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INNOVATIVE AIR CONDITIONING SYSTEM WITH RATIONAL DISTRIBUTION OF THERMAL LOAD

Abstract: *The efficiency of air conditioning (AC) systems depends on the operation of their air coolers at varying heat loads in response to current changeable climatic conditions. The intensity of heat transfer of refrigerant, evaporated inside air coils, drops at the final stage of evaporation, that is caused by drying out the inner wall surface. This results in lowering the overall heat transfer coefficient and reduction of air cooler efficiency in the whole. The concept of overfilling air coils that leads to excluding a dry-out of their inner surface and falling the overall heat transfer intensity at variation of refrigerant flows in response to change of current thermal load on air coolers is developed.*

Keywords: *air conditioning system, heat transfer coefficient.*

Introduction

Air conditioning (AC) systems have grown wide application practically in all fields of human activities: comfort AC in building [1-3] and technological AC in food processing [4-6], gas, oil [7-9] and other technologies, in transport applications, including railway [10-12] and ship [13-15].

They are applied as independent systems as well as subsystems in integrated energy plants (IEP) [16-18] for combined cooling, heat and power (CCHP) or trigeneration [19-21].

The efficiency of AC systems depends on the operation of the whole waste heat recovery complex including extracting the heat into atmosphere by cooling towers [22, 23] and utilizing the exhaust heat from combustion engines [24-26]. Complex waste heat recovery with deep exhaust heat utilization by applying low temperature condensing surfaces [27-29], thermopressor [30-32], ejector [33, 34] and other jet devices [35, 36] and turboexpander technologies [37, 38] are used to enhance the cooling potential and efficiency of AC systems.

The performance efficiency of air conditioning (AC) systems depends on the heat efficiency of their heat exchangers. A lot of publications are devoted to intensification of heat transfer in evaporators [39-41] and condensers [42-44], enhancement of hydrodynamics in minichannel heat exchangers [45, 46] to mitigate flow maldistribution [47, 48], application of two-stage cooling in AC systems for comfort and energetic application [49, 50].

In all cases AC systems operate at variable heat loads in response to actual climatic conditions [50, 51] or have to provide efficient performance of machinery or power plants and combustion engines at varying climatic conditions. Generally, an overall heat load of any AC system comprises the unstable heat load range, corresponding to ambient (outdoor) air processing with considerable heat load fluctuations in response to actual climatic conditions, and a comparatively stable heat load part for subsequent air cooling (subcooling) to a target temperature [49, 50]. The ambient air precooling mode with considerable heat load fluctuation needs load modulation, whereas the comparatively stable heat load range can be covered by operation of refrigerant compressor at about nominal mode.

In modern variable refrigerant flow (VRF) systems the load modulation is performed by varying refrigerant feed to air coolers [52, 53]. But the problem of inefficient operation of air coolers caused by dry-out of inner walls at the final stage of refrigerant evaporation remains unsolved [49].

The intensity of heat transfer of refrigerant, evaporated inside air coils of air coolers, drops at the final stage of evaporation, that is caused by drying out the inner wall surface while transition of refrigerant two-phase flow from annular to disperse (mist) flow. A sharp decrease in heat transfer coefficient to refrigerant at the final stage of its evaporation in compact air coolers results in lowering the overall heat transfer coefficient and reduction of air cooler efficiency.

A concept of efficient operation of air coolers due to incomplete refrigerant evaporation by injector recirculation of liquid refrigerant that excludes the final burn-out stage of evaporation with drop in intensity of evaporation heat transfer was considered in [49].

A new impulse for further realization of this concept is forced due to applying a circulation of liquid refrigerant, i.e. over filling all the coils of air cooler, that excludes a drop in intensity of evaporation heat transfer leading to overall heat transfer intensity decrease and, as result, the influence of variation in refrigerant flows in response to change of current thermal load on air coolers.

The aim of research is to developed a concept of incomplete refrigerant evaporation with overfilling air coils that leads to excluding a dry-out of their inner surface and falling the overall heat transfer intensity while variation in refrigerant flows in response to change of current thermal loads on air coolers.

Research Methodology

The main idea behind the rational designing and operation of ambient air conditioning systems to match current varying heat loads is sharing the overall heat load in unstable heat load range, corresponding to ambient air processing with considerable load fluctuations in response to actual climatic conditions, and a comparatively stable heat load part for subsequent air cooling (subcooling).

A rational design overall refrigeration capacity $Q_{0.10\text{rat}}$ for cooling ambient air for instance to t_{a2} = 10°C is selected to provide a maximum annular refrigeration energy generation according to AC duties and shared into a comparatively stable basic load and a remaining part for ambient air precooling at varying heat loads [49].

All the calculation results are presented for the refrigeration capacity in relative values of specific refrigeration capacity q_0 as the overall refrigeration capacity Q_0, kW, referred to the unit of air mass flow $G_a : q_0 = Q_0 / G_a$, kW/(kg/s), or kJ/kg; G_a – air mass flow in air cooler, kg/s.

With this the values of specific refrigeration capacity $q_{0.15}$ for cooling ambient air from its current temperature t_{amb} to the temperature t_{a2} = 15°C and $q_{0.10}$ for cooling ambient air to t_{a2} = 10°C and specific refrigeration capacity $q_{0.10-15}$ as their difference $q_{0.10-15} = q_{0.10} - q_{0.15}$ for subcooling air from t_{a2} = 15°C to t_{a2} = 10°C have been calculated for current climatic conditions.

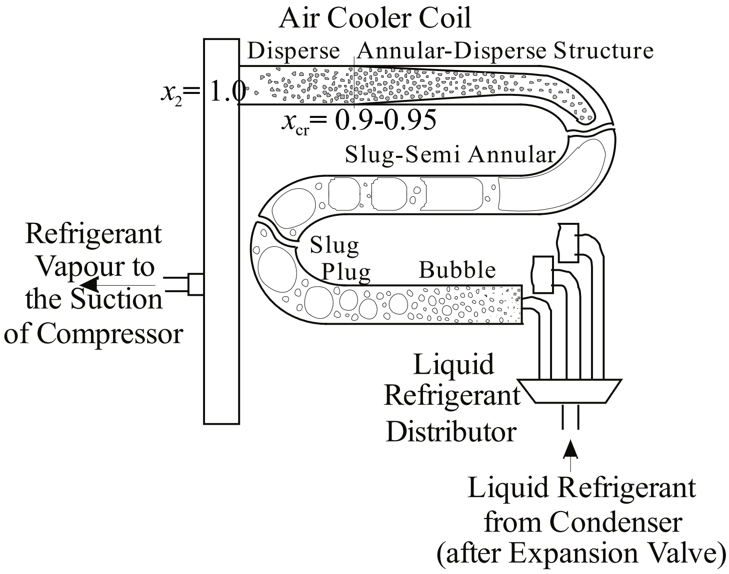
A remaining available part $q_{0.A10-15}$ for ambient air precooling at varying heat loads is calculated as difference $q_{0.A10-15} = q_{0.10} - q_{0.10-15}$.

