A design concept of main propulsion system with hydrostatic transmission gear for inland waterways ship

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ABSTRACT

This analysis is aimed at presentation of a design concept of ship's main propulsion system and determination of its total efficiency. To realize the task it is necessary to have a look at description of energy losses and efficiency of hydrostatic transmission gear, with taking into account its possible simplification. Image of mutual interaction of losses generated in all elements of hydraulic system appears very complex. In this paper energy balance of main propulsion system with hydrostatic transmission gear intended for a segment ship for inland waterways and coastal service, was analyzed under assumption of necessary simplifications.

Keywords : inland ships, hydrostatic propulsion system

INTRODUCTION

Selection of a ship's propulsion system should be preceded by an analysis which takes into account a.o. : types of today applied solutions, possibility of fulfillment of its demanded tasks, efficiency, initial and operational costs. Out of many analyzed propulsion systems applicable to ship operating in shallow inland waters, a system based on a hydrostatic transmission gear and fixed screw propeller ducted in a rotatable Kort nozzle, was selected to be designed. The system is consisted of a high-speed diesel engine which drives a variable delivery oil pump feeding hydraulic motor (-s) which directly or through a reduction gear drives a fixed screw propeller.

It should be mentioned that as compared with the hydraulic system the traditional propulsion system with fixed screw propeller has, apart from its higher simplicity and reliability, many important drawbacks such as : the necessity of application of a reversible reduction gear, a lower efficiency in distinctly varying operational conditions, as well as a lower possibility of ship speed control and power plant automation.

Hydraulic drives are widely used on contemporary merchant and naval ships. They serve for driving steering gears, windlasses, mooring, trawling and hoisting winches, closing devices of hatch covers, watertight and fire protecting doors, for driving ship-rolling fin stabilizers as well as fixed propellers of ship thrusters and low –power main propulsion systems.

Hydraulic drives are today applied not only to the mechanisms of to-and-fro-motion but wider and wider also to the rotary mechanisms. The reason is much smaller weight and inertia moment of hydraulic motor as compared with those of electric motor of the same output power.

Below are presented : schematic diagram, drive concept and energy balance of the main propulsion system with hydrostatic transmission gear intended for a segment ship provided for inland waterways and coastal service.

SCHEMATIC DIAGRAM OF SHIP'S MAIN PROPULSION SYSTEM

The schematic diagram of ship's main propulsion system with hydrostatic transmission gear is presented in Fig.1. It is supplemented with the drives of inclination mechanisms of Kort nozzles which fulfill function of rudders in the case in question.

The main propulsion system is composed of two independent sub-systems. Each of them is equipped with a high-speed diesel engine which directly drives a set of pumps as well as – through a reduction gear – an electric generating set, that makes operating with constant rotational speed necessary. The main pump of the system is of variable delivery and it feeds - in closed circulation - hydraulic motors of constant absorbing capacity. By changing geometrical working volume of the pump it is possible not only to change steplessly its capacity from 0 to Q_{max} , but also to reverse oil pumping direction, hence to control direction and rotational speed of the motors fed by the pump and of the screw propeller driven by the motors.

Closed-circulation drives usually have smaller dimensions as compared with open- circulation ones, and they operate more stable as during operation air does not get inside the main. Hence they are suitable for operating in automatically controlled systems.

Closed -circulation systems are as a rule applied where load changes its sense at changed direction of rotation, e.g. in the case of ship screw propeller. As in the closed- circulation systems the suction and pumping pipes interchange their functions they cannot be connected with open oil tanks, hence heat transfer conditions in them are worse than in open-circulation systems, hence it is as a rule necessary to apply oil coolers.

