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ENHANCING HEAT EFFICIENCY OF AIR COOLERS OF AIR CONDITIONING SYSTEMS BY INJECTOR REFRIGERANT CIRCULATION

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Abstract*: One of the most attractive reserves of enhancing the efficiency of air conditioning (AC) systems and their application in various fields consists in efficient operation of air coolers (refrigerant evaporators). A retrofit concept of efficient operation of refrigerant evaporators with incomplete refrigerant evaporation due to liquid refrigerant recirculation by injector (jet pump) has found a new impulse for further applications in outdoor air processing units (OAPU) to match varying outdoor heat loads according to actual climatic conditions and for indoor air coils to match varying indoor heat loads in ductless Variable Refrigerant Flow (VRF) AC systems. A proposed concept of enhancing heat efficiency of heat exchangers with boiling refrigerants inside channels is intended to solve the problem of uneven refrigerant distribution in inlet manifolds (headers) for microchannel heat exchangers or between refrigerant coils and of uneven outside air heat loads on refrigerant coils by over filling them through liquid refrigerant injector recirculation that provides excluding the final dry-out stage of refrigerant evaporation with low intensity of heat transfer. Thus, due to excluding the internal problem of refrigerant uneven distribution and low intensity of heat transfer of refrigerant evaporation in channels the general problem of enhancing the efficiency of heat exchangers with boiling refrigerants inside channels is transformed into the external problem of heat transfer on air side.*
Keywords: *air conditioning system, heat exchanger, phase change, refrigerant circulation, maldistribution, uneven heat load*

1. INTRODUCTION

The efficiency of air conditioning (AC) systems and their refrigeration machine performance depends on the operation of their air coolers (refrigerant evaporators) intended to match various heat loads. The intensity of heat transfer of refrigerant, evaporated inside channels, drops at the final stage of evaporation, that is caused by drying out the inner channel wall surface while transition of refrigerant two-phase flow from annular to disperse (mist) flow. In compact air coolers with finned tubes a sharp decrease in heat transfer coefficient to refrigerant at the final stage of its evaporation 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 [28].

The injector recirculation of unevaporated liquid refrigerant in air coolers is conducted by using a potential energy of high pressure liquid refrigerant, leaving a condenser, which is conventionally lost while its throttling to evaporation pressure in expansion valve.

A new impulse for further realization of this concept is forced due to applying a circulation of liquid refrigerant, that causes over filling all the coils of air cooler, i. e. due to incomplete refrigerant evaporation, that excludes a drop in intensity of evaporation heat transfer and, as result, any influence of inside tube heat transfer on the intensity of the overall heat transfer of air cooler.

Therefore there is no need to apply a traditional refrigerant distribution by throttling the refrigerant stream in a distributor at the inlet of air cooler for equalizing refrigerant distribution among coils due to much more higher pressure drop in a distributor head compared with its value for separate refrigerant coils, so far as over filling all the coils results in more higher intensity of refrigerant heat transfer compared with its air side value, that excludes any influence of inside tube heat transfer as more intensive.

So, there is no need for improving refrigerant distribution among coils by applying a refrigerant distributor as inlet manifold of air cooler as well as by applying inlet manifold with special distributing devices (different channel inlet holes, differ manifold cross sections and others).

The principle of over filling all the channels of air cooler by liquid refrigerant injector recirculation, that provides excluding the final stage of refrigerant evaporation with low intensity of heat transfer and, as result, excluding any influence of inside channel heat transfer, as more intensive, instead of refrigerant distribution in the inlet manifolds, is very useful for widespread microchannel plate evaporators with uneven refrigerant distribution in inlet manifolds as well as for traditional heat exchanges with uneven air velocity distribution.

A concept of efficient operation of air coolers with incomplete refrigerant evaporation due to liquid refrigerant recirculation by injector (jet pump) has found a new impulse for further applications in central Heating, Ventilation and Air Conditioning (HVAC) systems and their combined versions with indoor air conditioning system, as well as for outdoor air processing units (OAPU) to match varying outdoor heat loads according to actual climatic conditions and for indoor air coils to match varying indoor environments in ductless Variable Refrigerant Flow (VRF) systems.

The research is devoted to develop a general concept of incomplete refrigerant phase change through injector refrigerant recirculation, based on a retrofit concept of incomplete refrigerant evaporation, to spread it over new applications for outdoor air processing units to match varying heat loads according to actual climatic conditions and for indoor air coils to match varying indoor heat loads in ductless VRF AC systems, for widespread microchannel plates evaporators with uneven refrigerant distribution in inlet manifolds and others .

2. Literature Review

A lot of publications are devoted to enhancing the efficiency of AC systems by intensification of heat transfer processes in heat exchangers (evaporators and condensers) [4, 5, 9, 11, 28], through application of advanced refrigerant circulation scheme decisions [6, 8, 9, 11, 23-25, 29, 32, 33] and enhancing air procession by two-stage cooling developed for AC systems in comfort and energetic application [26, 27, 30, 31, 38].

Numerous researchers have studied the energy efficiency of the VRF system [10-14, 17, 41-43]. The VRF systems operate with high part-load efficiency, that results in high daily and seasonal energy efficiency, so as AC systems typically spend most of their operating hours in the range of 40% to 80% of maximum cooling capacity [12].

