WŁADYSŁAW NOWAK, Prof.,D.Sc.,M.E.
ROMAN SOBAŃSKI, Prof.,D.Sc.,M.E.
Technical University of Szczecin
Chair of Thermal Engineering

Backgrounds of thermodynamic design calculations for living accommodations in underwater research complexes

SUMMARY

The paper presents shortly characteristics of a hyperbaric environment, which saturated divers stay in, and of thermal-flow phenomena occuring in it. Such an environment is distinctly different from the earth atmosphere.

Thermophysical properties of compressed breathing mixtures can disturb, apart from their physiological impact on the human organism, its regular functioning , as well as facilities and systems which have to assure divers' comfort and safety.

The theoretical and experimental research works, performed by the Chair of Thermal Engineering, Technical University in Szczecin and reviewed in this paper, make it possible to highlight some poorly recognized thermodynamic processes which occur in compressed breathing atmospheres. Application of the results can be helpful in carrying out further research and designing new saturated diving systems.

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INTRODUCTION

Contemporary ocean technology, particularly sea bed exploration, is almost always linked with saturated diving which makes long staying of man under water possible. During saturated diving man respires with a gas mixture of a composition and pressure relevant to a diving depth. Main components of such breathing mixtures are light gases, neutral for human organism, such as helium, hydrogen, neon or argon, apart from a physiologically indispensable amount of oxygen.

In this special hyperbaric atmosphere, filling all accommodations in which divers stay, heat and mass exchange as well as medium flow takes place in different components of the installation aimed at assuring a microclimate suitable for personnel and its safety.

A literature search made by these authors showed that selection of breathing mixture thermal parametres and technical concepts for saturated diving systems is based in most cases on simplified thermodynamic calculations and on multi-year experiments carried out in natural or simulated hyperbaric environment, in particular. A general analytical description of heat phenomena in such installations is still lacking, although its knowledge would make possible to anticipate some situations already in the design phase without any need of carrying out long-lasting and costly experiments.

Research works performed by the Chair's staff showed that it is possible to create such description on the basis of the actual thermodynamic knowledge [1,2,3,4,5,6].

In this paper selected problems are presented which ought to be taken into account while carrying out stable state balance calculations aimed at determining a temperature of comfort and an energy demand for operation of an underwater habitat, decompression chamber or diving bell.

HYPERBARIC ENVIRONMENT PROPERTIES

Behaviour of compressed real gas mixtures differ from that of their seperate components. It concerns breathing mixtures containing helium or hydrogen in particular.

The distinct and usually nonlinear relationship of some thermophysical properties of breathing mixtures to their composition and thermal parametres, not always known to designers of underwater research installations, causes many unpleasant astonishments and unpredicted troubles.

For instance, the influence of pressure and helium content on the viscosity η, thermal conductance λ, Prandtl Number Pr, and specific heat at constant pressure c_p, as well as the specific enthalpy h, of the helium-oxygen mixture at 0°C is presented in Fig. 1.

The influence of pressure, temperature and the helium content X_{He}, of the helium-oxygen mixture on its specific enthalpy h is shown in Fig. 2.

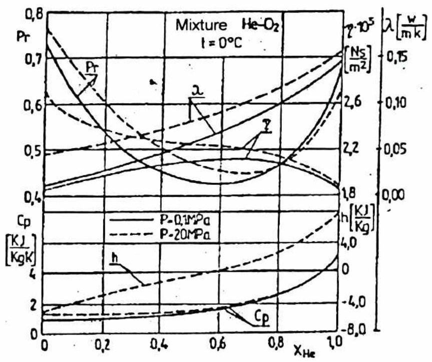

Fig.1. The influence of pressure and the helium content X_{He}, on the viscosity η, thermal conductance λ, Prandtl Number Pr, specific heat at constant pressure c_p, and the specific enthalpy h, of the helium-oxygen mixture at 0°C

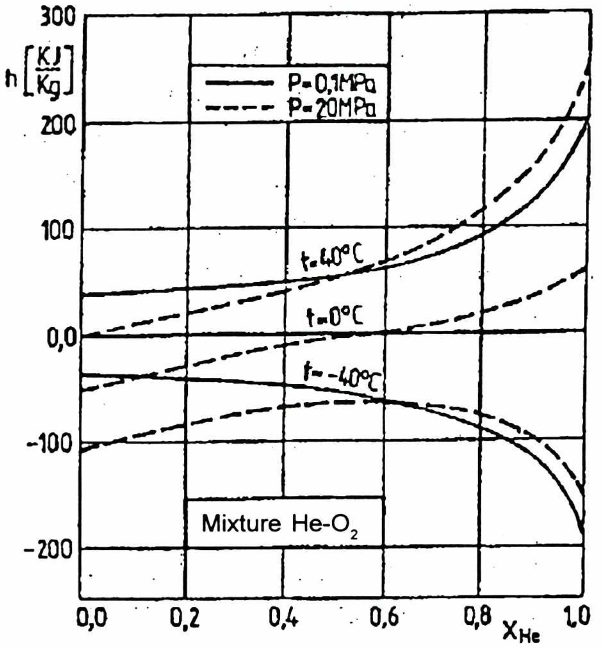

Fig. 2. The influence of pressure, temperature and the helium content X_{He}, of the helium-oxygen mixture on its specific enthalpy h

From these figures it can be observed that already a small change in gas mixture composition can influence substantially a given mixture property. An influence of pressure, at high helium content in particular, is usually small and may be neglected at a lower pressure (e.g. at P < 2 MPa) and in less exact balance calculations.

Experiments have confirmed the influence of pressure on convective heat exchange. For instance, the influence of pressure (of 0,1 ÷ 10 MPa) on Nusselt Number in forced laminar flow through a ring channel (within the range of Reynolds Number: $400 < Re < 2 \times 10^4$) is shown in Fig. 3. The influence of pressure on free convection is especially high close to the critical point, as it is indicated by the course of isobars in Fig. 4.

