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A STUDY OF NEW LOCAL HEATING AND AIR CONDITIONING SCHEMES BASED ON THE MAISOTSENKO CYCLE

O. Stupak
Postgraduate Student, Researcher*
E-mail: stupakalewka@gmail.com
A. Khalatov
Doctor of Technical Sciences, Professor,
Academician of NAS of Ukraine, Head of Department*
Head of Department**
E-mail: Artem.Khalatov1942@gmail.com
T. Donyk
PhD, Senior Researcher*
Senior Engineer**
E-mail: doniktv@ukr.net
O. Shikhabutinova
PhD, Senior Researcher*
Senior Engineer**
E-mail: oshikhabutinova@gmail.com
*Department of High-Temperature Thermogasdynamics
Institute of Engineering Thermophysics
of National academy of sciences of Ukraine
Mariyi Kapnist str., 2a, Kyiv, Ukraine, 03057
**Department of Power Systems Physics IPT
National Technical University of Ukraine
«Igor Sikorsky Kyiv Polytechnic Institute»
Peremohy ave., 37, Kyiv, Ukraine, 03056

Головною проблемою комунальної теплоенергетики є значні витрати енергоресурсів при виробництві теплової енергії в зимовий період та кондиціювання – в літній. Тому велике значення набувають енергоефективні системи локального теплопостачання та кліматичні системи тепло- та холодопостачання на основі поновлювальних джерел енергії. В останні роки широке розповсюдження отримали теплові насосі за циклом Ренкіна, які використовують енергію атмосферного повітря, ґрунту, стічних вод, а також системи кондиціювання за циклом Майсоценка з використанням психрометричної енергії оточуючого середовища. Теоретичний аналіз показує, що комбінація цих циклів дозволяє досягнути високої енергоефективності та створити принципово нові системи тепло- та холодопостачання приміщень. В даній роботі представлені результати порівняльного експериментального дослідження двох схем тепло- та холодопостачання, на основі комбінації циклів Майсоценка та Ренкіна. Для експериментального дослідження був розроблений експериментальний стенд комбінованого циклу теплової потужності 28 кВт з потужністю теплового насосу 3 кВт. конструкції стенда використаний серійний тепломасообмінний апарат за М-циклом, виготовлений компанією «Coolerado Corporation», США. Дослідження показали високу енергетичну ефективність обох схем теплопостачання, яка визначалася коефіцієнтом перетворення енергії COP 6,3–7,21 для першої схеми та 7,44–9,73 – для другої. При кондиціюванні приміщення тепловий насос Ренкіна не використовується, тому затрати енергії йдуть тільки на роботу вентилятора для прокачування повітря через тепломасообмінний апарат М-циклу та систему кондиціювання. Коефіцієнт перетворення енергії в цьому випадку склав величину 10,49–16,32

Ключові слова: теплопостачання, повітряне опалення, холодопостачання, тепловий насос, цикл Ренкіна, цикл Майсоценка

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1. Introduction

The problem of energy-efficient heating of living space in winter and conditioning in summer is one of the most difficult issues in the world as it is the most energy-intensive industry consuming about 55 % of primary energy resources. In Ukraine, the economic problems related to the unsatisfactory technical and physical condition of central heating are added. Therefore, the search for more efficient, environmentally friendly and cheap installations for local heat supply and combined climatic systems for heating and cooling is one of the primary problems in heat-and-power engineering.

Current conventional schemes of local air heat supply include various types of gas, induction, gas-generating, electrode, condensing, etc. boilers. Almost all of them use fossil fuels and thus are characterized by high costs.

The main advantage of modern boilers is high power at a minimum size [1]. However, it should be borne in mind that both natural and liquefied gas is fire- and explosion-hazardous. In addition, when air exchange is disturbed (due to the insufficient inflow or poor draft in the chimney), some of the combustion products may remain in the room. This is dangerous by carbon monoxide (CO) poisoning. Their disadvantages also include the need for constant professional preventive maintenance. Direct dependence of modern heating equipment on the availability of electricity is also added for the residents of remote areas.

Infrared heating has become more widespread recently. Such alternative heating sources are relatively inexpensive and very easy to install. The surface of the floor, furniture, and walls heated by infrared radiation warms the room air. Temperature distribution in the room is similar to that provided by underfloor heating systems: useless heating of air under the ceiling is minimized. As a result, to maintain subjective comfort, the actual temperature in the room can be reduced to 16–18 °C. As a result, lower indoor air temperature reduces heat loss by the building as a whole [2]. This type of heating has several significant disadvantages such as rapid temperature fall when the heater is turned off, uneven space heating, negative impact on the person at a long intensive influence, it is dangerous for children and pets, and the risk of fire.

Increasing competition is created by new energy-efficient local heat supply systems as well as climatic heating and cooling systems that use renewable energy sources. This trend is observed because of the ever-increasing cost of primary energy resources and the heat suppliedч by central heating systems as well as the obsolescence and energy consumption of existing heating systems. However, the cost of unconventional energy sources is still quite high and does not solve the problem of reducing the heat supply cost in a radical way. Therefore, improvement of existing and development of new energy-saving technologies remain to be important in the field of heating and cooling the living spaces.