A combination of HVAC system with roof top unit (RTU) used as the outdoor air processing (OAP) system in the VRF system and control strategies to enhance its energy performance were proposed [41, 43]. A comparisen of VRF system with conventional AC systems showed that the VRF system was more energy efficient than variable air volume (VAV) system by 22% and than the fan-coil plus fresh air (FPFA) system by 11.7% [41]. It was obtained that the multi-split VRF system saved more than 20% energy compared to a variable air volume (VAV) system [42].

The HVAC system that processed outdoor air loads by supplying refrigerant from the outdoor unit performed simultaneously as an outdoor unit in the VRF system in contrast with the OAP, which had compressors, was investigated [13]. In the VRF-OAP system, the multiple indoor units and the OAP were simultaneously connected to a single outdoor unit.

A higher energy reduction compared with conventional operation without refrigerant flow regulation is due to superlative applying the variable speed compressor in part load modes, especially when the outdoor air temperature was closer to its indoor set point.

The authors [14] on the base of field test results revealed that the design outdoor unit cooling capacity should be of around 70% excess to prevent a lack of indoor units cooling capacities.

It was also shown that negative impacts on the indoor comfort of the outdoor air not completely processed and introduced to the indoor environment were much greater than that of the indoor units processing the thermal loads of the indoor air [13]. Therefore, the refrigerant flow control in the VRF-OAP system has been designed to provide more flows to the OAP than to the indoor units and most of the refrigerant flows inside the system were introduced to the OAP.

3. Research Methodology

Proceeding from the approach that heat loads on refrigerant coils depend only on air side heat transfer as lower compared with refrigerant side, the problem of enhancing heat transfer in air coolers is solved through incomplete refrigerant phase change due to liquid refrigerant recirculation by injector to exclude the final stage of refrigerant evaporation with low intensity of heat transfer.

A proposed general concept of enhancing heat efficiency of heat exchangers with boiling refrigerants inside channels is intended to solve the problem of uneven refrigerant distribution in inlet manifolds for microchannel heat exchangers or between refrigerant coils (tubes, microchannel plates) and of uneven outside air heat loads on refrigerant coils by over filling all channels through liquid refrigerant jet pump recirculation that provides excluding the final stage of refrigerant evaporation with low intensity of heat transfer.

Thus, due to excluding the inner problem of refrigerant uneven distribution or low intensity of heat transfer of refrigerant evaporation in channels the general problem of enhancing heat efficiency of heat exchangers with boiling or condensing refrigerants inside channels is transformed into the outside problem of heat transfer on the air side to be solved as it is required and possible.

Applying the forced circulation of liquid refrigerant with over filling all air coils, operating with incomplete refrigerant evaporation, excludes a drop in intensity of evaporation heat transfer, caused by inner channel wall surface drying out while complete refrigerant evaporation, and its influence on intensity of the overall heat transfer.

Therefore there is no need to solve the problem of refrigerant distribution between air coils (channels), so as the problem has to be solved by overfilling all the air coils, that leads to heat loads on the air coils as depending only on limiting influence of less air side heat transfer intensity.

4. Results

4.1. Some fundamental approaches of intensification of heat transfer of refrigerant phase changes inside channels of air coolers and condensers

The accent is made on air coolers for which overall heat transfer is usually considered to be dependent only on intensity of external heat transfer on the air side and refrigerant phase change does not influence thermal efficiency of heat exchanger.

But 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 (Fig. 1).

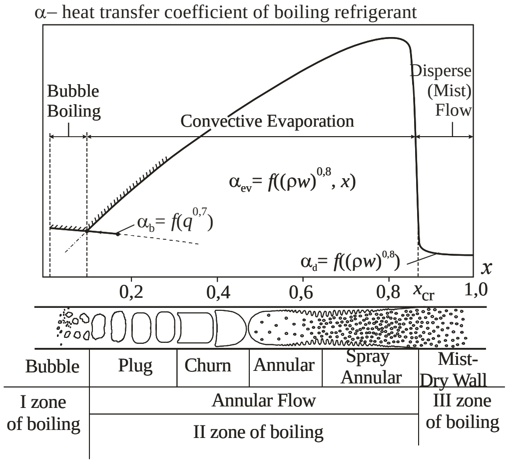


Fig. 1. Typical structures of inside tube refrigerant evaporation and behaviour of coefficients of heat transfer to refrigerant in bubble boiling zone α_b , convective evaporation zone α_{ev} and at the final stage of evaporation in disperse (mist) flow α_d with the vapour mass fraction *x*

This occurs due to inner channel wall surface drying out with transition of refrigerant two-phase flow from annular flow to disperse (mist) flow.

In compact air coolers with finned tubes the coefficient of heat transfer to refrigerant at the final stage of its evaporation is much lower than to air. This results in decrease in overall heat transfer coefficient (Fig. 2).

Typical structures of inside tube refrigerant evaporation and behaviour of coefficients of heat transfer to refrigerant in bubble boiling zone α_b , convective evaporation zone α_{ev} and at the final stage of evaporation in disperse (mist) flow α_d with the vapour mass fraction *x* are presented in Fig. 1.

