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Modelling of defects in a serial system of axial compressors

SUMMARY

A modelling method of the axial compressor that operates within marine gas turbine configuration is presented. The method makes it possible to determine an arbitrary working point on the compressor characteristics.

A concept is described of modelling the characteristics of the turbine engine axial compressors which work in series, with a given character of inter-blade gas channel fouling accounted for. The determined characteristics are suitable to carry out numerical simulation experiments by means of computers.

INTRODUCTION

One of the main operation problems of the marine gas turbine engines is to systematically evaluate the technical conditions of their gas passages in service. A set of technical structural features of the passages are simultaneously determined. The magnitudes, subject to changes during engine operation, are characterized by adequate changes of the measured thermogasdynamic parameters of the working medium flow. They result from the inevitable and continuous fouling and wear processes of gas path structural elements, which are always linked with engine operation in seaway conditions. In this case an important difficulty is to adequately interprete the results of experimental investigations, in order to provide a reliable diagnosis. It is possible to determine the so called operational tolerance area for each of the selected diagnostic parameters if a sufficiently large set of possible technical states of the gas passage is at disposal. In consequence many homogeneous engines should be examined during a sufficiently long operation time. A particular problem is to determine the extreme (failure) states as well as to account for individual features of each engine, impossible to be experimentally assessed by examining the real objects.

The state of affairs provokes looking for more and more effective, computer based diagnostic instruments. The technical state identification of the complex dynamic objects on the basis of computer simulation is a new trend of developing the technical diagnostics, called the simulation diagnostics. The ***defects***, in the sense of hypothetical or even extreme changes of the construction parameters of engine flow passages, generate, if appropriately modelled, a characteristic course of the energy processes which proceed in the engine. The ***symptoms*** of the simulated defects are obtained in this way. The searched diagnostic functions of ***defect-symptom*** relationships can be elaborated on the basis of simulation results and applied to assess a technical state of the selected functional modules of the engine flow passage. Obtaining such information by means of an active experiment is labour-consuming, expensive and, in the case of engines in service - quite impossible.

The paper presents a modelling method of the characteristics of turbine engine axial compressors arranged in the serial system, with an assumed character of inter-blade gas channel fouling. The elaborated mathematical model of the co-working compressors is the basis of a computer program for numerical simulation of turbine engine dynamic processes.

DEFECTS OF COMPRESSOR FLOW PASSAGE ELEMENTS

While operating the marine gas turbine, especially in heavy weather conditions, different substances contained in the sucked -in working medium, can be admitted to its flow part. The air which flows into the engine is really an aerosol in which solid and liquid particles of land and sea origin are contained. The sea water droplets together with salt particles solved in them penetrate at first the compressor interior (of compressor serial system) and intensely evaporate.

It reveals that the highest intensity of salt deposition already appears in 4 to 5 of the first stages where the temperature of the sucked-in aerosol reaches 350 to 380 K, when assuming that the working medium temperature increases by 15 to 20 K in each successive stage.

Endoscopic tests of the gas turbine engines, carried out onboard Polish Navy vessels by the author in the last years, confirmed the limited range of fouling by salt deposits [2]. The salt layer deposited on the controllable inlet guide vanes of LP compressor detected during endoscopic tests of an engine in service is shown in Fig.1.

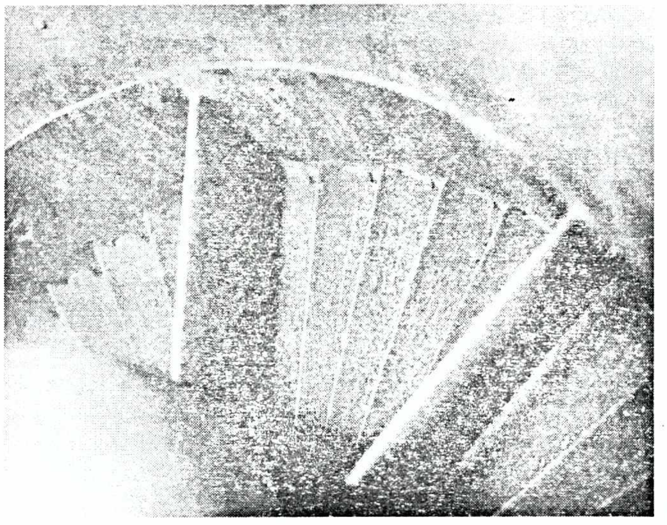

Fig.1. The salt layer deposited on the controllable inlet guide vanes of LP compressor