This study takes into account long term annual weather data collected in the weather datasets of various meteorological centres by using “on-line” programs like “mundomanz.com” or others.

Results of investigation

Typical structures of inside tube refrigerant evaporation and behaviour of refrigerant heat transfer coefficients α_a with the vapor mass fraction x are presented in Figure 1.

a)



b)

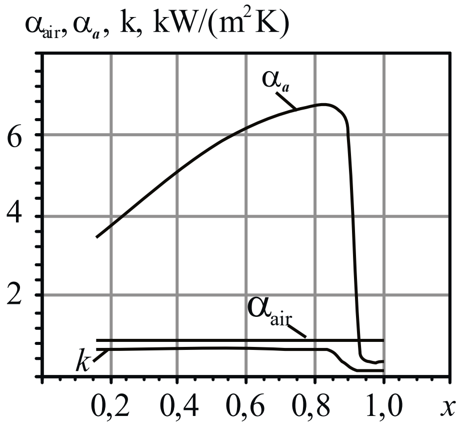


***Figure 1.** Typical structures of in tube refrigerant boiling (a) and variation of heat transfer coefficients to boiling refrigerant α_a and air α_{air} and overall heat transfer coefficient k with the vapor mass fraction x (b)*

The convective evaporation of refrigerant inside channels is characterized by sharp drop in intensity of heat transfer at the final stage of evaporation when so called burnout takes place. This occurs due to inner channel wall surface drying out with transition of refrigerant two-phase flow from annular-disperse flow to disperse (mist) flow (Fig. 1a).

In compact air coolers with finned tubes the coefficient of heat transfer to refrigerant α_a at the final stage of its evaporation is much lower than α_{air} to air. This results in decrease in overall heat transfer coefficient k (Fig. 1b).

Calculations are performed for the air cooler with plate finned tubes of 12 mm and 10 mm outside and inside diameters, air temperature at the inlet t_{air1} = 25°C and outlet t_{air2} = 15°C, refrigerant boiling temperature at the exit t_{02} = 0°C, refrigerant R142b.

Considerable lowering the heat transfer coefficient to refrigerant α_a which becomes lower than the heat transfer coefficient to air α_{air} and causes a decrease in the overall heat transfer coefficient k at burnout vapor fraction $x_{cr} \approx 0.9$ corresponding to drying the channel wall surface with the transition from annular to disperse flow that leads to the sharp decrease in the heat flux q.

To provide intensive heat transfer on all the length of air cooler coils it is necessary to exclude their ending post dry out sections, i.e. make the air coolers operate with incomplete boiling. The unevaporated liquid should be separated from the vapour in the liquid separator and directed again at the entrance of air cooler.

The results of thermal efficiency comparison of conventional air cooler with complete evaporation and superheated vapor at the exit and of advanced air cooler with incomplete evaporation are shown in Figure 2.

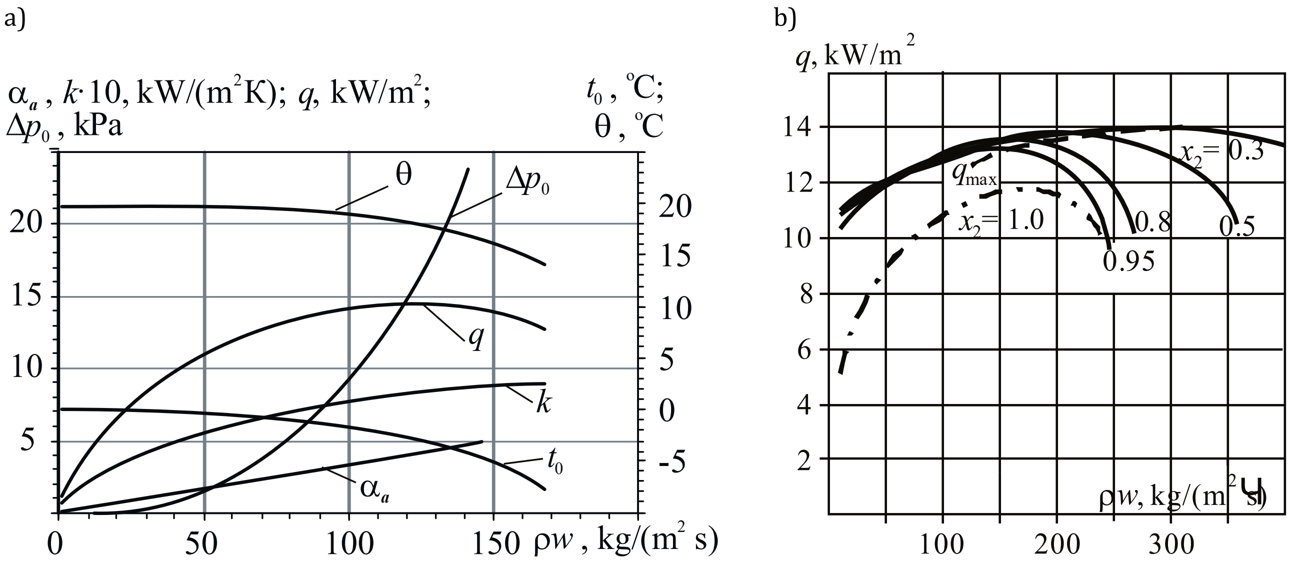


FIGURE 2. *Mean values of heat fluxes q, heat transfer coefficients to refrigerant α_a and overall heat transfer coefficients k, logarithmic temperature difference θ, refrigerant boiling temperature t_0 and pressure drop ΔP against refrigerant mass velocities ρw for complete evaporation (a) and heat fluxes q at mass vapor fraction x_2 at the outlet of air coil for incomplete refrigerant evaporation (b): R142b, t_{02} = 0°C; air velocity w = 6 m/s*

Thus, overfilling the air coils of the air cooler by liquid refrigerant provides an increase in heat flux *q* by 25%, ..., 40% compared with conventional complete refrigerant evaporation and enables a larger deviation of refrigerant mass velocities *ρw* from their optimum value, providing maximum value of heat flux *q*. This means that larger heat load changes are permitted, that give good perspectives for application of overfilling the air coolers by liquid refrigerant in ambient air conditioning systems characterized by considerable fluctuations of loading.