2. Literature review and problem statement

Heat pumps (HP), which use the energy of atmospheric air, soil, wastewater, and other renewable energy sources can be referred to though not new but promising means of heat supply. HP can solve current energy, environmental and socio-economic problems in many countries. This is an environmentally friendly type of heating, so it is not surprising to see a constantly growing trend of modernization of existing heating systems or the creation of new ones based on the heat pump equipment.

It is known that the efficiency of the heat pump installations in heating, ventilation, and air conditioning systems depends on the design temperature of the coolant in a particular heat supply system and temperature of a low-potential heat source. The HP efficiency is characterized by the coefficient of performance (COP). It represents the ratio of the heat obtained for heating or the heat Q removed in conditioning to the energy N spent on the operation of a compressor and other components of the heat pump [3–5]:

$$\mathrm{COP} = \frac{\text{benefit}}{\text{consumption}} = \frac{|Q|}{|N|}.$$

The heat pumps operating according to the Rankine cycle provide a high coefficient of performance in a nominal mode at a level of 3 to 5 per year. However, the loss of power and efficiency when temperature decreases is a significant disadvantage of these systems. It should also be noted that they have a high unit cost (over 1000 euros per 1 kW of installed thermal capacity) and a significant payback period of 8...10 years. Many studies [3–8] address the elimination of shortcomings of the use of heat pumps.

For example, [6] proposes to increase the efficiency of air heating systems by optimizing their automation system, provision of optimal operation of the heat pump heating system, and the engineering system. This optimization can reduce power consumption by more than 16 %. We can say that in terms of thermodynamics, the heat pump has reached its technological design limit and new promising ways should be found to improve the efficiency of the air heating systems based on it.

A heating tower heat pump (HTHP) was proposed in [7] as an alternative to the conventional airstream heat pump (ASHP) for heating and cooling. The HTHP features water cooling in summer and a special system to prevent icing in winter. An experimental study and modeling of conditions in buildings of Nanking, China, have shown that though the cost of the proposed scheme is higher, the average energy efficiency of this alternative system in summer is 23.1 % higher than that of a heat pump due to different approaches to water cooling. However, an increase in energy efficiency measures only 7.4 % in winter.

The studies conducted in [8] have shown that deep recuperation of exhaust heat can be achieved due to a proposed combined cooling and heating system: a dynamic exhaust low-temperature recuperative heating system. This solves the problem of icing of the heat pump evaporator and its low efficiency. COP has increased by 39.5 % and was at the level of 4.66. The efficiency of primary energy use was increased by 3.9 %. The authors have modeled the system and analyzed the impact of waste exhaust air to be recuperated, COP of the heat pump, and the impact of the degree of primary energy use. This approach used in the study makes it possible to conclude that the use of combined schemes and efficient heat recuperation systems and the heat pump systems can result in a significant COP increase.

The application of heat recuperation was studied in [9]. A heating system with a recuperation of fresh air with hot exhaust air was considered. A prototype was built and a parametric analysis was performed to verify the effect of temperatures and mass flows on the performance of the heat recuperation system. The paper presents the results of experimental studies in typical conditions that simulate the operation of heating, ventilation, and air conditioning systems in a cold climate. It has been shown that power up to 110 W enables savings in a cold climate and at low airflow rates of about 0.1 kg/s and 0.005 kg/s of hot exhaust and fresh cold air, respectively. Extrapolation calculations for higher airflow rates show that the economical power can exceed 1 kW.

Basically, the increase in efficiency of heat pumps is ensured by the modernization of their elements and structures, optimization of heat supply systems and mode parameters of these systems. As shown by the previous work of the authors [10], the Maisotsenko cycle (M-cycle in what follows) is the further effective development of the heat pump.

One of the important properties of the M-cycle is its high thermodynamic efficiency in the region of high ambient temperatures, that is, where the Rankine cycle has low efficiency [11–13]. Environmental safety, high efficiency, low unit cost, the design that does not contain complex components, low operating costs are the main advantages of the M-cycle based installations. Since all processes take place under atmospheric conditions, there is no problem of sealing the installation.

The M-cycle was implemented in a heat-and-mass exchanger of special design [14]. Heat-and-mass exchange processes in it are close to thermodynamically reversible processes. This makes it possible to obtain the maximum effect of air cooling at a minimum power consumption [11–13]. In all practical cases, atmospheric air or used gas moves through dry and wet working channels. Atmospheric air passes through the dry cooling channel in the air conditioning systems. The exhaust gas of power installations or a cooled working fluid (for example, in a condenser) move in power engineering and heat-and-mass exchange technologies. The National Renewable Energy Laboratory (NREL), USA, has confirmed that M-cycle based air conditioners consume almost 10 times less electricity than conventional air conditioners with a compression cycle.

The absence of a high-cost compressor and refrigerant [13] is another important factor in the M-cycle installations. Such installations use 100 % clean ambient air. Due to the fact that the apparatus based on the M-cycle use the potential energy of the environment (humid atmosphere air), the cost of the energy produced is significantly lower compared to other renewable energy technologies [11–13]. The efficiency of the heat-and-mass exchanger of indirect evaporative cooling (M-cycle) can be assessed by temperatures to which air is cooled, that is, the efficiency of cooling to the wet thermometer temperature and to the dew point temperature.

