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SYSTEM ANALYSIS OF THE SECONDARY CONDENSATION UNIT IN THE CONTEXT OF IMPROVING ENERGY EFFICIENCY OF AMMONIA PRODUCTION

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Наведені результати досліджень по ефективності процесів теплообміну технологічного комплексу вторинної конденсації типового для України агрегату синтезу аміаку. Виконана ідентифікація процесів теплообміну конденсаційної колони. Методом математичного моделювання здійснено синтез енергоефективного технологічного оформлення стадії вторинної конденсації із застосуванням лише тепловикористовуючих холодильних систем. Запропонована технологія забезпечує зниження споживання електроенергії і природного газу відповідно на 60 кВт·год/т NH_3 і 1,2 м^3/ т NH_3

Ключові слова: виробництво аміаку, вторинна конденсація, ідентифікація, математичне моделювання процесів теплообміну, енергоефективність

Приведены результаты исследований по эффективности процессов теплообмена комплекса вторичной конденсации типового для Украины агрегата синтеза аммиака. Выполнена идентификация процессов теплообмена конденсационной колонны. Методом математического моделирования осуществлен синтез энергоэффективного технологического оформления стадии вторичной конденсации с применением только теплоиспользующих холодильных систем. Предложенная технология обеспечивает снижение потребления электроэнергии и природного газа соответственно на 60 кВт·ч/т NH_3 и 1,2 м^3/т NH_3

Ключевые слова: производство аммиака, вторичная конденсация, идентификация, математическое моделирование процессов теплообмена, энергоэффективность

1. Introduction

The most important product of the chemical industry is synthetic ammonia. It is widely used in various areas and especially in the production of nitrogenous mineral fertilizers, which are a powerful lever in improving the yields of agricultural crops. This is, therefore, no coincidence that the volume of production of ammonia has been growing and reached 214 million tons in 2015, considerably exceeding the level of 2008 by almost 19 % [1]. In this case, the main raw material in the technology of production of ammonia now, and will remain over the next decades, is natural gas, which provides, if compared to heavy oil and coal as raw materials, a decrease in energy cost by 1.3 and 1.7 times, respectively [2].

Modern plants for the synthesis of ammonia are sophisticated energy technological complexes with a large number of branches that are closely interrelated. They are, in spite of the variety of hardware technological structures, employed catalysts and equipment for the implementation of processes, built in almost all countries by a unified ideology of the company “M. W. Kellogg & Co.” (USA). In this case, the unit of synthesis, where the production ammonia is actually obtained, is based on the traditional closed circulation scheme of Haber-Bosch [3].

The basic units for the nitric industry of Ukraine are those with medium synthesis pressure of capacity 1360 t/per day (series AM-1360), which are considerably inferior in power consumption to the technologies of such world-known manufacturers as “Haldor Topsoe” (Denmark), “Imperial Chemical Industries” (United Kingdom), “Kellogg Braun & Root (USA), by almost 25 % [4] and by electricity by more than 2 times. The latter is largely due to the use at the stage of condensation of energy-intensive turbo-compressor refrigeration unit with an electric drive, whose share of electricity is up to 40 % of the total consumption by the unit of synthesis. In this regard, improving the energy efficiency of condensation systems acquires special relevance in a general process of the modernization of domestic units for ammonia synthesis.

2. Literature review and problem statement

Technical and economic indicators (energy efficiency) of the process of extracting production ammonia are largely due to the use of one or another method, in particular absorption or condensation. The latter is more economical for the systems of synthesis at pressure over 10 MPa [4], which is typical for the production of ammonia that exists in Ukraine. The process of condensation takes place by cooling the circulation gas (CG), during which the elasticity of ammonia saturated vapor decreases, and hence its concentration in the gas mixture. In this case, excessive amount of ammonia in CG affects the increase in energy consumption on the pressurizing and circulation of the gas mixture [5]. In connection with this, the process of synthesis should be conducted at minimum content of ammonia in the gas at the inlet of synthesis column.

The extraction of production ammonia from CG in the cycle of synthesis is carried out in almost all technologies of the leading companies-manufacturers of ammonia units “Kellogg Braun & Root”, “Thyssen Krup Uhde”, “Haldor Topsoe” and others by using the turbo-compressor cooling systems [6]. Their efficiency under existing scheme of steam formation is due mainly to the use of compressor’s drive to separate medium pressure (4 MPa), which is obtained through the utilization of high-potential heat at the stages of reforming, and by conducting the synthesis of ammonia at low pressure of 10÷15 MPa [7]. The application of this cooling technology in the units that are operated in Ukraine and designed by manufacturers “TEC” (Japan) and “GIAP” (Russia) is impossible due to the higher pressure at some stages and, especially, at the stage of synthesis (exceeding 15 MPa). This would require a large amount of backward linkages in the unit of synthesis for the adjustment of hardware-technological design of the entire production. Given the economic condition of the country, we should not expect free financial resources for such adjustment. Under such conditions, as noted by authors in [8], it is advisable to conduct gradual modernization of separate units, and, in particular, a block of secondary condensation.