To prove a methodological approach to determine a design heat load, matching current changeable climatic conditions, the values of specific refrigeration capacity $q_{0.15}$ for cooling ambient air from its current temperature t_{amb} to the temperature t_{a2} = 15°C and $q_{0.10}$ for cooling ambient air from t_{amb} to t_{a2} = 10°C and specific refrigeration capacity $q_{0.10-15}$ as their difference $q_{0.10-15} = q_{0.10} - q_{0.15}$ for cooling air from t_{a2} = 15°C to t_{a2} = 10°C have been calculated for climatic conditions in Nikolaev region, southern Ukraine, in July 2015 (Fig. 3).

As can be seen from Figure 3, with cooling the ambient air from t_{amb} to t_{a2} = 15°C the fluctuations in the current heat load $q_{0.15}$ on the air cooler of AC system are very significant. But when air is being cooled from t_{a2} = 15°C to t_{a2} = 10°C, the fluctuations of the heat load on the air cooler of AC system $q_{0.10-15} = q_{0.10} - q_{0.15}$ are relatively small: from 10 kW/(kg/s) to 12 kW/(kg/s).

Obviously, the range of refrigeration capacity controlling according to heat load can be narrowed by dividing the current heat load range on the air cooler in two parts: the relatively stable basic part $q_{0.10-15} = q_{0.10} - q_{0.15}$ while cooling air from t_{a2} = 15°C to t_{a2} = 10°C, and its extremely unstable part $q_{0.15}$ of precooling the ambient air from its current temperature t_{amb} to t_{a2} = 15°C.

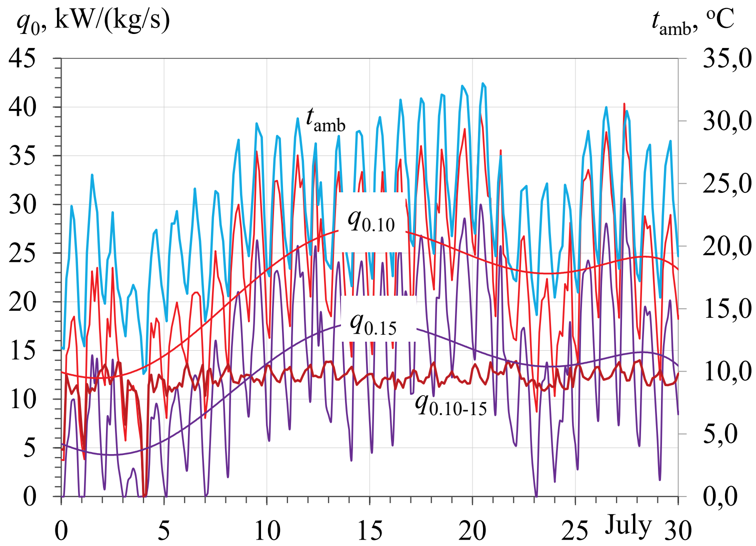


FIGURE 3. *Current values of specific refrigeration capacity $q_{0.15}$ for cooling ambient air from t_{amb} to t_{a2} = 15°C, $q_{0.10}$ for cooling ambient air from t_{amb} to t_{a2} = 10°C and refrigeration capacity $q_{0.10\text{-}15}$ for cooling air from t_{a2} = 15°C to t_{a2} = 10°C: $q_{0.10\text{-}15} = q_{0.10} - q_{0.15}$*

So, the stable heat load value $q_{0.10-15}$ is chosen as design basic part $q_{0.10-15} = q_{0.10} - q_{0.15}$ of the rational design total heat load $q_{0.10\text{rat}}$ about 34 kW/(kg/s), ..., 35 kW/(kg/s) on the whole air cooler of AC system determined according to maximum annual refrigeration energy generation [49].

The available rest part of the total heat load $q_{0.10\text{rat}}$ on the whole air cooler might be used for precooling the air from the current changeable ambient temperature t_{amb} to t_{a2} = 15°C and be determined according to a remained principle as available loads $q_{0.\text{A}10-15} = q_{0.10\text{rat}} - q_{0.10-15}$.

So, the total unstable current heat load $q_{0.10}$ for cooling ambient air from the changeable current ambient temperature t_{amb} to t_{a2} = 10°C can be covered by two stage ambient air cooling (Fig. 4).

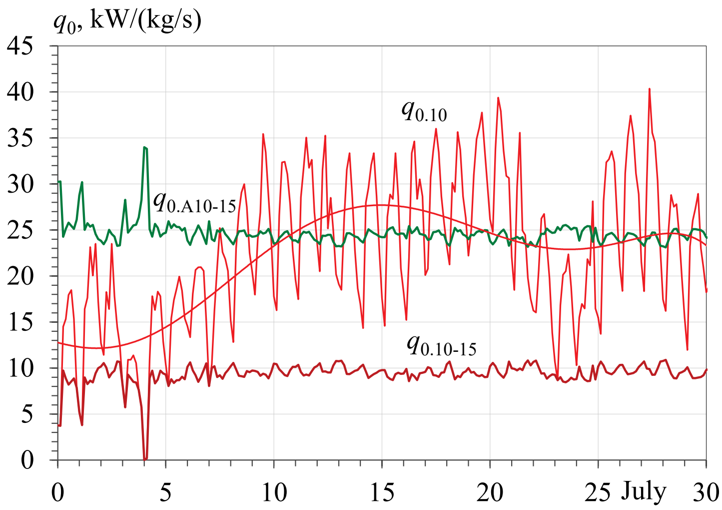


FIGURE 4. *Current values of changeable heat load $q_{0.10}$ for cooling ambient air from t_{amb} to t_{a2} = 10°C covered by available rest specific refrigeration capacity $q_{0.A10\text{-}15}$ and by basic specific refrigeration capacity $q_{0.10\text{-}15}$ for cooling air from t_{a2} = 15°C to t_{a2} = 10°C: $q_{0.10\text{-}15} = q_{0.10} - q_{0.15}$; $q_{0.A10\text{-}15} = q_{0.10rat} - q_{0.10\text{-}15}$*

As Figure 5 shows, the available rest specific refrigeration capacity $q_{0.\text{A}10-15}$ generally covers current heat loads $q_{0.15}$ for precooling the air from the ambient temperature t_{amb} to the temperature t_{a2} = 15°C, except a few the warmest quite short periods of daylight hours.