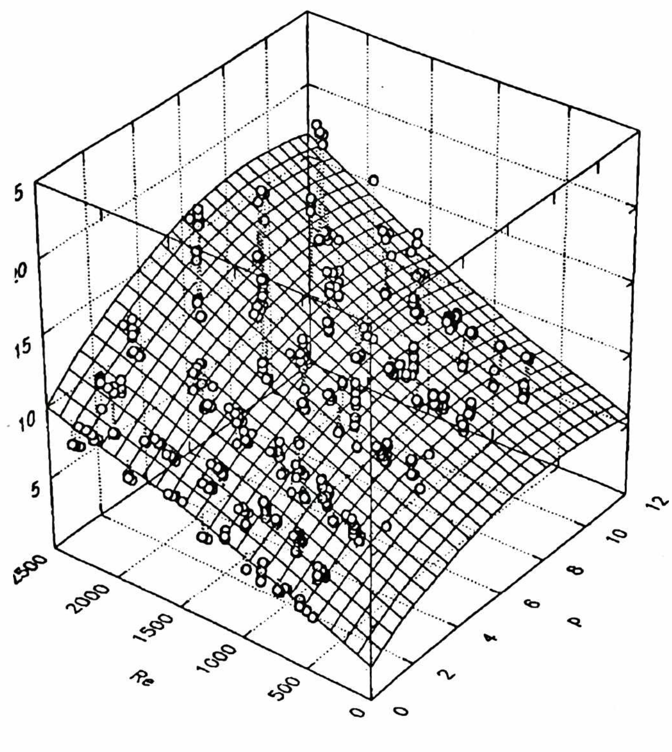

Fig. 3. The influence of pressure (of 0,1 ÷ 10 MPa) on Nusselt Number Nu, in the forced laminar flow through a ring channel (within the range of Reynolds Number: $400 < Re < 2 \times 10^4$)

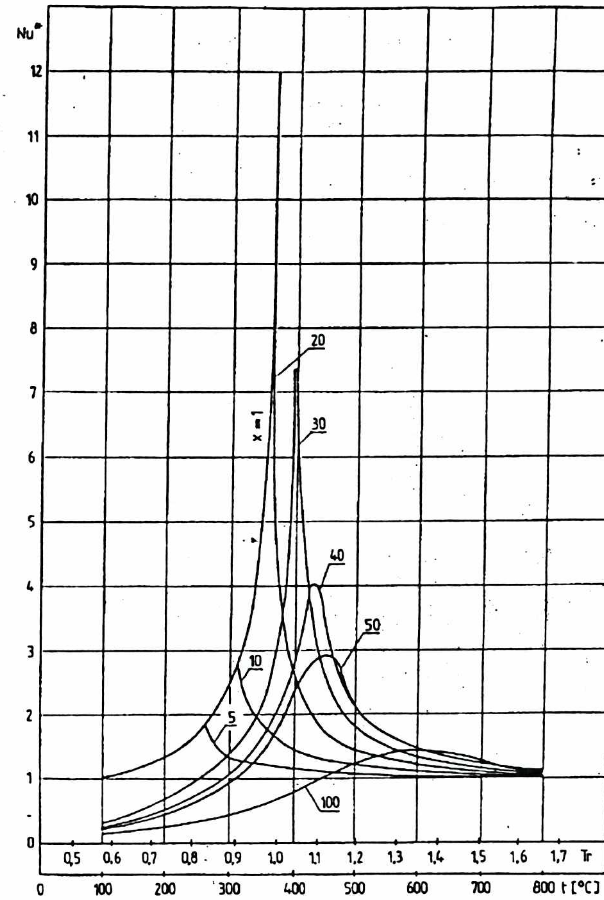

Fig. 4. Influence of pressure and temperature on the reduced Nusselt Number during the free convection close to the critical point of water

Gases for supplying breathing installations are stored in pressure cylinders and then choked to a required final pressure. Joule-Thomson's effect takes place during the choking, which changes a final gas temperature. While breathing mixtures choking the gas temperature may drop or rise depending on a gas composition. Joule-Thomson-effect neutralization heat should be also taken into account in the heat balance of an installation. The relationship of the Joule-Thomson-effect differential coefficient δ_i ,and the helium content X_{He} , pressure, and temperature of a helium-oxygen mixture is illustrated in Fig. 5.

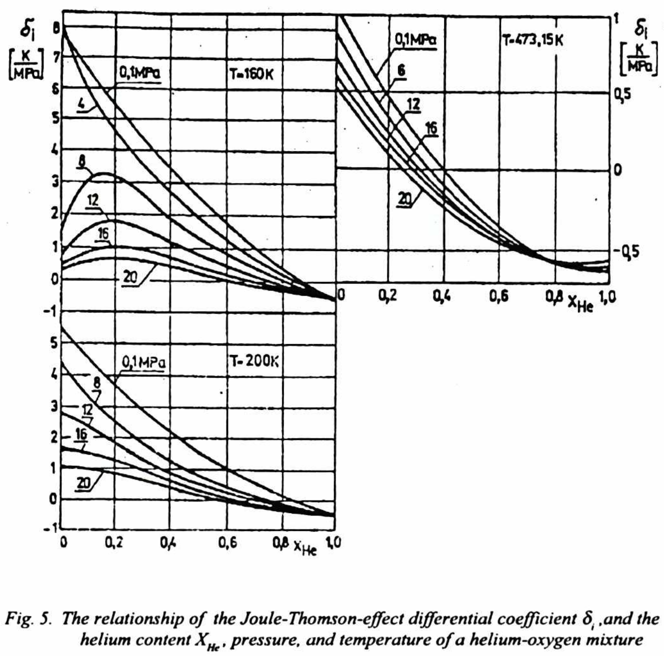

Fig. 5. The relationship of the Joule-Thomson-effect differential coefficient δ_i ,and the helium content X_{He}, pressure, and temperature of a helium-oxygen mixture

Distinct heat effects appear also while mixing compressed real gases. For instance, the mixing of helium and oxygen causes mixture temperature dropping which leads to taking heat from environment. This is a consequence of not complying with Dalton's and Amagat's laws which are valid for ideal gases, but for real gases under a low pressure only.

Calculation results of the temperature drop while adiathermically mixing helium and oxygen at constant pressure is shown in Fig. 6. They were obtained with the use of Beattie-Bridgeman's state equation which makes calculation of many thermodynamic properties of breathing mixtures (e.g. P, c_p, h, δ_i) possible with a sufficient accuracy. The calculation results have not been experimentally verified so far. However they are in a qualitative compliance with some published results of experimental research on similar mixtures (He - CO_2 , CO_2 - N_2 - H_2) as well as quite close to them quantitatively.

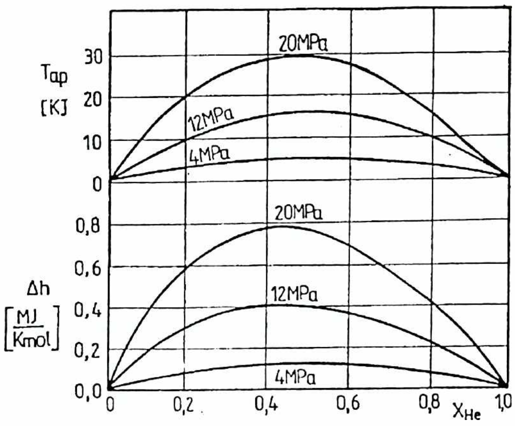

Fig. 6. Calculation results of the temperature drop during the adiathermic mixing of helium and oxygen at constant pressure

The heat of mixing may be a substantial component of system's heat balance in the case of preparing mixtures e.g inside a decompression chamber. The mixing heat of equimolar helium - oxygen mixture at P = 20 MPa and T = 273 K is of about 600 kJ/kmol.