When considering the present-day air conditioning systems, it can be concluded that almost all of them are based on the use of the Rankine cycle. The presence of a compressor and freon is the main cause of high power consumption by the equipment. Therefore, all conventional air conditioners work 85 % on already used air recycled in the room. This is the major disadvantage of heat pumps of the Rankine cycle.

Thermodynamic analysis has led the authors to an innovative idea of in-series application of Maisotsenko and Rankine cycles for more efficient use of the best qualities of these cycles and the creation of fundamentally new efficient systems for local heating and cooling of living spaces.

The M-cycle was implemented in a number of heat-and-mass exchangers of indirect evaporative cooling with a system of dry and wet channels of a small equivalent diameter [14–18]. In practice, the M-cycle was first used for air conditioning purposes (Fig. 1, *a*). Structurally, the heat-and-mass exchanger consists of a set of propylene plates which form a system of dry and wet channels (Fig. 1, *b*). The cooling effect is achieved due to the heat-and-mass exchange of air flows at a psychrometric temperature difference with the heating and humidifying one (working) flow and cooling another (auxiliary) flow [14].

To realize the processes of heat and moisture transfer with an indirect evaporative cooling of air, it is necessary to have two airflows:

– working airflow from which heat is removed by convection through the heat exchanger wall to the water film evaporated in the wet channel. This air is cooled with a decrease in its enthalpy, theoretically to the dew point temperature, and then delivered to the room;

– an auxiliary flow in which atmospheric air is pre-cooled in the dry channel and then heated and moisture saturated to 100 % in the channel with water evaporation. The process in the auxiliary flow proceeds with increasing air enthalpy but with maintaining its temperature. Therefore, as was defined in [19], this air is unsuitable for indoor air cooling and is discharged into the environment.

Fig. 1. Products of Coolerado Corporation [14]: *a* – air conditioner; *b* – Maisotsenko heat-and-mass exchange apparatus

The authors of [14–18] have conducted studies on the efficiency of the M-cycle in different heating systems [17], that is, ventilation and air conditioning systems [14, 15, 18].

The paper [11] presents in detail an overview of conditions and characteristics of indirect evaporative cooling in the M-cycle and predicts a significant growth of its application in the near future due to very low energy consumption and high efficiency.

Further studies of energy and exergy analysis and assessment of the stability of the new evaporative air cooling system based on the Maisotsenko cycle are presented in [15]. As a result of the conducted studies, optimum working temperatures of the M-cycle for cooling were obtained.

The study [16] is devoted to determining the possibility of using the M-cycle in various heating, ventilation, and air conditioning systems, cooling systems, and gas turbine cycles but unfortunately without concrete practical implementations.

The Maisotsenko cycle has a high potential for its implementation in a large number of technological processes: refrigeration and energy production, water regeneration, heat utilization, solar and wind power installations, thermochemical recuperation, and others. Experimental studies on the developed evaporative cooling systems according to the Maisotsenko cycle are given in [17] and in [18] for climatic conditions of the Multan region (Pakistan) and in a hot and arid Mediterranean climate, respectively. These studies provide examples of practical use of the M-cycle and show the variety of its applications. This allows us to assert the feasibility of further studies in the areas of the M-cycle implementation, especially taking into account the main advantages of the M-cycle installations.

Industrial cooling towers, air humidifiers, installations for obtaining fresh water from industrial liquids and seawater, air coolers at the inlet to the gas turbine, heat pumps [10] are at the stage of study and pilot design. They are just in the development phase today.

Analysis of published data and consideration of the M-cycle features allow us to conclude that it is possible to use this cycle in air recuperation based on an air heating system. Thermodynamic analysis has led the authors to an innovative idea of in-series application of Maisotsenko and Rankine cycles for more efficient use of best qualities of these cycles and the creation of fundamentally new efficient systems of local heat supply and cooling living spaces.

Based on this, a new scheme of air heat supply of living spaces based on a combined cycle was proposed in [20]. Results of the conceptual study presented in [21] have allowed us to conclude that it is possible to achieve higher values of the coefficient of performance which was also supported by patents [20, 22–24]. Further studies in this direction will make it possible to create fundamentally new systems of heating and cooling living spaces.

3. The aim and objectives of the study

The study objective is to perform a comparative experimental study of two new schemes of heat supply to living spaces based on a combination of Maisotsenko and Rankine cycles and to establish a possibility of their use for cooling living spaces in the summer season.

To achieve this objective, the following tasks were set:

– to develop two heat supply schemes based on combining the Maisotsenko and Rankine cycles;

– to design an experimental bench of the heat supply system for the needs of heating living spaces based on combining the Maisotsenko and Rankine cycles;

– to conduct a comparative experimental study of the efficiency of both heat supply systems for heating living spaces;

– to conduct an experimental study of the efficiency of the system for cooling and air conditioning of living spaces.

4. The combined Maisotsenko-Rankine cycle

The authors have considered promising heating and cooling schemes based on a combination of the above cycles.

