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HEAT PIPES, THEORY AND PRACTICE

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1. INTRODUCTION

Shared development of applied thermodynamics and heat transfer is a historic identity of the Engineering of the XX th century. Such a trend has provided a basis for emerging of innovative technologies, and this has caused a noticeable progress in the simulation and design of both macro- and micro-systems. Some applications have been expanded so fast that researchers lack time for sufficient analytical generalizations and explanations. That was a story of known heat pipe technology, referred to the research advancements in two-phase heat transfer, thermal design, and material science, and accelerated by space engineering, cryogenics, solar energetics, and microengineering.

The idea of heat pipes was first suggested by R.S. Gaugler in 1944 [1]. However, only after 1964, when G.M. Grover invented it [2] and its remarkable properties have been appreciated and serious developments began.

Heat pipes are highly efficient, two-phase heat transfer devices with effective thermal conductivities of up to about 500 times that of solid copper, depending on the application. A heat pipe usually begins as a tube, typically copper, that is given an internal wick structure. The tube is evacuated and backfilled with a small amount of working fluid and then sealed. A heat pipe is similar to a thermosyphon. It differs from a thermosyphon by virtue of its ability to transport heat against gravity by an evaporation-condensation cycle with the help of porous capillaries that form the wick. The wick provides the capillary driving force to return the condensate to the evaporator. The quality and type of wick usually determines the performance of the heat pipe, for this is the heart of the product. Different types of wicks are used, depending on the application for which the heat pipe is being employed. For a comprehensive summary of the available work on the theoretical principles and design considerations, reader is referred to the books by Dunn and Reay [3] and Ivanovskii *et al.* [4].

2. HEAT PIPE DESIGN AND OPERATIONAL CONSTRAINS

All heat pipes have three physical elements in common. These include an outer container, a small amount of working fluid, and a capillary wick structure. In addition to these basic components, heat pipes may also include gas reservoirs (variable conductance/diode heat pipes) and liquid or gas traps (diodes). Functionally, the heat pipe consists of three sections: evaporator, condenser section, and adiabatic region. The evaporator section is mounted to the heat-generating components, while the condenser is thermally coupled to a heat sink or radiator. The adiabatic section allows heat to be transferred from the evaporator to the condenser with very small heat losses and temperature drops. Figure 1 depicts the basic heat pipe.

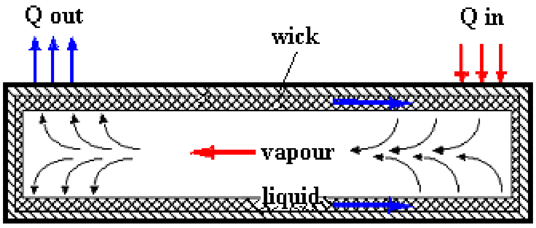

Figure 1. The basic heat pipe principle.

Inside the container there is a liquid under its own pressure, which enters the pores of the capillary material, wetting all internal surfaces. Applying heat at any point along the surface of the heat pipe causes the liquid at that point to boil and enter a vapor state. When this occurs, the liquid picks up the latent heat of vaporization. Then vapor of a higher pressure moves inside the sealed container to a colder location, where it condenses. Thus, the vapor gives up the latent heat of vaporization and moves heat from the input to the output end of the heat pipe.

Heat pipes can operate in the fixed conductance, variable conductance, or diode mode. The fixed conductance heat pipe can transfer heat in either direction and operates over broad temperature ranges, but has no inherent temperature control capability. Constant conduction heat pipes allow isothermalization of shelves, radiators and structures; spread heat from high heat dissipating components; and conduct heat away from heat producing devices embedded within instruments and satellites.

In the variable conductance heat pipe, a small quantity of non-condensable gas is loaded into the heat pipe. This mode of heat pipes can be used to control the temperature of the equipment within very narrow limits; control is possible to less than 1°C by using carefully designed techniques. This is accomplished by controlling the location of the non-condensable gas/vapor interface within the condenser end of the heat pipe, thereby varying the active length of the condenser and causing a modulation in the condenser heat rejection capability. Temperature control of the attached device is achieved by an active feedback system consisting of a temperature sensor at the heat source and a controller for a heater at the non-condensable gas reservoir. The heater causes the gas in the reservoir to expand, thus moving the gas/vapor interface. Diode heat pipes permit heat to flow in one direction and inhibit heat flow in the opposite direction.

Heat pipes can be designed to operate over a very broad range of temperatures from cryogenic (less than -243°C) applications utilizing titanium alloy/nitrogen heat pipes, to high temperature applications (more than 2000°C) using tungsten/silver heat pipes. There are many factors to consider when designing a heat pipe: compatibility of materials, operating temperature range, diameter, power limitations, thermal resistances, and operating orientation.

The most important heat pipe design consideration is the amount of power the heat pipe is capable of transferring. Heat pipes can be designed to carry a few watts or several kilowatts, depending on the application. If driven beyond its capacity, however, the effective thermal conductivity of the heat pipe will be significantly reduced. Therefore, it is important to ensure that the heat pipe is designed for safely transport of the required heat load. The maximum heat transport capability of the heat pipe is governed by several limiting factors, which must be addressed when designing a heat pipe. There are five primary limitations of heat transport inside heat pipe. These heat transport limits are a function of the heat pipe operating temperature. Each heat transport limitation is summarized in Table 1.

