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RESEARCH OF TECHNOLOGICAL SYSTEMS OF STEAM SUPPLY AND POSSIBLE WAYS OF INCREASING THEIR ENERGY EFFICIENCY ON THE EXAMPLE OF DEVICES FOR DEFROSTING WAGONS

Abstract. *Steam supply systems of industrial enterprises in various fields of activity remain relevant primarily due to the thermophysical features of the use of phase transition energy of water vapor in the process of its condensation. In addition, in the countries of Eastern Europe, traditionally a significant number of existing boiler houses are steam and the choice of energy carrier is determined precisely by the presence of a characteristic source. At the same time, cargo logistics processes are provided today largely by sea transport, and delivery to ports is traditionally carried out by rail, which in turn requires the use of a defrosting system and transshipment of bulk products. The paper presents the results of studies of energy consumption of devices for defrosting wagons of port infrastructure and proposes circuit thermal and mechanical solutions for the modernization of the steam supply system of defrosting house. According to the results of the study, the values of heat energy consumption per unit of output in the existing scenario and after modernization are given. The characteristic technological and thermotechnical features of the existing scheme of devices for defrosting wagons of the port infrastructure of Ukraine and measures for the modernization of such systems, the introduction of which will achieve a significant reduction in the consumption of* *thermal energy in shunting non-stationary modes of operation of defrosting houses, are given. The feasibility of using an infrared wagon heating system using steam registers is substantiated.*

Keywords: Steam supply system, energy efficiency, technological defrosting house, wagon defrosting device, heat loss of steam pipelines.

Introduction. There are few studies devoted to the rather narrow and geographically limited topic of energy efficiency of steam supply systems for defrosting wagons. First of all, this is due to the fact that defrosting houses with steam heating devices were actively used only in Eastern Europe, and the geography of research is mainly limited to the northern countries of this region. At the same time, the growth of shipping is an inherent process for actively developing and consuming humanity. Accordingly, the importance of the energy efficiency of all processes at all stages of shipping is difficult to deny [1-2]. At the same time, the delivery of goods by rail to ports remains an integral part of the logistics process [3-5].

Latest research and publications. There are conclusions that increased loading, particularly of coal and other bulk commodities to specialised marine terminals to reduce gondola turnover, shunting and marshalling operations, may lead to an increase in logistics turnover. Reference [6] provides data regarding the impact of the lack of wagon defrosters; it is stated that, among other things, this deficiency leads to slower wagon turnover.

The study [7] is devoted to improving the efficiency of the defrosting process of freight wagons. Various variants of convection heating are given and the performance of such variants is evaluated.

The paper [8] considers a numerical method for solving the problem of unsteady heat transfer during coal heating in a special defrosting device with the subsequent obtaining of a method allowing to reduce the time of coal heating due to the application of a special thermal regime. The attention is focused on time saving at coal heating by the value from 87,9 to 12,7 %, that is definitely very actual at shunting modes of work of the device of a greenhouse.

The research [9] gives directions and ways of modernisation of greenhouses and substantiation of the decision of expediency of transition from convective systems of heating of devices of defrosting of cars to systems of infra-red heating. Similar conclusions and emphasis on gas infra-red side screens and hearth burners are given in the study [10]. It is stated that the heating time can be in the range of 20 minutes.

The study [11] confirms the effectiveness of the solution of using pipe registers for extended objects, including wagon defrosting devices. The article also focuses on gas infrared heaters.

Thus, the relevance and object of the study are confirmed and require a detailed study, in particular, the study of the operation and thermal-mechanical scheme of steam heating systems of the existing wagon-defrosting devices of the port infrastructure. This study is primarily intended to focus on improving the energy efficiency of individual technological steam consumers of industrial enterprises, in particular, the defrosting devices in the port infrastructure.

The article presents the results of experimental studies of the steam supply system for greenhouses, in particular, the defrosting devices in the port infrastructure. The paper presents the results of experimental studies of the steam supply system of greenhouses and provides possible ways to improve the energy efficiency of wagon defrosting devices as significant steam consumers.

Main part. The object of this study was one of the ports of Ukraine, located on the Black Sea coast. In the total energy balance of the facility, the largest share is occupied by the central boiler house, represented by a combination of steam and hot water boilers with a total installed capacity of about 45 MW, of which 9 MW is the installed capacity of hot water boilers. At the same time, the share of the installed capacity of two devices for defrosting wagons (hereinafter, technological defrosting house) is up to 8.8 MW in full load mode (according to the enterprise).

One of the largest technological consumers of water vapor at the enterprise are two devices for defrosting wagons (loading one device - 20 wagons). As thermotechnical objects, defrosting devices are long frameless buildings with two spans and bearing longitudinal walls of concrete. The width of one span is 6 m. Side and ceiling smooth pipe steam registers are used as heating elements (Fig. 1).

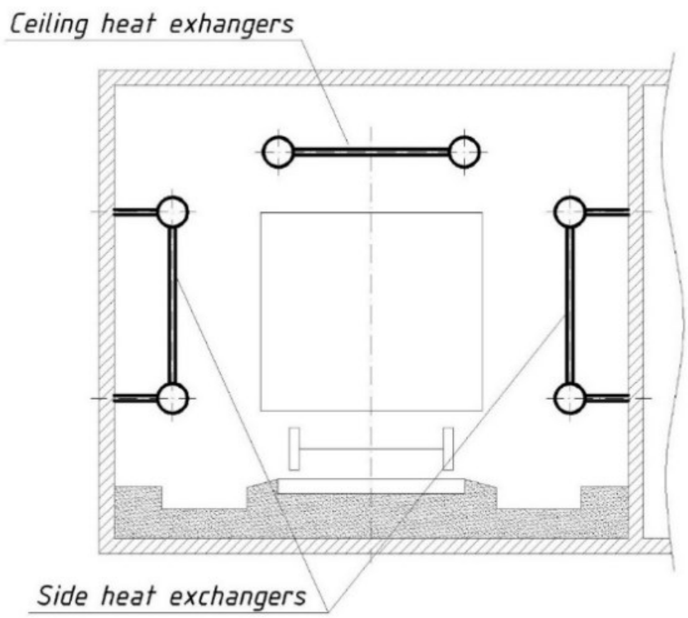

Fig. 1. Section of the defrosting house with an image of the location of the steam turbines registers

Steam registers provide intensive radiation supply of heat by heating frozen cargoes. In stationary working conditions, with a continuous flow of goods, the devices work effectively with an average specific consumption of 0.128...0.395 MW/wagon [8]. In order to avoid irreversible thermal effects on goods, surfaces and elements of wagons, the average air temperature in the working space of the defrosting house is maintained at 90... 95 °C.