4. 1. Heat supply

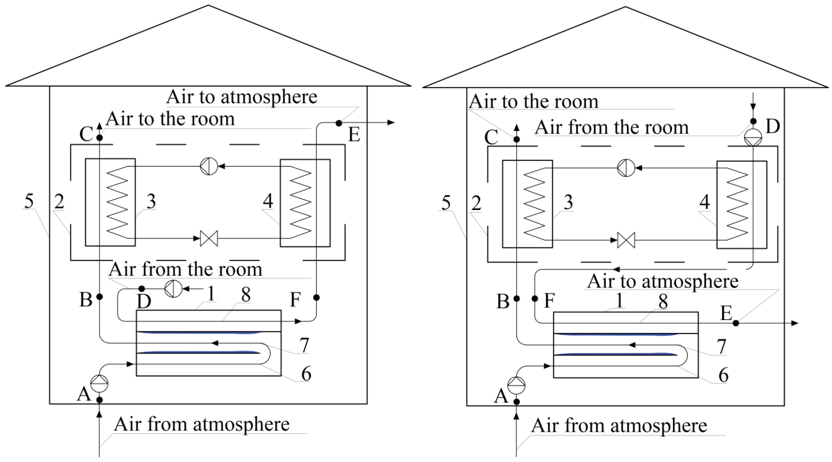
Two schemes of combined air heat supply presented in Fig. 3, *a, b* were developed and studied. They include a heat-and-mass exchanger (1) according to the Maisotsenko cycle and a standard air heat pump TNA-15 (2) according to the Rankine cycle [20, 22–24]. The ERV apparatus of Coolerado Corporation, USA, was used in the scheme as a heat-and-mass exchanger according to the Maisotsenko cycle.

Scheme 1 (Fig. 2). The installation works as follows. Air with environmental parameters (point *A*, Fig. 3) enters the M-cycle heat-and-mass exchanger (1). Air is cooled in the dry channel (6) and then enters the wet channel (7) where it is saturated with moisture (theoretically, up to 100 %) with an increase in its enthalpy due to the use of psychrometric energy of environment and recuperation of heat from the channel (8). Next, the air, saturated in the M-cycle apparatus (point *B*, Fig. 3), is directed to the condenser (3) of the Rankine heat pump where due to heating, it achieves air parameters that meet sanitary conditions [25] of the heating system (point *C*, Fig. 3). Air from the room (point *D*, Fig. 3, *a*) enters through the exhaust air duct the working channel (8) of the M-cycle heat-and-mass exchanger where it is cooled to the dew point at the channel outlet and sent to the evaporator (4) of the heat pump (point *F*, Fig. 3, *a*). After cooling in the evaporator (point *E*, Fig. 3, *a*), the air is released to the environment.

Fig. 2. Photograph of the installation of air heat supply according to the combined thermodynamic cycle with heat recuperation: 1 — the M-cycle heat-and-mass exchanger; 2 — the heat pump condenser; 3 — the heat pump evaporator; 4 — the heat pump compressor

Scheme 2. This scheme [20] was previously used in the conceptual study of the combined cycle [21]. According to this scheme, moist air from the M-cycle apparatus is also sent to the condenser (3) of the Rankine heat pump but after circulation and heating in the room (point *D*, Fig. 3, *b*), it enters the evaporator (4) of the heat pump where it is precooled (point *F*, Fig. 3, *b*) and then goes further into the working channel (8) of the M-cycle apparatus. After cooling the airflow in the channel (8) to the dew point, moisture condensation begins with a release of heat which enters the wet channel through the channel wall and is used to evaporate water and obtain saturated air. Next, the cooled air (point *E*, Fig. 3, *b*) is released into the atmosphere. This scheme provides a higher level of heat dissipation in the channel (8) and its regeneration due to moisture condensation in the intermediate section of the working channel (8).

The thermodynamic processes occurring in the installations are presented in the psychrometric diagram of moist air (Fig. 4). The processes in Fig. 4, *a, b* depict the data obtained in experimental studies (Table 1). Point *A* corresponds to the parameters of the incoming atmospheric air which is first cooled in the dry channel (6) of the M-cycle apparatus to the dew point temperature, then enters the wet channel (7) where it is saturated with moisture and heated along the saturation line (point *B*) due to the heat supply from the dry and working channels. Heat transfer from the colder dry channel is due to the psychrometric temperature difference between the air in the dry channel (6) and the water film on the wall surface of the wet channel (8). Heat transfer from the working channel (8) to the wet channel (7) occurs by the same principle of psychrometric temperature difference due to the cooling of exhaust air and condensation of water vapor. Due to humidification in the Maisotsenko heat-and-mass exchanger and heating of the air coming from the condenser of the Rankine heat pump (process *B*–*C*), the air parameters corresponding to the sanitary conditions are achieved. Warm air enters the room and mixes with room air (process *C*–*D*).



