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INJECTOR REFRIGERANT CIRCULATION FOR ENHANCING HEAT EFFICIENCY OF AIR COOLERS WITH UNEVEN HEAT LOAD DISTRIBUTION

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ABSTRACT

A concept of incomplete refrigerant evaporation with overfilling air coolers of air conditioning systems that leads to excluding a dry-out of inner surface of air coils and is realized through liquid refrigerant recirculation by injector (jet pump) is developed for air coolers operating at uneven heat load distribution as alternative approach of the heat load modulation through air velocity profile equalizing. A realization of proposed concept allows to solve the problem of uneven refrigerant distribution in manifolds of heat exchangers (including microchannel heat exchangers) and between refrigerant channels (coils, tubes, microchannel plates) and by uneven outside air heat loads or varying heat loading during operation according to actual climatic conditions due to excluding the final stage of refrigerant evaporation with extremely low intensity of heat transfer. It is quite effective for conventional heat exchangers as well as for microchannel ones.

Keywords: Air Cooler, Evaporation, Dry-out, Heat Load, Refrigerant Circulation, Injector

1. INTRODUCTION

The operation of air coolers (AC) of air conditioning systems (ACS) is characterized by uneven heat load distribution in space and time caused by uneven air velocity profile of air freezers (Paliwoda, 1972) and AC of ACS (Radchenko N., 2004) or by varying heat loading, caused by ambient air temperature fluctuation during operation AC at the inlet of gas turbines (GT) (Radchenko A. et. al., 2020; Trushliakov et. al., 2019), inlet and scavenge air AC of Gas Engines (GE) (Fordui et. al., 2020; Radchenko A. et. al., 2020) and AC for railway conditioners (Radchenko M. et. al., 2020; Trushliakov et. al., 2020) and marine engines (Konovalov et. al., 2020; Radchenko R. et. al., 2020) according to actual climatic conditions, or by refrigerant maldistribution in conventionsl heat exchangers (Mikielewicz D. et. al., 2016; Yang, 2017; Zhang, 2018) and microchannel heat exchangers (Bohdal T. et. al., 2015; Kumar S. et. al., 2019; Li, 2015; Mu, 2015; Tang, 2017) in manifolds (Dąbrowski et. al., 2017; 2019; Wang, 2011). Many researches deal with enhancing the efficiency of ACS by mitigating flow maldistribution (Anbumeenakshi et. al., 2016; Dąbrowski, 2020; Mikielewicz D. et. al., 2013; Tuo, 2013), by application of advanced refrigerant circulation (Bandel and Schlunder, 1973; Butrymowicz et. al., 2018; Chawla, 1973; Radchenko N., 2004; Smierciew et. al., 2014; Trushliakov et. al., 2020) and enhancing air processing through two-stage cooling developed for ACS in comfort and energetic application.

Their analysis shows that the intensity of heat transfer of refrigerant, evaporated inside channels, drops at the final stage of evaporation (Bandel, 1973; Chaddock and Varma, 1979; Chawla, 1973; Khovalyg and Baranenko, 2015; Piasecka, 2015), that is caused by drying out the inner channel wall surface while transition of flow from annular-disperse to disperse (mist) flow. In compact finned air coolers, a sharp decrease in heat transfer coefficient to refrigerant at the final stage of its evaporation results in falling the overall heat transfer coefficient and reduction of air cooler efficiency.

The Variable Refrigerant Flow (VRF) systems operate with high part-load efficiency, that results in high daily and seasonal energy efficiency, so as ACS typically spend most of their operating hours in the range of 40% to

80% of maximum cooling capacity (Goetzler et. al., 2007). The authors (Khatri et. al., 2017) 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 (Im et. al., 2016). Therefore, the refrigerant flow control in the VRF-OAP system has been designed to provide more flows to the outdoor air processing (OAP) than to the indoor units and most of the refrigerant flows inside the system were introduced to the OAP. Although the VRF system reduced energy consumption by 40% to 60% compared to that of central ACS, but the problem of inefficient operation of air coolers caused by dry-out of inner walls at the final stage of inside tube refrigerant evaporation followed by dropping the intensity of heat transfer still unsolved. Issuing from the priority of the outdoor air processing (Khatri et. al., 2017) the application of liquid refrigerant recirculation by injector, leading to overfilling the air cooler, would be quite reasonable.

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 (Radchenko N., 2004). The injector circulation of liquid refrigerant originaly was applied for equalizing the refrigerant distribution in air coolers and compact freezers instead of pump (Paliwoda, 1972). A new impulse for further realization of this concept is forced due to applying a circulation of liquid refrigerant to provide incomplete refrigerant evaporation and exclude 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, including influence of refrigerant maldistribution in manifolds of heat exchangers (microchannel heat exchangers) and between refrigerant channels (coils, tubes, microchannel plates).