As a coolant in devices for defrosting wagons, water vapor is used. From the boiler room, overheated water vapor enters the main steam pipeline with the parameters - overpressure 0.8 MPa, temperature 200... 220 °C. At the entrance to the defrosting house, the water vapor is reduced, the pressure after the reduction gear is 0.2... 0.3 MPa. Such reduction can potentially also be used as part of the substitution for low-power steam turbines with sufficient economic justification [9]. Superheated steam enters the side and ceiling steam screens from smooth pipes, where it cools and condenses. The system is designed in such a way that the formed condensate is supercooled below the saturation temperature, at a pressure set in the condensate pipeline (excess pressure in the condensate pipeline 0.18... 0.25 MPa, condensate temperature 102... 106 °C). Overpressure ensures continuous condensate return to the boiler house.

The principle thermal scheme of the existing heat supply system of steam defrosting house is shown in Fig. 2. Characteristic of the existing system:

1. Stable work is ensured in stationary operating mode.
2. The system is cheap and technically simple (with a minimum number of elements that require maintenance).
3. Condensate return due to back pressure in the system, no pumps are used to move condensate (no additional electrical energy is consumed).
4. Low economically justified specific energy costs per unit of "finished products".

Along with the above advantages, attention should also be paid to the fact that steam screens are not equipped with devices to prevent the flow of water vapor into the condensate collection and return system - steam traps. It is known that in steam condensate systems not equipped with steam traps, under non-stationary operating conditions of systems, it is possible to slip "acute" (not condensed) steam into the condensate collection and return system. The value of steam slippage can be from 5 to 30% of the nominal mass flow rate of the coolant [10-11].

In the case of condensate collection systems equipped with atmospheric tanks (a similar system is also organized in the thermal management of the enterprise under study), "hot" steam usually transits into the atmosphere (is lost). The second feature of the existing steam supply system is the limited ability to replace its thermal capacity.

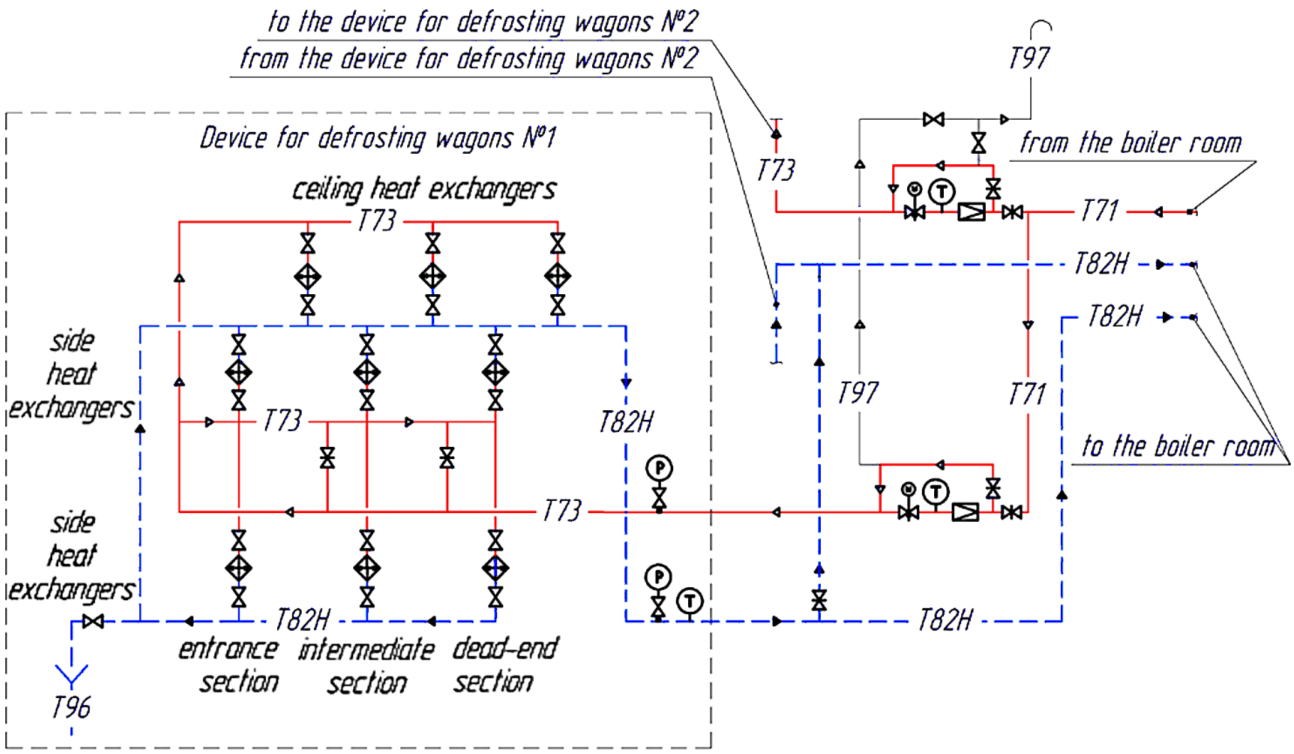

Fig. 2. Basic energy scheme of the existing steam supply system of process defrosting houses (T71 – pipeline of superheated water vapor (P*=0.8 MPa, T=200...220 ˚C); T73 – pipeline of moist saturated steam after reduction (P=0, 2...0.3 MPa); T82N – supercooled condensate pipeline (P=0.18...0.25 MPa, T=102...106 ˚C); T96 – drainage pipeline; T97 – pipeline for blowing ; (notations according to SSTU B A.2.4-8:2009; *- the amount of excess pressure is given)

The power of the system is determined not by technological necessity, but by the possibility of its reliable, trouble-free operation. Many years of experience in the operation of steam defrosting house have shown that for the reliable return of condensate to the boiler room, an excess pressure of 0.18...0.25 MPa must be maintained in the collecting condensate line. This condition leads to the fact that during periods of "hot" downtimes (when the defrosting house is "under steam" without wagons) it is not possible to reduce the temperature in the working volume below 90...95 °C When the time between "hot" downtimes is small, which is typical for stationary operating conditions, the share of such energy overspending in the total mass of usefully used energy is insignificant. If the time of "hot" downtimes turns out to be commensurate with the useful time, then the amount of energy overspending turns out to be correspondingly commensurate with the amount of usefully used energy. As one of the means of reducing heat consumption during periods of "hot" downtimes it is possible to recommend lowering the average air temperature in the working volume of the defrosting house, thereby reducing its thermal power

$$Q_{hp} = Q_H \frac{(t_h - t_{hp})}{(95 - t_{hp})}, \tag{1}$$

where Q_H is the nominal power of the defrosting house, [MW]; t_{hp} – average air temperature in the working volume of the defrosting house in the "hot" reserve [°C]; t_{oda} - average outdoor air temperature during the heating period [12] [°C]; 95 is the average air temperature in the working volume of the defrosting house under nominal operating conditions [°C].

When changing the value t_{hp} from 95 to 50 °C, the capacity of the defrosting device (in the "hot" reserve) is reduced by 50%. Heat energy specifications of the devices for defrosting wagons are presented in Tab. 1.