As evidenced by the analysis of the above technologies existing in Ukraine, increased pressure causes an increase in the capacity of compressors for gas synthesis. Under these circumstances, there is not enough water vapor to drive the compressor systems. In this case, the lack of its amount is obtained in an additional steam boiler through the consumption of up to 4 000 nm³/h of natural gas that significantly reduces the efficiency of these units. It is therefore not incidentally that in the units of synthesis by the technology of “GIAP” in the block of secondary condensation (Fig. 1) they used two ammonia absorption-refrigeration units (ARU) with total refrigeration capacity of 6.28 MW. The work of these units is enabled by the utilization of low-potential heat of converted gas and steam-gas mixture at temperatures of 135 °C and 126 °C, respectively. The technology of applying ARU is definitely more economical compared to the utilization of high-potential heat of water vapor for the separation of medium pressure in compressor cooling systems. However, providing the required refrigeration capacity of 11.16 MW for cooling CG to the necessary temperature of – 5 °C is impossible by using ARU only. This is due to the absence in a synthesis unit of utilization flows with temperature level above 100 °C, required for the provision of ARU operation through the supply of heat to its generator-rectifier. In this connection, they used a turbo-compressor refrigeration unit ATC with electric drive of capacity up to 4 000 kW·h [5], which reduces the cost of production.

A characteristic feature of this unit is the existence of a condensation column to provide for the completeness of cold recuperation. The column is a vertical high-pressure apparatus. Inside the case are a heat exchanger and a separator. As a result of applying the air cooling at the stage of initial condensation, the range of change in temperature and concentration of CG at the inlet to the inter-pipe space of the condensation column is 35÷45 °C and 9÷12 % by volume, respectively. After cooling to temperature 13÷23 °C, CG and partially condensed ammonia from the central pipe of the column proceeds to the pipe space of low-temperature evaporators ARU and ATC. In the evaporators, CG is cooled by ammonia boiling in the inter-pipe space under pressure not larger than 0.296 MPa. As a result of such changes in temperatures and concentrations, there is also a change in the temperature of cooling of CG from –5 °C to 5 °C. However, increase in the temperature even by 1 °C leads to an increase in the consumption of natural gas by 38 m³/h into additional steam boiler of receiving water vapor at pressure of 10 MPa, which supports the drive of compressor turbine of compression of ABC and CG [5]. A mixture of cooled gas and condensed ammonia enters by inverse flow a separation part of the column for the separation of liquid ammonia from gas. In the separating part of the column, there is a direct heat exchange between fresh ABC and CG. The resulting gas mixture goes through the basket with Raschig rings, where it is additionally separated from the drops of liquid ammonia. Then the gas mixture ascends the tubes of heat exchanger and cools the direct flow of CG. After the condensation column, CG with ammonia concentration up to 4.5 % by volume and temperature up to 38 °C, in line with a standard procedure, is directed through intermediate heat exchangers to the synthesis column.

The aforementioned indicates that the processes of heat exchange in the condensation column are rather complex and are accompanied both by the condensation of ammonia in the presence of a large amount of inerts at high pressure and its evaporation in the bottom part. However, experimental data on the coefficients of heat transfer and heat release are missing in the scientific periodicals. In this case, efficiency of the processes of heat exchange ultimately affects the distribution of temperatures and loading of refrigeration systems and, therefore, the energy efficiency of condensation unit as a whole.

Applying ATC in this circuit predetermines during operation an excessive consumption coefficient by electricity. Under such circumstances, an alternative to ARU and ATC is the steam-ejector refrigeration units (SRU). They make it possible, through the application of refrigerating agent with low boiling point, to utilize the low-potential heat of material flows with temperature level even below 90 °C. However, at present, in all units of synthesis without exception this heat is released in the air-cooling units into the environment [4]. In this case, only one article contains information about the possibility of using SRU with such cooling agents in relation to the units of ammonia synthesis. However, the proposed technology does not provide for a CG cooling temperature decrease to the standard level – 5 °C, only to 0 °C [9].

SRU compared with ARU and ATC is characterized by low thermal-dynamical efficiency. Its improvement can be obtained by increasing the pressure and the boiling temperature of coolant in evaporator [9]. Therefore, determining the position of SRU evaporator for cooling CG in the general scheme of a condensation unit acquires special importance. This requires conducting separate studies to define economic feasibility of applying the cycle of SRU. The studies are most effectively performed when using a method of mathematical modeling. This is a key method in a systematic approach towards the processes of creating not only effective hardware-technological design of objects, but control systems as well [10].

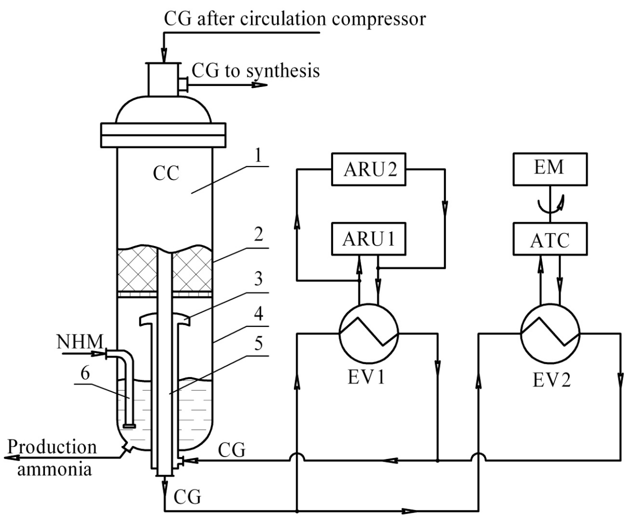


Fig. 1. Schematic of the unit of secondary condensation: CC — condensation column; CG — circulation gas; NHM — nitrogen-hydrogen mixture; ARU — absorption-refrigeration unit; ATC — ammonia turbo-compressor refrigeration unit; EV — evaporator; EM — electric motor; 1 — tubular heat exchanger with partitions; 2 — Raschig rings; 3 — bump stop; 4 — column case; 5 —central pipe with thermal insulation; 6 — NHM discharge pipe