A thresh value of vapour quality *x* between bubble (nucleate) boiling zone and convective evaporation (thresh bubble/convective vapour quality *x*) corresponds to equality of heat transfer coefficients calculated for both zones at this *x* [1, 7, 28]. The thresh bubble/convective vapour quality *x* is shifting in response to relation between heat flux *q* and mass flux (mass velocity) ρ*w*. As Fig. 1 shows, with rising mass flux ρ*w* the thresh point is shifting towards a decreased value of vapour quality *x* (to the inlet of channel) according to dominating a convective evaporation over nucleate boiling and it is inverted with increasing heat flux *q*.

In compact air coolers with finned tubes the coefficient of heat transfer to refrigerant at the final stage of its evaporation is much lower than that to air, related to the internal surface of tubes. This results in decrease in overall heat transfer coefficient (Fig. 2). The correlation between ΔL and Δx is calculated according to the heat balance for refrigerant boiling inside air coil. Calculations have been performed for the air cooler with plate finned tubes of 12 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 *R*142b.

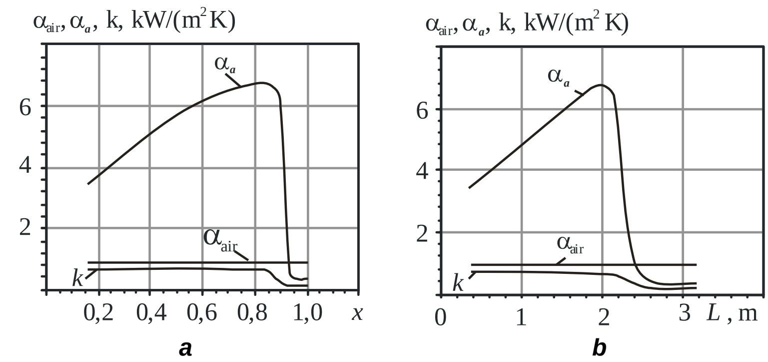


Fig. 2. Variation of heat transfer coefficients to refrigerant α_a and air α_{air}, overall heat transfer coefficient k with the vapour mass fraction x (***a***) and with the coil length L

As Fig. 1 and 2 show a sharp decrease in the heat flux q occurs at burnout vapour fraction x_{cr} ≈ 0,9 corresponding to drying the channel wall surface with the transition from annular to disperse flow. The reduction in q is caused by subsequent 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.

Taking into account that in the conventional air cooler with thermoexpansion valve the vapour at the exit of the air cooler should be superheated by 5…10 ºC a share of the surface, corresponding to the final stage of boiling and vapour superheating with extremely low intensity of heat transfer, is about 30 %.

It should be noted that a sharp decrease in the heat transfer coefficient to refrigerant α_a with the transition from annular to disperse flow takes place for most of refrigerants.

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 evaporated on the surface of regenerative heat exchanger (RHE) coil, mounted in the liquid separator, while subcooling the liquid refrigerant of a high pressure after the condenser, directing it again at the entrance of air cooler by injector (Fig. 3).

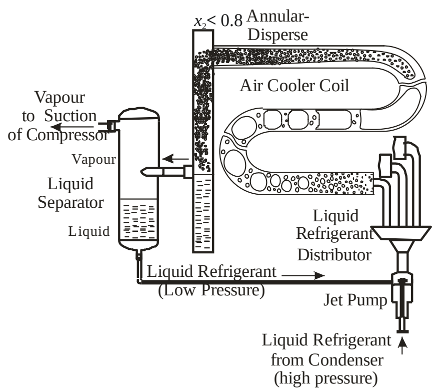


Fig. 3. Air coil contour with liquid refrigerant recirculation by injector

The injector uses the potential energy of refrigerant pressure drop from condensing to evaporation pressure, which is conventionally lost while throttling high pressure liquid refrigerant in thermo-expansion valve.

The thermal efficiency of the air coolers circuits is usually carried out at maximum heat flux q_{max}. For the refrigerant and air counter flow the correlation for heat flux relating to the internal tube surface area is expressed as

$$q = k\,\theta = \frac{1}{\dfrac{1}{\alpha_a} + \dfrac{1}{\alpha_{air}}} \cdot \frac{t_{air1} - t_{air2} + \Delta t_0}{\ln \dfrac{t_{air1} - t_{02}}{t_{air2} - (t_{02} + \Delta t_0)}},$$

where the refrigerant side heat transfer coefficient α_a may be calculated by equations [1, 7] and air side heat transfer coefficient α_{air} corresponds to the internal tube surface area, θ − logarithmic temperature difference; k − overall heat transfer coefficient.

The drop in the boiling temperature Δt_0 caused by the pressure drop for two phase flow ΔP is obtained from Clausius−Clapeyron relationship

$$\frac{dP}{dt_0} = \frac{r}{T(\upsilon_v - \upsilon_l)} \approx \frac{\Delta P}{\Delta t_0}.$$

The frictional pressure gradient for two phase flow dP/dL may be calculated according to the Lockhart-Martinelli-Nelson method [21, 22].

The existence of maximum heat flux q_{max} is caused by the following. With increasing mass velocity of refrigerant ρw the heat transfer coefficient to refrigerant α_a and overall heat transfer coefficient k increases. But the refrigerant pressure drop ΔP and corresponding refrigerant boiling temperature drop Δt_0 increases also. In conventional practice of optimum evaporator-air cooler designing the value of refrigerant boiling temperature t_{02} at the evaporator exit (compressor inlet) is fixed to keep the other points of refrigerant cycle invariable [28]. With fixed t_{02} the increase in Δt_0 causes the increase in refrigerant boiling temperature t_{01} at the evaporator inlet and decrease in logarithmic temperature difference θ between air to be cooled and boiling refrigerant as a result. Such opposite influence of the refrigerant mass velocity ρw upon k and θ causes the existence of maximum of function $q = k\theta$ at quite definite value of ρw. This value is considered as optimum $(\rho w)_{opt}$.