Table 1. Passport specifications of the devices for defrosting wagons

№	Name of the position	Unit	Defrosting house	
			№1	№2
1	Parameters of the coolant at the entrance:			
	Type of coolant		superheated steam	
	coolant pressure	MPa	0,8…0,85	
	coolant temperature	°C	200…220	
2	Parameters of the coolant after the reducer:			
	type of coolant		superheated steam	
	coolant pressure	MPa	0,2…0,3	
	coolant temperature	°C	200…220	
3	Nominal thermal power of the device	MW	2,7…8,8	
4	Heat carrier consumption in the nominal mode of operation	tons/hour	4,0…13,0	
5	Condensate parameters after the device:			
	pressure	MPa	0,18…0,25	
	temperature	°C	100…110	
6	Average specific costs per unit of production	kW/wagon	128...395	
7	Diameter of conditional passage of pipelines:			
	main steam pipeline	mm	300	
	workshop steam pipeline	mm	150	
	workshop condensate pipeline	mm	80	
	main condensate line	mm	80, 150	

Steam devices for defrosting cargoes are served by a main steam pipeline with a diameter of a conditional pass of Dn300 and a length 400 m Defrosting houses No. 1 and 2 have their own separate condensate pipelines (Fig. 2) with diameters of a conditional pass, respectively, Dn80 (L=400 m) and Dn150 (L=215 m). The method of laying the heating network is open, aboveground (on a flyover). Normative values of the linear density of the heat flow in the main steam pipeline and collective condensate pipelines [13-14] are given in Tab. 2.

Table 2. Normative linear density of heat flow from insulated surfaces of pipelines operating in outdoor air for less than 5000 hours

№ пп	Specifications of the pipeline		Linear norm of heat flow density, W/m
	Diameter of conditional passage, mm	Temperature of the environment in the pipeline, °C	
1	300	200	149
2	150	100	52
3	80	100	37
4	150	50	29
5	80	50	21

As you can see, on condition of equality of the actual value of the linear density of the heat flow from the insulated surface of the pipeline to the normative value (Tab. 2), in the nominal mode of operation of defrosting houses, the amount of heat loss should be less than 1.6% of the nominal power of the devices, and not be the determining factor in the main economy of value.

The structure of thermal energy consumption by a steam device for defrosting cargoes has the following form:

$$Q_{\text{B}} = Q_1 + Q_2 + Q_3 \tag{2}$$

where Q_{B} - the total thermal power, which is released for the technology of defrosting houses from the central boiler house of the enterprise, [MW]; Q_1 the thermal power required to maintain the working temperature in the defrosting house (this cost item includes both the useful heat used for heating and defrosting the cargoes, as well as the energy supplied to compensate for regulatory and unorganized losses in the device), [MW]; Q_2 - thermal power, necessary for compensation of heat losses in main heat networks, [MW]; Q_3 - additional, unaccounted costs associated with non-standard heat losses (for example, with leaks, flying steam, non-return of condensate, etc.), [MW].

Based on the task, the value of the total thermal power (Q_B) was determined using the instrumental method during the "hot" reserve period of the device. The value Q_2 was calculated taking into account the results of thermal imaging. The presence of unaccounted expenses was controlled, and when processing the results, their value was taken equal to zero $Q_3 = 0$. The quantity Q_1 was determined by calculation by known values Q_B, Q_2, Q_3.

In order to determine the current state of the steam supply and condensate collection system and to evaluating the effectiveness of the steam devices for defrosting cargoes №1 and 2, in March 2023, defrosting house #2 was conducted and removed to the "hot" reserve №2. At the time of measurements, a stationary temperature was established in the defrosting house, and stationary parameters were established in the steam supply and condensate collection system (Tab. 3).

Defrosting house №2 was operated without technological load - in "hot" reserve. The wind speed did not exceed 4 m/s, the relative humidity of the outside air was within 58-64%, the atmospheric pressure was equal to 763 mm.

The amount and parameters of the coolant in the steam supply and condensate collection system of steam device for cargoes defrosting №2 operated for established mode of operation during the "hot" reserve period.

The volumes of consumption and parameters of natural gas during the period of tests of defrosting house №2, and the temperature values of the outside air are presented in Tab. 4. The data is obtained from the commercial natural gas consumption accounting system of the enterprise. The system of commercial accounting of natural gas consumption is giving corrections the measured values and brings them to standard conditions [19].

A fragment of the daily water consumption (is given sampling period is 10 hours) and steam during the test period of defrosting house №2 is presented in Tab. 5. The data is obtained from the automated system of control and accounting of the parameters of the production processes and distribution of thermal energy of the enterprise.

A comparing of the results of measurements of a number of values obtained using an automated system of control and accounting of the parameters of the production processes and distribution of thermal energy has allowed to draw a conclusion about the incorrectness of the measurement of some parameters.

Table 3. The coolant parameters in steam supply and condensate collection system of steam device for cargoes defrosting №2

Time	Condensate pressure, MPa	Temperature zone 1, °C	Temperature zone 2, °C	Temperature zone 3, °C	Condensate temperature, °C	Steam consumption, tons/hour	Steam pressure, bar	Steam temperature, °C	Water consumption, tons/hour	Water pressure, MPa	Water temperature, °C
8:00	0.186	89.64	88.29	85.14	102.7	16.92	9.17	207.9	12.3	1.502	104.1
8:30	0.199	90.09	89.64	86.49	103.6	16.56	9.20	207.6	12.02	1.502	104.2
9:00	0.211	90.99	90.54	87.39	104.5	16.02	9.24	207.5	11.67	1.503	104.2
9:30	0.224	91.89	91.44	88.29	104.5	15.89	9.26	207.4	11.42	1.503	104.4
10:00	0.232	91.44	91.44	88.29	106.31	15.35	9.28	207.0	11.13	1.503	104.3
11:00	0.231	91.35	91.45	89.51	107.41	14.83	9.32	206.8	10.78	1.502	104.2

In particular, when comparing the mass flow rates of feed water, superheated steam, and the volumetric flow rate of consumed natural gas, it turned out that the devices for measuring the flow rates of feed water and superheated steam, installed on part of the pipelines of one of the boilers, show, firstly, different values, and secondly, these values correlate poorly with the values obtained from the commercial natural gas consumption accounting system. The measurement points required verification and could not be used for further research.