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 microchannel plate evaporators affected by uneven refrigerant distribution in manifolds and for outdoor air processing units to match varying heat loads according to actual climatic conditions and for indoor fan coils to match varying indoor heat loads in VRF systems, as well as for transport air conditioners.

2. THE RESULTS

2.1. General approaches and original ideas

The main idea behind the principle of rational operation of AC under heat load maldistribution is not to mitigate a maldistribution of external heat loads or refrigerant in the manifolds or between channels (plates), but to minimize the negative influence of uneven heat loads on the heat efficiency of AC to match current actual varying heat load distribution. The general approach proposed is intended to realize the increased heat transfer potential of AC, revealed due to uneven heat load distribution, without affecting the primary heat load maldistribution itself or together with its mitigating. Thus, in the case of external heat load maldistribution caused by uneven air velocity profile in AC it is reasonably to increase the intensity of internal heat transfer to refrigerant in channels to match the increased external heat transfer to air with increased velocity that results in arising the overall heat transfer intensity and heat flux instead of equalizing the air velocity profile, because the last one is accompanied by energetic expenses.

A liquid refrigerant recirculation in AC provides excluding burn-out boiling due to suppression bubble generation. It is worth mentioning that the length of the channels (plates) is reduced with liquid refrigerant recirculation due to increasing the heat flux and decreasing refrigerant vapour mass fraction. This enables to keep or even decrease the refrigerant pressure drop. Certainly, to cover the heat load the number of the channels (plates) should be increased. But in any case the overall AC surface is reduced (Radchenko N., 2004).

The application of liquid refrigerant recirculation by jet pump has to match actual changeable heat loads not by varying refrigerant feed to air coolers as in VRF, but supplying excessive refrigerant flow to air coolers with their overfilling at any increased load to provide efficient operation of air coolers without drying out the inner walls of air coils. In this sense the liquid recirculation system is self controlled due to the presence of linear receiver after condenser and a liquid separator/circulation receiver both functioning as refrigerant accumulators.

2.2. Underground of intensification of heat transfer of refrigerant boiling inside of air cooler channels

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 Fig. 1a. 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 flow to disperse (mist) flow.

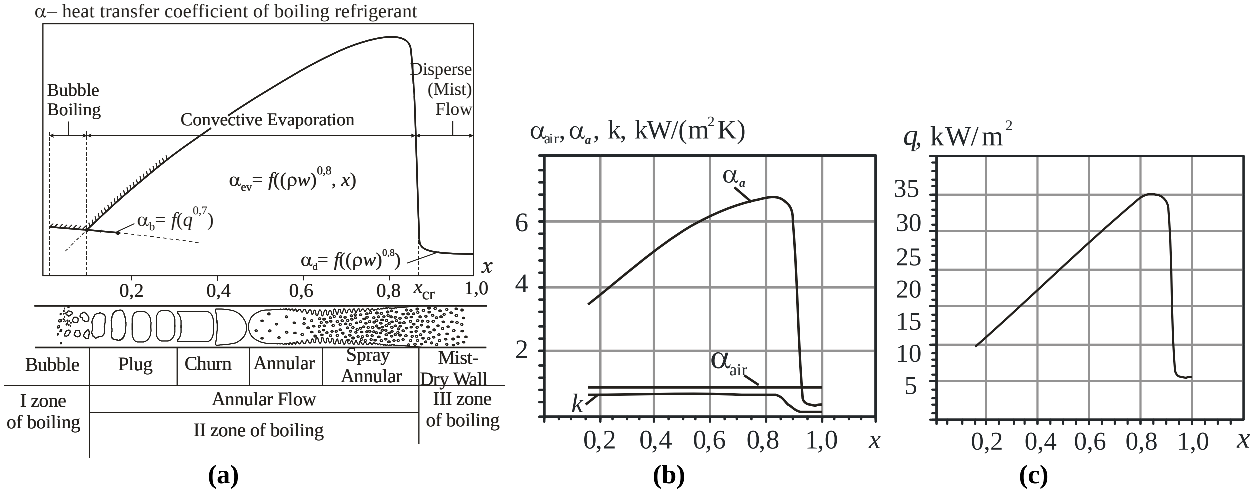


Figure 1: Typical structures of inside tube refrigerant evaporation and variations of coefficients of heat transfer to refrigerant in bubble boiling α_b , convective evaporation α_{ev} and disperse flow α_d (a), to boiling refrigerant α_a and air α_{air} and overall heat transfer coefficient k (b) and heat flux q (c) with vapor mass fraction x

