An Innovative Air Conditioning System for Changeable Heat Loads

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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. In general case, an overall heat load of any AC system comprises the unstable range, corresponding to ambient air processing with heat load fluctuations, and a comparatively stable part for subsequent air subcooling. Following from this approach, a rational design overall heat load is chosen to provide a maximum annular refrigeration capacity generation and divided into a comparatively stable basic part and a remaining part for ambient air precooling at changeable heat loads. 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 at about nominal mode. According to modern trend in AC systems the load modulation is performed by varying refrigerant feed to air coolers in Variable Refrigerant Flow (VRF) system. But with this 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 remains unsolved. As alternative approach of the heat load modulation in AC systems there is a concept of incomplete refrigerant evaporation with overfilling air coils that leads to excluding a dry-out of inner surface of air coils and is realized through liquid refrigerant recirculation by injector (jet pump).

Keywords: Air conditioning · Heat load · Refrigerant overfilling

1 Introduction

The performance efficiency of air conditioning (AC) systems depends on the heat efficiency of their air coolers. 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 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.

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In general case, 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. 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. But with this the problem of inefficient operation of air coolers caused by dry-out of inner walls at the final stage of refrigerant evaporation remains unsolved.

An alternative approach of the heat load modulation in AC systems is the concept of incomplete refrigerant evaporation with overfilling air coils due to liquid refrigerant recirculation by injector (jet pump) that leads to excluding a dry-out of inner surface of air coils.

2 Literature Review

A lot of publications are devoted to improving the performance of AC systems by enhancing heat transfer processes in evaporators and condensers [1–6], applying the energy efficient scheme decisions of refrigeration machines [7–10] and waste heat recovery refrigeration techniques [11–14], methods of modeling and controlling, experimental, monitoring and statistical methods [15–17].

As modern trend in AC system the VRF systems are considered [18–21]. The VRF system maintains the zone air temperature at the set-point by supplying adequate refrigerant to indoor fan coils to meet the space cooling load needs [22].

Most of articles studies have been conducted on solutions of efficient operation of the VRF system in actual buildings [23–27] and control strategies of the systems [28–34].

The performance evaluations showed that the VRF system reduced energy consumption by 40% to 60% compared to that of central AC systems [18]. But with this the general 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 remains unsolved.

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 [19]. 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 [18, 20, 21], 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.

3 Research Methodology

The main idea being 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 (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. The ambient air precooling mode with considerable heat load fluctuation needs load modulation, whereas the basic comparatively stable heat load range can be covered by operation of refrigerant compressor at about nominal mode.

The application of liquid refrigerant recirculation by jet pump is to match actual changeable heat loads not by varying refrigerant feed to air coolers but supplying excessive refrigerant flow to air coolers with their overfilling at any load to provide efficient operation of air coolers without drying out the inner walls of air coils. In this sense the liquid refrigerant recirculation system is self controlled due to the presence of linear receiver after condenser and a liquid separator/circulation receiver functioning as cooling capacity accumulators [9].

Following this approach, a rational design overall heat load is chosen to provide a maximum annular refrigeration capacity generation and shared into a comparatively stable and a remaining parts for ambient air precooling at varying heat loads [35, 36].

All the calculation results have been presented for the refrigeration capacity in related value – specific refrigeration capacity q_0 as the overall refrigeration capacity Q_0, kW, related 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 from t_{amb} to t_{a2} = 10 °C and specific refrigeration capacity $q_{0.10\text{-}15}$ as their difference $q_{0.10\text{-}15} = q_{0.10} - q_{0.15}$ for subcooling air from t_{a2} = 15 °C to t_{a2} = 10 °C have been calculated for current climatic conditions.

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

4 Results

4.1 Determining the Rational Design Heat Load

To determine the rational design heat load on ambient air conditioning system, matching changeable actual heat loads in response to current 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 have been calculated for all the year round.

The annual refrigeration capacity output in ratio value as total annual refrigeration capacity output $\sum(Q_0 \cdot \tau)$, kW·h, related to the unit of air mass flow rate: $\sum(Q_0 \cdot \tau)/G_a$, or $\sum(q_0 \cdot \tau)$, kW·h/(kg/s), or kJ·h/kg, where Q_0 – refrigeration capacity, kW;·τ – time duration, h; G_a – air mass flow rate in ambient air cooler, kg/s, in dependence on design specific refrigeration capacity $q_0 = Q_0/G_a$, kW/(kg/s), or kJ/kg, of installed refrigeration machine for temperatures of cooled air t_{a2} = 10, 15 and 20 °C and climatic conditions of Nikolaev region, Ukraine, 2015, are presented in Fig. 1.