a *b*

Fig. 3. The scheme of installation of air heat supply according to the combined thermodynamic cycle: *a* – with heat recuperation; *b* – with pre-cooling of air and heat recuperation; 1 – M-cycle heat-mass exchanger; 2 – Rankine cycle heat pump; 3 – the heat pump condenser; 4 – the heat pump evaporator; 5 – room; 6 – dry channel; 7 – wet channel; 8 – working channel

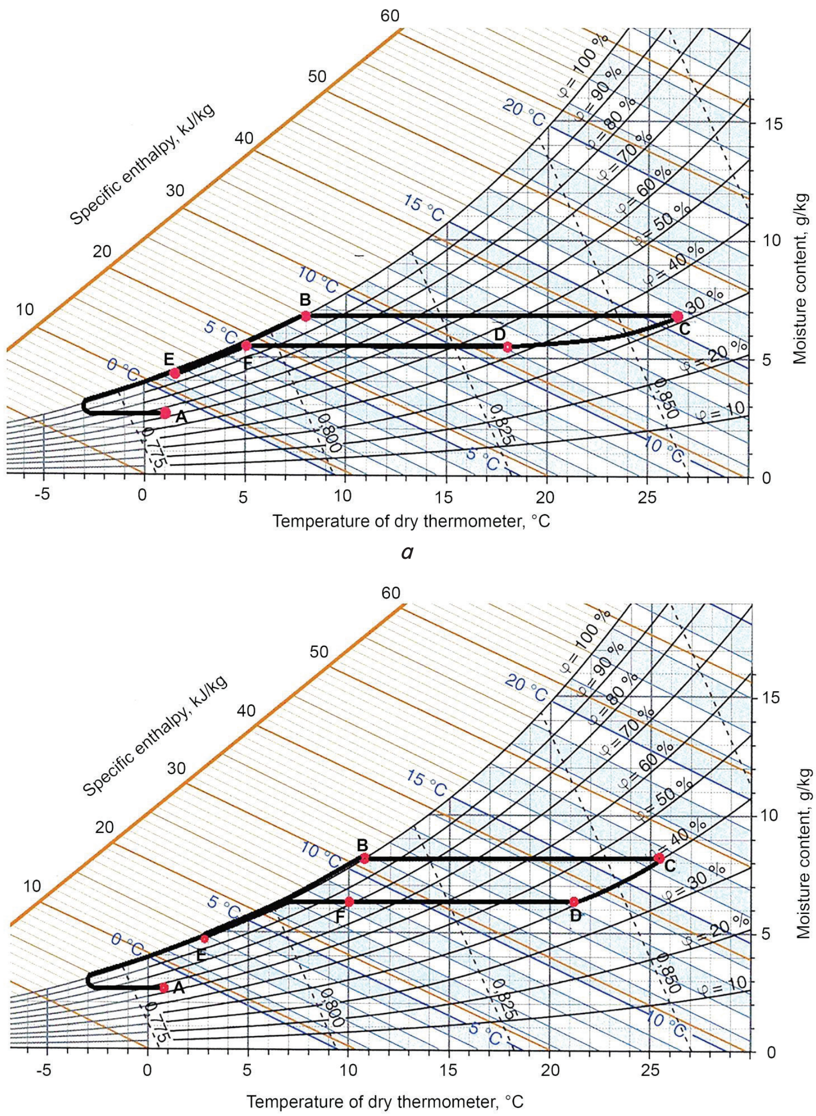


Fig. 4. Psychrometric diagrams of the combined thermodynamic cycle: *a, b* – scheme 1 and 2, respectively

The process D–F for scheme 1 (Fig. 4, *a*) is characterized by cooling the air from the room in the working dry channel and moisture condensation with subsequent cooling of air in the heat pump evaporator (process F–E). Whereas the process C–D for the scheme 2 (Fig. 4, *b*) consists in cooling the room air and the process D–F–E consists in cooling of the humid air from the Rankine heat pump evaporator (a part of the process D–F), cooling of air to the dew point temperature (point F) in the dry channel (8) and subsequent condensation of moisture in it (process F–E). Thus, the process D–F–E (except for a part of the process D–F) determines the heat of recuperation (return of heat) in the M-cycle apparatus.

The theoretical temperature of the air released from the room (point E) may correspond to the wet thermometer ambient temperature and even below it in some cases. This means that due to the low degree of irreversibility and the high degree of the cycle regeneration, efficiency of the M-cycle apparatus can approach one. As is known, the efficiency of the best industrial heat exchangers in «dry» apparatuses is not more than 0.7...0.75.

4. 2. Refrigeration for air conditioning

Two schemes (Fig. 3) can be used for cooling (air conditioning) in summer and a warm spring season. In this case, the Rankine heat pump is turned off and only the M-cycle heat-and-mass exchanger is used (Fig. 5).

During cooling, atmospheric air from the environment enters the M-cycle apparatus (1) in two separate flows (Fig. 3). One of the flows passes through the working channel (2) of the heat-and-mass exchanger where it is cooled and finally enters the room. The other flow passing through the dry channel (4) and the wet channel (3) is saturated with moisture, takes heat from the dry and working channel and is released into the atmosphere.