The first consideration in the identification of a suitable working fluid is the operating vapor temperature range. Within the approximate temperature band, several possible working fluids may exist, and a variety of characteristics must be examined in order to determine the most acceptable of these fluids for the application considered. The prime requirements are compatibility with wick and wall materials, good thermal stability, wettability of wick and wall materials, vapor pressure not too high or low over the operating temperature range, high latent heat, high thermal conductivity, low liquid and vapor viscosities, high surface tension, acceptable freezing or pour point.

In heat pipe design, a high value of surface tension is desirable in order to enable the heat pipe to operate against gravity and to generate a high capillary driving force. In addition to high surface tension, it is necessary for the working fluid to wet the wick and the container material, i.e. the contact angle should be zero or very small. The vapor pressure over the operating temperature range must be sufficiently large to avoid high vapor velocities, which tend to setup large temperature gradients and thus cause flow instabilities.

A high latent heat of vaporization is desirable in order to transfer large amounts of heat with minimum fluid flow, and hence to maintain low pressure drops within the heat pipe. The thermal conductivity of the working fluid should preferably be high in order to minimize the radial temperature gradient and to reduce the possibility of nucleate boiling at the wick or wall surface. The resistance to fluid flow will be minimized by choosing fluids with low values of vapor and liquid viscosities. Steps must also be taken to ensure the purity of the fluid charge. Chi [5] listed standard cleaning and filling methods for a variety of working fluid/wall material combinations. Special consideration must be given to the processing of heat pipes to be used at temperatures below 250K. As the temperature drops, the vapor pressure of the fluid falls off. This allows any non-condensable gas created by contamination to expand, thus creating an even larger problem.

Table 1. Heat transport limits

Heat transport limit	Description	Cause	Potential solution
Viscous	Viscous forces prevent vapor flow in the heat pipe	Heat pipe operating below recommended operating temperature	Increasing of heat pipe operating temperature or detection of alternative working fluid
Sonic	Vapor flow reaches sonic velocity when exiting heat pipe evaporator resulting in a constant heat pipe transport power and large temperature gradients	Power/temperature combination, too much power at low operating temperature	This is typically only a problem at start-up. The heat pipe will carry a set power and the large ΔT will self correct as the heat pipe warms up
Flooding	High velocity vapor flow prevents condensate from returning to evaporator	Heat pipe operating above designed power input or at too low an operating temperature	Increasing of vapor space diameter or operating temperature
Capillary	Sum of gravitational, liquid and vapor flow pressure drops exceed the capillary pumping head of the heat pipe wick structure	Heat pipe input power exceeds the design heat transport capacity of the heat pipe	Modification of heat pipe wick structure design or reducing of power input
Boiling	Film boiling in heat pipe evaporator typically initiates at 5-10 W/cm^2 for screen wicks and 20-30 W/cm^2 for powder metal wicks	High radial heat flux causes film boiling resulting in heat pipe dry out and large thermal resistances	Using of a wick with a higher heat flux capacity or spreading out of the heat load

3. WICK OR CAPILLARY STRUCTURE

The wick structure is the pumping system that moves the condensate from the condenser region to the evaporator region. This internal wick structure enhances the power handling capability as well as enables fluid return in horizontal and "against gravity" (evaporator above the condenser) orientations, with performance depending on the type of wick structure used. The performance of the wick is set by its pore radius and permeability. The pore radius determines the pumping pressure the wick can develop, whilst permeability determines the frictional losses of the fluid as it flows through the wick. There are several types of wick structures available, including swaged or extruded grooves, screen mesh, cables/fibers, and sintered powder metal. Figure 2 shows several conventional wick structures of heat pipes.

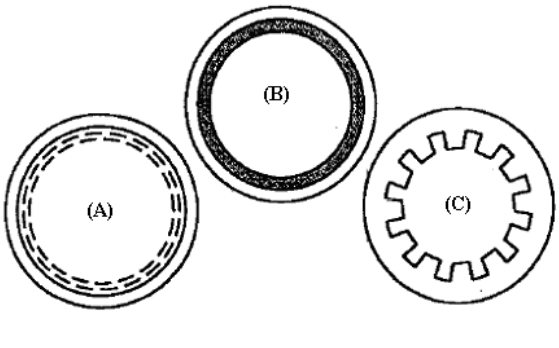

Figure 2. Typical homogeneous wick structures: (A) wrapped screen, (B) sintered metal, (C) axial grooves.

The above list of types of wick structures is in order of decreasing permeability and decreasing pore radius. However, the available capillary pressure generated by a wick increases with decrease in pore size. Width of the wick is its important feature, which must be optimized. The heat transport capability of the heat pipe is raised by increasing the wick width. The overall thermal resistance at the evaporator also depends on the conductivity of the working fluid in the wick. Other necessary properties of the wick are compatibility with the working fluid and wettability.

Grooved wicks have a large pore radius and a high permeability and, as a result, the pressure losses are low but the pumping head is also low. Grooved wicks can transfer high heat loads in a horizontal or gravity aided position, but cannot transfer large loads against gravity. An example is where the heat-input area of the heat pipe is placed physically above the cooled regions of the heat pipe.

Screen-mesh wick structures can be made with finer pores and, therefore, offer improved performance over simple-groove wick structures. Since the installation of the screen is an additional step in the fabrication of the heat pipe, and because the process can be tedious, screen-mesh wick heat pipes are slightly more expensive than groove-wick heat pipes.