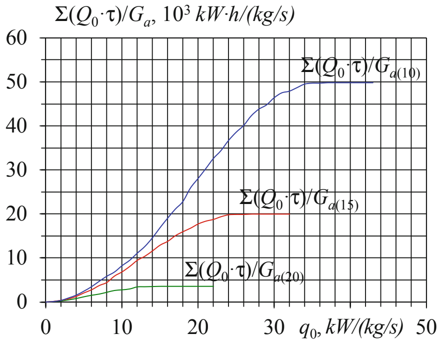

Fig. 1. Annual refrigeration capacity output in ratio values $\sum(Q_0{\cdot}\tau)/G_a$ (at unit air mass flow rate G_a = 1 kg/s) against design specific refrigeration capacity $q_0 = Q_0/G_a$ of applied refrigeration machine for temperatures of cooled air t_{a2} = 10, 15 and 20 °C: $\sum(Q_0{\cdot}\tau)/G_{a(10)}$ – at t_{a2} = 10 °C; $\sum(Q_0\ {\cdot}\tau)/G_{a(15)}$ – at t_{a2} = 15 °C; $\sum(Q_0\ {\cdot}\tau)/G_{a(20)}$ – at t_{a2} = 20 °C.

As Fig. 1 shows, the annual refrigeration capacity output $\sum(Q_0{\cdot}\tau)/G_{a(10)}$ for cooling air to the temperature t_{a2} = 10 °C at specific refrigeration capacity q_0 = 34 kW/(kg/s) is evaluated as $\sum(q_0{\cdot}\tau)_{10}$ = 60 MW·h/(kg/s) and achieved with high rates of its increments.

Because of sharply falling rate of arising the increments $\sum(Q_0{\cdot}\tau)/G_{a(10)}$ with arising a design specific refrigeration capacity q_0 the further increase in specific refrigerating capacity q_0 from 34 to 40 kW/(kg/s) does not result in appreciable increment in the annual refrigeration capacity output $\sum(Q_0{\cdot}\tau)/G_{a(10)}$. At the same time a subsequent increase in design refrigeration capacity q_0 of applied refrigeration machine causes considerable increase in its capital expense by 20…30%. Thus, the specific refrigeration capacity q_0 = 34 kW/(kg/s) is considered as rational one to calculate a full designed refrigeration capacity Q_0 of applied refrigeration machine according to the total air mass flow G_a, kg/s: $Q_0 = G_a \cdot q_0$, kW.

With this a specific refrigeration capacity q_0 is calculated as $q_0 = \xi \cdot c_a \cdot (t_{amb} - t_{a2})$, kW/(kg/s), or kJ/kg, where ξ – inversely proportional value of sensible heat rate calculated as a ratio of total heat removed from the wet air during cooling (an air enthalpy decrease including the heat of water vapor condensation) and sensible heat extracted; c_a – specific heat capacity of wet air.

4.2 Underground of Intensification of Heat Transfer of Refrigerant Boiling Inside Air Coils of Air Coolers

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. 2.

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. 2a).

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. 2b).

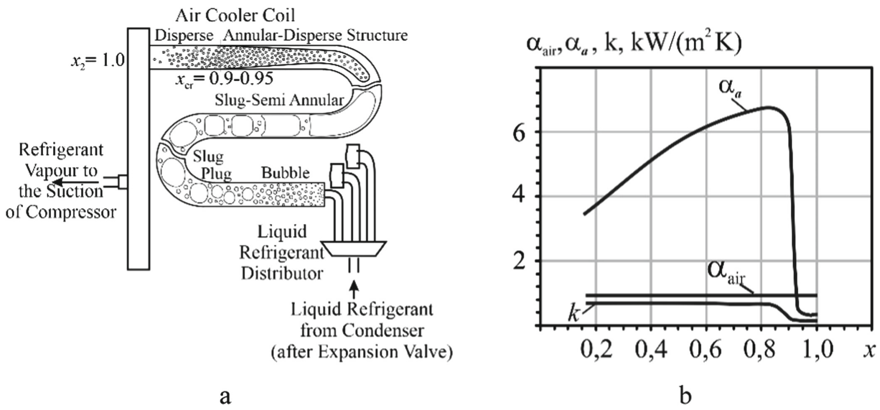

Fig. 2. Typical structures of inside tube refrigerant boiling (a) and variation of heat transfer coefficients to boiling refrigerant α_a and air α_{air} and overall heat transfer coefficient k with the vapor mass fraction x (b).

Calculations are 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 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.

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 °C, 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%.

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.

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 by injector [9].

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 $q_{max} = k \quad \theta$, where θ − logarithmic temperature difference; k − overall heat transfer coefficient.

The frictional pressure gradient for two phase flow was calculated according to the Lockhart-Martinelli-Nelson method.

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. 3a).