A relationship between the engine load change frequency and character of fouling was observed in result of systematic endoscopic checks. It was revealed that the transient processes (e.g. acceleration and deceleration) contributed in uniform distribution of deposits along the entire flow passage length.

Another phenomenon hazardous for engine reliability is the erosion due to the solid and liquid mineral particles contained in the sucked-in working medium. Fig.2 shows the edge of attack of the I - stage moving blade of HP compressor with the evident erosion wastage traces.

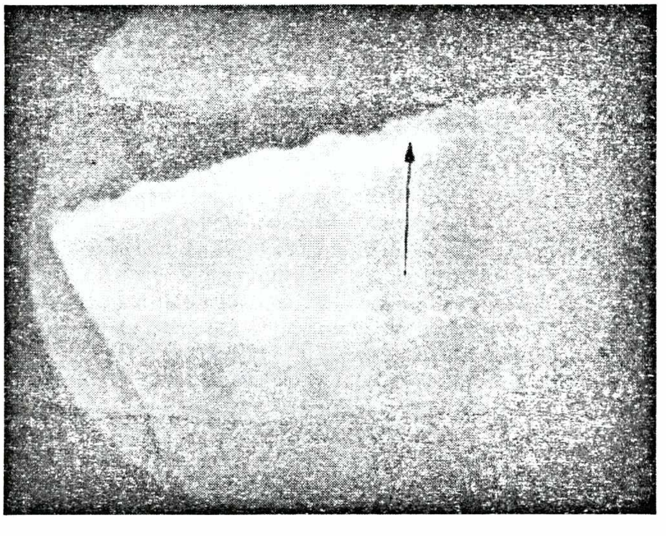

Fig.2. The edge of attack of the I - stage moving blade of HP compressor with the evident erosion wastage traces

Both the detrimental phenomena cause wearing the surface of flow-part blades, changes of blade geometry and flow passage shape, as well as growing surface roughness (as much as by several times). In consequence the hydraulic losses of working medium flow increase, engine's efficiency and performance parameters drop, all at its substantially worsened dynamic features.

MATHEMATICAL MODEL OF AXIAL COMPRESSOR

Possible operation states of an axial compressor are specified on the basis of its stationary characteristics experimentally determined by its manufacturer. Usually they are presented as the parameters related to the normal atmospheric conditions, i.e. comparable to and independent of widely defined environment parameters as follows:

$$\dot{m} = f(\pi_S^{\bullet}, n_S, \alpha_{KW}) \tag{1}$$

$$\eta_e^{\bullet} = f(\dot{m}, n_S, \alpha_{KW}) \tag{2}$$

The most popular and highly effective way of determining an axial compressor analytical model is that based on the least squares method and the polynomial multidimensional regression [1,4]. It makes possible to determine an arbitrary point on compressor characteristics with the guaranty that the values of deviations of the model from real object's behaviour are maintained within measurement error range. Usefulness of the proposed method was checked by applying it to two different axial compressors installed in DR76 engine. On the basis of the performed calculations and analyses the general model of each of the compressors was elaborated in the form of the following set of equations which approximate their universal characteristics:

$$\dot{m} = a_0 + a_1 n_S + a_2 \pi_S^{\bullet} + a_3 (n_S)^2 + a_4 n_S \pi_S^{\bullet} + a_5 (\pi_S^{\bullet})^2 \tag{3}$$

$$\eta_e^{\bullet} = b_0 + b_1 n_S + b_2 \dot{m} + b_3 (n_S)^2 + b_4 n_S \dot{m} + b_5 (\dot{m})^2 \tag{4}$$