The heat resistance of porous materials drops in an atmosphere containing helium or hydrogen. It concerns also clothing which leads in consequence to increasing heat loss out of diver's body to its environment and therefore to heat comfort deterioration. It is important to know the variations of diver's clothing heat conductivity in order to introduce relevant corrections into the equation of comfort in hyperbaric conditions.

The relationship of the relative clothing heat resistance, $\Lambda^*_{clm} = \Lambda_{clm} / \Lambda_{clp}$, the porosity ε, and the ratio λ*, of the gas mixture heat conductance λ_m and the atmospheric air heat conductance λ_p , is shown in Fig. 7. where:

Λ_{clm} - the clothing heat resistance in a breathing mixture
Λ_{clp} - the clothing heat resistance in the atmospheric air (both in the same units).

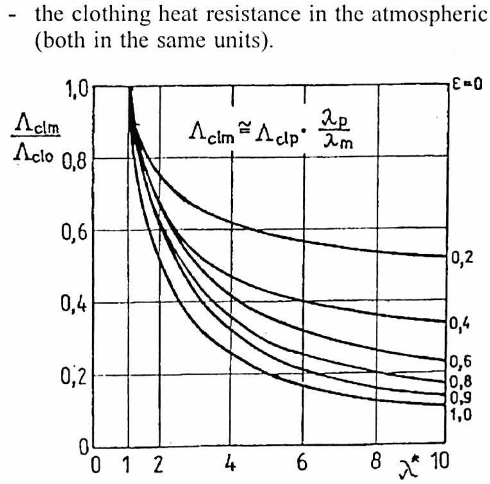

Fig.7. The relationship of the relative clothing heat resistance, $\Lambda^*_{clm} = \Lambda_{clm} / \Lambda_{clp}$, the porosity ε , and the ratio λ, of the gas mixture heat conductance λ_m and the atmospheric air heat conductance λ_p*

The results were achieved when assuming a porous body cellular model. In the calculations a fabric internal radiation, gas motion inside clothing and water vapour diffusion were omitted. The factors decrease the clothing total heat resistance. Having accounted for these simplifications, the extreme case corresponding to ε = 1 was introduced into the balance equations and the following relationship obtained:

$$\Lambda_{clm} \cong \Lambda_{clp} \frac{\lambda_p}{\lambda_m} \qquad (1)$$

It is easy to estimate a heat resistance of a given cloth in a breathing atmosphere of an arbitrary composition, if a clothing heat resistance in the atmospheric air is found from generally available tables and a heat conductance of a breathing mixture, intended to be applied, determined.

A diffusion resistance of the water vapour penetrating a usual clothing set is relatively low therefore all sweat, released to atmospheric air by man in heat comfort conditions, can evaporate. In hyperbaric conditions the sweat evaporation rate decreases along with rising pressure and the simultaneous rise of clothing diffusion resistance. The water evaporation rate in an environment of compressed breathing mixtures is 5 to 10 times lower than that in the atmospheric air, as it results from Webb's investigations. This is confirmed by the calculations and experiments carried out in The Chair of Thermal Engineering, results of which are presented in Fig. 8.

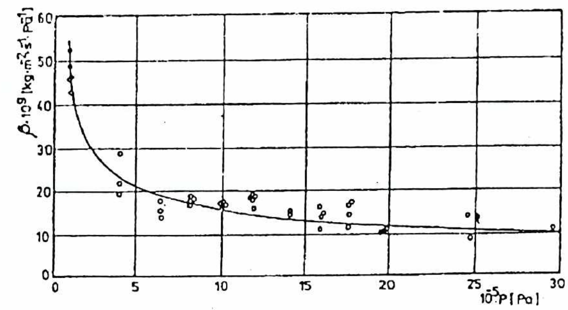

Fig. 8. Pressure influence on the water evaporation coefficient h (out of free surface to gas atmosphere)

The shortly presented properties of a hyperbaric environment in which divers stay differ substantially from those of natural atmosphere. The thermophysical properties of compressed breathing mixtures can disturb the regular functioning of human organism, apart from their physiological impacts on it, as well as of facilities and systems which have to assure comfort and safety of divers.

EQUATION OF COMFORT IN HYPERBARIC ACCOMMODATIONS

A heat comfort equation and general form of heat balance equation for an arbitrary hyperbaric accomodation for divers (underwater habitat, decompression chamber or diver's bell) was elaborated, accounting for the breathing mixture properties into particular items of heat balance in steady conditions.Comfort equation in a form proposed by Fanger was used in the calculations:

$$\dot{Q} - \left(\dot{Q}_0 + \dot{Q}_w\right) = \dot{Q}_p = \dot{Q}_r + \dot{Q}_k \qquad (2)$$

where:

$\dot{Q}$ - the heat flux transferred to environment by diver's organism, [W]
$\dot{Q}_0$ - the heat flux transferred to environment while breathing, [W]
$\dot{Q}_w$ - the heat flux transferred to environment during sweat evaporation on body surface,[W]
$\dot{Q}_p$ - the heat flux conducted by clothing, [W]
$\dot{Q}_k$ - the heat flux taken out of body surface due to convection, [W]
$\dot{Q}_r$ - the heat flux radiated to environment, [W]

The comfort equation, which contains the most important environment physical parametres ($P_m, \varphi, t_{mr}, w, X_i$) and those of clothing type and diver's motion activity (Λ_{clm}, q_M), makes it possible to calculate a comfort temperature of a gas atmosphere for saturated deep diving.

A comparison of the calculated comfort temperature with that determined experimentally for the same environment reveals good consistency of both values (e.g. t_{eksp}=30,8°C, t_{obl}=31,2°C) in spite of the comfort state felt subjectively by man. The results indirectly confirm correctness of the assumptions introduced while establishing the comfort equation. Information on the values and variabilities of particular heat fluxes being transferred to environment can be useful for physiologists. A typical heat comfort diagram is presented in Fig. 9.

