НАУЧНИ ТРУДОВЕ НА
УНИВЕРСИТЕТ ПО ХРАНИТЕЛНИ
ТЕХНОЛОГИИ - ПЛОВДИВ
2016 г.
ТОМ 63, КНИЖКА 1

SCIENTIFIC WORKS OF
UNIVERSITY OF FOOD
TECHNOLOGIES
2016
VOLUME 63 ISSUE 1

Study of Thermodynamic Parameters of the Mechanical Heat Pump System

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Running title: **Process of Water Vapor Compression – Thermodynamic Parameter Values**

Abstract

Object of the present study is an experimental determination of the values of thermodynamic parameters of mechanical heat pump system: isentropic efficiency of mechanical compressor, polytropic index in the process of compression and heat capacity of water vapor. A classic experiment by two significant factors with three levels of variation of temperature of secondary vapor and five levels of variation of compression ratio in order to obtain adequate regression equation is conducted. It was found experimentally that in heat pump system mechanical compressor has a isentropic efficiency ξ in the range of 0,143 to 0,288. Experimentally determined values of polytropic index in the process of water vapor compression is n = - 0,325 ± 0,006 and heat capacity of water vapor in the process at polytropic compression is c_n = 1,817 kJ / kgK ± 0,0005 kJ / kgK.

It was found experimentally that due to low dry matter content in wastewater thermodynamic parameters of the pilot heat pump system coincide with thermodynamic parameters of drinking water. For low energy consumption of the heat pump system it needs to operate at high temperatures of secondary vapor and low values of compression ratio in the mechanical compressor. Due to the low values of isentropic efficiency coefficient of used mechanical compressor value of polythropic index in the process of water vapor compression is negative.

Practical applications

The determination of thermodynamic parameters used to evaluate operating efficiency of mechanical pump system and to determine the effect of impurities in the pumped liquid water on the system.

Key words: efficiency of mechanical compressor, mechanical vapor recompression heat pump

Introduction

The object of study in the presented paper is heat pump system for treatment of waste water by the method of vapor recompression. When defining goal functions of this study are taken into account important to practice energy performance determining the effectiveness and practical application of the proposed system. The following thermodynamic parameters of working fluid of heat pump system are studied: isentropic efficiency coefficient of used mechanical compressor ξ, polytropic index in the process of water vapor compression n and heat capacity of water vapor in the process at polytropic compression c_n.

The main goal in conducting this study is to get information for research object by creating a mathematical relation (regression equation) of target function by the significant factors (dependent variables). Proper definition of the factors and the objective function is essential for the correct description for research object. They can be determined by preliminary experiments, expert evaluation or known theoretical dependencies.

Materials and methodology

For the purpose of the study is to use heat pump system for treatment of waste water by method of vapor recompression. Schematic diagram of the heat pump system is shown in Figure 1. Process diagram of the heat pump system is shown on Figure 2. The waste water in the initial state 0 at atmospheric pressure p_0 and the set initial temperature is throttled at a throttling valve TV1 to the set pressure of evaporation p_{ev} to state 1. Then throttling isobaric the water is heated to boiling condition of the regenerative heat exchanger RHE to state 2. In this state the water is fed into the heat exchanger HE where isobaric is heated without boiling to state 4. Due to the pressure maintained by the circulator CP this state is not boiling liquid. The water is throttled at a throttling valve TV2 to state of boiling liquid 5 with pressure of evaporation p_{ev}, which resulted one part from the waste water is self evaporated to secondary vapor in state 6 in separator S. As a result of physicochemical depression it has a lower temperature than the boiling point of the state 2. The resulting state secondary vapor 6 is compressed to a pressure of condensation p_{cond} of mechanical compressor MC by isentropic process ideally (state 7*) and no isentropic process real case to state 7. Hot vapor received in state 7 are fed into the heating section of the heat exchanger HE where isobaric condensed in two stages - from overheated to dry saturated vapor (state 7a) and dry saturated vapor to boiling liquid – state 8. Thus, the resulting condensate (purified water) is cooled to state 9 as isobaric given regenerative heat exchanger RHE inlet water. Cooled condensate is compressed in pump P to atmospheric pressure p_o in isentropic ideal process to state 10 * and no isentropic real process to state 10. The collected at the bottom of the separator S concentrated waste water in a fluidized state 2 is mixed with incoming non-concentrated wastewater in the same condition and is pumped by circulation pump CP to the set pressure in regenerative heat exchanger RHE through ideal isentropic process to state 3* and a real no isentropic process to state 3. In the created laboratory stand evaporate 5-16 kg/h of water from the incoming wastewater. Evaporation in the separator is carried out under vacuum in 15.5-25.0 kPa absolute pressure of the secondary vapor and condensation compressed vapor in a heat exchanger HE at 20.0-43.0 kPa absolute pressure. This pressure differential provides between 5 and 13 K temperature difference between boiling temperatures of secondary vapor and condensing vapor compression (in the absence of temperature depression). The pressures used in the absence of temperature depressions correspond to the saturation temperature at evaporation for 55 - 67°C and 60-78 °C with condensation. (Valchev&Nenov, 2014).