A main drawback of closed - circulation systems is the fast heating of circulating oil, which requires the generated heat to be absorbed to make long -lasting work under full load

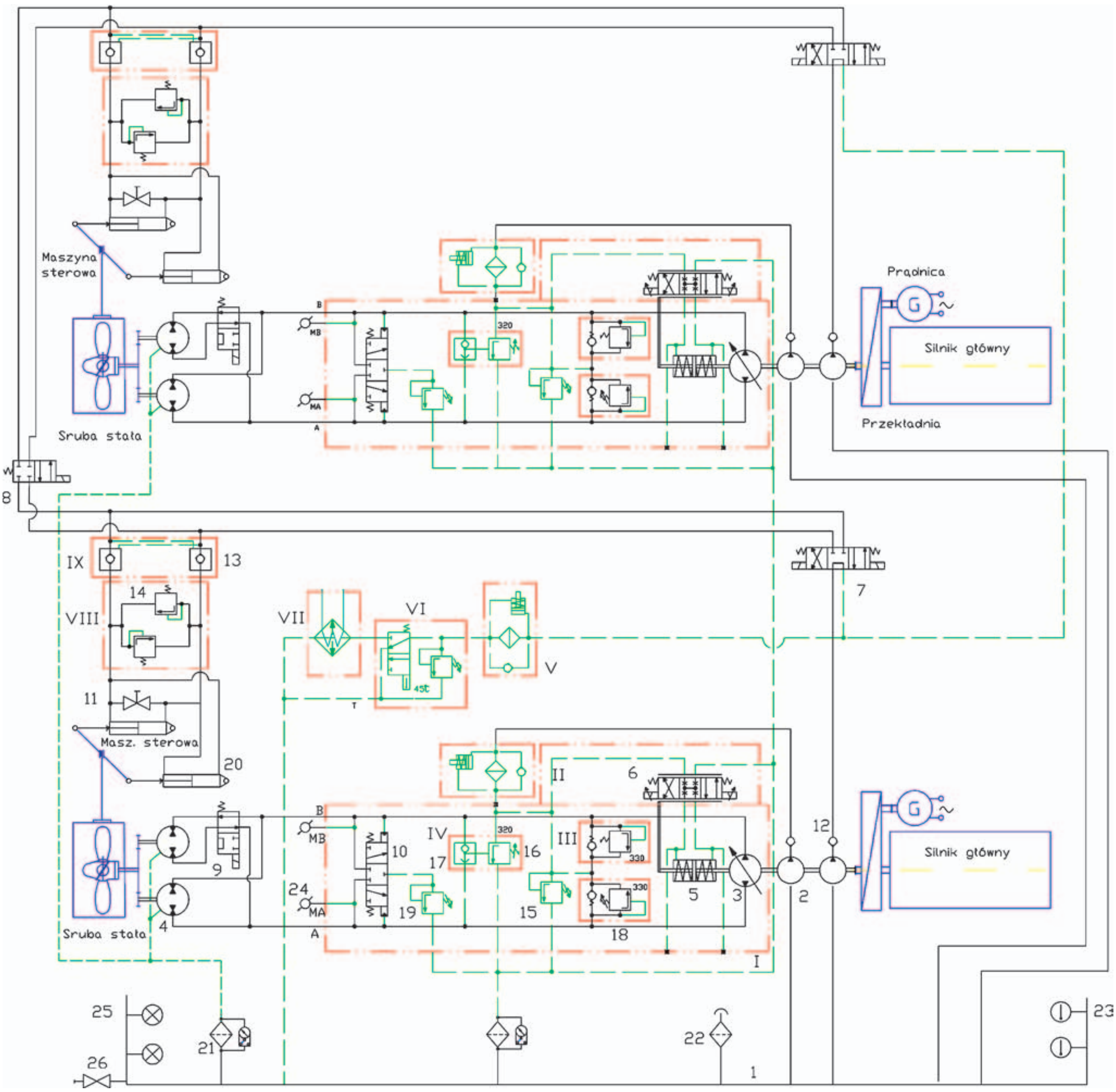

Fig. 1. Schematic diagram of ship main propulsion system and its control [1] .

impossible. Generally it is accomplished by means of continuous replacing a part of the heated oil taken from the main circulation by cleaned cool oil from the auxiliary reconditioning circulation.