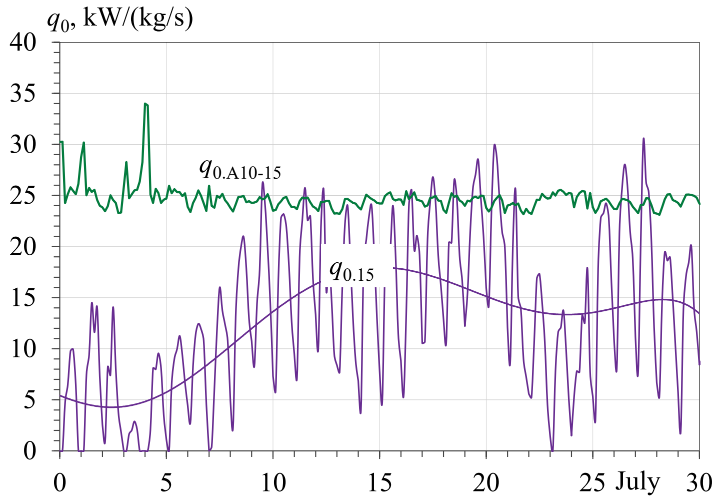


FIGURE 5. *Current values of changeable current heat load $q_{0.15}$ for cooling ambient air from current temperature t_{amb} to t_{a2} = 15°C covered by available rest specific refrigeration capacity $q_{0.A10-15}$*

It is quite reasonable to suppose that the less available rest range of refrigeration capacity for precooling of the ambient air with fluctuations of the current heat load on the air cooler, the lower energy losses caused by the operation of the compressor refrigeration machine in partial modes. But such supposal about more narrow available rest range of refrigeration capacity will be correct if the basic range of refrigeration capacity for futher deep cooling of air from a higher temperature, for example t_{a2} = 20°C, remains stable.

In order to make any conclusion the values of specific refrigeration capacity $q_{0.10}$ for cooling ambient air from its current temperatures t_{amb} to t_{a2} = 10°C are shared in two ranges: refrigeration capacities $q_{0.A10-20} = q_{0.rat} - q_{0.10-20}$ for precooling ambient air from t_{amb} to t_{a2} = 20°C and $q_{0.10-20} = q_{0.10} - q_{0.20}$ for further cooling air from t_{a2} = 20°C to t_{a2} = 10°C have been calculated (Fig. 6).

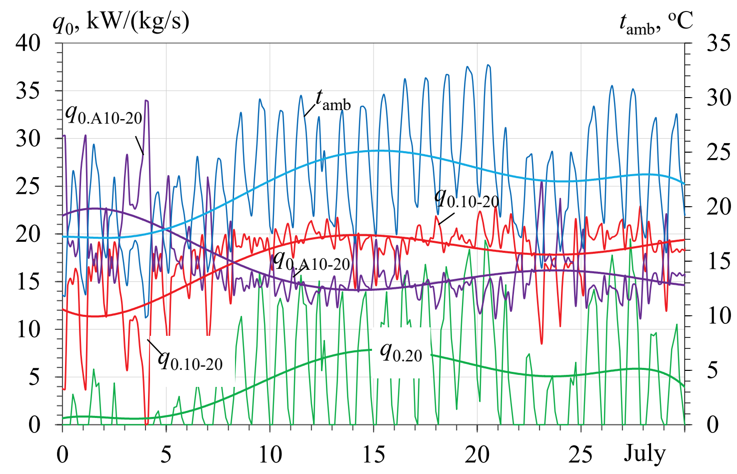


FIGURE 6. *Current values of specific refrigeration capacity $q_{0.20}$ needed for cooling ambient air from t_{amb} to t_{a2} = 20°C, refrigeration capacities $q_{0.A10-20}$ for cooling ambient air from t_{amb} to t_{a2} = 20° and $q_{0.10-20} = q_{0.10} - q_{0.20}$ for further cooling air from t_{a2} = 20°C to t_{a2} = 10°C: $q_{0.10-20} = q_{0.10} - q_{0.20}$; $q_{0.A10-20} = q_{0.rat} - q_{0.10-20}$*

As Figure 6 shows, with precooling the ambient air from t_{amb} to the temperature t_{a2} = 20°C the fluctuations in the current heat loads $q_{0.20}$ on the air cooler of the AC system are very significant. This is caused by daily ambient air temperate t_{amb} dropping lower 20°C (within 1-9 July) with corresponding falling down to zero of the refrigeration capacity $q_{0.20}$ needed. In its turn, this causes decreasing the refrigeration capacity $q_{0.10-20}$ required for further cooling air from t_{a2} = 20°C to t_{a2} = 10°C and leads to excess of the available rest refrigeration capacity $q_{0.A10-20}$ as compared with $q_{0.20}$ needed.

The excess of available design refrigeration capacities $q_{0.A10-20} = q_{0.10rat} - q_{0.10-20}$ can be used for deeper cooling ambient air, for instance to t_{a2} = 15°C, i.e. to cover heat load $q_{0.15}$ compared with designed $q_{0.20}$ (Fig. 7).