Values of the regression coefficients a_i and b_i were determined by using the Gauss-Markov theorem and searching for the minimum values of the following functionals:

$$J_{\dot{m}}(a_0, a_1, a_2, a_3, a_4, a_5) = \sum_{k=1}^{n} (\dot{m}_k - \bar{\dot{m}}_k)^2 \tag{5}$$

$$J_{\eta_e^{\bullet}}(b_0, b_1, b_2, b_3, b_4, b_5) = \sum_{k=1}^{n} (\eta_{ek}^{\bullet} - \bar{\eta}_{ek}^{\bullet})^2 \tag{6}$$

They are sums of squares of deviations of the model from real object's behaviour.

Adequacy of the compressor model was assessed by determining the following correlation coefficients of each isodrome:

$$r_m = \frac{\sum_{k=1}^{n} (\dot{m}_k - \dot{m})(\bar{\dot{m}}_k - \bar{\dot{m}})}{\sqrt{\sum_{k=1}^{n} (\dot{m}_k - \dot{m})^2 \sum_{k=1}^{n} (\bar{\dot{m}}_k - \bar{\dot{m}})^2}} \tag{7}$$

$$r_{\eta_e^{\bullet}} = \frac{\sum_{k=1}^{n} (\eta_{ek}^{\bullet} - \eta_e^{\bullet})(\bar{\eta}_{ek}^{\bullet} - \bar{\eta}_e^{\bullet})}{\sqrt{\sum_{k=1}^{n} (\eta_{ek}^{\bullet} - \eta_e^{\bullet})^2 \sum_{k=1}^{n} (\bar{\eta}_{ek}^{\bullet} - \bar{\eta}_e^{\bullet})^2}} \tag{8}$$

where:

$\dot{m} = \frac{1}{n}\sum_{k=1}^{n} \dot{m}_k, \ \eta_e^{\bullet} = \frac{1}{n}\sum_{k=1}^{n} \eta_{ek}^{\bullet}$ - arithmetic mean of model output

$\bar{\dot{m}} = \frac{1}{n}\sum_{k=1}^{n} \bar{\dot{m}}_k, \ \bar{\eta}_e^{\bullet} = \frac{1}{n}\sum_{k=1}^{n} \bar{\eta}_{ek}^{\bullet}$ - arithmetic mean of measurement output

The equations were supplemented by the boundary conditions which described the operation regions of the characteristics.

In Fig.3,4,5 and 6 the modelled characteristics of the LP and HP compressors are shown for the angle of the controllable inlet guide vanes α_{KW} = -10°. Values of the measurement points from producer's experimental tests were inserted onto the calculated compressor characteristics to demonstrate effectiveness of the modelling method.