Fibrous materials, like ceramics, have also been used widely. In general, they have smaller pores. The main disadvantage of ceramic fibers is that they have little stiffness and usually require a continuous support by a metal mesh. Thus while the fiber itself may be chemically compatible with the working fluids, the supporting materials may cause problems. More recently, interest has turned to carbon fibers as a wick material. Carbon fiber filaments have many fine longitudinal grooves on their surface, have high capillary pressures and are chemically stable. A number of heat pipes that have been successfully constructed using carbon fiber wicks seem to show a greater heat transport capability.

Sintered-powder metal wicks are porous metal structures, approximately 50 percent dense, which have small pore radius and relatively low permeability. Since the size of the particles used in forming the sintered structure can be varied, a tailored high-performance wick can be made using this process. Sintered-powder metal-wick heat pipes can be made to work in any orientation, even with the pipe vertical and the heat source at the top. This capability makes the sintered-powder metal-wick structure a very high-performance system, particularly suitable for applications on such as notebook computers where the use-orientation of the heat pipes is uncertain.

In the recent decades, advances in nanotechnology have made it increasingly practical to create high-porosity permeable structures with median surface pore diameters in the order of a few nanometers, see Smalley and Yakobson [6], Scharff [7] and Cohen [8]. For instance, chemical researchers at University of Illinois reported the use of so-called “kitchen chemistry” to synthesized micron thick porous cadmium telluride films comprised of straight evenly spaced pores of 7 to 8 nm in diameter. Similar structures used as capillary pumps can theoretically provide capillary pressures in the order of several MPa.

Hall [9] suggested improvement of the capillary capability by using an advanced wick structure - graded wick. While uniform wicks are easy to manufacture, they do not provide the maximum capillary capability required in many applications, especially for heat pipes that are operating against gravity and are long. Because the liquid vapor pressure differential changes continuously from the evaporator to condenser, a graded wick that corresponds to this change is able to provide the maximum capillary capability and the minimum liquid flow resistance.

4. APPLICATIONS

Until recently, the use of heat pipes has been mainly limited to space technology due to cost effectiveness and complex wick construction of heat pipes. There are several applications of heat pipes in this field, such as spacecraft temperature equalization component cooling, temperature control and radiator design in satellites. Other applications include moderator cooling, removal of heat from the reactor at emitter temperature and elimination of troublesome thermal gradients along the emitter and collector in spacecrafts.

Currently heat pipe technology has been integrated into modern thermal engineering designs, such as terrestrial thermal control systems, solar energetics, etc., see Faghri [10].

Perhaps the best way to demonstrate the heat pipes application to electronics cooling is to present a few of the more common examples. Currently, one of the highest volume applications for heat pipes is cooling the Pentium processors in laptops to reduce the working temperature for better efficiency. Due to the limited space and power available in notebook computers, heat pipes are ideally suited for cooling the high power chips. Fan assisted heat sinks require electrical power and reduce battery life. Standard metallic heat sinks capable of dissipating the heat load are too large to be incorporated into the laptop package. On the other hand, heat pipes offer a high efficiency, passive, compact heat transfer solution. Three or four millimeter diameter heat pipes can effectively remove the high flux heat from the processor. The heat pipe spreads the heat load over a relatively large area heat sink, where the heat flux is so low that it can be effectively dissipated through the notebook case to the ambient air.

Many research publications on different use of heat pipe technology are available within the mentioned engineering areas, while a number of papers devoted to specific applications in refrigeration and air conditioning, is very limited. However, refrigerating heat pipe has been suggested as a potentially compact, reliable system for the cold generation. Further development of the loop heat pipe principle assures amazing possibilities for various applications in refrigeration and air conditioning area. For example, heat pipe heat exchangers are commonly used as dehumidifiers at the numerous refrigerating vapor-compression units in the United States, see Chart and Sheldon [11], Mathur [12] and Meckler [13]. A development of the desiccant cooling systems releases principally new visions for the heat pipe technology.

In an air conditioning system, the colder the air as it passes over the cooling coil (evaporator), the more the moisture is condensed out. The heat pipe is designed to have one section in the warm incoming stream and the other in the cold outgoing stream. By transferring heat from the warm return air to the cold supply air, the heat pipes create the double effect of pre-cooling the air before it goes to the evaporator and then re-heating it immediately.

Activated by temperature difference and therefore consuming no energy, the heat pipe, due to its pre-cooling effect, allows the evaporator coil to operate at a lower temperature, increasing the moisture removal capability of the air conditioning system by 50-100%. With lower relative humidity, indoor comfort can be achieved at higher thermostat settings, which results in net energy savings. In general, for each 1oC rise in thermostat setting, there is a 7% savings in electricity cost. In addition, the pre-cooling effect of the heat pipe allows the use of a smaller compressor.

Recently, many ideas have been suggested to enhance thermal control systems by integrating loop heat pipes. In general, loop heat pipes differ from conventional heat pipes by its arrangement in a loop, high capillary wick pumping and low flow resistance, wick structure appears only in the evaporator, both vapor and liquid flow in the same direction. The loop heat pipe consists of sealed tubes connecting the evaporator, the heat source, with a condenser, and the heat sink. The working fluid circulates due to the capillary pressure gradient developed in the wick. It is important to distinguish between a loop heat pipe and a capillary pumped loop. The first appearance of the loop heat pipe was in the former Soviet Union in the early 1980s, see Maidanik *et al.* [14]. However, the capillary pumped loops developed in the United State around the same period. The major design distinction between these types of thermal control systems is the location of the compensation chamber. Nikitkin and Cullimore [15] presented a comprehensive analysis of the advantages and disadvantages of loop heat pipes and capillary pumped loops.

