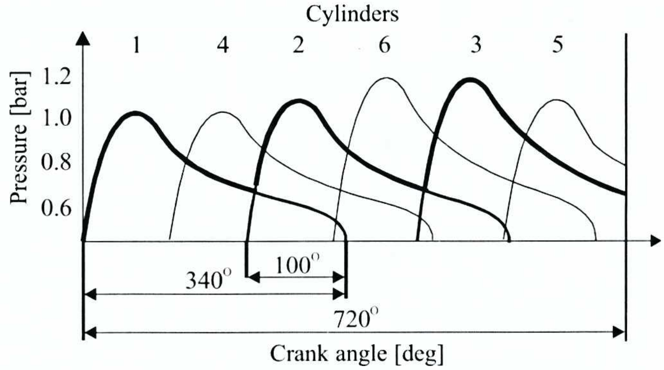

Fig.3. Course of exhaust gas changeable pressure in outlet passages of 6AL 20/24 SULZER diesel engine before the turbine

The measured parameters of 6AL20/24 SULZER diesel engine

Parameter		Measurement range	Measurement accuracy [%]	Gauge type
n^E	Engine rotational speed	800 rpm	0.1	inductive
n^{TC}	Rotational speed Of turbo-compressor rotor	40 000 rpm	0.5	magnetoelectric
$(P_a)_{ch}$	Charging air pressure	0.25 MPa	1.0	tensometric
R^{HB}	Response force of the hydraulic brake	1.2 kN	0.16	tensometric
$(T_a)^1_{in} \cdots (T_a)^6_{in}$	Air temperature at inlets to successive cyliders of the engine	373 K	0.5	resistive
$(T_a)^C_{in}$	Air temperature before the compressor	373 K	0.5	resistive
$(T_a)^C_{out}$	Air temperature behind the compressor	373 K	0.5	resistive
$(T_a)^{CO}_{out}$	Air temperature behind the cooler	373 K	0.5	resistive
$(T_{eg})^1_{in}$ (cyl.1,2,3) $(T_{eg})^2_{in}$ (cyl.4,5,6)	Exhaust gas temperature in collectors before the turbine	873 K	0.3	thermocouple
$(T_{eg})^1_{out} \cdots (T_{eg})^6_{out}$	Exhaust gas temperature at outlet from successive cylinders of the engine	873 K	0.3	thermocouple
$(T_{eg})^T_{out}$	Exhaust gas temperature behind the turbine	873 K	0.3	thermocouple
$(T_f)^E_{in}$	Fuel temperature at inlet to the engine	373 K	0.5	resistive
$(T_{ol})^E_{in}$	Oil temperature at inlet to the engine	373 K	0.5	resistive
$(T_{ol})^E_{out}$	Oil temperature at outlet from the engine	373 K	0.5	resistive
$(T_w)^{CO}_{in}$	Water temperature at inlet to the air cooler	373 K	0.5	resistive
$(T_w)^{CO}_{out}$	Water temperature at outlet from the air cooler	373 K	0.5	resistive
$(V_a)^C_{in}$	Volumetric rate of air flow before the compressor	2 600 m^3/h	1.5	Tensometric (von Karman effect)
$(V_f)^E_{in}$	Volumetric fuel flow rate to the engine	100 dm^3/h	0.5	flow measuring
$(V_{ol})^E_{in}$	Volumetric oil flow rate to the engine	25 m^3/h	0.5	flow measuring
$(V_w)^{CO}_{in}$	Volumetric rate of water flow through the air cooler	15 m^3/h	1.0	flow measuring

MODELLING OF PHYSICAL PROCESSES OCCURRING IN THE ENGINE TURBOCHARGING SYSTEM

In Fig.4 a simplified model of physical processes occurring in the turbocharging system of the considered diesel engine is presented. The model was elaborated on the basis of preliminary identification of dynamic features of working medium flow through gasdynamic coupling of the turbo-blower and engine after thorough analysis of the design structure of the charging system in question. For mathematical description of the unsteady processes in the system several dynamic modules were distinguished in it. For each of them input and output quantities being the system's state variables, were determined. Quantities describing changes of amounts of working medium substance and its internal energy accumulated within the gas flow passage spaces as well as changes of amount of kinetic energy accumulated in the turbo-blower rotor unit, were assumed the state parameters of the dynamic processes.