Each of the main engines drives one electric generating set and three pumps (no. 3, 2 and 2a) including two of variable delivery. In the main hydraulic system are installed : the pump no. 3 of variable delivery per revolution and variable pumping direction, two reversible motors no. 4 of constant absorbing capacity, the scavenging pump no. 2a, the main pump safety valves no. 18 and the valves, no. 15 and 16, of the scavenging filling-up pump, the check valves connected in series with the valves no. 18, as well as the filter no. 21.

Depending on pumping direction of oil either through the pipe A or B, clockwise or anti-clockwise direction of revolution of the motors no. 4 is obtained. It depends on pumping direction which of the safety valves no. 18 can be active.

The system of check valves ensures that the refreshing oil always is delivered to the low-pressure branch of the main circulation. In the low-pressure part some over-pressure usually equal to 0.5÷1.5 MPa, is maintained. The pressure prevents the suction piping from airing and cavitation as well as improves filling conditions of the working chambers of the pump no. 3. Hence a greater rotational speed of pump shaft can be used as compared with that in open-circulation systems.

The scavenging pump no. 2a pumps oil through the filter and check valves into the main. It ensures the reconditioning of oil in the main circulation system and the simultaneous supplementing of oil losses resulting from non-tightness of the pump no. 3, motors no.4 and fittings.

The overflow valve no. 15 is set-up at the opening pressure a little higher than that at the valve no. 19, hence the fresh oil is delivered through one of the blocks no. 18 to currently low--pressure branch of the main circulation. Whereas the valve no. 16, in cooperation with the distributor no. 6, limits capacity of the main pump in the case of a pressure increase in the pumping branch of the main propulsion hydraulic system.

The motors no. 4 drive – through the reduction gear – the fixed screw propeller ducted by the rotatable Kort nozzle having this way function of rudder. Stepless control of speed and direction of propeller rotation is accomplished by changing the capacity and pumping direction of the pump no. 3. To this end serves the hydraulic servo-mechanism of the pump, consisted of the setting cylinder no. 5 and the electro-magnetically controlled proportional distributor no. 6. Hence choice of direction of movement either forward or astern can be made by means of the latter distributing valve electrically and remotely controlled from cabin.

To ensure oil exchanging and cooling in the circulation, the so-called block of scavenging valves is used. The heated and contaminated oil is discharged to the tank by means of the distributor no. 10 and overflow valve no. 19. Change of pumping direction of the pump no. 3 (and direction of rotation of the motors no. 4) triggers automatically switch-over of the distributor no. 10 in such a way as to make oil exchange occurring always within the low-pressure branch of the circulation. Therefore the fresh oil cooled in cooler VII comes into circulation.

The pump no. 2 pumps oil – through the distributor no. 7 – – to the cylinders no. 20 of the Kort nozzle inclination system. The system is secured by two safety valves no. 14. They connect to each other both pipes of the cylinders in the case of exceeding the permissible force applied to the cylinders. In the central position of the distributor no. 7 the inflowing oil is directed through the unit of filter V and cooler VII, in the case when the oil were heated over 45°C.

Advantages of the closed-circulation system are the following :

- the possible changing of motion direction of consumer by means of the pump and
- the possible braking of drive by means of the pump and main motor.

Arrangement of basic units of the system

The ship's propulsion system is presented in Fig. 2, 3 and 4.