The value of the total thermal power (Q_{B}), which was released to the technology from the central boiler room of the enterprise during the period of the experiment, was calculated. The theoretical amount of thermal energy released during the combustion of natural gas was determined by the expression:

Table 4. The fragment of data on natural gas consumption during one day

Time	Gas temperature, °C	Absolute gas pressure, MPa	Gas volume, m^3 (reduced to standard conditions, 20 °C, 101325 Pa)	Outside air temperature, °C
01:00	-0.45	0.466	1751.5	
02:00	-0.66	0.472	1845.6	-5.6
03:00	-1.38	0.469	2120.2	
04:00	-1.08	0.475	1785.2	
05:00	-1.17	0.477	1749	-5.6
06:00	-2.01	0.468	2112.1	
07:00	-0.54	0.464	2053	
08:00	4.04	0.476	1230.6	-3.3
09:00	7.39	0.487	1375.9	
10:00	9.71	0.487	1600.1	
11:00	11.64	0.488	1551	0.3
12:00	13.64	0.493	1465.4	
13:00	16.32	0.499	954.2	
14:00	18.81	0.5	805.5	2.8
15:00	18.65	0.5	780.6	
16:00	17.67	0.501	761.6	
17:00	15.29	0.498	794.8	1.1
18:00	12.45	0.494	856	
19:00	9.33	0.49	969.5	
20:00	8.66	0.49	941.7	0.6
21:00	8.66	0.488	933.8	

$$Q = \frac{273.15}{293.15} Q_{\mathrm{w}}^{\mathrm{l}} \cdot V_{\mathrm{h}} \cdot 10^{-6} \tag{3}$$

where Q - is the theoretical amount of thermal energy released during the combustion of natural gas, [GJ/hour]; $273.15/293.15$ - a numerical coefficient that takes into account the change in volume during the transition from standard to normal conditions; $Q_{\mathrm{w}}^{\mathrm{l}}$ - lower working heat of combustion of natural gas, [kJ/nm^3]; V_{h} - hourly consumption of natural gas reduced to standard conditions [10], [m^3/hour] (Tab. 4); 10^{-6} - a numerical coefficient that takes into account the transition from [kJ] to [GJ].

Table 5. Dynamics of water and water vapor consumption during the day

Water				Third party consumer					Steam for own needs and production		
				D, 150		D, 100					
Time	P OCB, bar	T OCB, °C	PPV consumption, tons	Q steam, tons	P steam, bar	Q steam, tons	P steam, bar	Amount of thermal energy, MW	Q steam for production, tons	P steam for production, bar	Q steam for own needs, tons
1:00	2.4	100	0	2.59	9.11	0	3.57	0	9.43	8.23	2.37
2:00	2.4	100	0	2.61	9.15	0	3.55	0	14.21	8.89	2.26
3:00	2.4	100	0	2.6	9.14	0	3.52	0	19.87	9.01	2.23
4:00	2.4	100	0	2.59	9.09	0	3.48	0	19.93	10.72	2.14
5:00	2.4	100	0	2.59	9.09	0	3.46	0	20	15.53	2.08
6:00	2.42	100	2.21	2.58	9.09	0	3.48	0	20	15.58	2.02
7:00	2.49	100	6.08	2	9.26	0	3.47	0	19.78	15.1	1.91
8:00	2.4	100	0.03	1.96	9.42	0	3.47	0	20	16	2.19
9:00	2.41	100	0	1.98	9.54	0	3.53	0	20	16	2.37
10:00	2.4	100	0	2	9.61	0	3.57	0	4.6	5.37	2.44

The useful amount of thermal energy spent on obtaining superheated water vapor was determined by the expression:

$$Q_{\mathrm{SV}} = D_{\mathrm{SV}} \cdot (i_{\mathrm{SV}} + i_{\mathrm{SW}}) \cdot 10^{-3} \tag{4}$$

where Q_{SV} - is the thermal energy spent on obtaining superheated water vapor, [GJ/hour]; D_{SV} - mass consumption of superheated steam, [ton/hour]; i_{SV}, i_{SW} - enthalpy of superheated steam and feed water, respectively [kJ/kg] [20]; 10^{-3} - numerical coefficient, takes into account the transition [GJ/MJ].

The work efficiency of the steam generator, which is characterized by its efficiency - gross, was calculated according to the direct balance:

$$\eta = \frac{Q_{SV}}{Q} \quad (5)$$

Below is a graphic dependence of the work efficiency of the steam generator on the value of its useful power (Fig. 3).

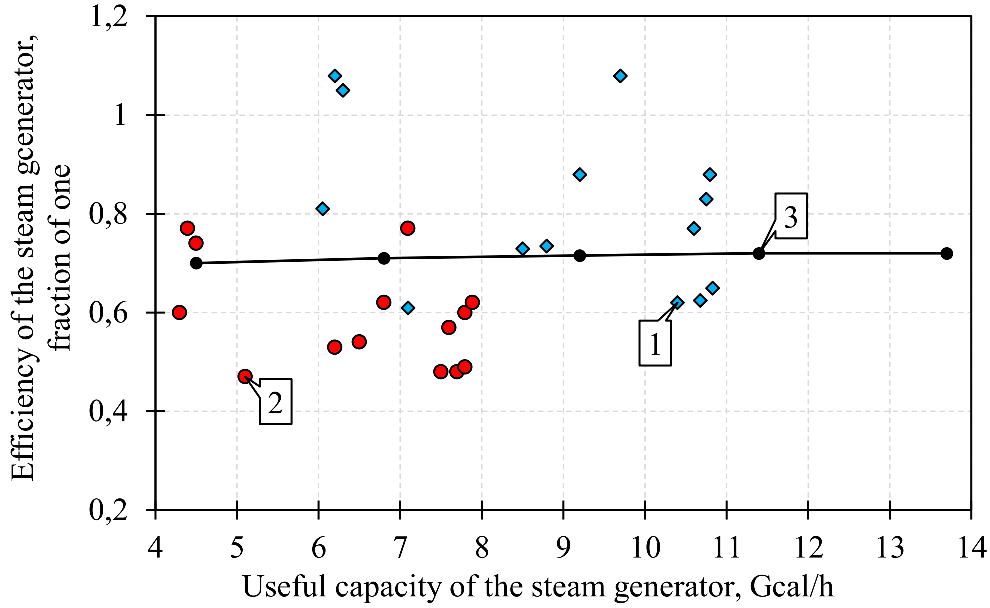

Fig. 3. Dependence of the efficiency of the steam generator on the amount of useful thermal power:
1 - dependencies (1-4); 2 - under the condition that the mass flow rate of steam is equal to the mass flow rate of feed water; 3 - average characteristic of the array of points 1

The useful power of the steam generator is laid down on the abscissa axis, $Q_{\text{ПП}}$, and its gross efficiency is on the ordinate axis. In blue (1) marked the values obtained according to dependencies (1-4). Red colour (2) indicates the values obtained under the condition that the mass flow rate of steam is equal to the mass flow rate of feed water. The line (3) is an average characteristic of the array of points (1), which in turn are calculated but based on measured steam flow rates (this is the reason why the graph shows physically incorrect indicators of steam generator efficiency more than 1 fraction of one). Values taken from the mode map of the boiler (provided by the company) are indicated in black. As you can see, values (2) are poorly correlated with data (1) and (3), which may indicate incorrect operation of the flow measuring device installed in the supply line. Value (1) have a large scatter of data, in addition, a number of values ($\eta > 1$) have unphysical results. This may indicate the function of the sensors in the control system (measurement and regulation) at the highest and most unfavourable error. Such results are very rare, which is inside the probability of such a high error.