4.1 Heat pipes as the thermal links in the refrigeration systems

A significant reduction of refrigeration energy used by the domestic refrigerators can be achieved by the replacement of the vapor-compression refrigerating machines (VCRM) by the refrigerators based on the low-potential thermal power systems such as vapor-ejector refrigerating machines (VERM) or absorption refrigerating machines (ARM), including absorption-diffusion apparatus (ADA).

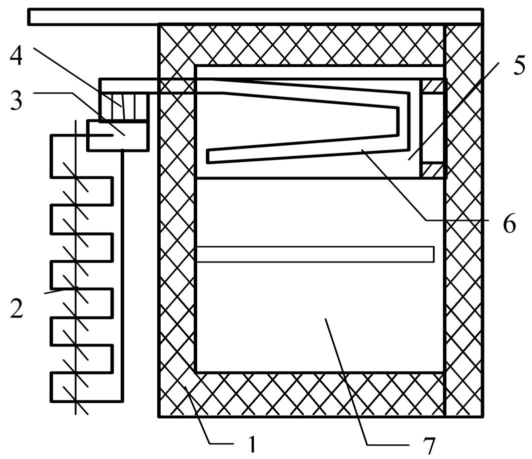

Figure 3. Schematic of thermoelectric refrigerator on the LHP base.
1 - insulation; 2 – “hot” radiator; 3 - LHP evaporator; 4 - thermoelectric device; 5 - freezing chamber; 6 - axial heat pipe; 7 - cold chamber.

It is known that the energy losses decrease in abovementioned cases could be obtained only if the low-potential thermal sources will be used for heating, as well as application of gas burning allows decreasing of power consumption in comparison with electricity.

Gas burning is a simplest heat load method. But its application for ARM or ADA associated with internal gas burning process inside or close to the refrigerating unit is hazardous matter.

The modern loop heat pipes could transfer heat up to thousands Watts to the distances up to ten meters. It was denoted that such a device could replace VCRM, ARM or VERM with the gas burning in domestic refrigerators, commercial refrigerating systems, etc. Nowadays these abilities could be easily implemented, which do not require of any additional research.

Another example of the LHP application is connected with substantial improvement of the thermoelectric refrigerator performance.

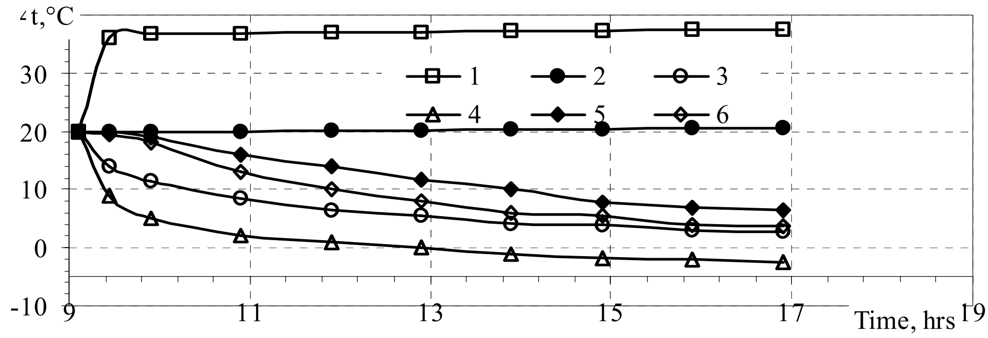

Figure 4. Refrigerator launch with food simulator
1 - LHP evaporator, 2 – ambience, 3 – “cold” chamber, 4 – “freezing” chamber, 5 - food simulator with thermal capacity of 9740 J/kg, 6 - food simulator with thermal capacity of 1470 J/kg

The idea of thermoelectric device application as the cold source for the domestic refrigerators is not new, but it wasn’t realized because of significant drawbacks, such as the problem of «turning off» of thermoelectric cold source. It was discovered that when thermoelectric cold source should be turned off, heat is transferring into internal volume of the cold chamber. The LHP used as the thermal link between the cold chamber and the cold attachment provides a strong thermal connection when thermoelectric device is on and very poor thermal contact when it is turned off. Thus, the LHP application as the thermal link easy ensures the matching heat transfer surface development and improves performance of the refrigerator. The Lavochkin Association designed and produced some samples of thermoelectric refrigerators (TER) on the LHP base.

One of TER designs is shown in Fig. 3. Figure 4 illustrates corresponding test results. It was essential to observe the substantial potential for decreasing of the temperature drop between the environment and LHP evaporator that considerably improved performance of the thermoelectric refrigerator.

4.2 Refrigerating heat pipes

The principles of the refrigerating heat pipes (RHP) are based on the combination of the conventional heat transfer methods and two-phase flow phenomena study. It should be mentioned that thermal energy transfer by means of the latent heat (two-phase flow) is the best method for the RHP. The different two-phase flowing principles were known and performed in conventional heat pipe technology, including application of capillary porous structures, pulsating heat pipes, use of electrohydrodynamic forces, etc. All of them could be accepted for the further developments of RHPs.