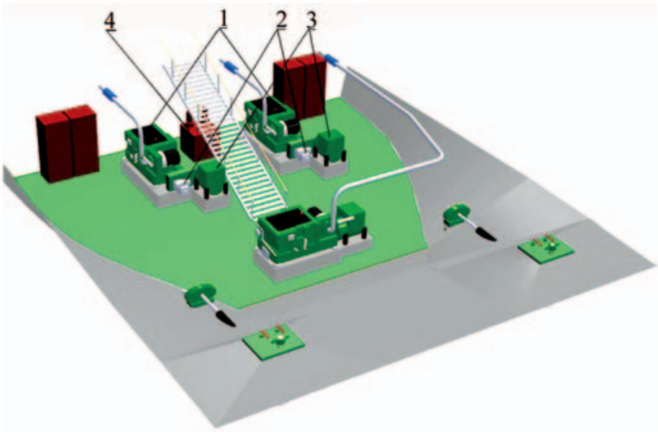

Fig. 2. *Aft view on the ship and its devices arranged in ship power plant .*

The main engines used to drive the pumps of the hydraulic system as well as the electric generating sets are IVECO CURSOR 300 Common Rail direct injection diesel engines (no.1 in Fig. 2). The engines develop 220kW (300 KM) output power at 2000 rpm speed.

The main engines serve for direct driving the Rexroth A4VSG 180 main hydraulic pumps (no. 2 in Fig. 2). They are multi-plunger (axial) pumps of inclining disk, which can operate at the nominal pressure up to 400 bar and maximum rotational speed equal to 2400 rpm, in closed circulation. The pump is integrated with the auxiliary pump of the scavenging circulation and control system.

Additionally, in the system the belt transmission (no.7 in Fig. 4) to drive Leroy Somer 42.2 VL8 electric generating set (no. 3) and the additional pump delivering oil to the hydraulic nozzle-rotation control system (no. 5 in Fig. 3), are provided for.

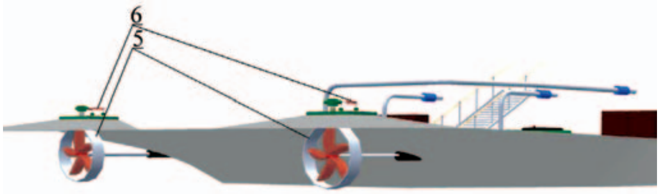

Fig. 3. *Aft view on the ship fitted with the propulsion system in question .*

The belt transmission (no. 7 in Fig. 4) is designed on the basis of Gates QuadPower II toothed vee-belts. Such construction eliminates practically entirely sliding the vee-belt wheels, which would be not acceptable because of the function of driving the electric generator, realized with its help.

The IVECO GE8210M22 electric generating set (no. 8 in Fig.4) was proposed to be used as an additional electric energy source. The diesel engine driving the electric generator is of direct injection (effected by BOSCH pump of P type). The generator is provided for operation both during ship sailing and staying (i.e. when its main engines are stopped).

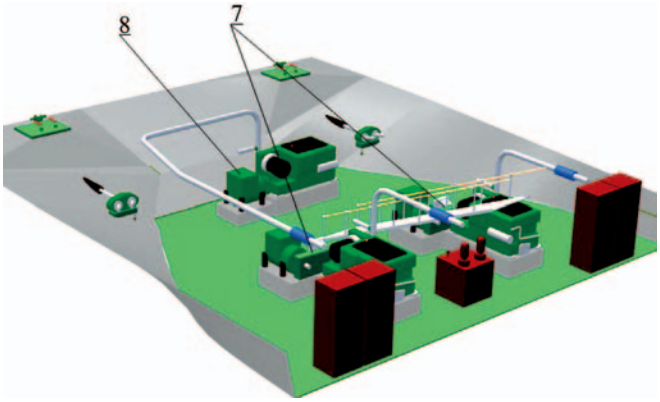

Fig. 4. *Fore view on the ship : schematic image of the devices arranged in ship power plant .*

Each of the fixed screw propellers of the ship is driven through the cylindrical gear (to decrease its mass and increase reliability) by two Rexroth A2FM 80 hydraulic motors (of multi-plunger (axial) construction, inclining cylinder block and constant absorbing capacity per revolution). The motors can operate under pressure up to 400 bar and 5000 rpm speed. The rotatable nozzle (no. 5 in Fig.3) is inclined by means of hydraulic cylinders.