In the presented model the phenomenon of gasdynamic and kinematic mutual interaction of the volume of the engine charging air passage and the engine speed governor, was also taken into account. Thermal state changes of the turbine and resulting changes of rotor radial clearances, hence changes of flow characteristics and turbine efficiency, were also accounted for in the model.

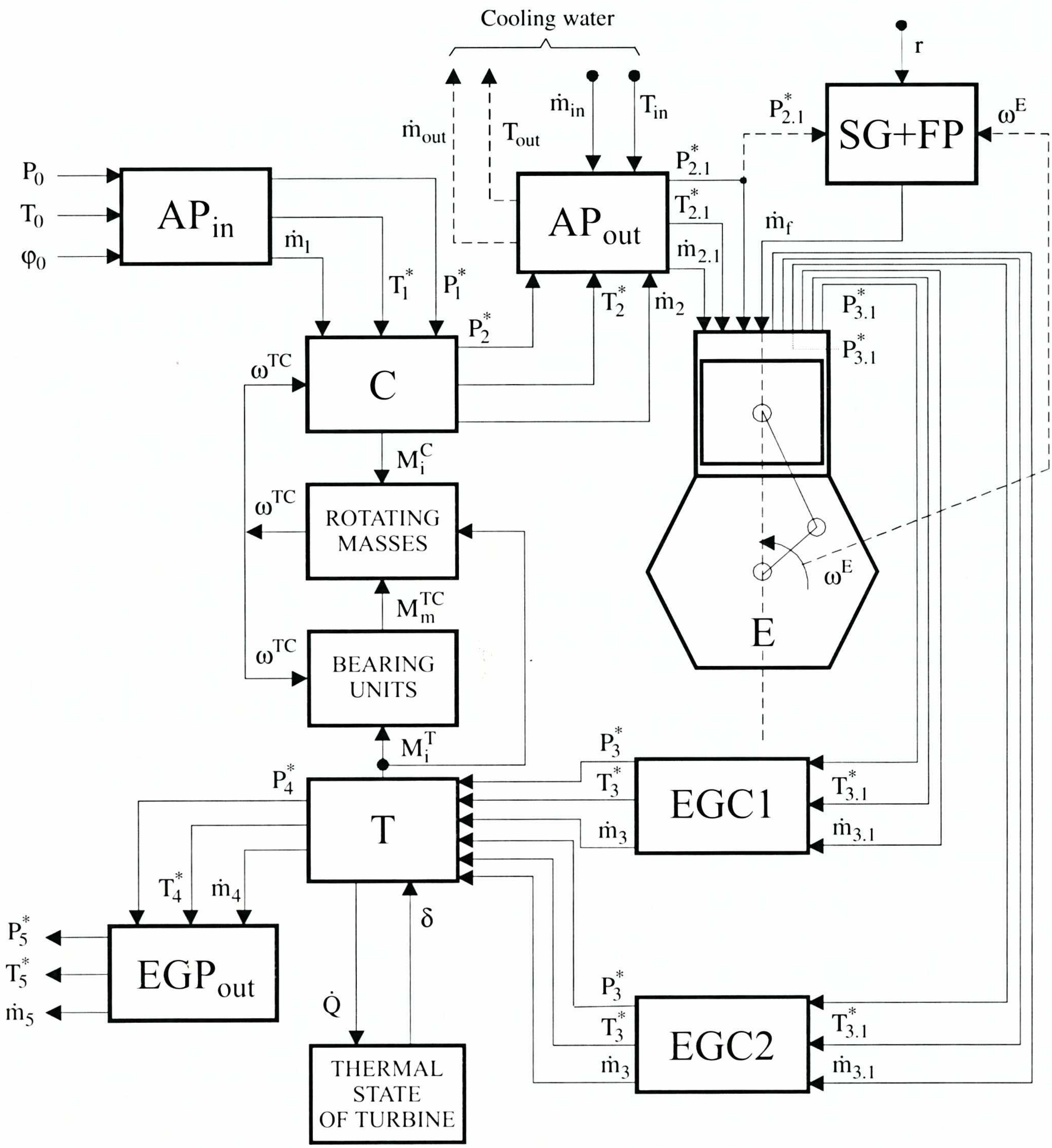

Fig.4. Schematic model of physical process of working medium flow through the air-charging system of the considered engine

MATHEMATICAL MODELLING

In the next stage of the research the above presented model of physical processes will be used as a basis for their mathematical modelling. The following equations will be developed :

- of instantaneous balance of mass and energy of the flowing working medium
- of instantaneous balance of angular momentum of rotating masses of the rotor unit
- algebraic nonlinear equations which characterize features of the flowing working medium, material properties and geometry of structure of the modelled object.