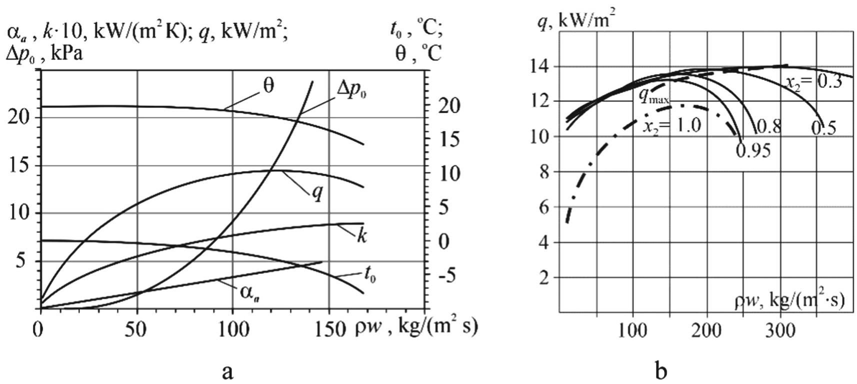

Fig. 3. Mean values of heat fluxes q, heat transfer coefficients to refrigerant α_a and overall heat transfer coefficients k, logarithmic temperature difference θ, refrigerant boiling temperature t_0 and pressure drop ΔP against refrigerant mass velocities ρw for complete evaporation (a) and heat fluxes q at mass vapor fraction x_2 at the outlet of air coil for incomplete refrigerant evaporation with liquid recirculation by injector (b): R142b, t_{02} = 0 °C; air velocity w = 6 m/s.

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. 3b. 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 vapor superheated in 10 °C for the throttle circuit and wetted wall with $x_2 < x_{cr}$ for the injector recirculation. In disperse mixture the vapor is superheated in 5 °C compared to the boiling temperature t_{02}.

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 (Fig. 3b).

As one can see from Fig. 3b, overfilling the air coils of the air cooler 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 permited, that gives good perspectives of injector liquid refrigerant recirculation in conditioning ambient and indoor air.

4.3 The Innovative Scheme Decisions of Air Conditioning Systems Intensification of Heat Transfer of Refrigerant Boiling Inside Air Coils of Air Coolers

Any pump circulation system 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.

Issuing from the priority of the outdoor air procession [18, 20, 21], the application of liquid refrigerant recirculation by injector, leading to overfilling the air cooler, would be quite reasonable first of all for Outdoor Air Processing (OAP) unit to provide a complete outdoor air procession to avoid introducing not completely processed outdoor air to the indoor environment with negative impacts on the indoor comfort (Fig. 4).

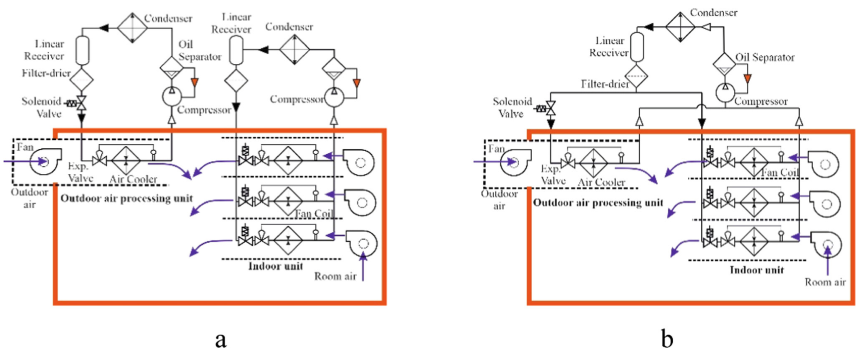

Fig. 4. The schemes of traditional ductless Variable Refrigerant Flow (VRF) system with Outdoor Air Processing (OAP) and Indoor Air Processing (IAP) units (a) and combined version (b).

The authors [18] 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. 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 $x_2 = 1/n \approx 0.6$ (Fig. 3b) due to liquid refrigerant recirculation by injector satisfies this requirements.

5 Conclusions

A method of rational designing of air conditioning systems by matching changeable heat loads corresponding current climatic conditions has been developed to enhance their operation efficiency. According to this method, a value of overall optimum design heat load is chosen to provide a maximum annular refrigeration capacity generation. This overall optimum heat load is divided into a stable basic heat load part, covered with operation of refrigeration machine in nominal mode, and a remaining heat load part for ambient air precooling at changeable heat loads, covered with operation of refrigeration machine in part-load modes. The remained part of current heat loads needs application of energy conserving technologies for example through using an excess refrigeration capacity accumulated at decreased current heat loads.

A proposed novel concept of enhancing heat efficiency of heat exchanges with boiling refrigerants inside channels is intended to solve the problem of changeable actual heat loads on ambient air coolers by over filling air coils through liquid refrigerant injector recirculation that provides excluding the final stage of refrigerant evaporation with low intensity of heat transfer.

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 coefficient.

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