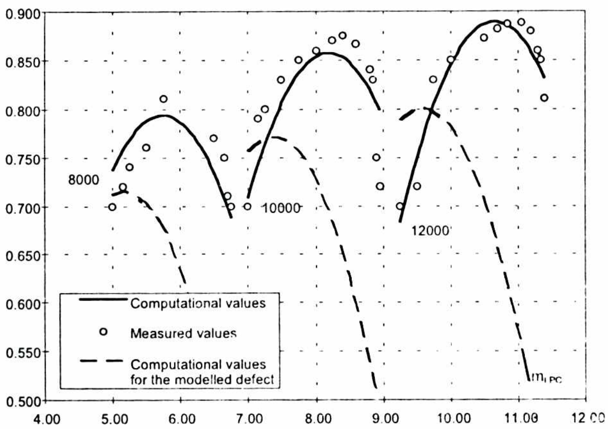

Fig.3. LP compressor characteristics: $\eta_c = f(n_{LP}, m_{LPC})$

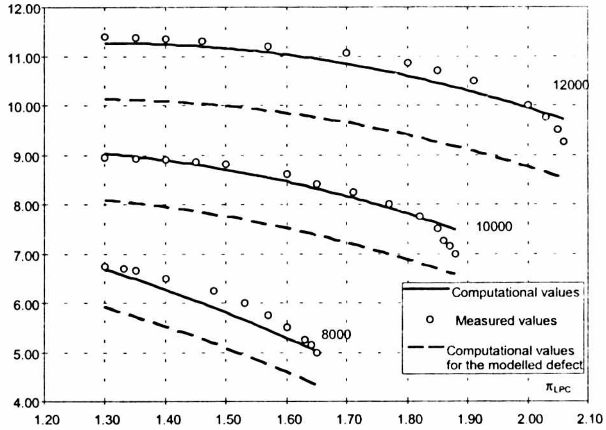

Fig.4. LP compressor characteristics: $m_{LPC} = f(n_{LP}, \pi_{LPC})$

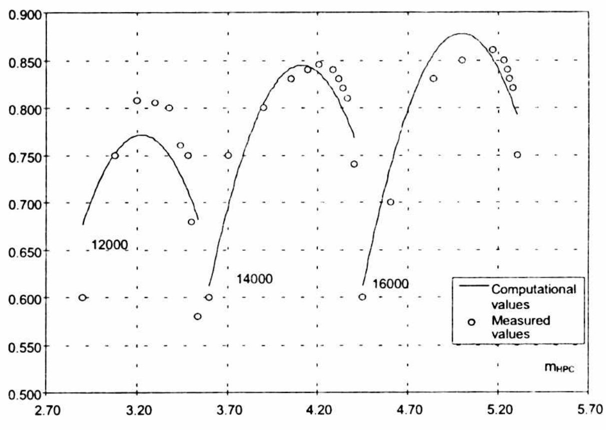

Fig.5. HP compressor characteristics: $\eta_c = f(n_{HP}, m_{HPC})$

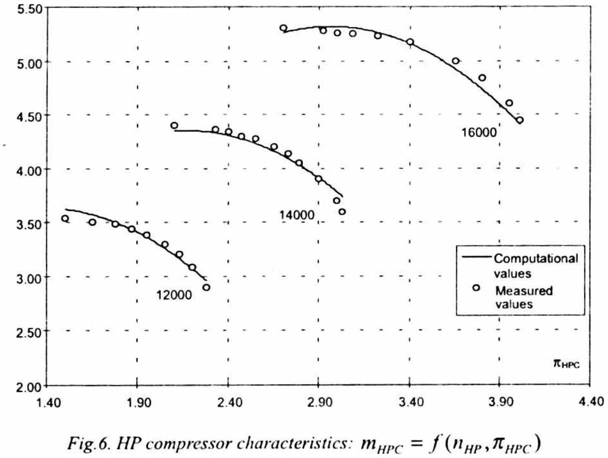

Fig.6. HP compressor characteristics: $m_{HPC} = f(n_{HP}, \pi_{HPC})$

Values of the regression coefficients of the equations which approximate the compressor characteristics are presented in Tab.1. Values of the correlation coefficients for some isodromes are given in Tab.2.

Tab.1. Values of the regression coefficients of the compressor model

No.	Regression coefficients	Of LP compressor model	Of HP compressor model
1	a_0	0.297927408	-4.983838915
2	a_1	0.151876709	0.106841430
3	a_2	-12.075520493	-5.600150728
4	a_3	-0.000760348	-0.000357375
5	a_4	0.115176331	0.039288583
6	a_5	-2.730237264	-0.823292787
7	b_0	-1.522442300	-4.418890500
8	b_1	0.069197400	0.109599500
9	b_2	-0.855382800	-3.752841200
10	b_3	-0.000582700	-0.000652500
11	b_4	0.015427600	0.047825200
12	b_5	-0.104657400	-0.901303600

Tab.2. Values of the correlation coefficients for some isodromes

No.	Compressor	Isodrome	$r_{\dot{m}}$	$r_{\eta_c^*}$
1	LP	8 000	0.98475	0.64864
2		10 000	0.97456	0.79395
3		12 000	0.97669	0.95989
4	HP	12 000	0.99997	0.92873
5		14 000	0.99997	0.94685
6		16 000	0.99998	0.95185

It can be concluded from the values of the correlation coefficients that the LP compressor efficiencies are the weakest correlated within the idle speed range. It results from a low number of the measurement points. An acceptable (or even very high in the case of HP compressor) , both qualitative and quantitative, consistency of the model is obtained in the upper operation ranges of the compressors where the number of measurement points is sufficient. The deviations of the calculated values from those experimentally determined by the producer are contained within the measurement error range. The number of terms of the approximating polynomial should be increased (at the expense of a higher labour demand for computation) to improve the quality of the elaborated model.