The upper operating level (T_h) corresponds to the temperature of the low-potential heat source. The lowest level (T_0) defines the temperature of refrigerating action, when the heat departure temperature level (T_c) coincides with the temperature of the surroundings. Thus, the principal temperature interrelation can be defined as $T_h > T_c > T_0$.

Heat Q_h that is transferred from high-temperature reservoir, serves for the completion of a refrigerating cycle with the refrigerating capacity of RHP Q_0, and in accordance to the first low of thermodynamics $Q_c = Q_0 + Q_h$. Finally, heat Q_c withdraws into the surroundings.

The main principles of analysis are the following:

1. Assume that the physical nature of the driving forces plays a major role in viability of the corresponding thermal control system (TCS).

2. The type of driving force could be identified from the analysis of the general momentum conservation equation. Gravitational forces, centrifugal forces, capillary forces, electro-hydrodynamic forces, etc. are conceivable, and every kind sets a different design of TCS.
3. Each TCS type can ensure heat transfer as the simplest purpose of its operation. Hence, it should be noticed that there is an opportunity to obtain a combined function of TCS such as refrigerating action, power generation, etc.
4. TCS will provide a combined functioning when the corresponding driving force will be sufficient for that.

Therefore, a comparison of the driving forces for an auxiliary energy potential can delight the workability of the current design of TCS to complete specific functions: refrigerating action, power generation, heat pumping, etc.

The maximum scale of each type of driving forces can be estimated with respect to the conditions of the corresponding model and the physical nature of these forces.

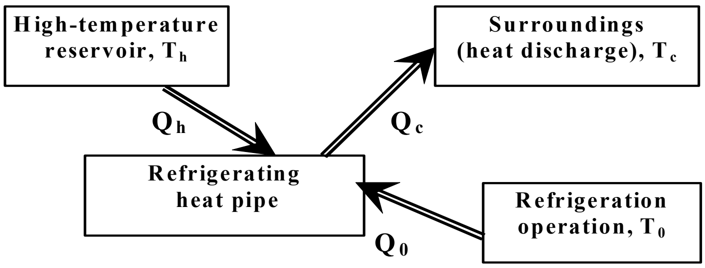

Figure 5. Operating levels of temperatures in RHP.

As an example, the maximum scale of capillary forces can be estimated when the minimum value of pore radius and surface tension of the particular working fluid are known. This last feature will depend on temperature level, the compatibility requirements, etc.

Assuming water as the working fluid and the minimum pore radius of one micron, the maximum scale of capillary forces will be equal to (1…1.5) 10^5 N/m^2. When the unit length will be estimated as 0.1…1 m, then the maximum scale of the density of capillary forces will be (1...1.5) *10^5 …10^6 N/m^3.

It could be suggested to make comparative analysis at the basis of the following formula:

$$Fv = \Delta P/L \tag{1}$$

The evaluation of the maximum achievable density of forces should be accomplished for the highest value of ΔP and minimum value of L.

Consider an opportunity of the cold generator design at the heat pipe base by using of hydrodynamical analysis. In the loops with the capillary pumping pressure drop can be developed by the surface tension forces, which scale essentially depends on the value of capillary radius, r_k:

$$F_v = 2\sigma/(r_k l) \tag{2}$$

Modern technologies of the capillary pumping yield to maintain a value of F_v ~10^6 N/m^3. High capacity values can be supplied with the osmotic pumping elements. The driving force for the process of the turn-out osmosis with ideal half-non-tight membrane can be defined by the following formula:

$$F_v = (1+a)RT\frac{x}{m} \tag{3}$$

As an example, for the water solution of salt with 2% concentration of $CaCl_2$, and temperature of 25^0 C, the value of ΔP can attain up to 20 MPa, when L = 0.02 m and $F_v \approx 10^9$ N/m^3.

Limited viability is distinguished in the gravitational refrigerating pipes, where $F = \rho' g$, and

$F_v \approx 10^4$ N/m^3. However, the density of this type of forces arranges a liquid circulation only. Thus, the operation of the refrigerating cycle is impracticable.

The flowing of liquid phase can be accomplished also by the effect of electric field. The density of the forces of such a nature should be determined by means of the next equation:

$$F_v = (\varepsilon' - 1)\frac{E^2}{L} \tag{4}$$

The density of forces in electrohydrodynamic pumps can be extended to $F_v \approx 10^4$ N/m^3. However, using of the cascade electrohydrodynamic pumps makes it possible to accomplish higher values of F_v.

For the rotating systems the centrifugal forces can be in charge for the working fluid circulation. The value of F_v will depend on liquid density, ρ', frequency of rotation ω, and difference of radiuses of the rotating surface, r_2 - r_1:

The combination of the mentioned physical phenomena presents a wide range of new possibilities for the application of heat pipes in TCS. The analogous estimation can be accomplished for the driving forces of a different physical nature and for new feasible applications of TCS. These predictions can be clearly noticed and summarised from the Table 2.