The results of thermal efficiency comparison of conventional air cooler with complete evaporation and superheated vapour at the exit and of advanced air cooler with incomplete evaporation due to liquid refrigerant recirculation by injector are shown in Fig. 4. The conditions at the air cooler outlet are the following: refrigerant R142b, refrigerant boiling temperature at the evaporator exit t_{02} = 0 °C,

There is a dry inner tube wall with a vapour superheated in 10 °C for the conventional throttle circuit and wetted wall with $x_2 < x_{cr}$ for the injector recirculation circuit. In disperse mixture the vapour is superheated in 5 °C as compared to the boiling temperature t_{02} .

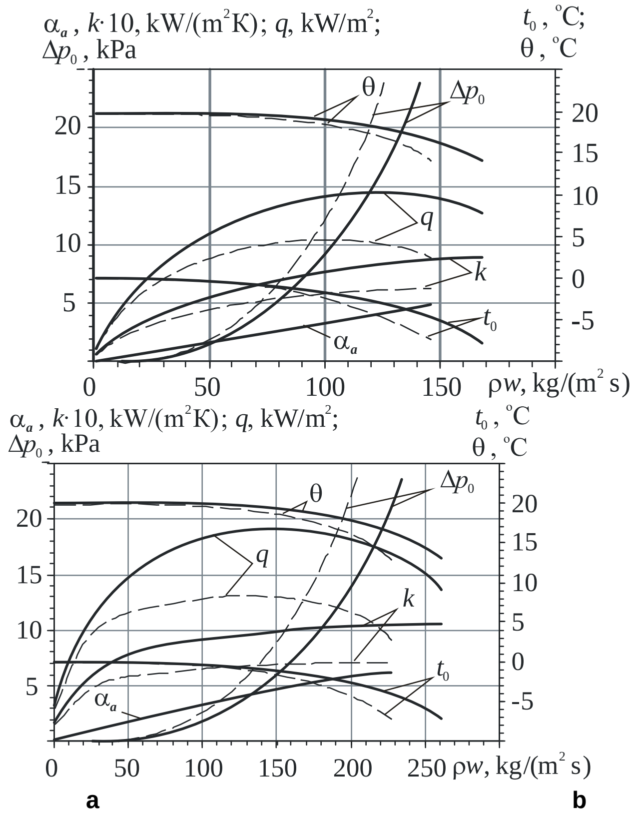


Fig. 4. Mean values of heat fluxes q, heat transfer coefficients to refrigerant α_a and overall heat transfer coefficients k , logarithmic temperature difference θ, refrigerant boiling temperature t_0 and pressure drop ΔP against refrigerant mass velocities ρw for different air velocities w and coefficients of ejection U: R142b, t_{02} = 0 °C; —— – w = 6 m/s; – – – w = 2 m/s;
a – U = 0 (conventional complete evaporation); b – U = 1 (incomplete evaporation)

As fig. 4 shows, with increasing the air velocities from w = 2 m/s to 6 m/s the influence of liquid refrigerant recirculation enlarges, that means its great efficiency for compact air coolers with high air velocities.

So, the recirculation of liquid refrigerant in the air cooler by injector provides an increase in heat flux *q* by 25…40 % compared with conventional complete refrigerant evaporation with superheated vapour at the exit.

As one can see from fig. 5, overfilling the air coils of the air cooler by recirculation of liquid refrigerant allows a larger deviation of refrigerant mass velocities ρ*w* from their optimum value, providing maximum value of heat flux *q*, that means that a larger refrigerant redistribution among air coils is permited and good perspectives of injector liquid refrigerant recirculation in microchannel evaporators with considerable refrigerant maldistribution in inlet and outlet manifolds.

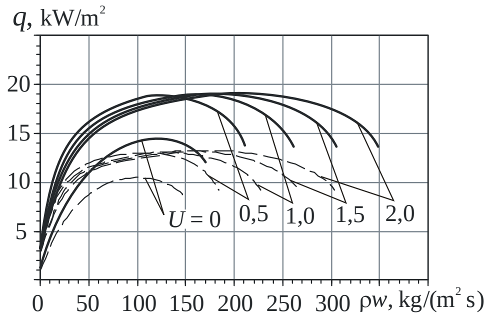
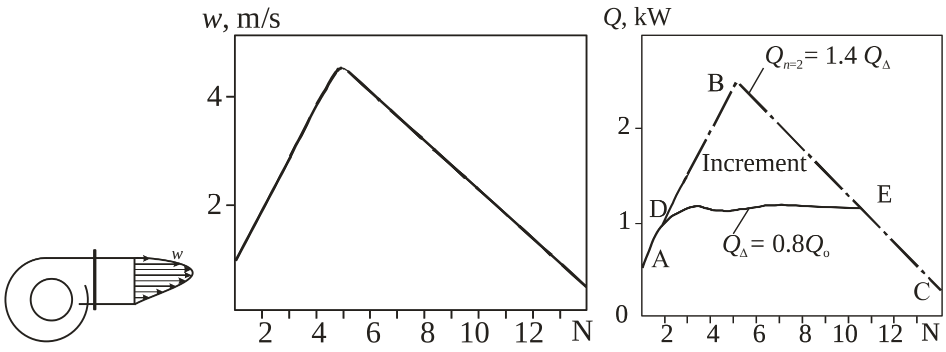