To get a reliable model in such a form it is necassary to assume at least 100 to 120 measurement points uniformly spread over the characteristics area. Good results can be obtained by „reading the points more densely" along the extreme segments of each isodrome.

DEPOSIT LAYER INFLUENCE ON CHANGE OF COMPRESSOR PARAMETERS

The corrosion and erosion wastage of the compressor flow passage cause, similarly as in the case of fouling, deformation of inter-blade channels and an increase of surface roughness. In consequence it leads to lowering the efficiency of the energy processes being performed in the compressor as well as to worsening the kinematic, dynamic and thermodynamic characteristics of the engine.

In the case of multi-shaft engines it is especially complicated to interprete the influence of the flow passage fouling on the characteristics of the co-working compressors. It results from the mutual interaction of the particular rotor units [3]. The interaction can be either of supporting or choking character, and its course is determined first of all by engine's design arrangement as well as by an assumed load control method [6,7,8].

The LP compressor is the most probable region of the multi-shaft engine flow passage which can be fouled. In consequence the mass of the compressor rotor grows and its speed drops. The active cross-section areas of the inter-blade flow channels decrease. The efficiency, compression and mass of the working medium being forced through the compressor as well as the compressor's stable performance margin drop. This appears to be a „quite different" flow machine. A „typically" deformed characteristics of the axial compressor serial system can be obtained (Fig.7), if it is assumed that the control system of the engine is able to maintain a constant speed of the rotors system, taking no account of technical state changes of the flow passages.