3. The aim and tasks of the study

The aim of present study is to develop energy-efficient hardware-technological design of the process of secondary condensation using the steam-ejector refrigeration plants. This will make it possible, by removing ATC from the circuit and lowering the temperature of secondary condensation, to reduce consumption of electricity and natural gas in additional steam boiler.

To achieve the set aim, the following tasks have to solve:

– to examine effectiveness of the processes of heat exchange in a condensation column with determining its basic indicator – coefficient of heat transfer;

– to establish in the process of solving the problem on identifying the mathematical model of condensation column the equation for determining the coefficients of heat transfer, heat exchange and the concentration of ammonia in the circulation gas at the outlet of a separation part;

– to analyze and identify material low-potential heat flows at the level of temperatures below 90 °C in the unit of synthesis to provide the operation of SRU;

– to perform the synthesis of hardware-technological design of the unit of secondary condensation and to determine effectiveness of its application.

4. Materials and methods of examining the processes of heat exchange

The studies were conducted under industrial conditions in the section of secondary condensation whose schematic is shown in Fig. 1, using a well-known method of passive registration experiment [11]. Collection of information about the parameters was carried out with the help of control tools of the microprocessor information-control complex TDS-3000 at the central point of control and laboratory analyses.

By using laboratory tests, we determined the content of ammonia in circulation gas by the known procedure, which is based on the absorption of ammonia by water with subsequent titration of ammonia water by sulfuric acid. By the amount of this acid at titration we established the concentration of ammonia. To separate the samples of gas, we used the Gaillard flasks of capacity 300 ml. Quantitative composition of the selected samples by their content of methane, argon, hydrogen and nitrogen was defined by using the chromatograph “Tsvet-102” in the central laboratory of the enterprise. The sampling for laboratory analysis was held once per shift since the frequency of collecting all other data on the parameters of work of the station of secondary condensation was also once per shift.

The resulting compilation of the array of experimental data was carried out after evaluating the convergence between material balance, heat fluxes of pipe Φ_P^{CC} and inter-pipe Φ_{IP}^{CC} space of tubular heat exchanger in the condensation column. In case the discrepancy between these fluxes exceeded 5 %, these modes were excluded from further consideration.

The efficiency of the heat exchange process was evaluated by coefficient of heat transfer K_E (W/m²·K), which was determined by the mean magnitude Φ of heat flux by equation:

$$K_E = \Phi / F \cdot \Delta\Theta_{MD}, \quad (1)$$

where $\Phi=0.5\cdot(\Phi_P^{CC}+\Phi_{IP}^{CC})$ is the heat flux of heat exchanger of the upper part of the condensation column, W; F=2180 m² is the surface of heat exchange; $\Delta\Theta_{MD}$ is the mean difference in temperatures, °C.

Calculation of heat flow Φ was carried out by a specially developed method. The necessity of its development is related to the absence in the industrial unit of devices for measuring the temperature of CG at the outlet of separation part of the column Θ_P^S. According to this procedure, temperature Θ_P^S. was determined by solving the system of equations of thermal balance of the separation part. The basic equations can be represented in the following form:

$$\Phi_{NHM} = M_{NHM} \cdot C_{NHM} \cdot \left(\Theta_{NHM}^S - \Theta_P^S\right); \quad (2)$$

$$\Phi_{NHM} = \Phi_{IP}^S + \Phi_{EV}^S; \quad (3)$$

$$\Phi_{EV}^{S}=G_{EV}^{S}\cdot r^{S}; \tag{4}$$

$$G_{EV}^{S}=V_{NHM}\cdot\frac{a_{NH_3}^{S}\cdot 0{,}771}{100-a_{NH_3}^{S}}; \tag{5}$$

$$\Phi_{IP}^{S}=M_{IP}^{S}\cdot C_{IP}^{S}\cdot\left(\Theta_{P}^{S}-\Theta_{P}^{EV}\right)+M_{EV}^{S}\cdot r^{S}+ +\left(M_{L}^{S}-0{,}5\cdot M_{EV}^{S}\right)\cdot C_{L}^{S}\cdot\left(\Theta_{P}^{S}-\Theta_{IP}^{EV}\right), \tag{6}$$

where Φ_{NHM}, Φ_{IP}^{S}, Φ_{EV}^{S} are the amount of heat from NHM used on heating CG from the evaporator, on cooling CG in the separation part and on the evaporation through heat transfer in a layer of liquid ammonia, W; Θ_{NHM}^{S} is the temperature of NHM at the inlet to the condensation column, °C; r^{S} is the heat of ammonia vapor formation, kJ/kg; M_{P}^{S} is the CG gas phase mass flow rate, kg/s; M_{EV}^{S}, G_{EV}^{S}, M_{L}^{S} is the mass flow rate of ammonia after evaporation in the flow of CG, obtained through the heat exchange in a layer of liquid ammonia and liquid ammonia in the flow of CG, respectively, kg/s; C_{NHM}, C_{IP}^{S}, C_{L}^{S} are the mean heat capacities of NHM, gas phase of CG and liquid ammonia from the evaporator, respectively, kJ/(kg·k).