Fig. 5. Mean values of heat fluxes *q*, heat transfer coefficients to refrigerant α_a and overall heat transfer coefficients *k* , logarithmic temperature difference θ, refrigerant boiling temperature t_0 and pressure drop ΔP against refrigerant mass velocities ρ*w* : R142b, t_{02} = 0 °C; ——— –w = 6 m/s; – – – –w = 2 m/s; U = 0 (conventionsl complete evaporation)

Overfilling the coils of air cooler due to liquid refrigerant recirculation of refrigerant reveals the reserves for subsequent decreasing the influence of uneven air velosity distribution on the heat efficiency of air cooler.

As an example, the air velocity *w*-profile at the inlet of air cooler and heat load *Q*-distributions among the air coils for conventional air cooler with complete refrigerant evaporation and advanced air cooler with refrigerant overfilling are presented in Fig. 6: ADEC – heat load *Q* distribution among air coils for conventional air cooler; ABC – heat load *Q* distribution for advanced air cooler with refrigerant overfilling; DBE – heat load *Q* increment due to air coils overfilling compared with heat load on conventional air cooler; Q_0 – overall heat load on air cooler with equal air velocity distribution; Q_Δ – overall heat load on conventional air cooler with angular Δ -air velocity profile; $Q_{n=2}$ – overall heat load on advanced air cooler with number of refrigerant circulation n = 2 and Δ-velocity *w* distribution.



a ***b***

Fig. 6. Air velocity *w*-profile at the inlet of air cooler (a), heat load *Q*-distributions among air coils (b): N – a number of air coil; ADEC – heat load *Q* distribution among air coils for conventional air cooler; ABC – heat load *Q* distribution for advanced air cooler with refrigerant overfilling; DBE – heat load *Q* increment; Q_0 – overall heat load on air cooler for equal air velocity distribution; Q_Δ – angular Δ -air velocity profile (a); $Q_{n=2}$ –refrigerant circulation with number of circulation n = 2 and Δ-velocity *w* distribution

As Fig. 6,b shows, in spite of increased air velocities at the inlet of coils number N =3-11 (according to angular Δ -air velocity profile) the heat loads on them remain practically constant in conventional air cooler, that is caused by high superheating of refrigerant vapour in them and considerably decreased heat transfer intensity to superheated vapour inside coils (lowered heat transfer coefficient to refrigerant α_a) and overall heat transfer intensity as result (Fig. 2), that limits the heat loads on these coils.

The incomplete evaporation due to refrigerant coils overfilling by liquid refrigerant recirculation leads to the full elimination of its inefficient final stage and to rising the intensity of overall heat transfer on the whole air cooler surface in response to increasing the air velosity at the inlet of the coils number N =3-11 (Fig. 6,a). As a result, the heat load *Q* distribution ABC among the air coils of the advanced air cooler with refrigerant overfilling follows strictly the angular Δ -air velocity profile with corresponding increment in heat load *Q* by about 40% compared with conventional air cooler (Fig. 6,b).

The principle of over filling all the channels by liquid refrigerant injector recirculation, providing excluding any influence of inside channel heat transfer, as more intensive is very useful for compact air coolers with finned coils where the coefficients of inside and outside heat transfer (to boiling refrigerant and air respectively) are closed. So, it is possible to increase their heat efficiency in response to rising the air velosity at the inlet by liquid refrigerant circulation with adequate number of circulation n = U+1 = 1/x_2 (Fig. 7). A complete evaporation of refrigerant in the conventional air cooler is characterized by n = 1/x_2 = 1, where x_2 – mass vapour fraction an the outlet.

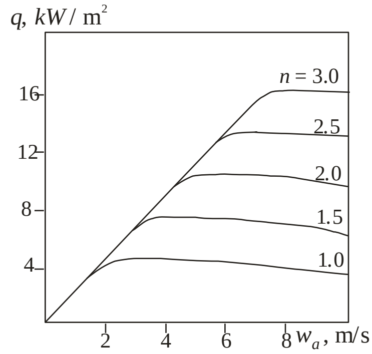


Fig. 7. The values of heat fluxes *q* in air coolers for various numbers of liquid refrigerant circulation *n* against inlet air velocities w_a: n = 1 – complete evaporation of refrigerant

The effect of intensification of heat transfer in evaporator due to elimination of dry wall regime can be evaluated by refrigeration coefficient (coefficient of performance of refrigeration system) ε. For the evaporator operated with wetted coil inner surface the increase in ε is achieved due to decrease in temperature differences between air and boiling refrigerant and a corresponding increase in the compressor suction pressure that provides an increment of ε by 15…20% [28].