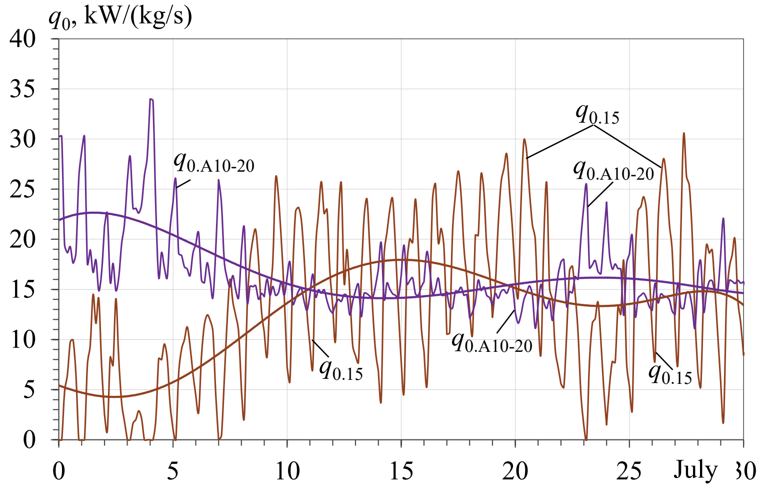


FIGURE 7. *Current values of specific refrigeration capacity $q_{0.15}$ needed for cooling ambient air from t_{amb} to t_{a2} = 15°C and available design refrigeration capacities $q_{0.A10-20} = q_{0.rat} - q_{0.10-20}$*

As it is seen, the available design refrigeration capacities $q_{0.A10-20}$ are considerably higher than refrigeration capacity $q_{0.15}$ needed for cooling ambient air to t_{a2} = 15°C within 1-9 and 22-25 July and lower along the rest daylight time. But large fluctuations of current heat loads $q_{0.15}$ reveal the possibilities to cover a deficit of available design refrigeration capacities $q_{0.A10-20}$ during daylight hours by its excess accumulated within night hours. This leads to reduction of installed refrigeration capacities by the values of $\Delta q_{0.15-20} = q_{0.15} - q_{0.20}$ with using as basic the reduced design refrigeration capacity $q_{0.10-15} = q_{0.10} - q_{0.15}$ instead of $q_{0.10-20} = q_{0.10} - q_{0.20}$ (Fig. 8).

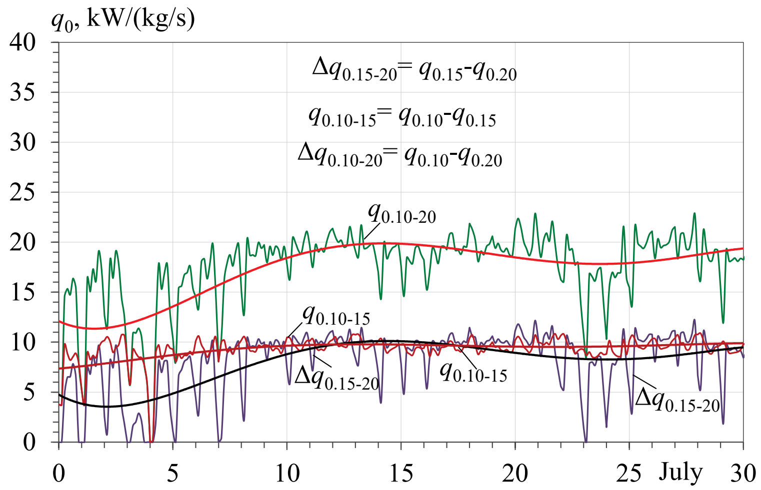


FIGURE 8. *Current values of basic comparatively stable specific refrigeration capacity $q_{0.10-15}$ for cooling air from 15°C to 10°C and $q_{0.10-20}$ for cooling air from 20°C to 10°C and decrease $\Delta q_{0.15-20}$ in design refrigeration capacity: $q_{0.10-15} = q_{0.10} - q_{0.15}$; $q_{0.10-20} = q_{0.10} - q_{0.20}$; $\Delta q_{0.15-20} = q_{0.15} - q_{0.20}$*

The same effect can be realized by decreasing the installed refrigeration capacities for ambient air precooling to t_{a2} = 15°C instead of t_{a2} = 20°C by the same values of $\Delta q_{0.15-20} = q_{0.15} - q_{0.20}$ with using the reduced design refrigeration capacity $q_{0.A10-20} = q_{0.10rat} - q_{0.10-20}$ instead of $q_{0.A10-15} = q_{0.10rat} - q_{0.10-15}$ (Fig. 9).

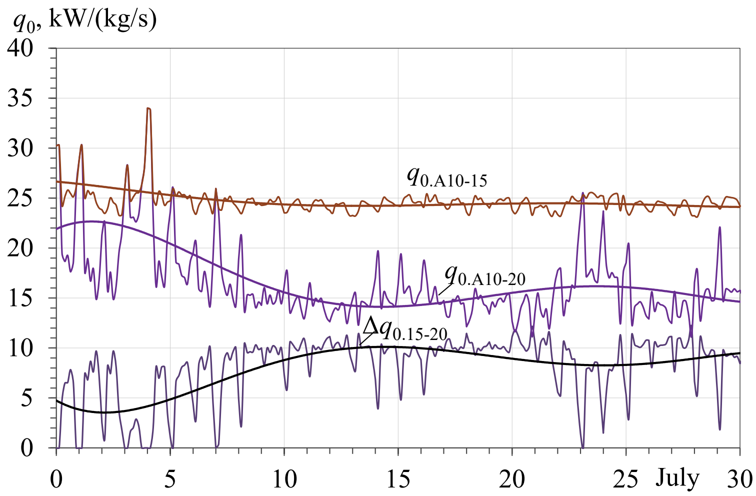


FIGURE 9. *Current values of available remained specific refrigeration capacity $q_{0.A10-20}$ for cooling ambient air 20°C and $q_{0A.10-15}$ for cooling ambient air from t_{amb} to t_{a2} = 15°C and decrease $\Delta q_{0.15-20}$ in design refrigeration capacity: $q_{0.10-15} = q_{0.10} - q_{0.15}$; $q_{0.10-20} = q_{0.10} - q_{0.20}$; $q_{0A.10-15} = q_{0.10rat} - q_{0.10-15}$; $q_{0.A10-20} = q_{0.10rat} - q_{0.10-20}$, $\Delta q_{0.15-20} = q_{0.15} - q_{0.20}$*

It is seen, that a decrease $\Delta q_{0.15-20} = q_{0.15} - q_{0.20}$ in design refrigeration capacity due to its rational distribution with cooling air from t_{a2} = 20°C to t_{a2} = 15°C through using excessive refrigeration capacity of the available remained design value $q_{0.A10-20} = q_{0.10rat} - q_{0.10-20}$ to cover current loads $q_{0.A10-15} = q_{0.10rat} - q_{0.10-15}$ for cooling ambient air to t_{a2} = 15°C instead of t_{a2} = 20°C is about $\Delta q_{0.15-20}$ = 10 kW/(kg/s), that is about 40% of decreased design value $q_{0.10rat} - \Delta q_{0.15-20} \approx$ 25 kW/(kg/s).