Regression models that describe the obtained set of values have a low coefficient of determination ($R^2 < 0{,}5$), which shows a large scatter of the data and a poor fit of the model. Considering of all the above, we assume that the useful amount of thermal energy (which is released by the central boiler room of enterprise for technology, own needs, and third-party consumers) is determined by the calculation method using data from the commercial natural gas consumption accounting system and data from the mode map of the steam generator. In addition, in the amount of 3% of the mass of the produced steam, the value of its continuous purging is taken into account.

$$Q_{\mathrm{SV}} = \frac{273{,}15}{293{,}15} Q_{\mathrm{W}}^{\mathrm{l}} \cdot V_{\mathrm{h}} \cdot 10^{-6} \cdot \eta \cdot (1 + 0{,}03)^{-1} \quad (6)$$

where $(1 + 0{,}03)$ – is a numerical coefficient that takes into account the value of continuous purging.

The amount of superheated steam that is released to consumers:

$$D_{\mathrm{SV}} = \frac{Q_{SV}}{(i_{\mathrm{SV}} + i_{\mathrm{SW}}) \cdot 10^{-3}}. \quad (7)$$

The results of calculations of quantities Q_{SV} (1) and D_{SV} (2) are presented in Fig. 4.

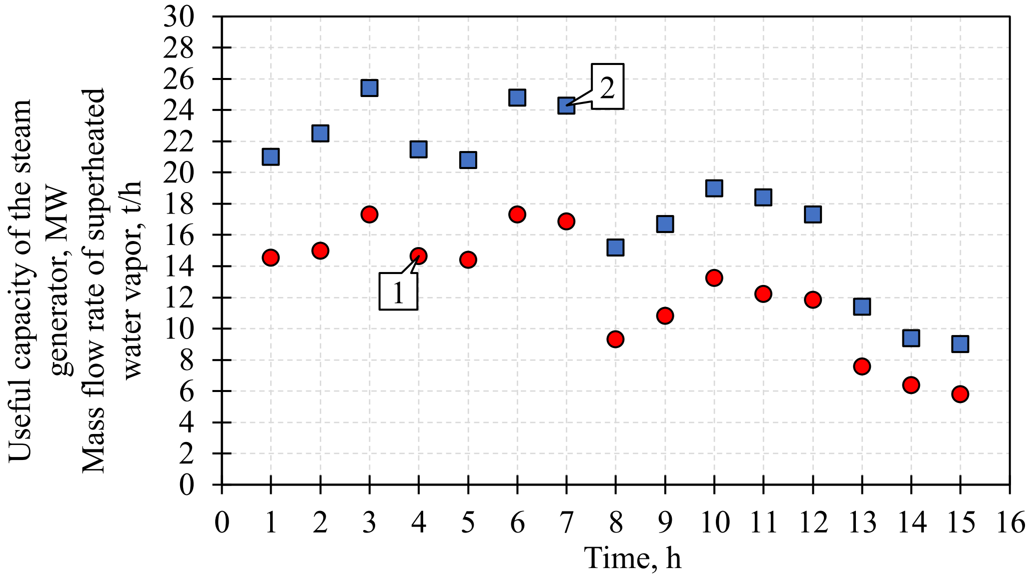

Fig. 4. Dynamics of changes in the useful thermal power of the object Q_{SV} and consumption of superheated water vapor D_{SV} at the time of measurements, 1 – useful thermal power; 2 – consumption of superheated water vapour

The main items of the balance of thermal energy costs during the measurements were:

- expenses for own needs (hot water preparation, feed water deaeration);
- production costs (devices for defrosting cargoes and other technological consumers);
- costs of third-party consumers;
- expenses for heat supply of houses and structures (water thermal network).

Subtracting from the value $Q_{\Pi\Pi}$ of current articles of thermal energy consumption, we determine the value released to the device for defrosting cargoes Q_{B}.

The calculation results are presented graphically (Fig. 5) as a function of the total mass flow rate of superheated steam (1), the mass flow rate of steam per defrosting house (2) and the dependence of the average temperature in the defrosting house on time (3). During the measurement period, the average mass consumption of superheated water vapor per defrosting house, during the "hot" reserve period, was 11.3 t/h. The obtained mass flow rate of superheated water vapor to the device for defrosting cargoes, in the "hot" reserve, approaches the value of the mass flow rate of the coolant when the defrosting house is operating in the nominal mode of operation (under load). The average value of the total thermal power (Q_{B}), which is released to the technological consumer in the "hot" reserve, was equal to 6.7 MW.

The value (Q_2) was determined by using the calculation-experimental approach, which is based on the results of the thermal imaging survey, which was carried out by the Dali TE thermal imager, and the calculation positions partially outlined in this article. The examples of processing the results of thermal imaging are in Fig. 6.

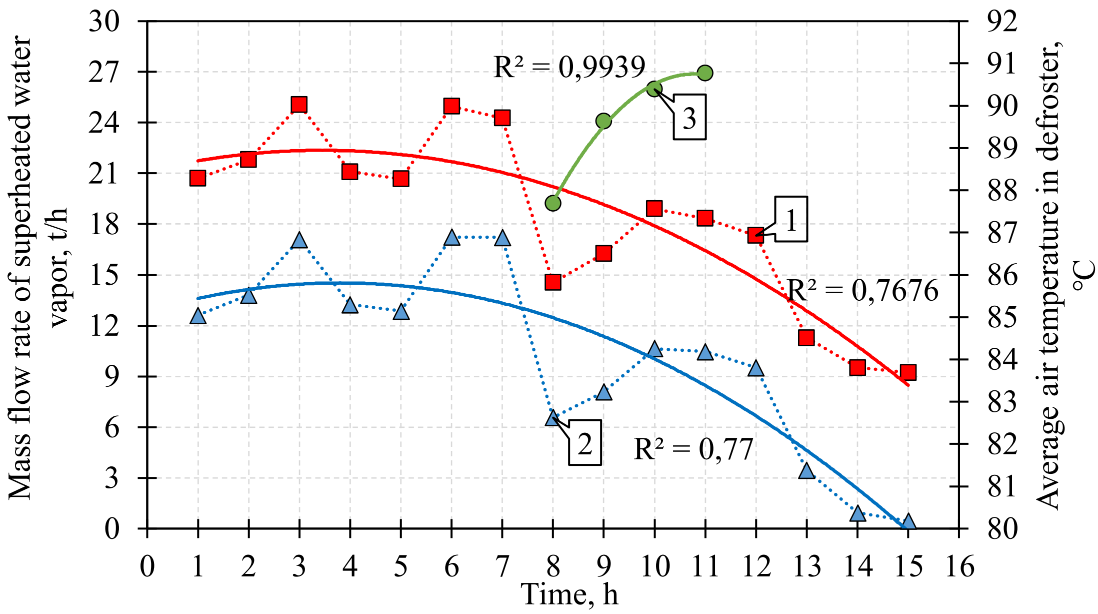

Fig. 5. Results of the calculation of the mass flow rate of superheated water vapor to the device №2 for defrosting cargoes