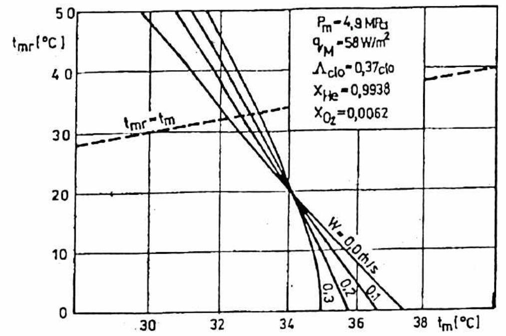

Fig. 9. Influence of the relative gas velocity w and the mean radiation temperature t_{mr} *on the comfort temperature* t_m *; the dash line relates to the case when* $t_m = t_{mr}$

When analyzing particular fluxes of the energy transferred to or out of a system, a heat balance equation for the system, e.g. a decompression chamber shown in Fig. 10, can be written as follows:

$$\sum_{i=1}^{n} \dot{H}_i + \dot{Q}_p + N\dot{Q} + \sum_{r=1}^{R} \dot{Q}_{jr} + \dot{H}_{m1} = \dot{H}_{m2} + \sum_{k=1}^{K} \dot{Q}_{Kk} \qquad (3)$$

where:

$\dot{Q}_p$ - the heat flux required to maintain comfort temperature.(Mixing heat and Joule-Thomson's effect neutralization heat is accounted for in this item.)

$N\dot{Q}$ - the heat flux transferred to environment by N diver's organisms staying in the chamber

$\sum_{r=1}^{R} \dot{Q}_{jr}$ - the heat flux transferred out from R devices installed in the chamber (lights, ovens, instruments etc)

$\sum_{i=1}^{n} \dot{H}_i$ - the entalphy flux of a fresh, n - component gas mixture supplied to the chamber

$\dot{H}_{m1}$, $\dot{H}_{m2}$ - the entalphy fluxes of the gas mixture supplied to and transferred from aregenerative system

$\sum_{k=1}^{K} \dot{Q}_{Kk}$ - the heat flux lost through K chamber items: casing, hatchways and portholes

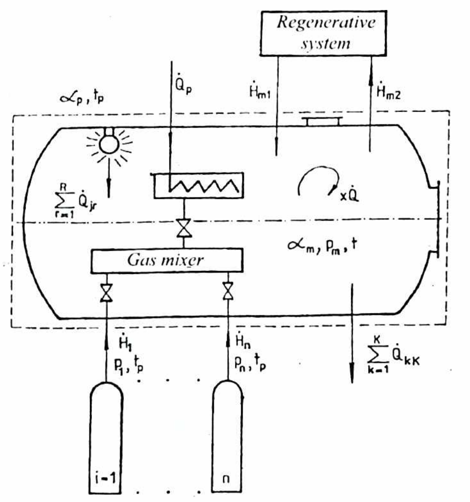

Fig. 10. Scheme of a decompression chamber with the marked balance screen and particular energy balance items

A computer software package was elaborated to solve effectively the confounding balance equations. It has been proved to be very useful in design and operation of saturated diving installations.

HEAT BALANCE OF DIVING BELL

A way of determining energy demanded for saturated diving is presented below using a diving bell as an example.

A quantity of electric energy which should be delivered to the diving bell can be determined on the basis of the stable state balance equation of the bell and divers under the conditions that one diver is inside the bell of 15°C temperature, but two divers are outside it, in the sea water of 0°C temperature and 37 promille salinity. The divers staying outside the bell are warmed with the use of heating water being in a closed circulation. The water, heated in a heater installed outside the bell, is delivered to a diver through one pipe and after being cooled is sent through the second pipe back to the heater. Each pipe is 25 m long. A breathing mixture delivered to the divers staying outside the bell is heated also in the heater to such a temperature to ensure an inlet temperature to a diver not lower than 20°C. Pipes delivering the breathing mixture are also 25 m long. The influence of a breathing mixture expansion in gas cyliders and that of choking in pressure reducing valves on mixture's temperature was neglected in calculations. The mixture temperature was assumed constant and close to that of the surrounding sea water. The electric energy required to drive bell's equipment and to carry out underwater works was also not accounted for.

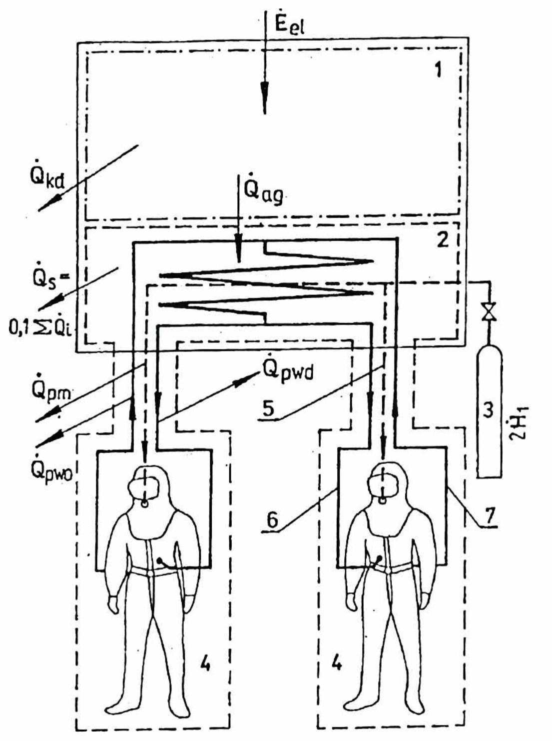

Fig. 11. Scheme of a diving bell with two divers outside the bell, the marked balance screens and particular energy balance items

-·-·- A heat balance barrier enclosing the diving bell interior without heater

- - - - A heat balance barrier enclosing the heater and two divers outside the bell

1 - diving bell, 2 - water heater, 3 - cylinders with breathing mixture, 4 - diver, 5 - mixture delivery pipe, 6 - heating water delivery pipe, 7 - heating water drain-off pipe

The electric energy delivered to the heater increases the entalphy of the breathing mixture supplied from the gas cylinder and covers the following loss items (Fig. 11.):

- the heat losses of two divers staying outside the bell
- the heat losses of the pipes of heating water circulating in the closed heater-divers system
- the heat losses in the pipes delivering the breathing mixture to the divers
- the heat losses of the heater itself (assumed 10% of all balance items)

Applying time related balance equation for a stable state:

$$\dot{E}_d = \dot{E}_w \tag{4}$$

to the problem in question a balance equation of a system containing a heater and two divers can be written as follows:

$$\dot{Q}_{ag} = 2\times 1,1\left[\dot{Q}_{ks} + \dot{Q}_{pm} + \dot{Q}_{pwd} + \dot{Q}_{pwo} + \left(\dot{H}_3 - \dot{H}_1\right) - \dot{Q}_{wn}\right] \tag{5}$$

where:

- $\dot{Q}_{ks}$ - the heat losses in the diving dress
- $\dot{Q}_{pm}$ - the heat losses in the pipe delivering breathing mixture to the diver
- $\dot{Q}_{pwd}$ - the heat losses in the pipe delivering hot water to the diving dress
- $\dot{Q}_{pwo}$ - the heat losses in the pipe draining warm water off the diving dress
- $\dot{Q}_{wn}$ - the heat released to environment by the organism of one diver staying outside the diving bell
- $\dot{H}_1$ - the enthalpy of the breathing mixture delivered to the diving bell
- $\dot{H}_3$ - the enthalpy of the wet breathing mixture drained off the diving dress

All the quantities are expressed in [W].