In the preliminary stage the following simplifying assumptions have been accepted to make formulation of the equations possible :

- the assumed functional modules are modelled as the dynamic objects of discrete parameters
- the sucked-in air parameters comply with ISA (International Standard Atmosphere) standard
- the flowing working medium is considered as a semi-ideal gas
- the combustion process in the engine cylinders is realized without any time-lag due to fuel oil transport between the fuel oil feeder of the engine control system and the injectors of the combustion chambers, as well as without delays due to its spraying, evaporation and ignition (of time constants below 25÷30 ms [3])
- mathematical description of the working medium parameters deals with the average total values of pressure and temperature at the selected control sections of the engine air-charging system on the assumption that changes of pressure and density of the working medium flowing through the system are small in relation to their mean values
- as time constants of working medium dynamic flow through blade passages of the compressor and turbine are very small (from 1 to 10 ms [4]) accumulation of the working medium can be neglected and the dynamic phenomena can be taken as a sequence of instantaneous steady states
- thermogasdynamic parameters at outlet from the accumulation spaces AP_{in}, AP_{out}, EGC1, EGC2 and EGP_{out} (see Fig.4) are determined with some delay in relation to those averaged inside the spaces, namely that equal to the time of flow of the working medium between inlet and outlet sections of the spaces
- changes of radial and axial clearances of the turbo-blower rotor unit due to centrifugal forces and thermal state changes of its elements are not taken into account in the modelled dynamic processes
- influence of working medium friction within the flow passages as well as heat exchange between the environment and working medium inside the passages are neglected.

FINAL REMARKS

- In the first stage of the research a model of physical processes which occur in the turbocharging system of a marine four-stroke diesel engine was elaborated. The considered system operates with pulsatory exhaust gas admission to the turbine and pulsatory intake of the air from the compressor by the engine cylinders.
- In the elaborated model the system was split into some functional modules important from the point of view of the realized gasdynamic processes for which also flow of input and output signals necessary to formulate energy balance was determined.
- Moreover, basic assumptions necessary to develop a mathematical model of the gasdynamic processes in question were presented.
- In the next stage of the research the mathematical model will be developed on this basis.

NOMENCLATURE for Fig.4

Acronyms

AP_{in} - air inlet passage to the compressor
AP_{out} - air passage to charge exchange unit of the engine
C - compressor
E - engine
EGC - exhaust gas collector (1-for 1,2,3 cylinders, 2-for 4,5,6 cylinders)
EGP_{out} - exhaust gas passage outlet
SG+FP - engine speed governor together with fuel oil pump
T - turbine

Parameters

$\dot{m}$ - mass flow rate
p - pressure
$\dot{Q}$ - heat flow rate
δ - radial clearance
ω - angular speed
M - torque
r - set point of engine speed governor
T - temperature
φ - relative air humidity

Upper indices

E - engine
T - turbine
$( )^*$ - of swelling parameters
C - compressor
TC - turbo-compressor

Lower indices

e - of environment
i - internal
m - of mechanical losses
f - of fuel oil
in - of inlet
out - of outlet

1, 2, 2.1, 3, 3.1, 4 and 5 - control intersection numbers

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Conference

Bearing Engineering '02

On 22÷24 May this year Jastrzębia Góra, a Polish sea-coast village hosted participants of 5th Scientific–Technical Conference on Theory and Practice of Bearing Engineering. It was arranged by The Regional Group of the Utility Foundations Section, Mechanical Engineering Committee, Polish Academy of Sciences.

70 participants from diferent domestic research centres heard out and discussed on 56 papers dealing with the following topics :

- diagnostics of bearing units; monitoring methods of their state regarding ultimate states
- designing of bearing systems with special accounting for dynamic behaviour of rotary machines, deformations and displacements of bearing sockets etc.
- construction, manufacturing, assembling and exploitation of bearings
- aiding methods and means of bearing design process
- research on features of bearings, bearing materials, lubricating substances, seals of movable elements, means and techniques of regeneration of movable units
- methods and devices for research on features of bearings and bearing systems
- tribology in the range of technically dry, mixed and liquid friction
- ecological aspects of bearing engineering
- new materials and techniques in bearing engineering.