Table 2. The scales of the driving forces for different types of two-phase thermal control systems

#	TCS type	Equations used for the evaluation of density of the driving forces	The maximum scale of density of the driving forces
1.	TCS with the forced convection	$\Delta P_{\max} / L$	10^8 N/m^3
2.	Gravitational TCS	$\rho' g$	10^4 N/m^3
3.	Capillary heat pipe TCS	$2\sigma / (R_{\min} L)$	10^6 N/m^3
4.	Centrifugal TCS	$4\pi^2 \rho w^2 \Delta R$	10^8 N/m^3
5.	Electrohydrodynamic TCS	$(\varepsilon' - 1)\frac{E_{\max}^2}{L_{\min}}$	10^4 N/m^3
6.	Pulsating TCS (pulsating heat pipes)	$kg\rho'$	10^5 N/m^3

The demanded scale range of the driving forces for the various operating modes such as heat transfer, refrigeration, and power generation can be given by the following grade:

- Heat transfer: from 10^2 N/m^3 (gas natural convection) and further without limitations.
- Refrigeration: from 10^5 … 10^7 N/m^3 and above.
- Power generation: from 10^7 and above.

The consequent approach can be used for the estimation of the required scale of the driving forces for the heat pumping systems, and for the different energy-technological units.

With respect to such a semi-theoretical analysis the relevance of conventional refrigerating heat pipes, pulsating TCS, centrifugal refrigerating heat pipes, etc. can be based and foreseen.

Hence, the analysis of driving forces should be settled as a significant stage in the grading of refrigerating heat pipes. The next notable stage, reasonably, should deal with the settlement of the refrigerating operability. Naturally, the cold generating devices, using heat as the primary energy source, are well-known, and they are vapor–evaporating refrigerating machines, absorption refrigerating machines with mechanical pumping, absorption–diffusion refrigerating machines (without mechanical pumping), combined refrigerating machines, etc.

The plot of the experimental unit and the test results for the absorption-diffusion refrigerating heat pipes (ADRHP) are shown in Figure 6. Evaporator, condenser, dephlegmator, absorber, thermosyphon vapor-generator were ordered, respectively, inside a pipe with diameter of 30 mm and length of 450 mm. Internal surfaces of the evaporator and condenser were totally layered by capillary-porous structure. The evaporator has been coupled with the absorber by a long channel 8. The cooled vapor-gas mixture turns-out to the evaporator by the channel 7. The dephlegmator serves for purification of the refrigerant from the absorbent. The refrigerating heat pipe has been filled with the water-ammonia solution with ammonia concentration of 0.35. As an inert gas helium or hydrogen can be used. Thus, an absorption-diffusion cycle has been completed for the given RHP.

It has been observed that sloping of the RHP from 90^0 up to 60^0 practically does not worse a refrigerating capacity. The performance has been kept for inclinations up to 30^0. At the test temperature level of the cold source minus 15 ^oC the achieved value of the refrigerating capacity was about 5W. The vapor generation in the thermosyphon was accompanied by concentration fluctuations of mixture. It was caused by the temperature pulsations. The maximum amplitude of temperature fluctuations has been defined in the vapor-generator. The working temperature in the evaporator was equal to minus 15 ^oC. However, the minimum temperature in the evaporator of minus 22 ^oC has been achieved. The RHP technology, design and performance improvement (first of all, this concern to the condenser and absorber) can essentially raise the refrigerating efficiency and decrease temperature level at the evaporator.

The vapor-ejector refrigerating heat pipe (VERHP) comprises a vapor-generator chamber, where the low-potential heat is transferring from the outside source to the internal liquid working fluid and transforming it into the vapor. The last one is flowing through the vapor line to the vapor-ejector jet. It contributes to the decreasing of the pressure in the evaporating chamber. Such a decrease in the saturation pressure justifies decreasing of the respective saturation temperature up to the value that is considerably less than the temperature of the surroundings. Thus, the refrigerating operation mode can be completed.

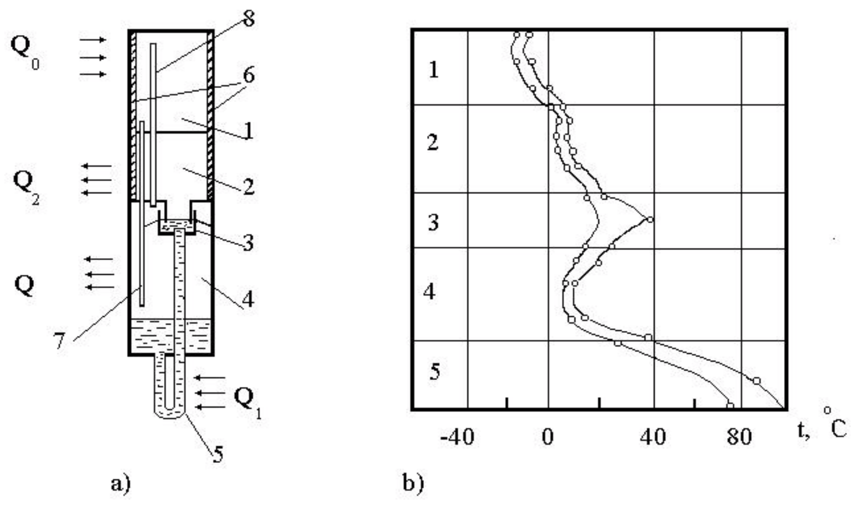

Figure 6. The absorption-diffusion RHP: (a) plot of experimental unit, (b) temperature distribution along the AD-RHP elements. 1- evaporator, 2 - condenser, 3 - dephlegmator, 4 - absorber, 5 - thermosyphon, 6 - capillary-porous structure, 7,8 - channels for the circulation the vapour-gas mixture.