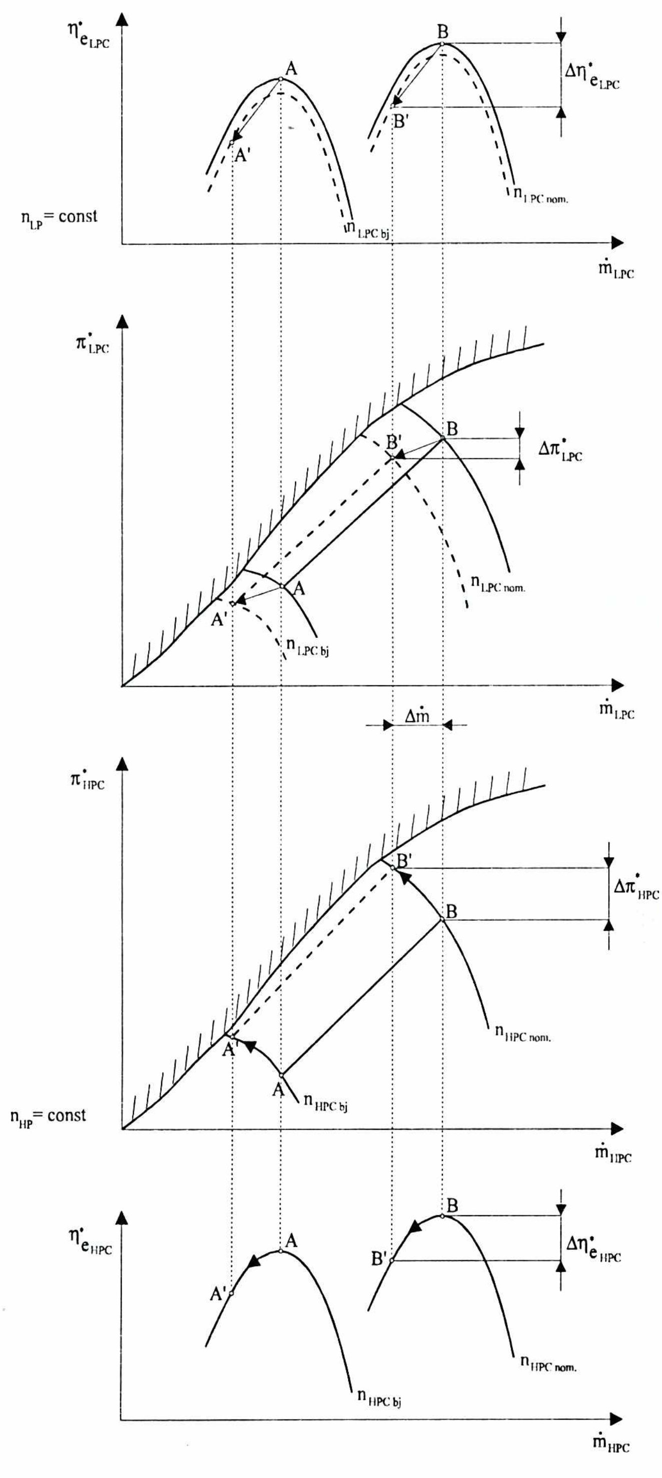

Fig.7. Shift of operating points of the LP and HP compressors in a two-shaft engine with a free power turbine caused by fouling of LP compressor inter-blade channels:
A, B - operating points of a „clean" system
A', B' - operating points of a system with fouled LP compressor

It can be observed in Fig.7. that the constant-speed lines of LP compressor become stratified, i.e. they are shifted left. The LP compressor characteristics belong to a different compressor. The so defined defect of LP compressor defines new operation conditions for the „clean", gasodynamically coupled with it, HP compressor. Choking of its inlet cross-section, at the maintained constant speed of it, triggers shifting a given operating point on the characteristics to left, i.e. towards compensation of intensity of working medium flow. If the assumed control program assures keeping $n_{HP} = const$, the compression of HP compressor grows up to fulfilment of the condition (9):

$$\dot m_{LPC} - \dot m_{HPC} = 0 \tag{9}$$

The vector translation procedures were applied to the regression function diagram as well as iterative procedures [5] to mathematically map the LP compressor defect in a form adequate to the deformed characteristics of the co-working compressors. In result the broken lines depicted on the LP compressor characteristics (see Fig.1 and 2) calculated with the assumed 10% drop of its efficiency and working medium mass flow and 2% drop of its compression, are obtained.

CONCLUSIONS

- The least squares method and the polynomial multidimensional regression method can be applied as an effective mathematical instrument for modelling the axial compressor characteristics. Effectiveness of the instrument is confirmed by the high values of correlation coefficients obtained for a relatively simple form of the polynomials approximating the experimental results.
- On this basis the computer program was elaborated which makes it possible to determine an arbitrary point on the compressor's characteristics, within measurement error accuracy. The program was used to numerically simulate the unsteady energy processes which proceed in the flow passages of the marine gas turbine.
- The proposed method of modelling the compressor defect is based on the application of simple analytical transformations due to which an earlier developed initial form of regression function can be deformed. The vector translation procedures were used to shift the functional diagram in the analyzed case of the uniformly fouled inter-blade channels. The stratification effect of isodrome characteristics is a measure of the modelled compressor defect confirmed by service tests.

NOMENCLATURE

a_i, b_i	-	regression coefficients, (i=0, 1,...5)
r	-	correlation coefficients
HP, HPC	-	high pressure compressor #
LP, LPC	-	low pressure compressor #
J	-	functional
n	-	shaft rotational speed, if marked with an appropriate index
m	-	mass flow rate #
α_{KW}	-	inlet guide vanes angle
η_e	-	efficiency #
π	-	compression ratio
Δ	-	change of a magnitude
x	-	any dashed value => measured value

Note: the above specified notations marked „ # " are also used as indices

Other indices:

bj	-	idle
k	-	number of segments of component sum equation (k=1,2...n)
s	-	compressor;
*	-	total value of a parameter
nom	-	nominal value of a parameter

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Appraised by Adam Charchalis, Prof.,D.Sc.,M.E.