4.2. ENHANCING A PERFORMANCE EFFICIENCY OF REFRIGERANT AIR COOLERS OF DUCTLESS VARIABLE REFRIGERANT FLOW (VRF) AIR CONDITIONING SYSTEM AT CHANGEABLE HEAT LOADS BY INJECTOR REFRIGERANT CIRCULATION

The central Heating, Ventilation, and Air Conditioning (HVAC) systems and their combined versions with indoor air conditioning system typically operate in the range of 40% to 80% of design capacity [12].

So, the approach to enhance the performance efficiency of AC system has to provide ambient air procession to match actual changeable cooling loads in response to current climatic conditions to shorten running of refrigeration compressor in partial modes.

Any pump circulation system, for example with injector as jet pump, operates at changeable cooling capacities according to current heat loads on evaporators-air coolers with changing the refrigerant volumes in liquid separators after evaporators-air coolers and in linear receiver after condenser: with its rising in liquid separators and its lowering in linear receiver at decreasing heat loads on evaporators-air coolers and vice versa.

A retrofit concept of efficient operation of refrigerant evaporators (air coolers) with incomplete refrigerant evaporation due to liquid refrigerant recirculation by injector (jet pump) has found a new impulse for further applications in air coolers of outdoor air processing (OAP) units to match varying outdoor heat loads according to actual climatic conditions and for indoor air coils to match varying indoor heat loads in ductless Variable Refrigerant Flow (VRF) AC systems.

A higher energy reduction compared with conventional operation without refrigerant flow regulation is due to superlative applying the variable speed compressor in part load modes, especially when the outdoor air temperature was closer to the indoor temperature set point.

The authors [14] on the base of field test results revealed that the design outdoor unit cooling capacity should be of around 70% excess to prevent a lack of indoor units cooling capacities.

It was shown that negative impacts on the indoor comfort of the outdoor air not completely processed and introduced to the indoor environment were much greater than that of the indoor units processing the thermal loads of the indoor air [13]. Therefore the refrigerant flow control in the VRF-OAP system has been designed to provide more flows to the OAP than to the indoor units and most of the refrigerant flows inside the system were introduced to the OAP.

Issuing from the priority of the outdoor air procession [13, 14], the application of liquid refrigerant recirculation by injector, leading to overfilling the air cooler, would be quite reasonable for Outdoor Air Processing (OAP) unit to provide and complete outdoor air procession to avoid introducing of not completely processed outdoor air to the indoor environment with corresponding negative impacts on the indoor comfort (Fig.8, 9).

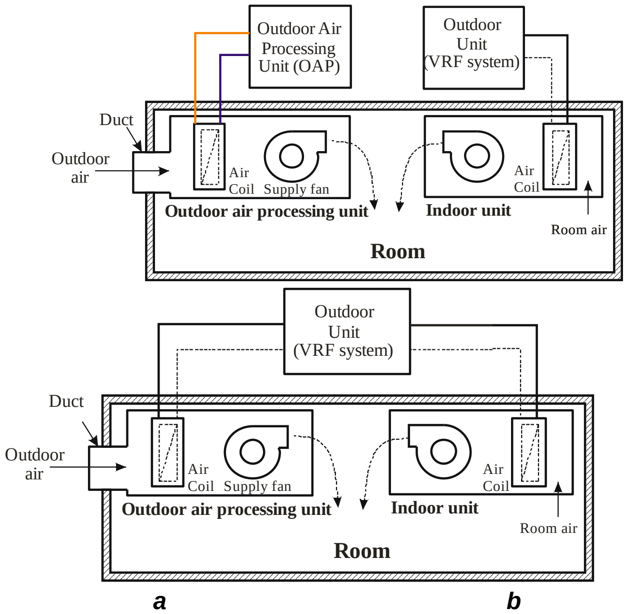


Fig. 8. The traditional schemes of ductless Variable Refrigerant Flow (VRF) systems with Outdoor Air Processing (OAP) and Indoor Air Processing (IAP) units separated (***a***) and combined (***b***)

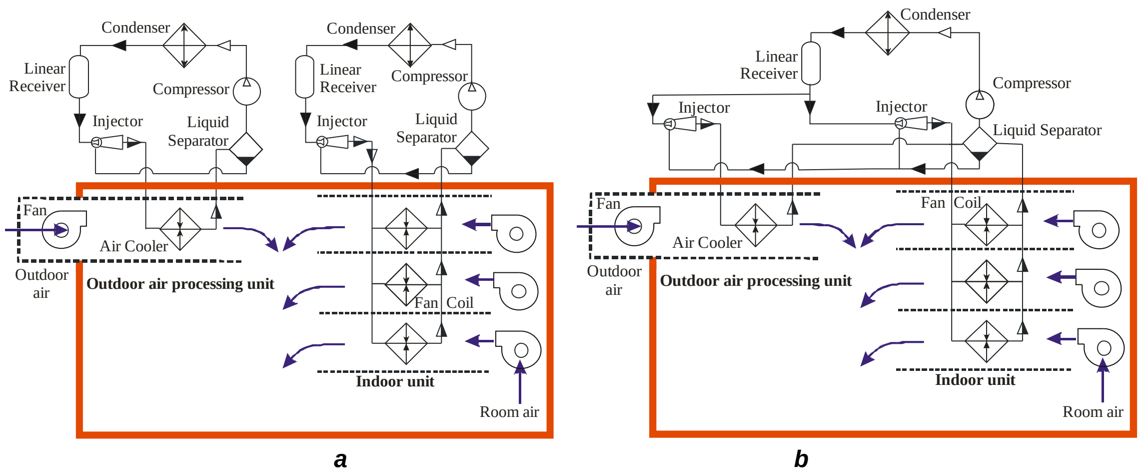


Fig. 9. The developed schemes of ductless Variable Refrigerant Flow (VRF) systems with Outdoor Air Processing (OAP) and Indoor Air Processing (IAP) units separated (***a***) and joint (***b***)