As Figure 2b shows, overfilling the air coils of the air cooler by liquid refrigerant allows the variation of refrigerant flows in response to change of current thermal loads on air coolers without considerable drop in heat flux enlarged by about 30% to 50%.

Thus the realization of a concept of incomplete refrigerant evaporation with overfilling air coils enables a larger deviation of refrigerant flows in ambient air coolers of AC system according to current heat load variation without noticeable decrease of heat flux.

Conclusions

A developed concept is intended to enhance the heat efficiency of air coolers at varying heat loads by over filling all the air coils that provides excluding the inner wall surface drying out at the final stage of refrigerant evaporation with low intensity of heat transfer that takes place while complete refrigerant evaporation in conventional air coolers of AC systems.

The overfilling of air coils allows enlarged variation of refrigerant flows in response to change of current thermal loads on air coolers without noticeable drop in heat flux.

The realization of concept of overfilling air coils permits enlarged variation of refrigerant flows in response to change of current thermal loads on air coolers without noticeable drop in heat flux and enables to cover current overloading the air coolers of AC system through increasing refrigerant flows and using the excessive refrigeration energy accumulated at lowered thermal loads.

The effect gained due to overfilling air coolers consists in reduction of installed (design) refrigeration capacity by about 15% to 20% due to covering the currant cooling capacities needed for cooling ambient air to 15°C by installed cooling capacity designed for cooling air to 20°C.

The main idea behind the principle of rational designing and operation of ambient AC systems to match current varying heat loads is sharing the overall heat load in unstable heat load range, corresponding to ambient air processing with considerable heat load fluctuations in response to actual climatic conditions, and a comparatively stable heat load part for subsequent air cooling to a target temperature.

The overfilling all the air coils by liquid refrigerant enables to match actual changeable heat loads not by varying compressor refrigerant capacity but through supplying the excessive refrigerant accumulated at lowered thermal loads on air coolers.

The air coolers based on the principle of overfilling can find a wide application in ambient air conditioning with great fluctuations of current heat loads to cover a deficit of installed (design) refrigeration capacity at arised current heat loads through using its excess at lowered loads and thereby reduce installed (design) refrigeration capacity by about 20%.

REFERENCES

[1] Rodriguez-Aumente P.A., Rodriguez-Hidalgo M.C., Nogueira J.I., Lecuona A., Venegas M.C., *District heating and cooling for business buildings in Madrid*, Applied Thermal Engineering, Vol. 50, 2013, pp. 1496-1503.

[2] Ortiga J., Bruno J.C., Coronas A., *Operational optimization of a complex trigeneration system connected to a district heating and cooling network*, Applied Thermal Engineering, Vol. 50, 2013, pp. 1536-1542.

[3] Trushliakov E., Radchenko A., Radchenko M., Kantor S., Zielikov O., *The Efficiency of refrigeration capacity regulation in the ambient air conditioning systems* [in:] Ivanov V. et al. (eds.), *Advances in Design*, Simulation and Manufacturing III (DSMIE 2020), LNME, Springer, Cham 2020, pp. 343-353.

[4] Freschi F., Giaccone L., Lazzeroni P., Repetto M., *Economic and environmental analysis of a trigeneration system for food-industry: A case study*, Appl. Energy, Vol. 107, 2013, pp. 157-172, doi:10.1016/j.apenergy.2013.02.037.

[5] Trushliakov E., Radchenko A., Forduy S., Zubarev A., Hrych A., *Increasing the Operation Efficiency of Air Conditioning System for Integrated Power Plant on the Base of Its Monitoring* [in:] Nechyporuk M., Pavlikov V., Kritskiy D. (eds.), ICTME (ICTM 2019), AISC, Springer, Vol. 1113, Cham 2020, pp. 351-360, https://doi.org/10.1007/978-3-030-37618-5_30.

[6] Radchenko A., Scurtu I-C., Radchenko M., Forduy S., Zubarev A., *Monitoring the efficiency of cooling air at the inlet of gas engine in integrated energy system*, Thermal Science, 2020, OnLine-First Issue 00, pp. 344-344, https://doi.org/10.2298/TSCI200711344R.

[7] Popli S., Rodgers P., Eveloy V., *Trigeneration scheme for energy efficiency enhancement in a natural gas processing plant through turbine exhaust gas waste heat utilization*, Appl. Energy, Vol. 93, 2012, pp. 624-636, doi:10.1016/j.apenergy.2011.11.038.

[8] Popli S., Rodgers P., Eveloy V., *Gas turbine efficiency enhancement using waste heat powered absorption chillers in the oil and gas industry*, Applied Thermal Engineering, Vol. 50, 2013, pp. 918-931, doi:10.1016/j.applthermaleng.2012.06.018.

[9] Radchenko A., Radchenko N., Tsoy A., Portnoi B., Kantor S., *Increasing the efficiency of gas turbine inlet air cooling in actual climatic conditions of Kazakhstan and Ukraine*, AIP Conference Proceedings, Vol. 2285, 2020, No. 030071, https://doi.org/10.1063/5.0026787.

[10] Radchenko M., Radchenko R., Tkachenko V., Kantor S., Smolyanoy E., *Increasing the Operation Efficiency of Railway Air Conditioning System on the Base of Its Simulation Along the Route Line* [in:] Nechyporuk M. et al. (eds.), ICTME (ICTM 2019), AISC, Springer, Vol. 1113, Cham 2020, pp. 461-467, https://doi.org/10.1007/978-3-030-37618-5_39.

[11] Radchenko N., Radchenko A., Tsoy A., Mikielewicz D., Kantor S., Tkachenko V., *Improving the efficiency of railway conditioners in actual climatic conditions of operation*, AIP Conference Proceedings, Vol. 2285, 2020, No. 030072, https://doi.org/10.1063/5.0026789.

[12] Radchenko M., Mikielewicz D., Tkachenko V., Klugmann M., Andreev A., *Enhancement of the operation efficiency of the transport air conditioning system* [in:] Ivanov V. et al. (eds.), ADSM III (DSMIE 2020), LNME, Springer, Cham 2020, pp. 332-342.

[13] Radchenko R., Pyrysunko M., Radchenko A., Andreev A., Kornienko V., *Ship engine intake air cooling by ejector chiller using recirculation gas heat* [in:] Tonkonogyi V. et al. (eds.), AMP, InterPartner-2020, LNME, Springer, Cham 2021, pp. 734-743.