Used in the laboratory pilot heat pump system compressor to compress the water vapor is “lobe” type. In its compression is performed by volumetric principle. For this model compressor factory work description is missing. It is therefore committed capture the relation between the flow rate of the working fluid pressure created and consumption of electricity. Completion is committed to working fluid air. The inlet air temperature during the measurement is t_1 = 23,4 °C ± 0,8 K and atmospheric pressure p_1 = 99,8 ± 0,05 kPa. Measurements were made at compressor operating in a stationary temperature regime and constant temperature of machine oil in it. On created laboratory heat pump system were conducted experimental studies for treatment of various types of waste water through a continuous process. Five different types of origin wastewater are studied - from boiler blowdown, from refrigerant condenser blowdown, from CIP system of milk factory, from washing vehicles and from washing of catering equipment.

For determination of thermodynamic parameters a classic experiment by two significant factors with three levels of variation of temperature of secondary vapor and five levels of variation of compression ratio in order to obtain adequate regression equation is conducted. When the experiment is conducted factor temperature of secondary vapor is fixed and varied with five levels of the factor compression ratio. The order of experiment is random to avoid the influence of systematic and random errors resulting from the order of their execution.

Isentropic efficiency coefficient of used mechanical compressor ξ is defined as the ratio of the theoretical electrical power required |Nt| in an isentropic process and the actual one |N| in real, noisentropic process (Kimenov, 1977):

$$\xi = \frac{|N_t|}{|N|} \tag{1}$$

In this dependence theoretical electrical power is calculated by the equation of energy balance of mechanical compressor isentropic process:

$$|N_t| = \dot{m}_k .\left[h_{cv}(p_{cond}, s^{"}(p_{ev})) - h^{"}(p_{ev})\right], \tag{2}$$

where $\dot{m}_k$ - mass flow of water condensate, kg/s,

$h_{cv}(p_{cond,} s^{"}(p_{ev}))$ is the specific enthalpy of compressed vapor at compressor isentropic process, kJ/kg. Defined by established scaled h, s - diagram for vacuum vapor condensing pressure and specific entropy equal to that of dry saturated vapor at boiling pressure before compressor.

$h^{"}(p_{ev})$ - specific enthalpy of the secondary vapor before the compressor, kJ/kg.

Polytropic index in process of water vapor compression n is determined by the dependencies for polytropic process of compression of ideal gas (relation between temperature and pressure of water vapor before and after the compression process) of an expression converted to type:

$$n = \frac{1}{1 - \dfrac{\ln \sigma}{\ln \dfrac{(t_{cv} + 273{,}15)}{(t_{sv} + 273{,}15)}}} \tag{3}$$

where σ is the compression ratio of compressor,-.

t_{cv} - temperature of compressed vapor, °C.

t_{sv} - temperature of secondary vapor, °C.

Heat capacity of water vapor in the process at polytropic compression c_n is determined by dependence (Kimenov, 1977):

$$c_n = c_\vartheta . \frac{k - n}{1 - n} \tag{4}$$

where c_ϑ is heat capacity of water vapor in isohoric process.

In this temperature range c_ϑ = 1,46 kJ / kgK.

k - isentropic indicator of water vapor. In this temperature range k = 1,324.

Results

From the studies can be summarized these results:

1. It was found experimentally that in the test heat pump system mechanical compressor has a isentropic efficiency coefficient of water vapor compresson ξ in the range of 0,143 to 0,288.
2. Experimentally determined values of the polytropic index in the process of water vapor compression with a mechanical compressor n = - 0,325 ± 0,006.
3. Experimentally determined values of heat capacity of water vapor in the process at polytropic compression c_n = 1,817 kJ / kgK ± 0,0005 kJ / kg.K.