The electric energy amount delivered to the diving bell can be determined by using the following diving bell balance equation:

$$\dot{E}_d = \dot{Q}_{ag} + \dot{Q}_{kd} - \dot{Q}_w + \left(\dot{H}_3 - \dot{H}_1\right) \tag{6}$$

where:

- $\dot{Q}_{ag}$ - the electric energy demand of the diving bell
- $\dot{Q}_{kd}$ - the heat emitted from the diving bell to the surrounding sea water
- $\dot{Q}_w$ - the heat released to environment by the organism of the diver staying in the diving bell

After substitution of (5) to (6) the energy delivered to the diving bell is expressed as follows :

$$\dot{E}_d = 2\times 1,1\left[\dot{Q}_{ks} + \dot{Q}_{pm} + \dot{Q}_{pwd} + \dot{Q}_{pwo} + \left(\dot{H}_3 - \dot{H}_1\right) - \dot{Q}_{wn}\right] + \dot{Q}_{kd} - \dot{Q}_w + \left(\dot{H}_3 - \dot{H}_1\right) \tag{7}$$

A calculation procedure of the particular heat balance components is given in [1] where known expessions for determining quantity of gas mixture entalphy, of heat penetrating multi-layer bulkheads as well as of penetrating heat in double - medium recuperators are used. In [1] a way of determining physical and thermal properties of breathing mixtures, which appear in the above mentioned expressions, is also given.

Total electric energy demand of a diving bell $\dot{E}_{el}$ [W], in some selected conditions

[W]	P=2,55 MPa				P= 1,57 MPa			
	0_2 =0,065 He=0,785 N_2= 0,15			0_2=0,065 He=0,935	0_2=0,106 He= 0,644 N_2=0,25			0_2=0,106 He=0,894
	t_{wg}=35°C Δt = 1K	t_{wg}=40°C Δt = 6K	t_{wg}=41°C Δt = 4K	t_{wg}=41°C Δt = 4K	t_{wg}=35°C Δt = 1K	t_{wg}=40°C Δt = 6K	t_{wg}=41°C Δt = 4K	t_{wg}=41°C Δt = 4K
$\dot{Q}_{ks} - \dot{Q}_{wn}$	1720	2032	2090	2744	1334	1569	1616	2520
$\dot{Q}_{pm}$	2226	2226	2226	2392	4202	4202	4202	5113
$\dot{Q}_{pwd}$	1524	1911	1882	1871	1527	1940	1898	1876
$\dot{Q}_{pwo}$	1452	1478	1593	1607	1446	1451	1574	1600
$\dot{H}_3 - \dot{H}_1$	781	781	781	736	539	539	536	496
$\dot{Q}_{ag}$	16947	18542	18858	20570	19906	21342	21617	25531
$\dot{Q}_{kd} - \dot{Q}_w$	1075	1075	1075	1123	865	865	865	831
$\dot{E}_{el}$	18803	20398	20714	22429	21310	22746	23018	26858

In table, the total electric energy demand of a diving bell, $\dot{E}_{el}$, is exemplified where all basic balance components for the pressure values 1,57 MPa and 2,55 MPa, with two - and three - component gas mixtures taken into account, are specified too. In extreme cases the total electric energy demand reaches the following values:

- for two-component gas mixtures under 1,57 MPa pressure:

$$\dot{E}_{el\max} = 26\ 858\ \ W$$

- for a three- component gas mixture under 2,55 MPa pressure:

$$\dot{E}_{el\min} = 18\ 803\ \ W$$

Heat losses in the pipe draining hot water off the diving dress, $\dot{Q}_{pwo}$, vary in a rather narrow range from 1451 W to 1607 W. Heat losses in the pipe delivering hot water to the diving dress are more diversified and vary from about 1525 W for 35°C mean temperature of the water in the diving dress to about 1940 W for 40°C mean temperature.

Heat losses in the pipe delivering breathing mixture are most diversified; the lowest values of them are at the higher pressure 2226 W. Such losses grow significantly up to 5113 W at 1,57 MPa when changing the gas mixture composition and lowering pressure. At 1,08 MPa and 0,59 MPa pressure the losses reach 60 kW and the relevant gas temperature make it impossible to put this variant into practice.

For this reason an earlier proposed concept to heat breathing mixture in a heater placed within the diving bell has been abandoned at all. A solution which was applied is to heat breathing mixture just close to a diver by supplying hot water of an appropriately higher temperature. A heat exchange scheme for a system composed of a gas mixture heater and a diver is shown in Fig. 12.

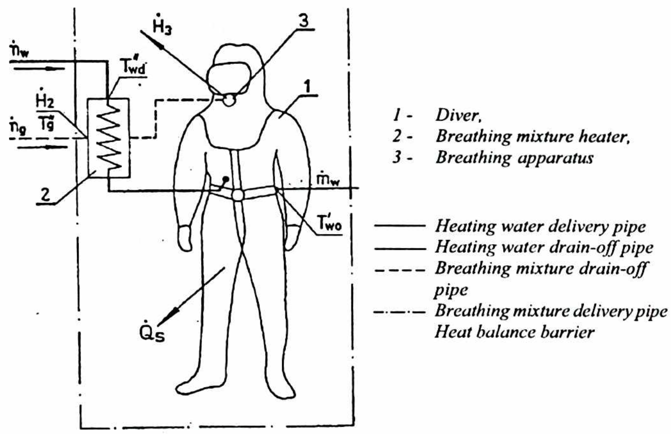

Fig. 12. Heat exchange scheme for a system composed of a gas mixture heater and a diver

In the diving bell, losses of the heat transferred to sea water are lower at low diving depths (618 W) and higher at large diving depths (1336 W) and are of the same order for two- and three - component gas mixtures.

Electric energy of an amount determined by (4), which covers all the system's losses, is delivered to the diving bell in order to maintain constant temperature inside the bell and to ensure suitable conditions to divers staying outside the bell.