[14] Radchenko R., Kornienko V., Pyrysunko M., Bogdanov M., Andreev A., *Enhancing the Efficiency of marine diesel engine by deep waste heat recovery on the base of its simulation along the route line* [in:] Nechyporuk M. et al. (eds.), ICTME, AISC, Springer, Vol. 1113, Cham 2020, pp. 337-350.

[15] Radchenko M., Radchenko R., Kornienko V., Pyrysunko M., *Semi-empirical correlations of pollution processes on the condensation surfaces of exhaust gas boilers with water-fuel emulsion Combustion* [in:] Ivanov V. et al. (eds.), DSMIE 2019, LNME, Springer, Cham 2020, pp. 853-862.

[16] Forduy S., Radchenko A., Kuczynski W., Zubarev A., Konovalov D., *Enhancing the fuel efficiency of gas engines in integrated energy system by chilling cyclic air* [in:] Tonkonogyi V. et al. (eds.), Grabchenko's ICAMP, InterPartner-2019, LNME, Springer, Cham 2020, pp. 500-509, https://doi.org/10.1007/978-3-030-40724-7_51.

[17] Radchenko A., Mikielewicz D., Forduy S., Radchenko M., Zubarev A., *Monitoring the fuel efficiency of gas engine in integrated energy system* [in:] Nechyporuk M. et al. (eds.), ICTM 2019, AISC (2020), Springer, Vol. 1113, Cham 2020, pp. 361-370, https://doi.org/10.1007/978-3-030-37618-5_31.

[18] Elberry M., Elsayed A., Teamah M., Abdel-Rahman A., Elsafty A., *Performance improvement of power plants using absorption cooling system*, Alex. Eng. J., Vol. 57, 2018, pp. 2679-2686, doi:10.1016/j.aej.2017.10.004.

[19] Suamir I., Tassou S., *Performance evaluation of integrated trigeneration and CO_2 refrigeration systems*, Appl. Therm. Eng., Vol. 50, 2013, pp. 1487-1495, doi:10.1016/j.applthermaleng.2011.11.055.

[20] Al-Ibrahim A.M., Varnham A., *A review of inlet air-cooling technologies for enhancing the performance of combustion turbines in Saudi Arabia*, Appl. Therm. Eng., Vol. 30, 2010, pp. 1879-1888, doi:10.1016/j.applthermaleng.2010.04.025.

[21] Trushliakov E., Radchenko M., Portnoi B., Tkachenko V., Hrych A., *Analysis of operation of ambient air conditioning systems with refrigeration machines of different types* [in:] Nechyporuk M., Pavlikov V., Kritskiy D. (eds.), ICTME – 2020, ICTM 2020, Lecture Notes in Networks and Systems, Springer, Vol. 188, Cham 2021, pp. 545-555.

[22] Radchenko A., Stachel A., Forduy S., Portnoi B., Rizun O., *Analysis of the Efficiency of Engine Inlet Air Chilling Unit with Cooling Towers* [in:] Ivanov V., et al. (eds.), *Advances in Design, Simulation and Manufacturing III (DSMIE 2020)*, Lecture Notes in Mechanical Engineering, Springer, Cham 2020, pp. 322-331.

[23] Radchenko M., Portnoi B., Kantor S., Forduy S., Konovalov D., *Rational thermal loading the engine inlet air chilling complex with cooling towers* [in:] Tonkonogyi V. et al. (eds.), *Advanced Manufacturing Processes II*, InterPartner 2020, Lecture Notes in Mechanical Engineering, Springer, Cham 2021, pp. 724-733.

[24] Kornienko V., Radchenko M., Radchenko R., Konovalov D., Andreev A., Pyrysunko M., *Improving the efficiency of heat recovery circuits of cogeneration plants with combustion of water-fuel emulsions*, Thermal Science, Vol. 25, Issue 1, Part B, 2021, pp. 791-800, doi: 10.2298/TSCI200116154K.

[25] Kornienko V., Radchenko R., Konovalov D., Andreev A., Pyrysunko M., *Characteristics of the rotary cup atomizer used as afterburning installation in exhaust gas boiler Flue* [in:] Ivanov V. et al. (eds.), ADSM III (DSMIE 2020), LNME, Springer, Cham 2020, pp. 302-311.

[26] Kornienko V., Radchenko R., Mikielewicz D., Pyrysunko M., Andreev A., *Improvement of characteristics of water-fuel rotary cup atomizer in a boiler* [in:] Tonkonogyi V. et al. (eds.), AMPII, InterPartner-2020, LNME, Springer, Cham 2021, pp. 664-674.

[27] Kornienko V., Radchenko R., Stachel A., Andreev A., Pyrysunko M., *Correlations for pollution on condensing surfaces of exhaust gas boilers with water-fuel emulsion combustion* [in:] Tonkonogyi V. et al. (eds.), AMP, InterPartner-2019, LNME, Springer, Cham 2020, pp. 530-539.

[28] Kornienko V., Radchenko R., Bohdal Ł., Kukiełka L., Legutko S., *Investigation of condensing heating surfaces with reduced corrosion of boilers with water-fuel emulsion combustion* [in:] Nechyporuk M. et al. (eds.), ICTM 2020, LNNS, Springer, Vol. 188, Cham 2021, pp. 300-309.

[29] Kornienko V., Radchenko M., Radchenko R., Bohdal Ł., Andreev A., *Thermal characteristics of the wet pollution layer on condensing heating surfaces of exhaust gas boilers* [in:] Ivanov V., et al. (eds.), ADSM IV (DSMIE 2021), LNME, Springer, Cham 2021, pp. 339-348.

[30] Konovalov D., Trushliakov E., Radchenko M., Kobalava G., Maksymov V., *Research of the aerothermopresor cooling system of charge air of a marine internal combustion engine under variable climatic conditions of operation* [in:] Tonkonogyi V. et al. (eds.), ICAMP, InterPartner-2019, LNME, Springer, Cham 2020, pp. 520-529.

[31] Konovalov D., Kobalava H., Radchenko M., Scurtu I.C., Radchenko R., *Determination of hydraulic resistance of the aerothermopressor for gas turbine cyclic air cooling* [in:] TE-RE-RD 2020, E3S Web of Conferences, Vol. 180, No. 0101231, 2020.