Discussion

It was found that for five types of wastewater, operating parameters such as pressure, temperature, flow rate and electrical consumption for compression statistically not different to each other or from those working with drinking water. This fact can be explained by the low concentration of solids in water probes (0.08 to 0.27%) and the proximity of their thermophysical characteristics and thermodynamic parameters to those of the drinking water.

In determining isentropic efficiency coefficient of used mechanical compressor ξ we examined the dependence of objective function (Y_1) from significant factors temperature of exhaust vapor (X_1) and the compression ratio in mechanical compressor (X_2). The results of triplicate measurements and average value are listed in Table 1.The graphical relation between the isentropic efficiency coefficient of used mechanical compressor and the temperature of exhaust vapor and the compression ratio in mechanical compressor is shown on Figure 3. Analysis of these dependencies graphic shows heterogeneity of mathematical dependencies for different parameter values temperature of exhaust vapor (about 65.7 °C it has a pronounced maximum, while the others are monotonously decreasing). (Aybar, 2002), (Ettouney, 2006). From the characteristics of the compressor shown in Table 2 is a logical existence of a maximum in the isentropic efficiency coefficient as a function of the compression ratio. In the set characteristics stand out low values of the isentropic efficiency coefficient of used mechanical compressor. The maximum value when working with atmospheric air is ξ = 0,292.

In determining polytropic index in the process of water vapor compression n experimental specified values of temperature and water vapor pressure before and after the mechanical compressor are examined. A dependence of objective function (Y2) of the factors temperature of secondary vapor (X3) and the compression ratio in the mechanical compressor (X4) is received. The results of triplicate measurements and average value are listed in Table 3. The graphical relation between the polytropic index and temperature of secondary vapor and the compression ratio in the mechanical compressor is shown in Figure 4. The data obtained show that the value of polytropic index in a study interval is constant, so it can be presented as an average value of 15 levels of variation after the relevant statistical processing. (Raichkov, 2001). The resulting negative value of the the polytropic index can be explained by the theory outlined in (Christians, 2012). According to her polytropic index n of the expressions of the first law of thermodynamics and equations of ideal gas can deduce the following relationship:

$$n = (1-k).\frac{\partial q}{|\partial l_t|} + k \qquad (5)$$

where $\frac{\partial q}{|\partial l_t|}$ is the ratio of the balance deposited or discharged specific heat and imported technical work of the working fluid in mechanical compressor. Under this dependence negative values for polytropic index are obtained large amounts of imported heat into mechanical compressor compared with worthwhile work on compression fluid (first addend as negatively exceeds the second one). So in case studies on polytropic index in the process of water vapor compression indicator isentropic water vapor k = 1,32 and imported heat mechanical compressor repeatedly exceeding net power is to receive these negative indicator of polythropic index. Physical explanation for this is the low values of isentropic efficiency coefficient of used mechanical compressor.This means that a larger part of the deposited compressor technical work is converted to heat. It is perceived by the working fluid and cause increased absolute value of the first addend according to (5). So in the end can be summed up that negative values of polythropic index in the process of water vapor compression correspond to low values of the coefficient of isentropic index of mechanical compressor /low efficiency of mechanical compressor/.

In determining heat capacity of water vapor in the process at polytropic compression c_n are used experimental specified values of temperature and water vapor pressure before and after the mechanical compressor. A dependence of objective function (Y3) of the factors temperature of secondary vapor (X5) and the compression ratio in the mechanical compressor (X6) is received. The resulting of heat capacity of water vapor in the process at polytropic compression obtained indicate that this specific heat capacity in the study interval is constant, so it can be presented as an average value of 15 levels of variation after the relevant statistical processing. The results are listed in Table 4. The resultant positive value of magnitude study shows that research model mechanical compressor with the introduction of technical work and convert part of it into heat temperature of the vapor rises. This can be confirmed by the measured values of the temperature of compressed water vapor.

Conslusions

1. It was found experimentally that due to low dry matter content in wastewater thermodynamic parameters of the pilot heat pump system coincide with thermodynamic parameters of drinking water.
2. It was found experimentally that for low energy consumption of the heat pump system it needs to operate at high temperatures of secondary vapor and low values of compression ratio in the mechanical compressor.
3. It was found experimentally that due to the low values of isentropic efficiency coefficient of used mechanical compressor value of polythropic index in the process of water vapor compression is negative.

Acknowledgments

The pilot heat pump system is built with financial help on Project 10/13 on Fund "Science" at the UFT – Plovdiv and Department of Heat Engineering at the UFT – Plovdiv.