The oil tank (no.4 in Fig.2) is located under stairs.

ENERGY EFFICIENCY OF HYDROSTATIC SYSTEM

One of the crucial features characterizing the hydrostatic system is its energy efficiency defined as the ratio of the output power obtained from the engine (-s) and the power delivered to the pump shaft. Apart from economical and energy consequences, drop of the efficiency results also in an increase of oil temperature and associated change of its viscosity, which influences the system's operational characteristics. Hence sometimes it can constitute the factor which decides on possibility of application of such system in a given case.

In many cases energy losses are mutually connected, e.g. mechanical friction in gaps is usually associated with leakages and both the phenomena mutually interact. Therefore unambiguous separation of particular parameters from each other and determination of their impact on run of the phenomena, e.g. in a working pump, often meet serious difficulties. The general principle that the total efficiency is the product of partial efficiencies, being in force in mechanics, in hydraulics, should be considered as a simplification due to occurrence of various interactions and feedbacks difficult for investigation and complicating the principle. Under necessary simplifications, was analyzed energy balance of the main propulsion system with hydrostatic transmission gear, intended for a segment ship for inland waterways and coastal service.

ANALYSIS OF ENERGY LOSSES AND EFFICIENCY OF SHIP MAIN PROPULSION SYSTEM WITH HYDROSTATIC TRANSMISSION GEAR

Structure of hydraulic system greatly impacts its efficiency. Its influence is usually considered under assumption of an ideal pump and motor and that real energy losses occurring in the pump and motor will result in a further proportional decrease of total efficiency of the system.

However, image of mutual interaction of losses in all elements of the hydraulic system appears much more complicated. Instantaneous energy efficiency of pump, for instance, results, apart from other factors, first of all from an applied structure of hydraulic motor control system.

The closed-circulation systems find wide practical application, and in their work it is necessary to supplement oil leakage occurring in the main propulsion system of propeller as well as to exchange heated oil circulating in the system.

Calculations of hydrostatic transmission gear and selection of its elements for a given drive are difficult because of many possibilities of such selection. Input data for calculations of the hydrostatic gear are the following :

- the maximum rotational speed of ship's propeller, n_{2max} = = 12.5 rps [750 rpm]
- the pump shaft rotational speed n_1 = 33.3 rps [2000 rpm]
- the propeller shaft power N_1 = 150 kW
- the maximum pressure in the system, p_1 = 27 MPa.

In elements of hydraulic system occur the losses which functionally depend a.o. on working liquid viscosity, as well as the energy losses which do not practically depend on viscosity.

In order to simplify the problem of viscosity influence on energy performance of the hydraulic system, the kinematic viscosity ν_n = 35 mm^2s^{-1} was chosen as the reference level recommended by producers from the point of view of the functioning of hydraulic elements produced by them.

From the schematic diagram presented in Fig. 1. it results that in the hydrostatic propulsion system of the ship the transmission gear controlled by changing geometrical working volume of the pump which cooperates with two motors of a constant absorbing capacity, is applied.

As above mentioned, in the hydrostatic gear the controllable plunger axial pump with inclining cylinder block, operating in the closed -circulation system with possible change of oil pumping direction, is applied. This is Rexroth A4VSG 180 pump of 180 cm^3 displacement volume. The demanded theoretical capacity of the pump, Q_{Pt} , is equal to 0.00594 m^3/s [356.4 dm^3/min]. Whereas from the characteristics of the real capacity of the pump, Q_P , in function of working pressure (catalogue data) it results that the real capacity is equal to 0.00558 m^3/s [335 dm^3/min] at the pressure p = 27 MPa, and from the pump characteristics at 27 MPa pressure – the displacement efficiency of the pump, η_{Pv} , equals 0.91. Hence the power P_{ep} delivered by the pump amounts to 165.7 kW.