Next, by the defined magnitude of temperature Θ_{P}^{S}, we calculated the values of heat fluxes along the inter-pipe Φ_{IP}^{CC} and pipe Φ_{P}^{CC} space by the equations that are rather well verified in practice:

$$\Phi_{IP}^{CC}=M_{IP}^{G}\cdot C_{IP}^{CG}\cdot\left(\Theta_{IP}^{CC}-\Theta_{IP}^{EV}\right)+M_{C}\cdot r_{IP}+ +\left(M_{L}^{CC}-0{,}5\cdot M_{C}^{CC}\right)\cdot C_{L}^{CG}\cdot\left(\Theta_{IP}^{CC}-\Theta_{IP}^{EV}\right); \tag{7}$$

$$\Phi_{P}^{CC}=M_{P}^{G}\cdot C_{P}^{CG}\cdot\left(\Theta_{P}^{CC}-\Theta_{P}^{S}\right)+M_{L}^{S}\cdot\left(i_{V}^{CC}-i_{L}^{S}\right); \tag{8}$$

$$M_{C}^{CC}=0{,}771\cdot V_{IP}^{CG}\cdot\left(a_{NH_3}^{IN}-a_{NH_3}^{OUT}\right); \tag{9}$$

$$a_{NH_3}^{IN}=\frac{P_{NH_3}^{IN}}{P_{CG}}; \tag{10}$$

$$a_{NH_3}^{OUT}=\frac{P_{NH_3}^{OUT}}{P_{CG}}, \tag{11}$$

where M_{C}^{CC}, M_{L}^{CC}, M_{L}^{S}, are the mass flow rate of condensated ammonia, liquid ammonia at the inlet to the inter-pipe and pipe space in a heat exchanger, kg/s; M_{P}^{G}, M_{IP}^{G}, are the CG gas phase mass flow rate in the pipe and inter-pipe space, kg/s; C_{L}^{CG}, C_{IP}^{CG}, C_{P}^{CG}, are the mean heat capacities of liquid ammonia in the inter-pipe space, gas phases of CG in the inter-pipe and pipe space of a heat exchanger, respectively, kJ/(kg·K); r_{IP} is the heat of condensation of ammonia, kJ/kg; i_{L}^{S}, i_{V}^{CC} is the enthalpy of liquid and vapor of ammonia at the inlet and outlet to the pipe space of heat exchanger, kJ/kg; $P_{NH_3}^{IN}$, $P_{NH_3}^{OUT}$ is the partial pressure of vapor of ammonia in CG at the inlet and outlet to the inter-pipe space, respectively, MPa.

5. Results of examining the indicators of effectiveness of the processes of heat exchange

By the results of experimental research, we compiled a data set on 120 regimes in the operation of a condensation column, some of which are presented in Table 1.

Table 2 give the results of calculation of the real coefficient of heat transfer K_E by the above algorithm; the numbers of modes correspond to those in Table 1.

In order to solve the problem on the identification of processes of heat exchange, we calculated the heat transfer coefficients from the pipe α_P, inter-pipe α_{IP} space. These calculations are carried out by the commonly known Krausold equations. We also computed heat transfer factor K_P. The equations accepted for design take the following form:

$$\alpha_{P}=A\cdot W_{P}^{0,8}\cdot d_{IN}^{-0,2}; \tag{12}$$

$$\alpha_{IP}=1{,}3\cdot A\cdot\varepsilon_{\varphi}\cdot W_{IP}^{0,56}\cdot d_{EX}^{-0,44}; \tag{13}$$

$$A=16{,}28\cdot\frac{\lambda_{CG}}{\mu_{CG}^{0,8}}\cdot\left(\frac{Pr}{0{,}73}\right)^{0,4}; \tag{14}$$

$$W_{P}=\frac{M_{P}^{CG}}{S_{P}}; \tag{15}$$

$$W_{IP}=\frac{M_{IP}^{CG}}{S_{IP}}; \tag{16}$$

$$S_{P}=0{,}785\cdot d_{IN}^{2}\cdot n; \tag{17}$$

$$S_{IP}=\pi\cdot D_{MD}\cdot H\cdot\left[1-\left(d_{EX}/t\right)\right]; \tag{18}$$

$$K_{P}=\frac{1}{\left(1/\alpha_{IP}+R_{T}^{P}+1/\alpha_{P}\right)}, \tag{19}$$

where W_P, W_{IP} are the weight speed of CG in the pipe and inter-pipe space per unit of surface, kg/m²·s; d_{IN}, d_{OUT} are the inner and outer diameter of heat exchanging pipes, m; ε_{ϕ}=0,67 is the correction factor for the angle of attack; λ_{CG} is the thermal conductivity of CG, W/m·K; μ_{CG} is the dynamic viscosity of CG, Pa·s; Pr is the Prandtl criterion; M_{P}^{CG} is are the CG mass flow rate in the pipe and inter-pipe space, kg/s; n=7808 pcs. is the number of heat exchanging pipes; H=0.34 m is the height of partition; D_{MD}=1.87 m is the average diameter of the inter-pipe space; t=0.02 m is the step of a bundle of pipes; $R_{T}^{P}=0{,}000354$ m²·K/W is the total thermal resistance of contamination and the walls of pipes accepted in the design.