[32] Kobalava H., Konovalov D., Radchenko R., Forduy S., Maksymov V., *Numerical simulation of an aerothermopressor with incomplete evaporation for intercooling of the gas turbine engine* [in:] Nechyporuk M., Pavlikov V., Kritskiy D. (eds.), *Integrated Computer Technologies in Mechanical Engineering – 2020*, ICTM 2020, Lecture Notes in Networks and Systems, Springer, Vol. 188, Cham 2021, pp. 519-530.

[33] Butrymowicz D., Gagan J., Śmierciew K., Łukaszuk M., Dudar A., Pawluczuk A., Łapiński A., Kuryłowicz A., *Investigations of prototype ejection refrigeration system driven by low grade heat*, HTRSE-2018, E3S Web of Conferences 70, 2018, p. 7.

[34] Smierciew K., Gagan J., Butrymowicz D., Karwacki J., *Experimental investigations of solar driven ejector air-conditioning system*, Energy and Buildings, Vol. 80, 2014, pp. 260-267.

[35] Trushliakov E., Radchenko M., Bohdal T., Radchenko R., Kantor S., *An innovative air conditioning system for changeable heat loads* [in:] Tonkonogyi V. et al. (eds.), ICAMP, InterPartner-2019, LNME, Springer, Cham 2020, pp. 616-625.

[36] Radchenko N.I., *On reducing the size of liquid separators for injector circulation plate freezers*, International Journal of Refrigeration, Vol. 8, No. 5, 1985, pp. 267-269.

[37] Radchenko A., Andreev A., Konovalov D., Zhang Qiang Z., Zewei L., *Analysis of ship main engine intake air cooling by ejector and turbocompressor chillers on equatorial voyages* [in:] Nechyporuk M., Pavlikov V., Kritskiy D. (eds.), *Integrated Computer Technologies in Mechanical Engineering – 2020*, ICTM 2020, Lecture Notes in Networks and Systems, Springer, Vol. 188, Cham 2021, pp. 487-497.

[38] Radchenko M., Mikielewicz D., Andreev A., Vanyeyev S., Savenkov O., *Efficient ship engine cyclic air cooling by turboexpander chiller for tropical climatic conditions* [in:] Nechyporuk M., Pavlikov V., Kritskiy D. (eds.), *Integrated Computer Technologies in Mechanical Engineering – 2020*, ICTM 2020, Lecture Notes in Networks and Systems, Springer, Vol. 188, Cham 2021, pp. 498-507.

[39] Bohdal T., Kuczynski W., *Boiling of R404A refrigeration medium under the conditions of periodically generated disturbances*, Heat Transf. Eng., Vol. 32, 2011, pp. 359-368.

[40] Mikielewicz D., Klugmann M., Wajs J., *Flow boiling intensification in minichannels by means of mechanical flow turbulising inserts*, International Journal of Thermal Sciences, Vol. 65, 2013, pp. 79-91.

[41] Mikielewicz D., *Analytical model with nonadiabatic effects for pressure drop and heat transfer during boiling and condensation flows in conventional channels and minichannels*, Heat Transfer Engineering, Vol. 37, Issue 13-14, 2016, pp. 1158-1171.

[42] Kuczyński W., Charun H., *Experimental investigations into the impact of the void fraction on the condensation characteristics of R134a refrigerant in minichannels under conditions of periodic instability*, Arch. Thermodyn., Vol. 32, 2011, pp. 21-37, doi:10.2478/v10173.

[43] Bohdal T., Sikora M., Widomska K., Radchenko A.M., *Investigation of flow structures during HFE-7100 refrigerant condensation*, Arch. Thermodyn., Vol. 36, 2015, pp. 25-34, doi:10.1515/aoter-2015-0030.

[44] Kuczyski W., Charun H., Bohdal T., Kuczynski W., *Influence of hydrodynamic instability on the heat transfer coefficient during condensation of R134a and R404A refrigerants in pipe minichannels*, Int. J. Heat Mass Transf., Vol. 55, 2012, pp. 1083-1094, doi:10.1016/j.ijheatmasstransfer.2011.10.002.

[45] Dąbrowski P., Klugmann M., Mikielewicz D., *Selected studies of flow maldistribution in a minichannel plate heat exchanger*, Archives of Thermodynamics, Vol. 38, 2017, pp. 135-148.

[46] Kumar R., Singh G., Mikielewicz D., *A new approach for the mitigating of Flow Maldistribution in Parallel Microchannel Heat Sink*, Journal of Heat Transfer, Vol. 140, 2018, pp. 72401-72410.

[47] Kumar R., Singh G., Mikielewicz D., *Numerical study on mitigation of flow maldistribution in parallel microchannel heat sink: channels variable width versus variable height approach*, Journal of Electronic Packaging, Vol. 141, 2019, pp. 21009-21011.

[48] Dąbrowski P., Klugmann M., Mikielewicz D., *Channel blockage and flow maldistribution during unsteady flow in a model microchannel plate heat exchanger*, Journal of Applied Fluid Mechanics, Vol. 12, 2019, pp. 1023-1035.

[49] Radchenko A., Trushliakov E., Kosowski K., Mikielewicz D., Radchenko M., *Innovative turbine intake air cooling systems and their rational designing,* Energies, Vol. 13, Issue 23, 2020, No. 6201, doi:10.3390/en13236201.

[50] Radchenko R., Radchenko N., Tsoy A., Forduy S., Zybarev A., Kalinichenko I., *Utilizing the heat of gas module by an absorption lithium-bromide chiller with an ejector booster stage*, AIP Conference Proceedings, Vol. 2285, 2020, No. 030084, https://doi.org/10.1063/5.0026788.

[51] Radchenko N., Trushliakov E., Radchenko A., Tsoy A., Shchesiuk O., *Methods to determine a design cooling capacity of ambient air conditioning systems in climatic conditions of Ukraine and Kazakhstan*, AIP Conference Proceedings, Vol. 2285, 2020, No. 030074, https://doi.org/10.1063/5.0026790.

[52] Zhu Y., Jin X., Du Z., Fang X., Fan B*., Control and energy simulation of variable refrigerant flow air conditioning system combined with outdoor air processing unit*, Appl. Therm. Eng., Vol. 64, 2014, pp. 385-395.

[53] Zhou Y.P., Wu J.Y., Wang R.Z., Shiochi S., *Energy simulation in the variable refrigerant flow air-conditioning system under cooling conditions*, Energy Build., Vol. 39, 2007, pp. 212-220.