Lowering or stopping electric energy supply will cause values of system's thermal parametres to differ from those assumed. The following considerations are made to determine changes of temperature inside the bell in the function of time in the case of a break-down stopping of the electric energy supply. It is then assumed that a bell would partly emerge over water surface causing heat exchange conditions to change. This is a rather complex task to establish exactly such a relationship. A simplified way of solving the problem under many simplifying assumptions and with the use of a simple analytical model is given in [1].

Calculation results of the temperature changes inside the bell in the function of time and for various values of the insulation thickness δ_i, are presented in Fig. 13.

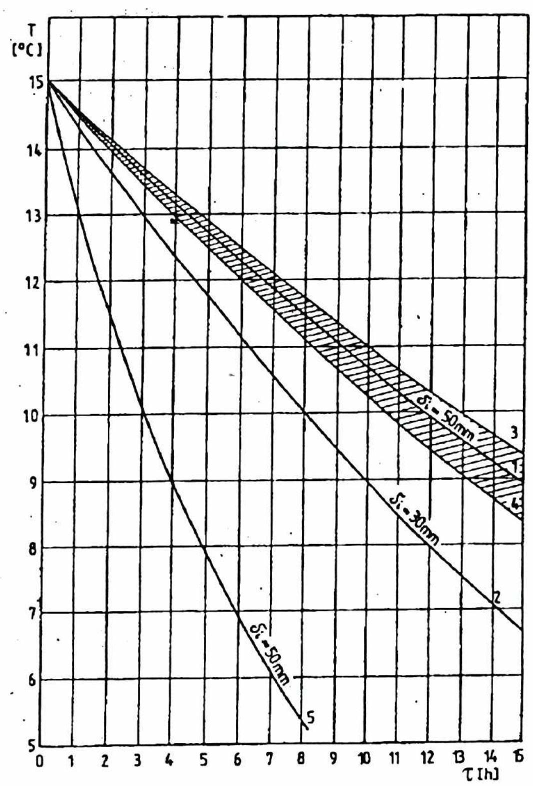

Fig. 13. Temperature changes inside the bell in the function of time for various values of the insulation thickness δ_i, occuring while the electric supply breaking-down

A difference between the temperatures calculated for two insulation thicknesses, 30 mm and 50 mm, grows with time and at τ=15 h reaches 25,2 % in respect to the temperature for 50 mm insulation thickness and when assuming ±10% error in determination of the heat capacities W and W_s (the curves 3 and 4). Temperature differences at τ=15 h, calculated in respect to the correctly determined heat capacity values, are 5,2% and -6,2% respectively. The curve 5 illustrates a temperature change inside the diving bell with non-insulated hatches and portholes.

BREATHING MIXTURES FLOW BETWEEN FINITE VOLUME TANKS

Real thermodynamic processes, which occur in particular system's components during emptying a decompression chamber with simultaneous draining-off breathing mixture through piping to an intermediate tank, are very complex. That is why their exact mathematical description is difficult and modelling of them requires some simplifying assumptions.

Solutions of various cases, comprising mathematical description, are given in [1]. These are solutions of:

- ☐ thermodynamic processes occuring in particular tanks, described with the use of energy balance, mass balance as well as thermal state equations
- ☐ fluid flow in piping, described with the use of equations of motion, of mass flow continuity of energy balance as well as thermal state equation

The following simplifying assumptions were adopted to solve the problems:

- flows occuring between two finite volume tanks which are linked thogether by a pipe of constant cross-section are considered with flow and local resistance taken into account
- isothermal process occurs in the first tank during decompression, due to continuous delivering of an appropriate amount of heat, but adiathermal one in the second tank during its filling (i.e. without any heat exchange with environment). This assumption is acceptable in spite of that the second tank is not insulated, because of a short time of filling the tank.
- breathing mixtures, flow of which between the tanks is considered, are assumed to be homogeneous gases complying with the conditions for ideal ones; they are further called „fluids"
- only flows which occur after propagation of pressure wave through piping connecting the tanks are considered
- transient flows, which can occur due to pressure changes in both tanks, are assumed quasi-steady ones which can be described by steady-state flow equations, viz. by substituting a transient fluid flow by a series of momentary consecutive steady-state flows

Pressure losses due to friction and local drag should be accounted for while considering a medium flow through a pipe between tanks. A model appropriate to a given length of the pipe can be selected for the flow calculations. A flow process at high velocities, occuring in very short pipes of neglectable friction losses, can be assumed adiathermal. In this case relationships valid for mass flux during adiathermal expansion can be applied together with a relevant flow rate value selected with the flow local drag accounted for.

A flow process in very long pipes and with low velocities can be sometimes assumed isothermal. This is justified only if heat exchange with environment is of such intensity that a medium's temperature is able to reach that of the environment. Such models can be used to calculate flow time extreme values only, because they were established for flow conditions highly different from the real ones.

In further considerations, real drag of flow and local drag was accounted for under the assumption that there is no heat exchange with environment, i.e. that flow through a pipe can be approximated by the irreversible adiathermal process. A flow process may be considered under these conditions as polytropic expansion. A polytropy exponent value can be determined with the use of a relevant relationship and all losses in the pipe in question taken into account, together with that due to flux contraction and then expansion.

A mass flux value during flow between two infinite volume tanks can be calculated by using the following expressions:

$$\dot{M} = A\frac{P_1}{RT_1}\sqrt{\frac{2\kappa}{\kappa-1}RT_1}\sqrt{\left(\frac{P_2}{P_1}\right)^{\frac{2}{z}} - \left(\frac{P_2}{P_1}\right)^{\frac{z+1}{z}}} \tag{8}$$

which is valid for subcritical flows, i.e. when:

$$\frac{P_2}{P_1} \geq \left(\frac{2}{z+1}\right)^{\frac{z}{z-1}} \tag{9}$$

and

$$\dot{M}_{\max} = A\frac{P_1}{RT_1}\sqrt{\frac{2\kappa}{\kappa-1}RT_1}\sqrt{\left(\frac{2}{z+1}\right)^{\frac{2}{z-1}}\frac{z-1}{z+1}} \tag{10}$$

which is valid for supercritical flows, i.e. when:

$$\frac{P_2}{P_1} \leq \left(\frac{2}{z+1}\right)^{\frac{z}{z-1}} \tag{11}$$

The expressions which are valid for a fluid flow between two infinite volume tanks can be adjusted to a fluid flow between two tanks of finite volumes. For this purpose a thermal state equation is applied to two consecutive flow states in both tanks as well as an adiathermal equation to filling the second tank. The latter equation takes the following form under the above adopted conditions:

$$\frac{dP_2}{P_2} = \frac{dT_2}{T_2\left(1 - \dfrac{T_2}{\kappa T_d + \dfrac{w_d^2}{2c_v}}\right)} \tag{12}$$

This is the most general equation of transformation which can ocurr in a adiathermally filled tank with a fluid kinetic energy at tank inlet taken into account. A detailed analysis of various flow cases can be found in [1].