In order to assess inconsistency between the estimated and actual indicators of efficiency of the heat exchange process, we determined total thermal resistance by formula:

$$R_{T}^{E}=\frac{1}{K_{E}}-\left(\frac{1}{\alpha_{P}}+\frac{1}{\alpha_{IP}}\right). \tag{20}$$

Results of calculations by equations (12)–(20) are compiled in Table 3; the numbers of modes correspond to those in Table 1.

Table 1

Experimental data on the operation modes of a condensation column

Parameter title			Number of modes						
			1	2	3	4	5	6	7
Circulation gas at the inlet to the column	Consumption V_{IP}^{CG} 10^3, nm^3/h		639.23	621.25	627.08	625.53	621.59	613.08	613.08
	Pressure P_{CG}, MPa		24.0	22.4	23.0	23.0	22.8	22.2	22.2
	Temperature Θ_{IP}^{CC} °C		37	42	35	40	39	34	40
	Concentration, % by volume	hydrogen $a_{H_2}^{IP}$	55.7	55.9	55.7	56.5	54.4	55.2	55.6
		methane $a_{CH_4}^{IP}$	8.4	8.0	8.0	8.2	8.2	8.2	7.8
		nitrogen $a_{N_2}^{IP}$	18.9	19.6	20.0	19.0	19.5	18.7	18.9
		argon a_{Ar}^{IP}	6.9	6.8	7.0	7.9	7.6	7.6	7.3
		ammonia $a_{NH_3}^{IP}$	10.1	8.6	9.3	8.4	10.3	10.3	10.4
Nitrogen-hydrogen mixture at the inlet to the separator	Consumption $V_{NHM} \cdot 10^3$, nm^3/h		174	175	175	176	174	173	169
	Pressure P_{NHM} , MPa		24.2	23.2	24.0	23.5	23.2	23.2	23.2
	Temperature Θ_{NHM}^{S}, °C		35	36	35	41	43	36	40
Temperature of CG at the inlet of the evaporator, Θ_{IP}^{EV}, °C			16	19	13	17	16	12	17
Temperature of CG at the outlet of the evaporator, Θ_{P}^{EV}, °C			–0.5	–2	–2	–5	–6	–7	–4
Circulation gas at the outlet of the column	Consumption $V_{P}^{CG} \cdot 10^3$, nm^3/h		774.2	771.35	770.72	776.14	758.08	747.58	745.12
	Temperature Θ_{P}^{CC}, °C		25	24	22	24	24	21	24
	Concentration, % by volume	hydrogen $a_{H_2}^{P}$	62.2	62.3	61.7	62.4	62.3	61.9	62.8
		methane $a_{CH_4}^{P}$	7.2	6.8	6.9	6.9	7.1	7.0	6.5
		nitrogen $a_{N_2}^{P}$	21.3	21.3	21.9	20.6	20.5	21.2	20.8
		argon a_{Ar}^{P}	6.0	5.9	6.0	6.6	6.6	6.6	6.3
		ammonia $a_{NH_3}^{P}$	3.3	3.7	3.5	3.5	3.5	3.3	3.6

Note: composition of NHM (% by volume) – $a_{H_2}^{NHM}=76,3$; $a_{CH_4}^{NHM}=0,4$; $a_{N_2}^{NHM}=23,2$; $a_{Ar}^{NHM}=0,1$

Table 2

Results of calculation of the coefficient of heat transfer based on experimental data

Parameter title			Number of modes						
			1	2	3	4	5	6	7
Heat fluxes, MW	Separation part	Φ_{NHM}^{S}	2.128	2.302	2.233	2.733	2.861	2.512	2.535
		Φ_{IP}^{S}	0.523	0.488	0.523	1.000	1.139	0.895	0.814
		Φ_{EV}^{S}	1.605	1.826	1.721	1.732	1.721	1.616	1.709
	Heat exchanger	Φ_{P}^{CC}	9.687	10.595	9.687	10.932	10.978	10.187	10.443
		Φ_{IP}^{CC}	9.874	11.129	10.048	10.804	10.850	9.955	10.932
The temperature of CG at the outlet of the separation part, Θ_{P}^{S}, °C			0.89	–0.68	–0.61	–2.23	–2.84	–4.5	–1.75
Discrepancy in heat balance, %			1.9	4.9	3.8	1.7	1.1	2.2	4.7
Heat transfer coefficient K_E, $W/m^2 \cdot K$			340.3	271.9	349.9	292.1	304.2	322.3	290.1
Condensated ammonia consumption M_C, t/h			14.44	18.11	14.6	16.96	16.6	14.43	17.22

Table 3

Efficiency indicators of the heat exchange process based on the calculated equations

Parameter title	Number of modes						
	1	2	3	4	5	6	7
Coefficient of heat exchange α_P, W/m²·K	1790.12	1780.65	1801.8	1797.95	1764.68	1761.93	1727.63
Coefficient of heat exchange α_{IP}, W/m²·K	1454.49	1418.31	1447.51	1443.06	1460.13	1431.65	1423.98
Coefficient of heat transfer K_P, W/m²·K	645	636.91	645.39	644.32	642.89	636.91	635.52
Total thermal resistance R_T^E, m²·K/W	0.00169	0.00241	0.00161	0.00217	0.00203	0.00183	0.00216

Tables 2, 3 show that the coefficient of heat transfer under real conditions K_E is almost two times less than the coefficient of K_P, calculated by the equations accepted in design. Such a mismatch, according to existing theoretical provisions [12], may be predetermined by the existence of additional condensation thermal coefficient of resistance.