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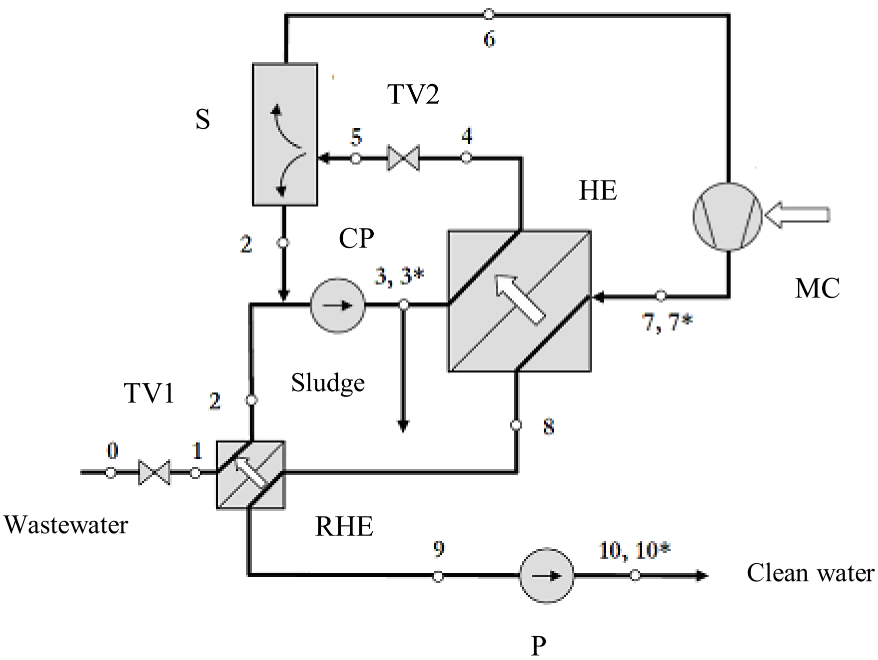

Figure 1. Schematic diagram of mechanical heat pump system

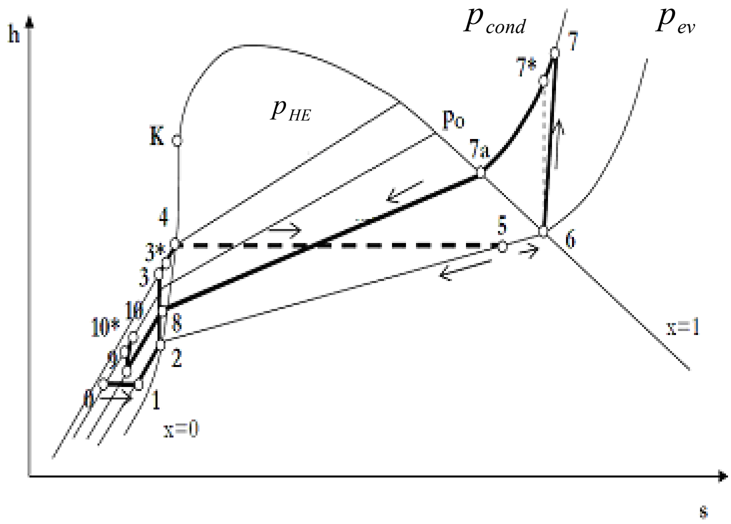

Figure 2. Process diagram of mechanical heat pump system

Table 1. Plan matrix with varying levels and values of the objective function isentropic efficiency coefficient at mechanical compressor

№	X_1	X_2	$Y_{1,1}$	$Y_{1,2}$	$Y_{1,3}$	Y_1
	^oC	-	-			
1	54,7	1,290	0,169	0,164	0,167	0,167
2	56,0	1,394	0,163	0,160	0,170	0,164
3	55,3	1,500	0,164	0,164	0,155	0,161
4	54,7	1,613	0,153	0,154	0,150	0,152
5	55,3	1,719	0,144	0,138	0,148	0,143
6	60,1	1,300	0,203	0,210	0,195	0,203
7	60,6	1,390	0,193	0,202	0,202	0,199
8	59,0	1,526	0,186	0,197	0,196	0,193
9	59,0	1,632	0,182	0,202	0,182	0,189
10	60,1	1,750	0,166	0,168	0,164	0,166
11	65,9	1,288	0,259	0,271	0,278	0,269
12	65,0	1,400	0,271	0,297	0,297	0,288
13	66,7	1,481	0,274	0,289	0,295	0,286
14	65,9	1,615	0,253	0,264	0,259	0,259
15	65,0	1,712	0,219	0,236	0,241	0,232