Current reports

EXPERIMENTAL SYSTEM OF NAVIGATION DIRECTIONAL POINTS

In the case of navigation area periodical icing it is necessary to remove (in autumn) and set again (in spring) the seamarks which route the fairway. The system designed and manufactured at Faculty of Maritime Technology, Technical University in Szczecin, makes it possible to precisely renew the earlier exactly routed water lane and practically prevents from its distortion e.g by the fishing nets.

Arrangement of the system

The system consists of two subsystems installed on board a floating unit:

A. The satellite navigation subsystem used to determine preliminary position of a directional point by means of two basic devices:

- GPS Magellan receiver
- DBR differential module.

The GPS receiver makes use of the signals emitted by at least 3 navigation satellites. It is generally known that errors are deliberately introduced to some of the signals, due to which position can be settled within 100 m accuracy only. Thererfore it is necessary to use some additional devices to increase the accuracy.

The main element which provides the possibility is the land-based differential station which determines a correction for the received satellite signal and emits its by radio. Such correcting signal is received by the DBR differential module which introduces the correction to the information on positions of the directional points (obtained from GPS receiver). It makes possible to fix the positions with the accuracy of 5 m.

B. Magnetometric subsystem aimed at improving the position determination accuracy of directional points to 1 m. It consists of :

- PULSE 10 magnetometer
- bottom-fixed response buoys.

The magnetometer (its coil) generates magnetic field. If any metal objects are within its range vortex currents are generated in them and in result the magnetic field which deforms the coil field. The deformation is recorded by the magnetometer and visualized on a needle indicator or by an acoustic signal.

The principle was utilized to design the bottom response buoy which can precisely and permanently fix a position of a directional point. The buoy (Fig.1) is placed on the water area bed (or under a bottom deposit upper layer to protect it against damaging by nets, anchors etc) and it is so designed as to ensure the maximum disturbance of the signal generated by the magnetometer coil which is fixed at the submerged float connected to the surface float by means of the boom (Fig.2).

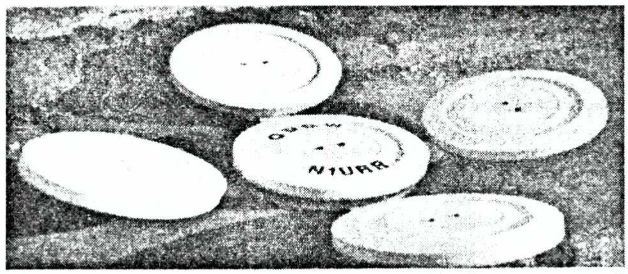

Fig.1. Bottom-fixed response buoy

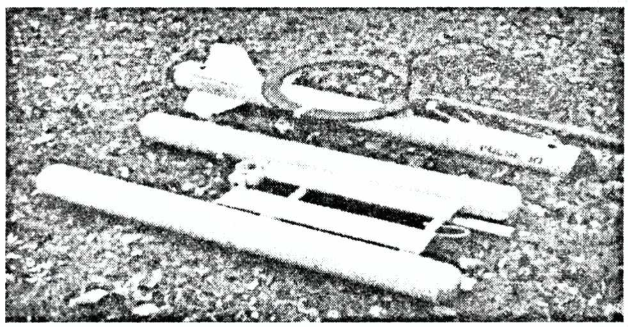

Fig.2. The submerged and surface floats

Preliminary results of application

Preliminary results of the system's application to the Szczecin - Świnoujście fairway (through Dąbie Lake), with the use of the land-based differential station in Dziwnowo, are in compliance with the design assumptions. Response of its users is also positive despite that the system radically changes the today routine of fairway routing.

Further tests to be carried out during consecutive navigation seasons are expected to improve the system operation and lead to its new upgraded versions.

The system was designed and manufactured by the following team:

Lech Tołkacz, D.Sc.,M.E. (team's leader)
Władysław Skórski, M.Sc.,E.E.
Mariusz Matejski, M.Sc.,M.E.
Zbigniew Szymczyk, Technician.