6. Discussion of results of examining the effectiveness of processes of heat exchange in a technological complex of secondary condensation

The received calculated indicators given in Tables 2, 3 demonstrate that there is a non-accidental dependence between general coefficient of thermal resistance R_T^E and the consumption of condensated ammonia M_c. According to the results of processing these indicators by the least square method, we received the following equation:

$$R_T^E = 2{,}102 \cdot 10^{-5} \cdot M_C^2 - \\ -0{,}0004674 \cdot M_C + 0{,}0040679. \qquad (21)$$

Error of calculation by equation (21) does not exceed 6 %.

By using the package of programs Statistica, we derived an equation to calculate the concentrations of ammonia in CG at the outlet of pipe space of the condensation column. Determining this equation was carried out by processing experimental data, some of which are given in Table 1.

$$a_{NH_3}^{P} = -7{,}78 + 0{,}02441 \cdot V_{NHM} + 0{,}01176 \cdot V_{IP}^{CG} + \\ +0{,}0327 \cdot \left(\Theta_P^{EV} + 273\right) + 0{,}085 \cdot a_{NH_3}^{IP} - 0{,}0635 \cdot P_{CG}, \qquad (22)$$

where V is the consumption, nm³/s.

Error of calculations by equation (22) is 5 %.

Equations (21) and (22), received by the results of identification of the processes of heat exchange, together with equations (1)–(18) and (20) make up a mathematical model of the condensation column.

Table 4 gives some results of calculating the target indicators by the modes of operation of a condensation column, obtained in the process of mathematical modeling. The numbers of modes in Table 4 correspond to those in Table 1.

A comparison of the experimental data (Table 3) and those derived in the process of modeling (Table 4) indicates that the error in calculations does not exceed the accuracy of approximation by equations (21) and (22), that is 6 %. Such convergence allows us to draw a conclusion about the possibility of applying the mathematical model of condensation column for the synthesis of technological complex of secondary condensation with improved energy efficiency.

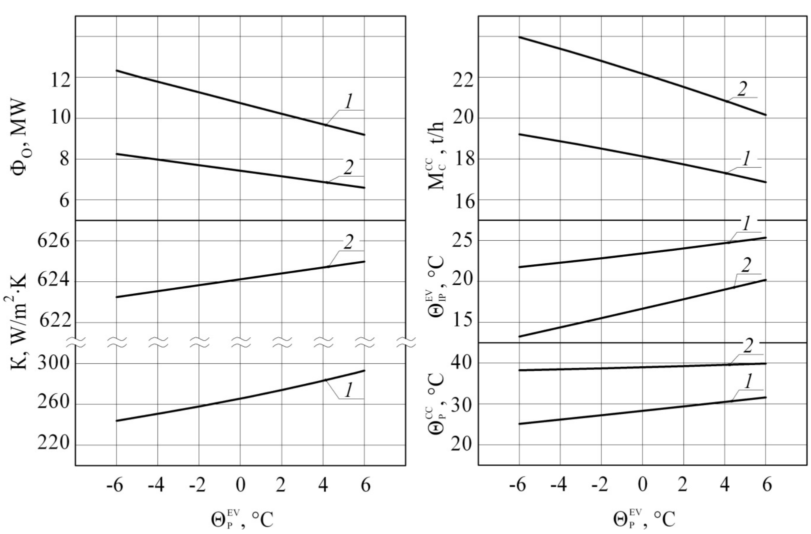
Fig. 2 shows selected results of mathematical modeling of the condensation column. They make it possible to quantitatively assess the increase in thermal load on the evaporators under real conditions of operation compared to the estimated ones during design. Dependences, shown in Fig. 2, are received at the CG maximum temperature at the inlet to condensation column of 45 °C.

According to Fig. 2, maintaining the required level temperature of 5 °C necessitates increasing the cooling capacity of refrigeration systems by 4 MW relative to 8.12 MW defined by the design. Such an increase is linked to a decrease in the real heat transfer coefficient K_E to 247.7 W/m²·K compared to the default coefficient K_P=623.4 W/m²·K, which is caused by the existence of additional condensation thermal resistance due to the condensation of ammonia M_C. Because of this, there is an increase in temperature Θ_{IP}^{EV} at the inlet of the evaporator with 13.8 °C to 22 °C and lowering the temperature Θ_P^{CC} at the outlet of pipe space of the condensation column from 38.4 °C to 25.7 °C.