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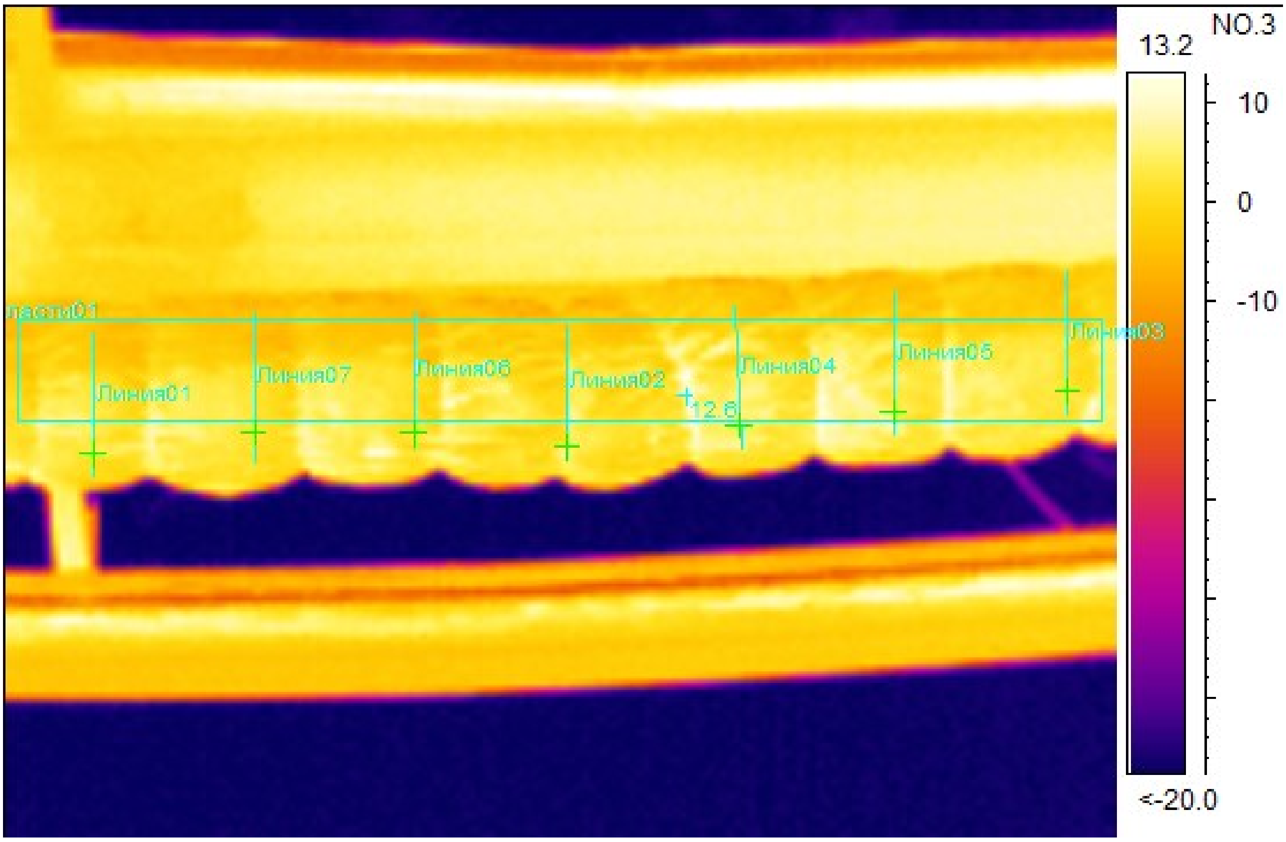

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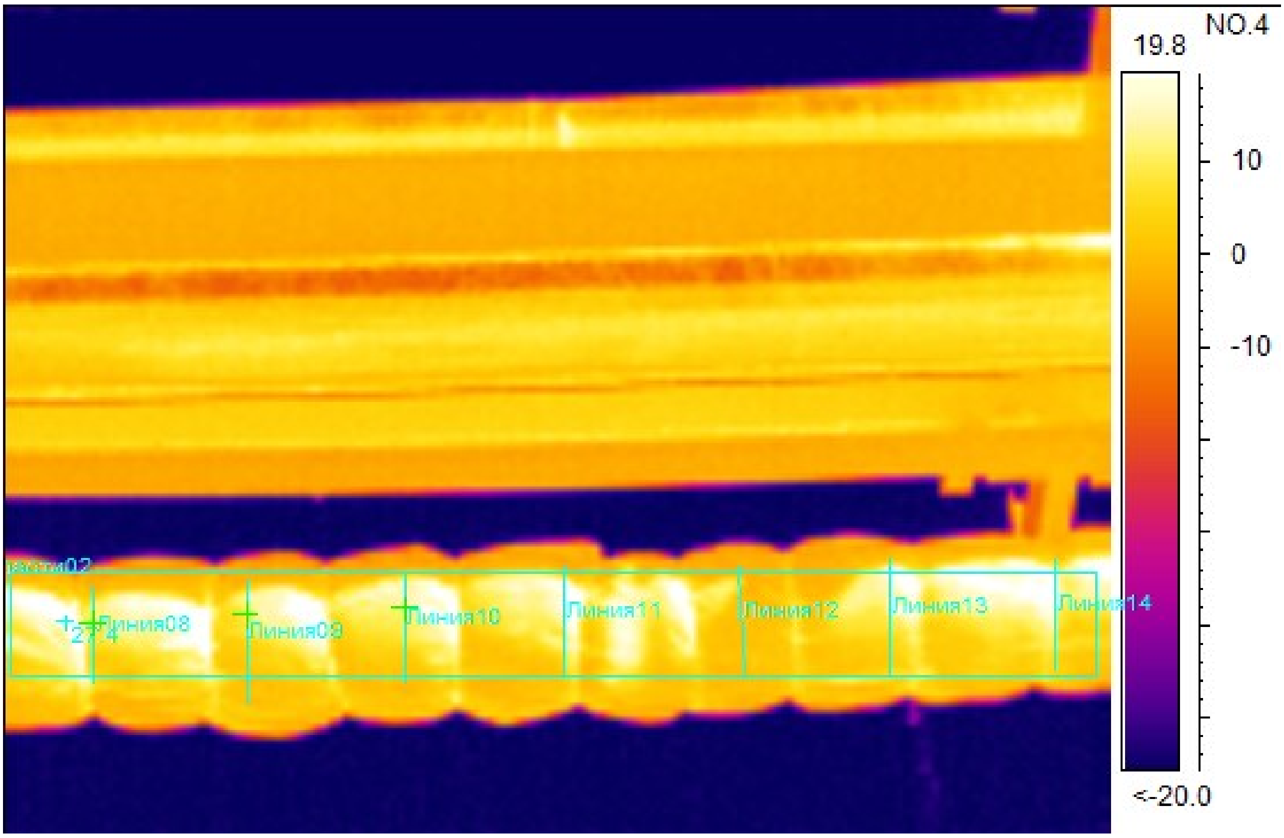

Fig. 6. Example of thermograms for sections of the condensate line from defrosting house №2

The surface of the pipeline was divided into sections located in the planes perpendicular to the axis of the pipeline (Fig. 6, a, б), the averaging the temperature was conducted the surface on each section (τ).