Then two hydraulic motors, also produced by Rexroth, were selected for each of the two propulsion systems of the ship. These are A2FM 80 motors of 365 Nm torque at 27 MPa working pressure. The minimum rotational speed n_{min} at which shaft rotational speed is still maintained, amounts to 0.8 rps [50 rpm]. For the selected high-speed motor, high efficiency can be obtained at the speed within the range of 10 ÷ 56 rps [600÷ ÷ 3400 rpm]. The total efficiency η_s read from the efficiency characteristics of the motor operating with the rotational speed of 35 rps [2100 rpm] at 27 MPa working pressure, is equal to 0.94.

Hence the power P_M of one hydraulic motor obtained from calculations is equal to 76 kW.

The ship screw propeller was assumed to rotate with the rotational speed of 12.5 rps [750 rpm, therefore application of a mechanical reduction gear is necessary. Of course, the propulsion efficiency would be then lowered by the mechanical gear efficiency η_{prz} which can be assumed equal to 0.97.

The total propulsion efficiency of the ship, η, being the product of the total efficiencies of the pump, motor and mechanical gear, is equal to 0.82.

After preliminary determination of sizes of consumers and approximate arrangement of hydraulic elements on the ship it was possible to select diameters of piping and nominal diameters of valves and fittings. Usually, the principle of selecting uniform nominal diameters of piping, valves and other devices for one pipeline, is used. The selection of nominal diameter of piping and control elements is very important as the diameter decides on flow losses hence also on the whole device efficiency. If the nominal diameter is too small the flow losses are high and the whole system efficiency – low. A too large nominal diameter would not provide any significant increase of the system efficiency and simultaneously would lead not only to increased cost of the installation but also to difficulties in arranging particular devices on the ship. Hence to reach a reasonable compromise is necessary.

By assuming the liquid velocity v_t equal to 4 m/s in the pumping pipeline the nominal diameter D_t = 40 mm was selected for it. As the closed circulation is taken into account the return pipeline diameter is selected equal to that of pumping one.

If TOTAL Azola 46 oil (of 873.3 kg/m^3density in 20°C) is chosen and the limit design temperature range from -10°C to +60°C is assumed, then to determine oil viscosity in the limit temperatures is possible, namely : ν_{-10} = 400 cSt and ν_{60} = 20 cSt, respectively. Then the efficiency which can be obtained by the system at the viscosity of 35 $mm^2s^{-1,}$ amounts to 0.82.

SUMMARY

- An applied speed control structure of hydraulic motor (-s) is the crucial factor which decides on an instantaneous value of energy efficiency of pump or whole hydrostatic drive. The known descriptions of energy losses in pump serve mainly to explain complexity of the phenomena which occur during operation of displacement machines. They have a simplified form as to grasp all factors deciding on the losses is not possible. Simultaneously, the form does not allow to describe global image of relationships of particular kinds of losses and working parameters of the system.

- To obtain description of hydraulic system's efficiency in function of load and speed of motor it is necessary, in the displacement machines, to separate mechanical losses from pressure ones which depend in different ways on pressure level and flow rate of working medium. The system's efficiency model which takes into account influence of working medium viscosity, makes it necessary to apply such description of displacement losses, in which proportions of the main elements of the losses, i.e. layered leakages and turbulent ones are taken into consideration [5].

- The efficiency of the described hydraulic propulsion and control system for the inland waterways passenger ship operating under full load, will be equal to 0.82. However under lower load the total system's efficiency will be lower. Hence a power surplus of the diesel engine driving the pump can be taken over by the generator.
- On the basis of the elaboration [2] were considered successive concepts of ship propulsion systems, e.g. that fitted with azimuthing propeller. The considered concepts of hydraulic propulsion system of the ship did not influence energy balance of the system significantly.