Table 4

Basic target indicators of operating modes of a condensation column according to the results of mathematical modeling

Indicators of heat exchange efficiency	Number of modes						
	1	2	3	4	5	6	7
Heat flux of separation part Φ^S, MW	2.162	2.298	2.248	2.729	2.78	2.534	2.536
Heat flux of heat exchanger Φ^{CC}, MW	10.22	10.772	9.927	10.855	10.929	10.604	10.690
Heat transfer coefficient of heat exchanger K, W/m·K	332.51	275.67	339.29	289.8	296.31	324.48	289.61
Consumption rate of circulation gas at the outlet of pipe space $V_P^{CG} \cdot 10^3$, nm^3/h	777.67	770.59	771.89	775.93	758.78	749.59	745.42
Temperature of circulation gas at the outlet of pipe space Θ_P^{CC}, °C	22.8	25.5	21.5	23.7	22.9	20	24.5
Temperature of circulation gas at the inlet of pipe space Θ_P^S, °C	0.396	–0.575	–0.774	–2.194	–2.948	–4.8	–1.8
Temperature of circulation gas at the inlet of evaporator Θ_{IP}^{EV}, °C	15.17	19.8	13.3	16.9	15.8	10.4	17.6
Concentration of ammonia in the circulation gas at the outlet of pipe space, $a_{NH_3}^P$, % by volume	3.7	3.6	3.65	3.47	3.59	3.56	3.64



a *b*

Fig. 2. Dependences of certain indicators of the heat exchange efficiency in a condensation column and cooling capacity of refrigeration systems on the cooling temperature of circulation gas in evaporators based on original data of mode No. 5 in Table 1: 1 – at real coefficient K_E; 2 – at design coefficient K_P: *a* – dependence of cooling capacity Φ_O and heat transfer coefficient K on the cooling temperature of CG Θ_P^{EV}; *b* – dependence of the amount of condensated ammonia M_C^{CC}, temperature of CG at the inlet of evaporator Θ_{IP}^{EV} and temperature at the outlet of pipe space in the condensation column Θ_P^{CC} on the cooling temperature of CG in evaporators Θ_P^{EV}

Dependences, received in Figs. 3, 4, allow us to specify the conditions (shown dotted) to provide not only for the exclusion of ATC from the operating scheme of the synthesis unit, but also for the reduction in the cooling temperature of CG in evaporators by two ARU only to the required level – 5 °C. In this case, in order to reduce thermal load on the evaporators by temperature, it is necessary to install an additional heat exchanger for the deeper recuperation of cold.

An analysis of dependences shown in Fig. 3, 4 indicates that at temperature $\Theta_{IP}^{CC} = 30$ °C of CG at the inlet of condensation column, it is possible to stabilize the cooling temperature at the required level – 5 °C by two ARU only. In this case, their cooling capacity must be even lower than the existing one, that is, 6.02 MW. The fulfillment of this condition, even by installing an additional heat exchanger according to Fig. 4, is impossible. That is why it is more expedient at much smaller heat exchange surface of the additional heat exchanger F_{AD}=1150 m^2 to install a high-temperature evaporator before the condensation column. Boiling temperature of ammonia in its inter-pipe space does not exceed 24 °C and pressure of 0.9915 MPa makes it possible to reduce the temperature of direct flow of CG from 35 °C to 30 °C. Temperature regime of boiling in the evaporator can be enabled by connecting it to the cycle of steam ejector refrigeration unit (SRU), as shown in Fig. 5.

The magnitude of injection coefficient of SRU jet compressor was determined by algorithm [9], sufficiently tested in practice. As a result of applying in SRU the air cooling devices, the injection coefficient was established by the magnitude of attainable compression pressure whose value was limited by 1.6 MPa. This makes it possible to provide high temperature (40 °C) of ammonia vapor condensation after a jet compressor even in summer.

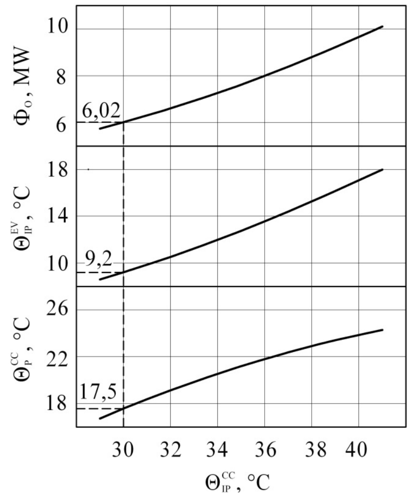


Fig. 3. Effect of temperature of the circulation gas at the inlet of a condensation column Θ_{IP}^{CC} on the distribution of temperatures at the inlet of evaporators Θ_{IP}^{EV}, at the outlet of pipe space of a condensation column Θ_{P}^{CC} and cooling capacity Φ_O of the evaporators at constant cooling temperature $\Theta_{P}^{EV} = -5$ °C

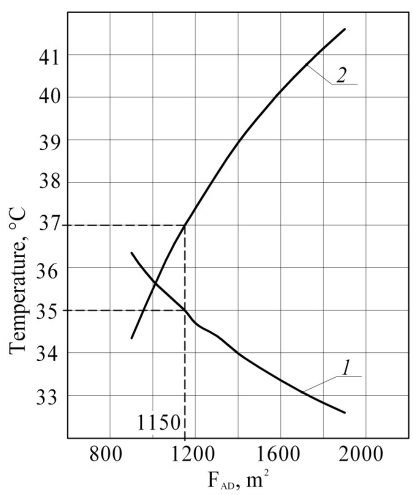


Fig. 4. Effect of the heat exchange surface of additional heat exchanger F_{AD} on the distribution of temperatures of the circulation gas at its outlets at maximum thermal load of the circulation gas by temperature at the inlet Θ_{IP}^{IN} =45 °C: 1 — temperature of the circulation gas at the outlet of inter-pipe space Θ_{IP}^{OUT}; 2 — temperature of the circulation gas at the outlet of pipe space Θ_{P}^{OUT}