4.3. ENHANCING HEAT EFFICIENCY OF MICROCHANNEL EVAPORATORS BY INJECTOR REFRIGERANT CIRCULATION

In spite of peculiarities in physical mechanism of refrigerant evaporation in high efficient microchannel evaporators, the sharp drop in intensity of boiling refrigerant heat transfer takes place at the final dry-out stage of evaporation with drying the wall of channels too [1, 7, 15, 35, 36].

In general case the refrigerant evaporation in micro- and conventional channels are influenced by the same mechanism of changes in impulse and heat: heat transfer in bubble flow influenced generally by heat flux q, whereas convective evaporation (slug, slug-semi annular, semi annular, annular flows) is in strong dependency of heat transfer on mass flux (mass velocity) ρw and vapour quality x and in dry-out zone of superheating vapour at the final stage of evaporation – in dependency of heat transfer only on mass flux ρw [1, 7, 15].

According to the investigation of refrigerant evaporation in microchannels [15] the transient variation of annular sub structures within convective evaporation was observed at vapour quality x = 0.15−0.5 and a dry-out of liquid film on the wall was observed at x above 0.5, that means that negative influence of the final dry-out on heat transfer in microchannel evaporators is considerably higher compared with conventional air coolers.

As Fig. 1 shows, a thresh value of vapour quality x between bubble (nucleate) boiling zone and convective evaporation is shifting in response to relation between heat flux q and mass flux ρw towards a decreased value of vapour quality x (to the inlet of channel) according to dominating a convective evaporation over nucleate boiling and in inversed direction with increasing heat flux q. With account of such behavior of thresh point of bubble/convective vapour quality x the differ influence of heat flux q, mass flux ρw and vapour quality x on the heat transfer intensity should be analyzed.

Thus, it is quite reasonable to suppose, that an increase in refrigerant flow by recirculation leads to rising the refrigerant boiling heat transfer intensity due to excluding dry-out inefficient zone that lowers the overall heat transfer, and to enhancing the efficiency of microchannel evaporators from the point of heat transfer.

Majority of the studies point out the lack of influence of the refrigerant mass flux ρw and vapour quality x on the heat transfer coefficients [2, 3, 16, 18, 37, 40], i.e. the small contribution of the convective evaporation. Those conclusions should be corrected taking into account the behavior of thresh point of bubble/convective vapour quality x to consider differ influence of heat flux q and mass flux ρw in each zone.

The other advantage of applying the refrigerant recirculation consists in suppressing the bubble generation, initiated by heat flux q and leading to instabilities of two-phase flow. The instabilities of two-phase flow in microchannels are caused by their too small diameters, limiting the fast growing bubbles at increased heat flux [15, 35, 36], that can lead to instantaneous dry-out of the wall with sharp drop in heat transfer intensity and even to reversing the flow towards the inlet with blocking the refrigerant feed to the channels.

An example of variation in mass flux and heat flux leading to unstable boiling with reversing the refrigerant flow in the channel with diameter d_h = 540 μm is presented in Fig. 10 [15].

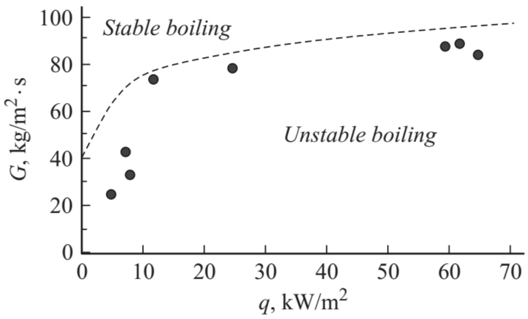


Fig. 10. The variation in mass flux and heat flux in stable and unstable boiling of refrigerant R134a in channel of d_h = 540 μm

As Fig. 10 shows, to avoid the instabilities of two-phase flow with its reversing and sharp drop in heat transfer intensity, caused by instantaneous dry-out of the wall, increasing of heat flux is to be followed by rising the mass flux. Issuing this it is reasonable to increase the mass flux through liquid refrigerant recirculation.

The principle of over filling all the channels by liquid refrigerant injector recirculation, providing excluding the final dry-out stage of refrigerant evaporation with low intensity of heat transfer is very useful for microchannel evaporators, characterized by uneven refrigerant distribution in inlet manifolds, instead of refrigerant redistribution in inlet manifolds with the use of special devises [19, 20, 34, 39].