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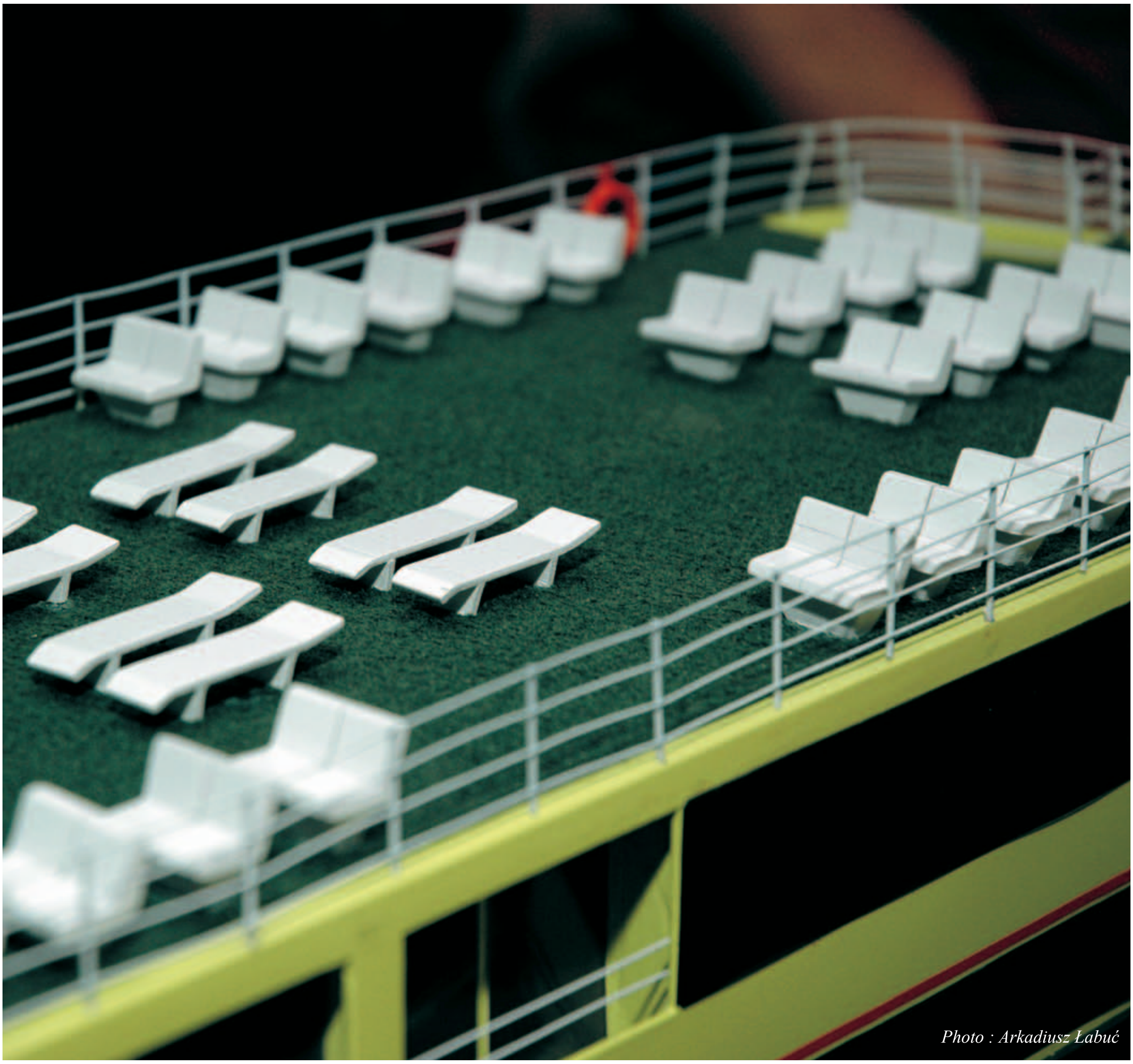

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