According to the results of calculations, the magnitude of injection coefficient is not less than 0.4. At such coefficient, 20 t/h of working vapor from a SRU steam generator with pressure 3 MPa will suffice for the injection of 8 t/h of ammonia vapor from a high-temperature evaporator. This will provide for cooling capacity at the level of 2.48 MW. The total amount of vapor of coolant and working vapor to the air condensers will reach 28 t/h; their condensation can be enabled by three condensers with power consumption of 600 kW·h. In order to receive working vapor in the amount of 20 t/h, 515 t/h of monoethanolamine (MEA) solution of the first flow of separation of MEA purification through the steam generator will suffice, that is 20·974,4/3,78·(85–75)= =515 t/h. Numerical values in the process of this calculation are as follows: specific vaporization enthalpy of ammonia at temperature 65 °C and pressure 3 MPa=974,4 kJ/kg; specific heat capacity of the MEA solution=3.78 kJ/kg·k; input and output temperature of the MEA solution=85 and 75 °C, respectively. Such utilization of the heat of MEA solution reduces the loading on air coolers of this solution and heat emissions into the atmosphere.

The technological design presented allows is to reduce the total cooling capacity at the stage of secondary condensation from 11.16 MW to 8.5 MW. This is predetermined by the deeper recuperation and utilization of low-potential heat in the steam-ejector refrigeration unit with the level of flow temperature of the MEA solution up to 90 °C. In this case, as a result of the exclusion of ATC from the operational circuit of the unit, there will follow a decrease in energy consumption by 3.4 thousand kW·h. Despite such a reduction in the cooling capacity, the reduction and stabilization of the cooling temperature of CG is achieved in the evaporators from 0 °C to –5 °C at its maximum temperature at the inlet of a condensation column. Such maximum loading is characteristic of the work of the synthesis unit for 4 months in the spring and summer period. Reducing the temperature during this time by 5 °C will also reduce the consumption of natural gas by 190 m^3/h into additional steam boiler for receiving water vapor pressure with 10 MPa.

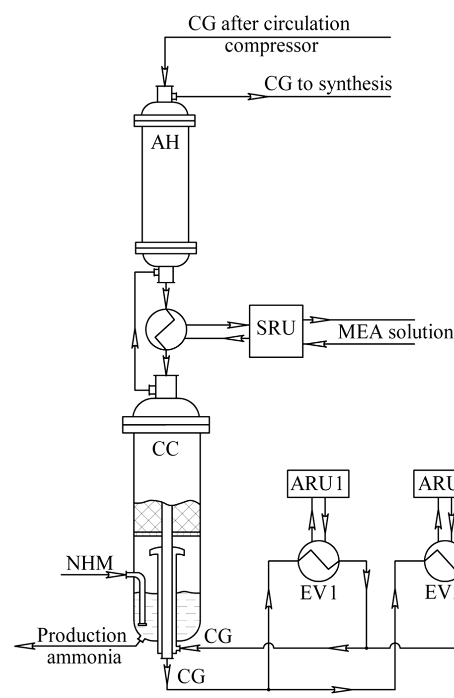


Fig. 5. Hardware-technological design of the complex of secondary condensation with improved energy efficiency: CC — condensation column; CG — circulation gas; NHM — nitrogen-hydrogen mixture; ARU — absorption-refrigeration unit; SRU — steam ejector refrigeration unit; EV — evaporator; AH — additional heat exchanger

The advantage of the proposed technology is providing the CG cooling temperature mode only by employing the heat-using refrigeration systems that utilizing heat of both low and super-low potential. Creating such a technological complex provides for the possibility to replace powerful turbo-compressor refrigeration systems with a steam drive that are difficult to operate. In this case, the saved discharged water vapor with pressure 4 MPa can be used to generate electricity. The following studies will be conducted in the direction of further improvement of the condensation processes in order to reduce the load on the compressors of CG compression by lowering the temperature.

7. Conclusions

1. The research conducted identified indicators of effectiveness of the processes of heat exchange in a condensation column, such as heat fluxes, temperature and concentration in its separation and heat-exchanging parts. We established a pattern in the influence of condensation thermal resistance on the coefficient of heat transfer, which is characterized by the inversely proportional dependence.

2. We identified the processes of heat exchange in a condensation column, the results of which defined the equation for calculating the coefficients of heat exchange, heat transfer and the concentration of ammonia in the circulation gas at the outlet of a separation part. It is shown that such equations, in contrast to those well-known, consider condensation thermal resistance in the process of heat exchange and external material-thermal load on the condensation column.

3. We synthesized the hardware-technological design of the complex of secondary condensation by using heat-consuming refrigeration units only, whose operation is enabled by the utilization of low and super-low potential of heat from the material flows in the synthesis unit.

4. We identified for the proposed technology efficiency indicators for the reduction in total cooling capacity, specific energy consumption by electricity and natural gas, which are 2.66 MW, 60 kW·h/t NH_3 and 1.2 m^3/t NH_3, respectively.

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