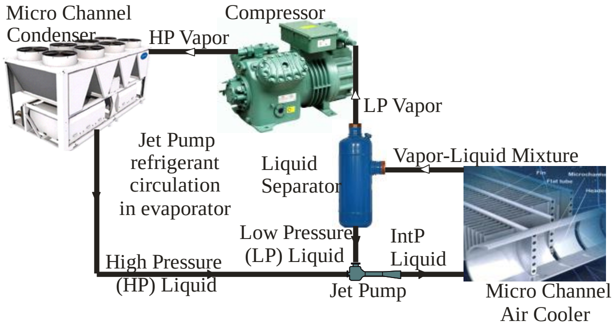


Fig. 11. A developed circuit of vapour compression chiller with liquid refrigerant recirculation in microchannel evaporator by injector

Overfilling the evaporators (Fig. 11), that leads to increasing the intensity of boiling heat transfer on all length of their channels due to incomplete refrigerant evaporation, excludes any influence of refrigerant boiling heat transfer coefficient, as more high value compared with air side heat transfer coefficient, on intensity of the overall heat transfer as result.

So, instead of solving the problem of refrigerant distribution between microchannel plates, the problem should be solved by overfilling all the microchannels, that leads to dependence of microchannel plate heat loads only on limiting air side heat transfer intensity.

5. Conclusions

A developed concept of enhancing heat efficiency of evaporators with boiling refrigerants inside channels is intended to solve the problem of uneven refrigerant distribution in inlet manifolds of microchannel heat exhanges as well as the problem of uneven outside heat loads on refrigerant coils by over filling all the channels (coils) through liquid refrigerant injector recirculation that provides excluding the final dry-out stage of refrigerant evaporation with drying the inside wall, that leads to sharp drop in evaporation heat transfer and lowering the intensity of overall heat transfer.

Therefore there is no need to solve the problem of equalizing the refrigerant distribution between air microchannels plates (coils) by applying special devices in inlet manifolds (headers) of microchannel evaporators, as well as equalizing air velocity profiles so as overfilling all the channels (coils) provides maximum heat fluxes at any uneven refrigerant distribution and air velocity profile.

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Підвищення теплової ефективності повітроохолоджувачів систем кондиціонування інжектором

Анотація: *Один з найпривабливіших резервів підвищення ефективності систем кондиціонування та їх застосування в різних областях полягає в ефективній роботі повітряних охолоджувачів (випарників холодоагенту). Концепція доопрацювання ефективної роботи випарників холодоагенту з неповним випаровуванням холодоагенту за рахунок рециркуляції рідкого холодоагенту інжектором (реактивним насосом) знайшла новий імпульс для подальших застосувань у зовнішніх повітряних переробних установках, щоб відповідати різним нагріванням зовнішнього тепла відповідно до фактичних кліматичних умов Умови в приміщенні відповідали різним тепловим навантаженням у приміщеннях в системах кондиціювання без змінного холодильного потоку. Запропонована концепція підвищення теплової ефективності теплообмінників з киплячими холодоагентами всередині каналів розроблена для вирішення проблеми нерівномірного розподілу холодоагенту у впускних колекторах (головках) для мікроканальних теплообмінників або між котушками холодоагенту та нерівними зовнішнім боковим нагріванням повітря на змійовиках холодоагенту шляхом переповнення їх за допомогою рециркуляції рідкого холодоагенту, що забезпечує виключення кінцевої стадії висихання випаровування холодоагенту з низькою інтенсивністю передачі тепла. Таким чином, за рахунок виключення внутрішньої проблеми нерівномірного розподілу холодоагенту та низької інтенсивності передачі тепла випаровуванням холодоагенту в каналах загальна проблема підвищення ефективності теплообмінників киплячим холодоагентом всередині каналів перетворюється на зовнішню проблему передачі тепла на повітряній стороні.*

Ключові слова: *система кондиціонування, теплообмінник, зміна фази, циркуляція холодоагенту, порушення розподілу, нерівномірне теплове навантаження*

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ТЕПЛОВИКОРИСТОВУЮЧА ЕЖЕКТОРНА ХОЛОДИЛЬНА МАШИНА ІЗ ГАЗОДИНАМІЧНИМ ОХОЛОДЖЕННЯМ РОБОЧОГО ТІЛА

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Анотація. *Підвищити ефективність ежекторних холодильних машин можна за рахунок використання термопресора, забезпечуючи при цьому охолодження на певну різницю температур. Аналіз схеми ежекторних холодильних машин з використанням термопрессора показує, що найбільший тепловий коефіцієнт при максимально можливих температурах кипіння в генераторі* t_g *мають холодоагенти: R142b, R600, R1233zd(E), R245fa.*

Ключові слова*: ежектор; термопресор; холодоагент.*

Вступ.

Тепловикористовуючі холодильні машини, на відміну від класичних і найбільш розповсюджених парокомпрессорних холодильних машин, виробляють холод споживаючи при цьому низькопотенційну теплову енергію.

Останнім часом, одним з перспективних напрямів є використання тепловикористовуючих холодильних машин для систем кондиціювання повітря та рефрижерації у складі установок комбінованого виробництва енергії (когенерації, тригенерації) для суднових та стаціонарних енергетичних установок. Серед тепловикористовуючих холодильних машин застосовують наступні: абсорбційні холодильні машини; турбокомпресорні холодильні машини; ежекторні холодильні машини [1, 2].

Перспективним є застосування ежекторних холодильних машин (ЕХМ), які по енергетичній ефективності хоча і поступаються абсорбційним холодильним машинам [1], але перевершують по масогабаритним показникам. В якості робочого тіла (холодоагента) можуть застосовуватися вода або низькокиплячі речовини (хладони). Хладонові ЕХМ вигідно відрізняються відсутністю вакууму, можливістю