A threshold value of vapour quantity x between bubble (nucleate) boiling and convective evaporation (threshold bubble/convective vapour quantity x) corresponds to equality of heat transfer coefficients calculated for both zones at this x (Radchenko N., 2004). The threshold bubble/convective vapour quantity x is shifting in response to relation between heat flux q and mass flux (mass velocity) ρw. As Fig. 1a shows, with rising mass flux ρw the threshhold point is shifting towards a decreased value of vapour quantity 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 evaporation is much lower than to air. This results in decrease in overall heat transfer coefficient (Fig. 1b). A sharp decrease in heat flux q occurs at burnout vapour fraction $x_{cr} \approx 0{,}9$ (Fig. 1c) and is caused by falling the heat transfer coefficient to refrigerant α_a and the overall heat transfer coefficient k as result. Calculations are conducted for the air cooler with plate finned tubes of 12 and 10 mm outside and inside diameters, air temperature at the inlet 25 °C and outlet 15 °C, refrigerant boiling temperature at the exit 0 °C, refrigerant R142b. It should be noted that a sharp decrease in heat transfer coefficient α_a with the transition from annular to disperse flow takes place for most of refrigerants.

Taking into account that in the conventional air cooler with thermoexpansion valve the vapor 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 vapor superheating with extremely low intensity of heat transfer, is about 30 %. 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.

2.3. Enhancing heat efficiency of air coolers with uneven heat load distribution by refrigerant recirculation

The thermal efficiency of the air coolers is usually estimated by heat flux $q = k\ \theta$, where θ − logarithmic temperature difference; k − overall heat transfer coefficient. 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. 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 $(\rho w)_{opt}$ (Fig. 2a).

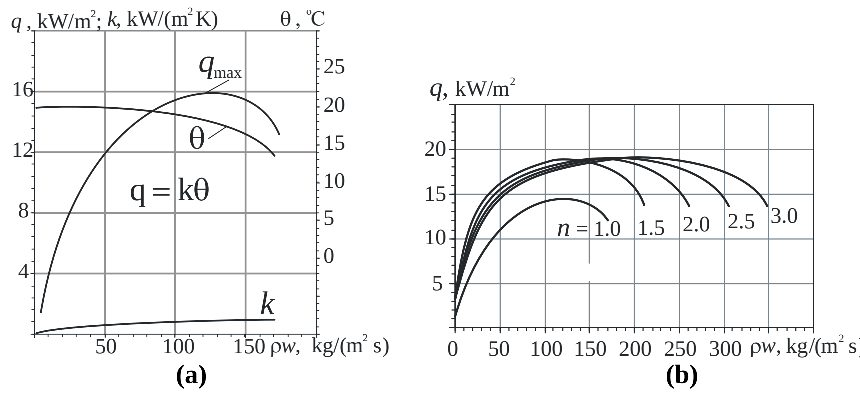


Figure 2: Variation of overall heat transfer coefficient k, logarithmic temperature difference θ and heat flux q (a) and heat fluxes q (b) against refrigerant mass velocities ρw in AC: n = 1.0 – conventional AC with complete evaporation; n > 1.0 – incomplete evaporation with liquid refrigerant recirculation

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 due to liquid refrigerant recirculation by injector are shown in Fig. 2b. When the refrigerant is completely evaporated in AC: vapor is superheated in 5 °C in disperse flow, the circulation number is $n = 1/x_2 = 1$, where x_2 – is the mass vapor fraction of the refrigerant at the outlet. The boiling temperature of R142b at the exit t_{02} = 0 °C; air velocity w = 6 m/s.

As Fig. 2b shows, the recirculation of liquid refrigerant in the AC by injector provides an increase in heat flux q by 25…40 % compared with conventional complete refrigerant evaporation. It is also seen, that overfilling the air coils of AC by recirculation of liquid refrigerant enables a larger deviation of refrigerant mass velocities ρw from their optimum value, providing maximum value of heat flux q, that means that a larger heat load changes are permitted, that gives good perspectives of injector liquid refrigerant recirculation for AC with heat load and refrigerant maldistribution.

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. 3: 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.

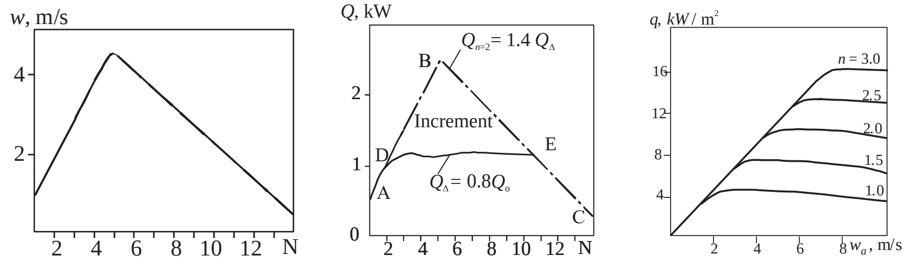


Figure 3: Air velocity *w*-profile at the inlet of air cooler (a), heat load *Q*-distributions among air coils (b) and heat fluxes *q* for various numbers of circulation *n* against inlet air velocities w_a (c): N – a number of air coil; ADEC – heat load *Q* distribution among air coils for conventional 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. 3b 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. 1), that limits the heat loads on these coils.