Table 2. Operating parameters of the mechanical compressor for fluid - air

Δp	σ	$V_1.10^3$	|N|	|N_T|	ζ
kPa	-	m^3/s	W	W	-
0,2	1,002	27,2±1,6	570±10	5	0,009
3,2	1,032	27,3±1,8	610±13	85	0,140
7,7	1,077	23,2±1,4	690±13	173	0,251
10,6	1,106	21,2±1,3	760±14	216	0,284
13,5	1,135	19,0±1,3	830±16	242	0,292
17,2	1,172	16,2±1,0	920±17	260	0,283
21,3	1,213	12,6±0,5	1030±20	247	0,240
26,3	1,264	8,0±0,3	1170±21	189	0,162

Data are presented as average value ± standard deviation

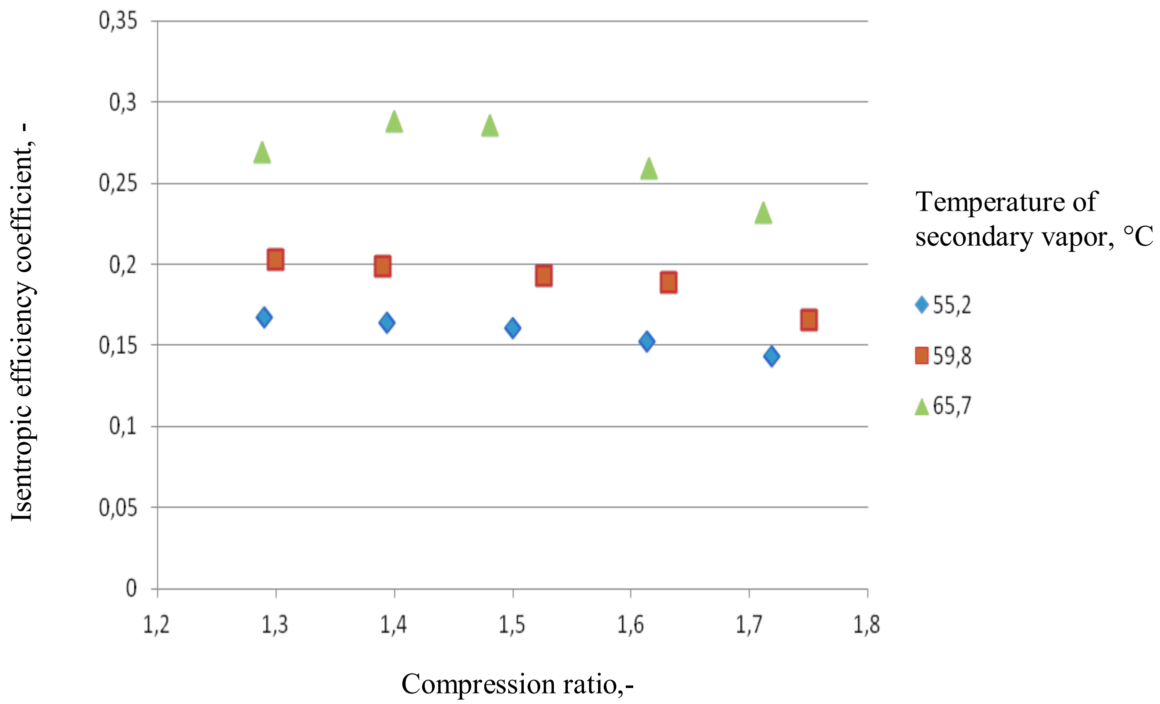

Figure 3. Depending factor isentropic efficiency coefficient at mechanical compressor of the compression ratio of water vapor and temperature of secondary vapor

Table 3. Plan matrix with varying levels and values of the objective function polythropic index in the process of water vapor compression