The expression (12) can be written, in the case of adiathermal fluid flow through a piping between two tanks when assuming isothermal transformation occurence in the first tank, i.e. when $T_1 = T_{10} = idem$, and the flow velocity at pipe inlet $w_1 = 0$, and with an energy balance equation of the piping taken into account, as follows:

$$\frac{P_2\Theta_0}{P_{20}\Theta} = \frac{\kappa - \Theta_0}{\kappa - \Theta} \qquad (13)$$

where:

$$\Theta = \frac{T_2}{T_{10}} \qquad \Theta_0 = \frac{T_{20}}{T_{10}}$$

and a final pressure in the second tank can be determined by using :

$$P_{2K} = P_{20} + \kappa\frac{V_1}{V_2}(P_{10} - P_{1K}) \qquad (14)$$

A tank emptying time can be calculated with the use of the above given relations provided a flow is considered as polytropic expansion.

If a quasi-stable, supercritical polytropic flow is assumed, an outflow time, which has to pass to change pressure in tank from P_{10} to $P_{1K} = P_{11}$, can be determined with the use of the following expression:

$$\tau = \frac{V_1}{AB\phi_k}\ln\frac{P_{10}}{P_{11}} \qquad (15)$$

where:

$$B = \sqrt{\frac{2\kappa}{\kappa - 1}RT_1} \qquad \phi_k = \sqrt{\left(\frac{2}{z+1}\right)^{\frac{2}{z-1}}\frac{z-1}{z+1}}$$

If pressure in a tank being filled reaches and then exceeds a critical value, a subcritical flow will occur. If a quasi-stable subcritical polytropic flow is assumed, an outflow time is determined by the following expression:

$$\tau = \frac{1}{AB}\int_{P_{11}}^{P_{12}}\frac{dP_1}{P_1\phi\left(\frac{P_2}{P_1}\right)} \qquad (16)$$

where:

$$\phi = \sqrt{\left(\frac{P_2}{P_1}\right)^{\frac{2}{z}} - \left(\frac{P_2}{P_1}\right)^{\frac{z+1}{z}}}$$

which can be integrated with the use of numerical methods only and with (14) applied to express dependence of P_2 on P_1.

Pressure equalization in both tanks may happen after subcritical flow. A final pressure value can be calculated by applying (14), having substituted $P_{1K} = P_{2K}$.

FINAL REMARKS

The presented characteristics of the hyperbaric environment, which saturated divers stay in, and of thermal-flow phenomena occuring in it indicate that the environment is distinctly different from the earth atmosphere.

Thermophysical properties of compressed breathing mixtures can disturb the regular functioning of human organism, apart from their physiological impacts on it, as well as of facilities and systems which have to assure comfort and safety of divers.

The theoretical and experimental research works, performed by the Chair of Thermal Engineering, Technical University in Szczecin, make it possible to highlight some poorly recognized thermodynamic processes which occur in compressed breathing atmospheres.

The works contain also some guidelines which can make carrying out further investigations and designing new saturated diving systems easier.

NOMENCLATURE

A	pipe cross-section net area, m^2		
c_p	specific heat at constant pressure, J/(K kg)		
h	specific enthalpy, J/kg	$\dot{H}$	medium flux enthalpy, W
R	gas constant (individual), J/(K kg)		
l	linear dimension, m	m	mass, kg
$\dot{M}$	mass flux in sub-critical flows, kg/s		
$\dot{M}_{max}$	max. mass flux in super-critical flows, kg/s		
P	pressure, Pa	$\dot{Q}$	heat flux, W
q_M	metabolic heat flux density, W/m^2	q	heat flux density, W/m^2
T	absolute temperature, K		
T_r	relative temperature T/T_K	t	temperature, °C
t_{mr}	radiation mean temperature, °C	t_m	mixture temperature, °C
W	heat capacity of diving bell interior, J/K		
W_s	heat capacity of bell's steel plating, J/K	$\dot{W}$	flux heat capacity, W/K
X	gas wetness level, kg H_2O/kg (of dry gas)		
w	mixture relative velocity, m/s		
X_i	molar content of ith gas component	x	steam dryness level
$\dot{E}_d$	energy delivered to a system, W		
$\dot{E}_w$	energy derived from a system, W		
z	polytropic transformation exponent		
α	surface film conductance, W/(K m^2)		
β	water evaporation coefficient (off free surface to gas atmosphere), kg/(m^2 s Pa)		
δ	insulation thickness, m		
δ_i	differential coefficient of Joule-Thomson's effect, K/Pa		
ε	porosity		
η	dynamic viscosity coefficient, kg/(m s)		
Θ	dimensionless temperature	κ	ideal gas isentropic exponent
λ	heat conductance, W/(m K)		
λ_m	breathing mixture heat conductance, W/(m K)		
λ_p	atmospheric air heat conductance, W/(m K)		
Λ_{clm}	clothing heat conductance in a breathing mixture, clo		
Λ_{clp}	clothing heat conductance in the atmospheric air, clo		
Λ^{*}_{clm}	relative clothing heat conductance $\frac{\Lambda_{clm}}{\Lambda_{clp}}$		
ν	kinematic viscosity coefficient, m^2/s	τ	time, s
φ	relative humidity, %		

CRITERION NUMBERS

Nu	Nusselt Number $\frac{\alpha l}{\lambda}$		
Nu^{*}	relative Nusselt Number at constant temperature $\frac{Nu_p}{Nu_a}$		
Nu_p	Nusselt Number at the pressure P		
Nu_a	Nusselt Number at the atmospheric pressure		
Re	Reynolds Number $\frac{wl}{\nu}$	Pr	Prandtl Number $\frac{c_p\eta}{\lambda}$

BIBLIOGRAPHY

1. Nowak. W. (coordinator) et al.:„Studium wybranych zagadnień dotyczących przepływu mieszanek gazowych, wymiany ciepła i klimatyzacji w systemach prac podwodnych". Opracowanie Zakładu Techniki Cieplnej IBM, Szczecin, 1977÷1978 (Internal publ. of Thermal Engineering Division, Machine Construction Institute, Technical University of Szczecin)
2. Sobański R.: „Termodynamika mieszanin oddechowych i ich własności cieplne". Prace Naukowe P.S., Nr 213, Szczecin, 1982 (Scientific Publications of Technical University of Szczecin)
3. Sobański R.(coord.) et al.: „ Badania problemów cieplno-fizycznych mieszanin oddechowych dla nurków saturowanych ". Opracowanie KTC Politechniki Szczecińskiej, Szczecin 1985 (Int. publ. of Chair of Thermal Engineering , Technical University of Szczecin)