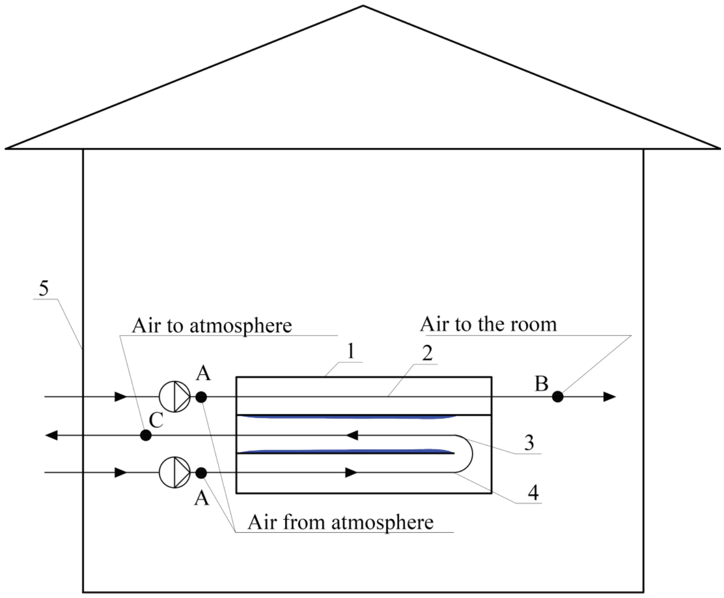


Fig. 5. The scheme of the M-cycle air cooling installation: 1 — the M-cycle heat-and-mass exchanger; 2 — the working channel; 3 — the damp channel; 4 — the dry channel; 5 — the room

Theoretically, air can reach the dew point temperature at the outlet of the channel (2) due to the sufficiently large length of this channel but it is more appropriate to create apparatuses that cool the air slightly below the wet thermometer temperature.

5. The results obtained in the experimental study of the new schemes of local heat supply and cooling based on the Maisotsenko cycle

5. 1. Heat supply

An experimental bench of the combined cycle of thermal power of 28 kW in which power of the Rankine cycle heat pump is 3 kW was designed and constructed for experimental study of both schemes. The bench design uses a serial M-cycle heat-and-mass exchanger manufactured by Coolerado Corporation, USA, (Fig. 2).

The measurement system included determination of all parameters which are necessary for determining the efficiency of the M-cycle apparatus and total efficiency of the installation (preparation of air to sanitary norms). The parameters include temperature, relative humidity, and mass airflow. For this purpose, standard resistance (copper) temperature transducers and hygrometers (standard converters of humidity of capacitive type) were installed in air ducts in control points (*A, B, C, D, E, F*). Thermo anemometer was installed at the entrance to the M-cycle apparatus to measure the velocity field and determine the mass airflow (points *A*, *E*, Fig. 3). Maximum measurement errors were, respectively, 1.7 % for air temperature; 4.0 % for relative humidity; 2.5 % for specific enthalpy; 1.0 % for flow rate; 1.4 % for mass airflow; 3.9 % for heat flux; 2.0 % for electric power of the fan and compressor of the heat pump; 5.9 % for COP.

Experimental studies were conducted in the 2017–2019 winter seasons at various values of temperature and humidity. The study results are presented in Table 1 where t is temperature, φ is relative humidity, h is the enthalpy of air. The results of calculations included determination of the efficiency of the Maisotsenko heat-and-mass exchanger in terms of temperature:

$$\eta^t = \frac{t_D - t_E}{t_D - t_A} \quad (1)$$

and efficiency in terms of air enthalpy because the apparatus directly depends on the change in the phase state of water vapor in the air:

$$\eta^h = \frac{h_D - h_E}{h_D - h_A}. \quad (2)$$

The overall efficiency of the combined cycle installation was determined using the COP:

$$\mathrm{COP} = \frac{Q_M + Q_{HP}}{N_M + N_{HP}}, \quad (3)$$

where $Q_M = G \cdot (h_B - h_A)$ is the heat flow that enters the airflow in the dry and wet channels of the M-cycle apparatus, W; $Q_{HP} = G \cdot (h_C - h_B)$ is the heat flow coming to the airflow from the heat pump condenser, W; N_M is the fan power for pumping air through the M-cycle heat-and-mass exchanger, W; N_{HP} is the heat pump power, W; $G = \rho \cdot \overline{w} \cdot f$ is mass airflow, kg/s; ρ is the air density, kg/m^3; $\overline{w}$ is the average airflow rate in the air duct, m/s; f is the cross-sectional area of the air duct, m^2.

The mass airflow rate was 0.33 kg/s for all experiments. The calculation results are given in Table 2.