№	**X_3**	**X_4**	**$Y_{2,1}$**	**$Y_{2,2}$**	**$Y_{2,3}$**	**Y_2**
	°C	-	-			
1	54,7	1,290	-0,328	-0,318	-0,328	-0,325
2	56,0	1,394	-0,326	-0,319	-0,319	-0,321
3	55,3	1,500	-0,323	-0,323	-0,317	-0,321
4	54,7	1,613	-0,325	-0,325	-0,325	-0,325
5	55,3	1,719	-0,325	-0,325	-0,325	-0,325
6	60,1	1,300	-0,320	-0,330	-0,320	-0,323
7	60,6	1,390	-0,322	-0,329	-0,322	-0,324
8	59,0	1,526	-0,322	-0,322	-0,327	-0,324
9	59,0	1,632	-0,321	-0,331	-0,331	-0,328
10	60,1	1,750	-0,324	-0,328	-0,328	-0,327
11	65,9	1,288	-0,323	-0,333	-0,313	-0,323
12	65,0	1,400	-0,324	-0,331	-0,338	-0,331
13	66,7	1,481	-0,324	-0,330	-0,336	-0,330
14	65,9	1,615	-0,323	-0,323	-0,323	-0,323
15	65,0	1,712	-0,324	-0,320	-0,316	-0,320

$$n = -0{,}325 \pm 0{,}006$$

Data are presented as average value ± standard deviation

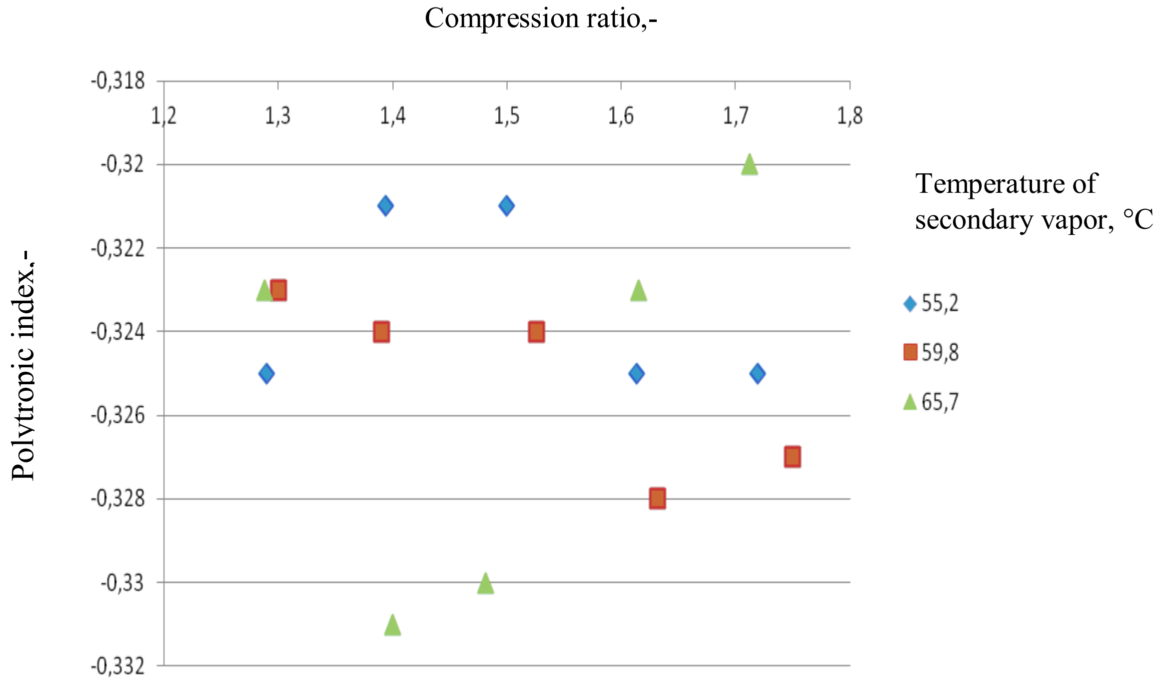

Figure 4. Depending factor polythropic index in the process of water vapor compression of the compression ratio of water vapor and temperature of secondary vapor

Table 4. Values of heat capacity at the process of polythropic compression of water vapor

№	X_5	X_6	Y_3
	^{o} C	-	kJ/kgK
1	54,7	1,290	1,817
2	56,0	1,394	1,818
3	55,3	1,500	1,818
4	54,7	1,613	1,817
5	55,3	1,719	1,817
6	60,1	1,300	1,818
7	60,6	1,390	1,817
8	59,0	1,526	1,817
9	59,0	1,632	1,816
10	60,1	1,750	1,816
11	65,9	1,288	1,818
12	65,0	1,400	1,815
13	66,7	1,481	1,816
14	65,9	1,615	1,818
15	65,0	1,712	1,818

$$c_n = 1{,}817 \text{ kJ / kgK} \pm 0{,}0005 \text{ kJ / kg.K}$$

Data are presented as average value ± standard deviation