The distribution of temperatures on the surface of the pipeline is uneven nature, the deviation of extreme temperature values from their average value exceeds the value of $\Delta\tau = 0{,}5 \dots 1{,}0$ °C, determined by the formula Eq. 8 and accepted as the limit when determining the presence and sizes of defective areas.

$$\Delta\tau = \frac{\Delta t}{R_0} \cdot \frac{1-r}{\alpha_3 \cdot r}, \tag{8}$$

where $\Delta\tau$ – is the temperature difference between the surfaces of the base and defective areas, [°C]; Δt – calculated temperature pressure, defined as the difference between the internal temperature of the medium in the pipeline and the external air temperature under the calculated conditions of insulation design, [°C]; R_0 – calculated value of the design heat transfer resistance of the basic section [m^2°C/W]; r – is the relative resistance to heat transfer, which is taken as 0.85 in the evaluation calculations [23]; α_3 – is the heat transfer coefficient of the external surface of the pipeline, calculated taking into account the operating conditions of the surface at the time of measurement, [W/(m^2·°C)].

Similar actions were conducted during the analysis of thermograms of the main steam pipeline. A fragment of the initial data and the results of the calculations of the specific indicators of heat loss through openly laid down steam pipelines and condensate pipelines are presented below (Tab. 6).

As can be seen from the obtained results, the path heat losses in main pipelines are different. If for the main steam pipeline, they satisfy the regulatory requirements (the measured heat flow is ½ of the normative value), then the actual losses in the main condensate pipeline from defrosting house №2 are 1.5 times higher than the normative ones.

The number of losses due to movable pipeline supports is taken into account by introducing an additional equivalent length of the pipeline, by analogy with the hydraulic method of equivalent lengths [22]. The measurements showed that one support of the OPP2 type is thermally equivalent to the normative losses of 1 linear meter of the pipeline. The total number of supports installed on the main steam pipeline is equal to 30 units, installed on the main condensate pipeline – 30 units.

The total heat losses in the main steam and condensate pipelines at the time of the measurements were:

$$Q_2 = (400 \cdot 0{,}5 + 30 \cdot 1) \cdot 150{,}3 + (215 \cdot 1{,}5 + 30 \cdot 1) \cdot 52{,}9 =$$
$$= 34569 + 18647 = 53216\ W.$$

Table 6. The results of heat loss calculations in steam and condensate pipelines

№	Name of the pipeline	Pipeline surface temperature, °C							τ_m, °C	t, °C	ε	t_T, °C	w, m/s	δ, mm	$D_{ЗН}$, m	$\alpha_з$, W/(m²°C)	q^P_L, W/m	$(q_L)_t$, W/m	q_F, W/m²	ε_q, %	Note
		1	2	3	4	5	6	7													
1	Condensate pipeline (defrosting house №2)	4,3	6,1	3,3	6	4,7	2,8	3	4,3	0,3	0,96	100	3	80	0,31	18,1	70,7	52,9	72,6	33,7	Linear losses
2		10,3	9,1	11,6	15,2	6,1	13,1	12,3	11,1	0,3	0,96	100	3	80	0,31	18,2	191,2	52,9	196,3	261,3	
3	Steam pipe (defrosting house №2)	4	2,9	3,4	3,8	4,4	5	4,6	4,0	0,3	0,96	200	3	100	0,5	16,9	98,3	150,3	62,6	-34,6	Linear losses
4		5,1	5,9	2,8	2,3	2,5	3,3	3,9	3,7	0,3	0,96	200	3	100	0,5	16,8	89,6	150,3	57,0	-40,4	
5	Steam pipe (defrosting house №2)	0,9	0,5	0,6	1	1,9	0,7	2,6	1,2	0,3	0,96	200	3	100	0,5	16,8	23,0	150,3	14,7	-84,7	Removal 90°
6		3,1	2,2	1,8	2,1	0,9	1,5	2,2	2,0	0,3	0,96	200	3	100	0,5	16,8	44,2	150,3	28,1	-70,6	Linear losses

Using the dependence of Eq.1 the power (Q_1) of the defrosting house in the "hot" reserve was defined:

$$Q_1 = 6{,}70 - 0{,}05 = 6{,}65\ \text{MW}.$$

Thus, the measurements carried out during the start-up and operation of the unit for defrosting wagons №2 in the "hot" reserve showed that the "idle" power of the unit - 6.65 MW (about 11 tons of steam per hour) - is close in its size to the nominal power. A similar picture is typical for large thermal objects (for example, chambers for heat-humidity treatment of reinforced concrete structures), in which the regenerative heat consumption for heating and maintaining parameters in the chambers is many times higher than the useful used heat.

The variant of modernization of the scheme of the energy principle of the defrosting house according to the conclusions described above is shown in Fig. 7.