The vapor generated in vapor-generating and evaporating chambers flows to the condensing chamber, where the latent heat of the vapor flow transfers through the walls of the chamber to the surroundings, and as a result the vapor condenses. By use of the capillary-porous structure, part of the condensed flow exhausts to the evaporating chamber. Similar, other fraction of this flow removes to the vapor generating chamber via porous structure.

The absorption refrigerating heat pipes can operate with both binary and tripartite working fluids. It is known that capillary-porous structures have been used in the refrigerating pipes, basically, to accomplish the flowing of the working fluid through the heat exchange surface and to enhance the heat transfer. The compactness of the refrigerating heat pipes contributes sufficiently to their performance. Surely, it depends on the appropriate selection of the type and parameters of the capillary-porous structure in use.

Goncharov *et al.* [16] reported results of the experimental modeling of the above mentioned system. The set-up allowed modeling of various VERHP designs: VERHP in the frequent operation mode of vapor-generator; VERHP with capillary pump on continuous operation mode of vapor-generator.

Before the operating the entire system has been vacuumed. Toluene has been selected as the working fluid, because it provides the functioning of vapor-ejector refrigerator at the temperature of the evaporating surface equal to or less than 0°C. The thermocouples have been located at whole surfaces of the test unit, and inside of some elements of the piping system. The visualization of experiments allowed designation of two basic operating modes of the evaporator: operating mode with unsteady phase interface, and operating mode with steady phase interface.

Authors of [17, 18] have discussed advantages of both possible applications of VERHP:

1. Heat pump TCS, appropriate for the aerospace conditions.
2. Automobile air conditioning system.

As an example, the ratio between the temperature drop $\Delta T_0 = T_e - T_o$ and the value of $\mu_2 = 1/COP = Q_g/Q_o$ for the optimal VERHP with respect to the space applications have been considered. This ratio for working temperature T_o= 173K could be approximated as $\Delta T_0 = 25 \log (Q_g / Q_o)$.

With accounting of the mentioned conditions, the operating modes for several temperature levels of the heat discharge have been analyzed:

- Thermoelectric refrigerating unit (with various numbers of stages at each temperature level).
- Combination of the vapor-ejector refrigerating machine with the loop heat pipe (with various numbers of stages, including a single-stage, two-stage and three-stage units; with a single-cascade and two-cascade designs).

Mathematical model for the estimation of the optimal values of the cycle pressures, ejector-ejector geometrical parameters, number of stages, etc. has been settled on the basis of the first law of thermodynamics. It was determined that for any feasible temperature conditions the application of VERHP design is more attractive than thermoelectric refrigerating units, including supposable combined designs. The outcomes of the basic thermogasodynamic analysis have been approximated by the amount of refrigerating operation temperature drop for the heat rates ratio, Q_h/Q_o. Such an approximation can be used for the estimation of the optimal temperature levels.

With respect to the appropriateness for the aerospace applications, as the primary optimum criteria one should consider the minimum mass. Hence, proposed approach considers a mass of the entire TCS as the goal function, which has been accounted for the value of so-called equivalent mass (its value was sized to the energy losses).

Present methodology supports analyzing of conventional TCS with the only one heat removal mechanism (VERHP, thermoelectric or passive), and combined systems having high temperature level unit and the low temperature level performed by VERHP, etc. For the last case the entire temperature drop, ΔT_0, could be divided into the "high" stage, ΔT_{o1}, and to the "low" stage, ΔT_{o2}. For the estimation of both optimal temperatures drops the following two equations can be presented:

$$\left[\frac{\partial(\ln \mu_2)}{\partial \Delta T_{o2}}\right]\left(T_r^4 - T_e^4\right) - 4T_r^3 = 0 \quad , \tag{5}$$

$$\left[\frac{\partial(\ln \mu_1)}{\partial \Delta T_{o1}}\right]\frac{\left(T_\gamma^4 - T_e^4\right)}{\mu_2}\left(1 + A_0\left(T_\gamma^4 - T_e^4\right)\right) - 4T_r^3 = 0 \quad , \tag{6}$$

where $A_0 = k_r \varepsilon_r \sigma_0 / \rho_r \delta_r$.

For the optimal thermoelectric units the value of μ_1 could be calculated as follows,

$$\mu_1=0.18\Delta T_{o1}-1. \tag{7}$$

Substituting equations for μ_1 and μ_2 into the system (6) is rearranging as,

$$\Delta T_{o2}=10.8\ln\mu_2 . \tag{8}$$

Thus, the following equations allow evaluation of an optimal radiator temperature and a value of μ_2, respectively:

$$T_r^4 - T_e^4 - 42.2T_r^3 = 0 \tag{9}$$

$$0.49 + A_0\left(T_r^4 - T_e^4\right)/\mu_2 = 0 \tag{10}$$

The last formula for actual values of A_0 gives $\mu_2 < 0$. It predicts an inexpedient application of the thermoelectric refrigerator at the considering VERHP design. It means that when any alternate passive system will have a temperature drop higher than the value 10-15 K, the application of VERHP can not be approved.