Table 1

Input experimental data and calculation results

No.	Parameters		The installation scheme									
			1	2	1	2	1	2	1	2	1	2
1	Ambient air properties (p. *A*)	t, °C	–10	–5	–9	–0.8	–6	0.9	–3	1.5	1	4
		φ, %	63	80	58	65	50	63.5	65	60	60	70
2	Properties of air after passing moist channels of the M-cycle apparatus (p. *B*)	t, °C	4	3.5	4	6	4	11.4	7.3	8.7	8	8.7
		φ, %	100	100	100	100	100	100	100	100	100	100
3	Ventilation discharge from the room (p. *D*)	t, °C	16	18	15	18.4	16	21.2	19	18.8	18	16.4
		φ, %	35	45	40	36	40	42.7	35	43	42	52
4	Air properties after passing the M-cycle apparatus (p. *F*)	t, °C	0	7.3	1	7	1	10	4	9	5	8
		φ, %	100	90	100	77	100	87	100	82	100	84
5	Properties of the air discharged into atmosphere after passing the M-cycle apparatus (p. *E*)	t, °C	–3.5	–0.2	–2.5	0.7	–6	3.9	0	4.2	2	5.2
		φ, %	100	100	100	98	100	100	100	100	100	100

Table 2

Input experimental data and calculation results

No.	Parameters	The installation scheme									
		1	2	1	2	1	2	1	2	1	2
1	Efficiency of the M-cycle apparatus in terms of temperature, %	62.3	79.1	58.1	92.2	68.0	85.2	68.1	84.4	72.0	90.3
2	Efficiency of the M-cycle apparatus in terms of enthalpy, %	49.2	71.9	44.4	77.6	52.1	70.2	48.0	61.1	50.3	66.7
3	COP	7.21	7.44	7.03	7.46	7.03	9.73	6.84	8.39	6.3	8.74

The efficiency of the M-cycle apparatus in terms of temperature was from 58.1 to 72.0 % for scheme 1 (Table 2) and from 79.1 to 92.2 % for scheme 2. Lower efficiency for scheme 1 is explained by the failure to achieve condensation of water vapor in the channel (8). Since temperature and enthalpy of air at the outlet of the M-cycle apparatus were above atmospheric conditions in all cases, the requirements for complete cooling of the flow in dry channels of the M-cycle apparatus have not been fulfilled. The coefficient of performance was 6.3–7.21 for scheme 1 and 7.44–9.73 for scheme 2. This is explained by the fact that additional heat for the circuit 2 was obtained in the working channel (8) due to pre-cooling of the exhaust air by the heat pump evaporator (4) and condensation of moisture in the channel (8).

5. 2. Air cooling

Energy for air conditioning was consumed exclusively by the fans for air transportation, so the installation efficiency when working for the needs of refrigeration was determined by the coefficient of performance COP from the equation:

$$\text{COP} = \frac{Q_{ERV}}{N_{ERV}} = \frac{(h_A - h_B)\cdot m}{N_{ERV}}. \quad (4)$$

In the experimental study in the 2018 summer season, the airflow rate was 0.816 kg/s or 2,300 m^3/h. Experiments have shown (Table 3) that the COP value of the M-cycle heat-and-mass exchanger varied from 10.49 to 16.32 at a relative humidity of 19.6 to 38.8 %. Some temperature difference at points A and C was caused by the imperfect nature of the Maisotsenko cycle.

Experimental studies were conducted with air conditioning without air recirculation, that is, with a supply of completely fresh air from the environment in contrast to conventional air conditioning systems in which the degree of recirculation is not less than 80 %.

The COP cycle increased significantly with air recirculation. Even higher efficiency can be obtained if the room is cooled in a closed air loop, such as at service or data centers.

6. Discussion of the results obtained in the study of new schemes of local heat supply and cooling based on the Maisotsenko cycle

The high efficiency of the heat supply system (Table 2) is explained by the combination of the M-cycle heat-and-mass exchanger and the Rankine cycle heat pump.

Due to the receipt of additional energy from the environment and almost complete utilization of heat emissions by the M-cycle apparatus (Table 2), air from the environment is heated and humidified in the cold season. Air is further heated by the heat pump to the conditions specified by sanitary norms.

From the point of view of the operation, the presented schemes for heating have restrictions on ambient temperature as the Maisotsenko apparatus is structurally executed as a heat-and-mass exchanger with a system of dry and damp channels. Walls of wet channels are covered with water which can freeze at very low temperatures. However, since the water layer is very thin and actively evaporates, the apparatus works in a normal mode, even at temperatures down to –12 °C. For the needs of air conditioning in rooms, the efficiency of the Maisotsenko apparatus deteriorates at relative humidity more than 80 %. At relative humidity more than 95 %, the apparatus will not work as the cooling process occurs due to the evaporation of the water film in the airflow. The evaporation process is weak or virtually absent at this relative humidity. Also, the constant need to feed the installation with water imposes restrictions on operation in countries where water is in deficiency. Water is evaporated and discharged. Its consumption is at a level of 6 g for every 1 m^3 of air. It is possible to create a closed-loop installation or return water but this measure reduces the COP.

The disadvantage is the small array of experimental data obtained and the fact that all experimental studies were conducted only within the Kyiv region. Performing an experimental study on a larger scale and in various global climatic conditions is a promising way to improve the results.

Elaboration of new schemes and the use of a more advanced heat pump and the structurally better Maisotsenko apparatus built from modern materials is the direct achievement of this study.