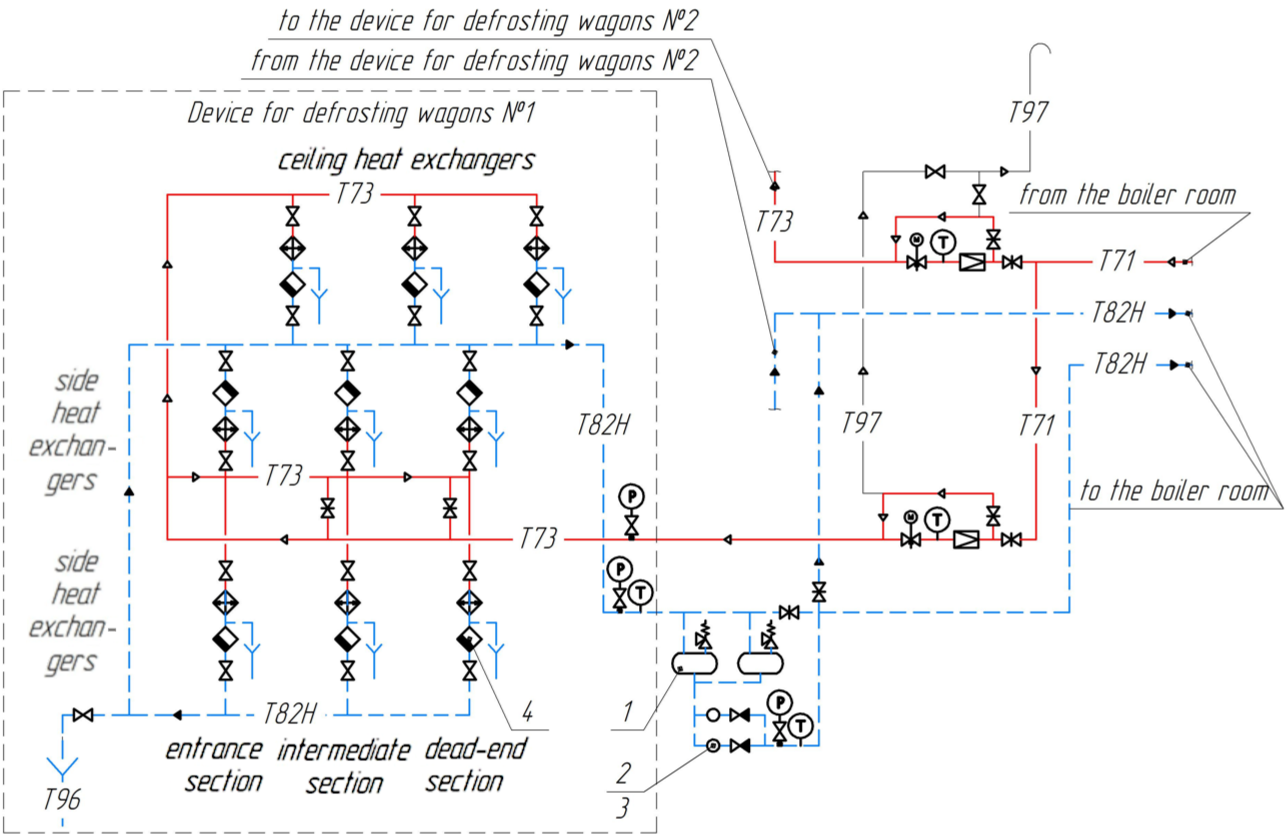

Fig.7. Basic energy diagram of the modernized steam supply system of technological defrosting houses (modernization positions: 1 – condensate collecting tank, 0.005 MPa; 2 – Ks-12-50 condensate transfer pump; 3 – device for frequency regulation of the Ks-12-50 Danfoss VLT 2800 A3 pump drive, 4 – Spirax Sarco FT14HC steam trap). Other designations correspond to the designations given in Fig. 2)

Characteristic of the upgraded system:

1. Operation in stationary and periodic modes ("hot" steam supply to the condensate pipeline is excluded, it becomes possibility to adjusting the system power).

2. Transportation of subcooled condensate is provided by a transfer pump (eliminating the need to maintain a minimum overpressure in the system).

3. The thermal resistance of the enclosing structures of the building has been increased (the amount of recuperative heat loss through the enclosure is reduced).

4. Heat losses in steam and condensate pipelines are reduced (due to the increase in the thermal resistance of the heat-insulating layer, the amount of path heat losses in the system is reduced).

The changes made both in the design of the heat supply system and in the design of the defrosting house allow to maintain the value of specific energy consumption at the level of 0.11...0.34 Gcal/(h·wagon) in the mode of its periodic operation. Additional measures for the modernization of the object allow to obtain the value of specific energy consumption of less than 0.11 Gcal/(h·wagon), namely:

a) modernization of steam and condensate management with laying near the main steam line of the condensate pipeline-satellite (allows to refuse the practice of "hot" reserves of technological equipment);

b) partial replacement of steam heating devices to gas infrared systems (it is important to follow the balance here, since as a rule, steam boiler houses of industrial facilities have a backup fuel supply, which allows increasing the facility's energy independence);

c) transition from convective heating devices to infrared (allows to keep the air temperature in the defrosting house building lower).

Conclusions. The main condition that ensures the reduction of heat consumption consists either in reducing the parameters in the working mode of the unit during the periods of "hot" reserve, or in the complete shutdown of the defrosting house in the absence of a useful load. In the first case, there will be a power reduction, directly proportional to the decrease in the temperature pressure, the implementation of such a scenario is possible when switching from a gravity condensate return system to a system with forced condensate return. The second case, in addition to the transition to a system with forced return of condensate, requires additional modernization of the main steam pipeline with the laying of a condensate pipeline-satellite, necessary for the constant removal of accompanying condensate and prevention of the danger of "freezing" of the system. Besides, additional insulation is needed by the condensate pipes from the defrosting house №2. The proposed variant of modernization of the scheme of the energy principle of the defrosting house gives higher effectiveness and durability.

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ДОСЛІДЖЕННЯ ТЕХНОЛОГІЧНИХ СИСТЕМ ПАРОПОСТАЧАННЯ ТА МОЖЛИВІ ШЛЯХИ ПІДВИЩЕННЯ ЇХ ЕНЕРГЕТИЧНОЇ ЕФЕКТИВНОСТІ НА ПРИКЛАДІ ПРИСТРОЇВ ДЛЯ РОЗМОРОЖУВАННЯ ВАГОНІВ

Анотація. *Системи паропостачання промислових підприємств різних сфер діяльності залишаються актуальними насамперед через теплофізичні особливості використання енергії фазових переходів водяної пари в процесі її конденсації. Крім того, в країнах Східної Європи традиційно значна кількість діючих котелень є паровими і вибір енергоносія визначається саме наявністю характерного джерела. Разом з тим, процеси вантажної логістики сьогодні забезпечуються переважно морським транспортом, а доставка в порти традиційно здійснюється залізничним транспортом, що, в свою чергу, вимагає використання системи розморожування і перевалювання сипучих продуктів. У статті наведено результати досліджень енергоспоживання пристроїв для розморожування вагонів портової інфраструктури та запропоновано схемні тепломеханічні рішення для модернізації системи паропостачання розморожувальної камери. За результатами дослідження наведено значення витрат теплової енергії на одиницю продукції за існуючого сценарію та після модернізації. Наведено характерні технологічні та теплотехнічні особливості існуючої схеми пристроїв для розморожування вагонів портової інфраструктури України та заходи з модернізації таких систем,* *впровадження яких дозволить досягти значного зниження споживання теплової енергії в маневрових нестаціонарних режимах роботи розморожувальних камер. Серед цих заходів: модернізація пароконденсатного господарства шляхом підвищення термічного опору теплоізоляції конденсатопроводів та встановлення насоса, а також часткова заміна парових пристроїв обігріву на газові інфрачервоні системи. Доцільність застосування інфрачервоної системи обігріву вагонів разом із паровими регістрами обґрунтована.*

Keywords: система паропостачання, енергетична ефективність, технологічний тепляк, пристрій розморожування вагонів, тепловтрати паропроводів.