4. Nowak.W. (coord.) et al.: „ Zjawiska cieplno-fizyczne zachodzące w mieszankach oddechowych w środowiskach hiperbarycznych". Opracowanie KTC P.S., Szczecin, 1986÷1990 (Int. publ. of Chair of Thermal Engineering, Technical University of Szczecin)
5. Nowak.W. (coord.) et al.: „ Konwekcja swobodna, wymuszona w hiperbarycznych atmosferach, badanie właściwości cieplno-fizycznych mieszanin oddechowych oraz odparowania cieczy w atmosferach o podwyższonym ciśnieniu ". Opracowanie KTC P.S., Szczecin, 1991÷1992 (Int. publ. of Chair of Thermal Engineering, Technical University of Szczecin)
6. Sobański R.(coord.) et al.: „ Badania właściwości cieplno-fizycznych gazów oraz konwekcyjnej wymiany ciepła w sprężonych mieszaninach gazowych". Opracowanie KTC P.S., Szczecin, 1987÷1990 (Int. publ. of Chair of Thermal Engineering, Technical University of Szczecin)
7. Stachel A.: „ Wpływ ciśnienia na warunki wymiany ciepła w zakresie wymuszonego ruchu laminarnego i przejściowego w kanale pierścieniowym ". Praca doktorska, Wydział Mechaniczny Politechniki Szczecińskiej, Szczecin, 1993 (Doctor's Thesis, int. publ. of Mechanical Faculty, Technical University of Szczecin)
8. Kozak T., Majchrzycka A.: „ Ventilation of the hyperbaric chambers: ventilation for moisture removal ". International Scientific Conference on „Indoor Air Quality-From Science to Practice", Warsaw, 1993

Miscellanea

35 years of the Central Maritime Museum in Gdańsk

The highest respect should be paid to all those persons, both employees and volunteers who together with the official efforts contributed to the achievement of the today rank and shape of the Central Maritime Museum. In the days of 1960, when the museum was established as an autonomous department of the then Pomeranian Museum, probably nobody expected the plentiful achievements in developing museum activities reached during the past 35 years.

Some years later already the number and value of the collected exponates and dynamic organizational and exhibitional activity of the department well exceeded its formal status, therefore it was transformed first into a self-contained cultural institution and then, in 1970, into the Central Maritime Museum (CMM).

The very beginnings of the museum are bound with the famous Gdańsk Crane the rebuilt compartments of which were the museum site. During the next 25 years CMM's position grows and grows both in this country and abroad. Indeed, many specialty and familiarizing exhibitions, conferences and meetings as well as worthy publications and scientific reports can be found in the history of museum's development. In the field of underwater archeology museum's own research section gained great success. The same can be said about the results of activity of the archive section documenting Polish maritime economy achievements in a form e.g. of the Polish man-of-sea archive or the Polish ship design archive. CMM carries out a very vivid and fruitful cooperation with the foreign maritime museums (especially those located around the Baltic Sea), which is respectfully acknowledged by its partners.

Simultaneously CMM still dynamically develops its organization. Today CMM it is not only a complex of the museum buildings which are the rebuilt old grain stores located along the Motława river bank, but also a huge institution comprising the Vistula Museum in Tczew, the Fishery Museum at Hel, the museum-ship SOŁDEK and the famous tall-sail training ship DAR POMORZA which is of a high merit in building maritime culture and awareness of Poles.

While speaking about CMM history a great tribute should be paid to the late prof. Przemysław Smolarek due to his exceptional pioneer contributions in creating and developing the museum.

Current *reports*

TECHNICAL UNIVERSITY OF GDAŃSK
MECHANICAL FACULTY
CHAIR OF COMBUSTION ENGINES AND COMPRESSORS

Data base for determining real operation conditions of the ship power plant machinery and equipment

In the designing of various ship power plants a great, often decisive role play such factors as: traditionalism, application of the „similar ship" approach, shipowner's pressure, designer's tendency to secure oneself against responsibility, some stiff requirements of classification societies or a yielding response to shipyard's expectations, but above all it is low knowledge of real conditions in which the designed ship has to operate in the future.

Therefore some contradiction can be observed between the very advanced calculation methods, complex laboratory tests and engineering processes applied in designing and building particular machines and installations and a rather crude approach used while collecting the machines and devices into a complete power plant system which finally appears not very modern. Traditional methods of selecting power plant machines and equipment components are based on deterministic models and on the assumed operation conditions which never appear simultaneously.

To improve that improper state of the matter a research project was undertaken by the Chair of Combustion Engines and Compressors, Mechanical Faculty, Technical University of Gdańsk, aimed at elaboration of scope and organization foredesign for the collecting and methodical processing of the data which determine real operation conditions of machines and equipment installed in ship power plants of various types.

For this purpose collecting the data from ships in service was based, due to practical matters, on the characteristics and parameters available from such standard shipboard records and documents as :

- ship's log books and reports
- reports on non-periodical tests and measurements
- production test reports
- „as-built"-ship technical documentation
- classification technical documentation
- guaranty engineer's reports
- reports on research and observations of ships in service
- students' shipboard training reports

Investigations of the ship machinery and equipment operation characteristics are concerned a.o. with the following:

- operation states' structure and balance
- operation loadings on the propulsion system components and electric generating sets
- fresh water supply economy
- real operation time periods of the machinery in various operation states
- loadings and operation times of the shipboard processing machinery
- atmospheric and marine environment conditions
- characteristics of the most important working media, e.g. fuel and lubricating oils

The commercial computer software FOX Pro 2.5 PL was used to build the data bank. Data set formats of dbf type with the assigned names: „ships", „ watches", „ manoeuvres „ were created by using the software module.

Data processing can be carried out with the use of the calculation sheets and standard statistical programs. In this way real operation characteristics, important correlation factors and real values of various design coefficients can be determined to make possible, a.o.:

- selection of the proper contracting parameters of the main engine and auxiliary engines
- selection of the proper number and operation characteristics of heat exchangers and tanks
- determination of the real value of power plant efficiency
- modelling of the noxious substances content in exhaust gases emitted from ships.

The project has been financially supported by The Scientific Research Committee and Andrzej Balcerski, Assoc. Prof., is the research team leader.