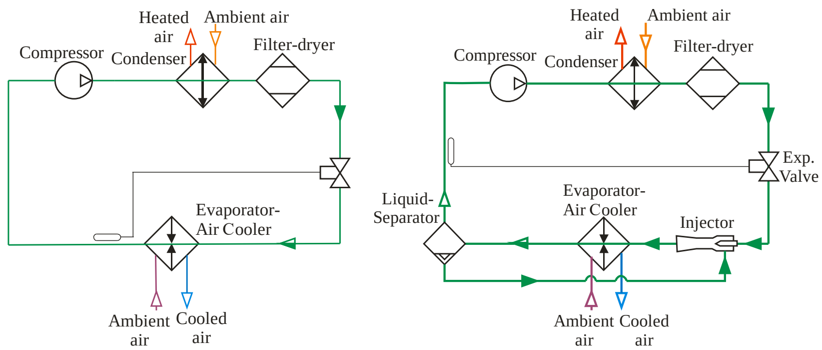
The incomplete evaporation due to refrigerant coils overfilling by liquid refrigerant recirculation leads to the full elimination of its inefficient final stage and to rising the intensity of overall heat transfer on the whole air cooler surface in response to increasing the air velocity at the inlet of the coils number N =3-11 (Fig. 3a). 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. 3b).

The principle of over filling the channels allows to increase the heat fluxes *q* in response to rising the air velocity at the inlet of AC by liquid refrigerant circulation with adequate number of circulation n = U+1 = 1/x_2 (Fig. 3c).

2.4. The refrigeration system with injector liquid refrigerant recirculation in air coolers

To ensure intensive heat transfer along the entire length of AC channels, the final stage of evaporation with disperse flow along dry inner walls of the channels should be excluded, i.e. the operation of the air cooler with incomplete evaporation has to be provided. Unevaporated liquid is separated from the vapor in the liquid separator and returned back to the air cooler for evaporation by injector (Fig. 4). 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 expansion valve.

Any pump circulation system operates at changeable cooling capacities according to current heat loads on evaporators-air coolers (E-AC) with changing the refrigerant volumes in liquid separators E-AC and in linear receiver after condenser with their varying issuing from heat loading of E-AC. This system provides effective processing air at actual changing heat loads in response with current climatic conditions. The authors (Khatri et. al., 2017) on the base of field test results of VRF revealed that the design outdoor unit cooling capacity should be of around 70% excess to prevent a lack of indoor units cooling capacities. This means that a number of refrigerant circulation n in air cooler of about n = 1.7−1.8 with mass vapor fraction at the outlet of air coil x_2 = 1/n ≈ 0.6 (Fig. 2b) due to liquid refrigerant recirculation by injector satisfies this requirements.



a b

Figure 4: The schemes of conventional refrigeraion system (a) and system with injector recirculation of liquid refrigerant in air cooler (b)

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 (Khovalyg and Baranenko 2015), 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. It is worth mentioning that with liquid refrigerant recirculation the length of the channels (plates) is reduced due to increasing the heat flux and decreasing refrigerant vapour mass fraction. This enables to keep or even decrease the refrigerant pressure drop. Certainly, to cover the heat load the number of the channels (plates) should be increased. But in any case the overall AC surface is reduced.

Overfilling the evaporators, that leads to increasing the intensity of boiling heat transfer on all length of their channels due to incomplete refrigerant evaporation, excludes 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. Therefore this principle is very useful for microchannel evaporators, characterized by refrigerant maldistribution in manifolds, and might be combined with applying the other methods of its mitigating.

3. CONCLUSIONS

A developed concept of enhancing heat efficiency of air coolers is intended to solve the problem of uneven outside heat loads as well as of refrigerant maldistribution in manifolds and between channels (coils, channel plates) by over filling all the channels through injector liquid refrigerant recirculation that provides excluding the final dry-out stage of refrigerant evaporation, that leads to sharp drop in evaporation heat transfer and falling the intensity of overall heat transfer and heat flux as result.

The application of liquid refrigerant recirculation by jet pump allows to match actual changeable heat loads not by varying refrigerant feed to air coolers as in VRF, but supplying excessive refrigerant flow to air coolers with their overfilling at any peak load to provide efficient operation without drying out the inner walls of air coils. In this sense the liquid refrigerant recirculation system operates as self controlled due to the presence of linear receiver and a liquid separator receiver both functioning as cooling capacity accumulators.

The principle of over filling all the channels of air coolers excludes any influence of inside channel heat transfer, is very useful for microchannel evaporators with uneven refrigerant distribution in manifolds as well as for conventional heat exchanges with uneven air velocity distribution.

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