Table 3

The results of experimental study in the 2018 summer season

No.	Ambient air properties			Fresh air properties at the exit from the working channel of the M-cycle apparatus (before entering the room)			Damp air properties at the exit from the M-cycle apparatus (discharge to the environment)			COP
	p. A			p. B			p. C			
	t, °C	φ, %	h, kJ/kg	t, °C	φ, %	h, kJ/kg	t, °C	φ, %	h, kJ/kg	
1	26.9	38.8	49	18	63.6	39	23	100	64	11.66
2	29	30	48	21	46	39	22.5	100	66	10.49
3	32.2	19.6	48	17.6	44.8	34	25	100	76	16.32
4	27	35	49	19	60	40	23.5	100	67	10.49

7. Conclusions

1. Two new schemes of air heat supply installations according to the combined cycle have been developed. They include a heat-and-mass exchanger according to the Maisotsenko cycle and an air heat pump according to the Rankine cycle.

2. An experimental bench of the heat supply system for the needs of

room heating has been designed on the basis of combining the Maisotsenko and Rankine cycles.

3. Experimental studies confirmed the high energy efficiency of both schemes: COP was 6.3–7.21 for scheme 1 and 7.44–9.73 for scheme 2. The higher values of COP according to scheme 2 are due to cooling of the air in the heat pump evaporator before recuperation in the M-cycle heat-and-mass exchanger and moisture condensation in the working channel. In general, the high COP of both schemes is explained by the use of the psychrometric energy of the environment and the high degree of heat regeneration in the M-cycle heat-and-mass exchanger.

4. When conditioning air in the combined scheme, the Rankine heat pump was not used, so the energy was consumed only by the fan to pump air through the M-cycle heat-and-mass exchanger. Experimental studies of the heat-and-mass exchanger operating according to the Maisotsenko cycle have shown record COP values: 10.49–16.32. This figure is much higher than that with conventional compression heat pumps in which COP is 2 to 5.

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DEVELOPMENT OF A SYSTEM FOR ESTIMATING AND FORECASTING THE RATIONAL RESOURCE-SAVING OPERATING MODES OF TPP

O. Chernousenko
Doctor of Technical Sciences, Professor*
E-mail: chernousenko20a@gmail.com

D. Rindyuk
PhD, Associate professor*
E-mail: rel_dv@ukr.net

V. Peshko
PhD, Senior Lecturer*
E-mail: vapeshko@gmail.com

V. Goryazhenko
Postgraduate Student*
E-mail: vgandcohookah@gmail.com

O. Chernov
Postgraduate Student*
E-mail: wist1706@gmail.com

*Department of Cogeneration Installations of Thermal and Nuclear Power Plants
National Technical University of Ukraine «Igor Sikorsky Kyiv Polytechnic Institute»
Peremohy ave., 37, Kyiv, Ukraine, 03056

Світова енергетична криза та екологічні проблеми спонукали стрімке збільшення долі генерації електроенергії відновлювальними джерелами. Проте, режими роботи таких енергопотужностей є досить нерівномірними протягом доби. Так, для енергетики більшості країн, що розвиваються, та частини розвинених країн такі коливання в загальному балансі енерговиробітку призводять до примусового обмеження потужності обладнання на ТЕС, або й до повної зупинки енергоблоків.

Недостатня кількість маневрових потужностей в загальному енергогенеруючому балансі є характерною рисою енергосистем таких країн як Російська Федерація, Україна, Македонія, Болгарія, Румунія, Аргентина та низки інших. Нестача таких потужностей часто компенсується за рахунок пиловугільних блоків потужністю 200–300 МВт, що залучаються до роботи в напівпікових та пікових режимах. Дане обладнання проектно не пристосовано до такої роботи. Тому для запобігання передчасного зношення генеруючого устаткування ТЕС пропонується розробка режимного методу управління ресурсом.

На базі технічного аудиту експлуатаційної документації генеруючих компаній, запропоновано метод, спрямований на прогнозування раціональних ресурсоощадних режимів роботи високотемпературних елементів енергетичного обладнання шляхом оптимізації дольового відношення числа пусків устаткування з різних теплових станів. Сформульовано задачу оптимізації, яка полягає у визначенні такого розподілу конструктивно-технологічних параметрів процесу, що забезпечують максимальне збереження ресурсу обладнання. Як цільову функцію задачі оптимізації обрано залишковий ресурс. Розроблений метод представлено у вигляді комплексної системи оцінки та прогнозування раціональних режимів експлуатації високотемпературних елементів ТЕС, що дає можливість визначення індивідуальних ресурсних показників на протязі всього часу експлуатації обладнання для усіх можливих майбутніх комбінацій режимів роботи та згенерувати прогнози для тисяч різних варіантів режимної експлуатації енергоблоку, з розрахунком ресурсних показників для кожного з них

Ключові слова: теплова електростанція, енергоблок, пошкоджуваність, залишковий ресурс, прогнозування, оптимізація

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1. Introduction

An important issue related to the energy systems of many countries, including parts of Europe, the CIS, Argentina, and South Africa, is the lack of peak capacities in the overall generating energy balance. As a result, there is a need to involve base and semi-base TPP energy units to control power in the grid. However, such equipment is not designed to operate under frequent variable modes. Violating the service regulations, caused by the necessity of excessively frequent launches and stops of power units, leads to a sharp decrease in the lifetime of the specified power units [1].