The similar theoretical analysis has been completed for the application of VERHP as the basic unit for the automobile air conditioner. It was established that use of the vehicle exhausting flow as the external thermal source is more effective than by means of the cooling system of motor. Furthermore, the optimal temperature drop values have been calculated for main heat exchangers, i.e. for the vapor-generator, condenser and evaporator. Finally, it has been proved analytically that water is the most appropriate working fluid for the analyzed VERHP with respect to the customary operating conditions of the air conditioning systems.

4.3 Refrigerators and refrigerating chambers with the heat transfer panels

Major types of domestic refrigerators contains two chambers: «cold» chamber and «freezing» one, and have been equipped with one stage vapor compressor. The "cold action" took place in the evaporator. A refrigerator thermal control system is based on the "relay" principle.

Normally, "turning" set orders to compressor to react consequently to the signals of temperature sensors. A temperature sensor could be positioned on the evaporating surface, inside "freezing" or "cold" chambers.

First case is the best method for the stabilization of evaporator temperature, but it was poorer for the stability control of the average temperatures of "freezing" or "cold" chambers. Second and third positions provide the best mode for the chambers' average temperature stabilization, but the energy losses are significantly bigger. The energy losses with the "cold action" generation depend on the ratio of vapor compressor "turning on" time τ_1 and "turning off" time, τ_2.

On the other hand, the value τ_1/τ_2 notably depends on the quality of thermal link between the surface of evaporator and "cold" chamber or "freezing" chamber volumes.

Conventional thermal links are based on the natural or forced air circulation in the internal volumes of the refrigerator. Forced convection is commonly used in "no frost" domestic refrigerators. However, natural and even forced convection in the domestic refrigerators do not ensure high heat transfer performance without undertaking of temperature drop values between internal volumes of "cold" and "freezing" chambers and evaporator surfaces. These temperature drop decreases could be achieved when the high-effective "thermal bridges" between the surfaces of evaporator and chambers would be established.

Heat transfer panels (HTP) could perform a significant assignment when their internal thermal resistance in the heat pipes or thermosyphons will be properly low. In this case the considerable decrease of energy losses might be possible. There are different designs of HTP. One of the original related engineering solutions was patented [19]. Two-phase HTP design is presented in Fig. 7.

A value of energy savings depends on the temperature sensor location, HTP surface value, thermal conductivity of the link surface, etc. Study of the two-phase HTP effect on thermal regimes of the dual-chamber of "no frost" domestic refrigerator has been completed on the basis of steady state analysis of thermal regime. Such an analysis results with the following equation:

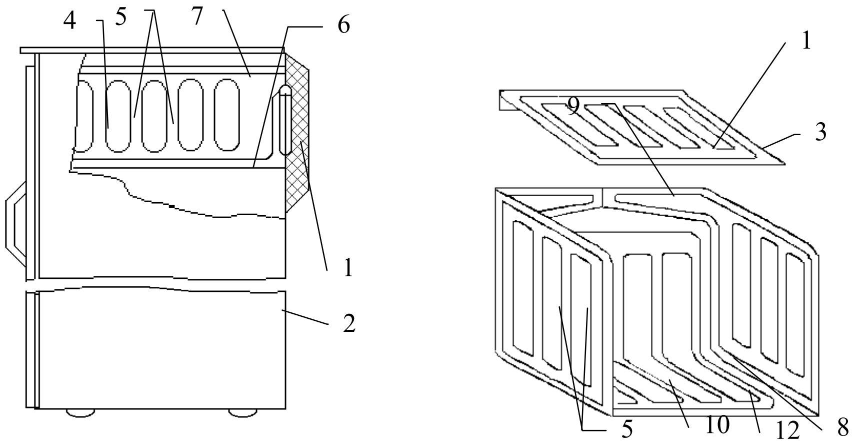

Figure 7. Absorption - diffusion refrigerator with panels and refrigerating box scheme.
1 - evaporator; 2 - refrigerating box back panel; 3 - vaporization - condensation panels; 4 - refrigerating box; 5, 8 - vertical channels; 6, 7, 9, 10 - lower and upper collectors; 11 - back part of refrigerating box; 12 lower part of refrigerating box.

$$t_s - t_{s0} = \frac{Q_0}{G_a c_{pa}} \left[\frac{1}{\exp(\alpha_{e0} F_{e0} / G_a c_{pa}) - 1} - - \frac{1}{\exp(\alpha_H F_p / G_a c_{pa}) - 1} - \frac{G_a c_{pa}}{k_C S_C} \right] \tag{11}$$

Here G_a, c_{pa} - mass flow rate and specific heat capacity of the circulated air in the refrigerator internal volumes; Q_0 - nominal cold productivity; α_{e0}, α_H, and k_C - heat transfer coefficients from evaporating surface to the air, from refrigerant inside HTP to the air and from evaporator to HTP through the contact, correspondingly;
F_{e0}, F_P, S_C - working surfaces: of evaporator, HTP and a contact zone, correspondingly; t_S, t_{S0} - average saturation temperature values in the evaporator with HTP, and the evaporator without HTP. For example, assuming the following parameters of "no frost" refrigerator: functional volume – 0.5 m^3; air mass flow rate - G_a~0.06 kg/s; α_{e0}= α_p= 10.0 W/m^{2}K; F_p= 3.6 m^2; S_c= 0.2 m^2 and k_c= 500 W/m^{2}K, and Q_o= 200 W, a saturation temperature difference given by formula (11) is equal to 13 K that corresponds to the energy savings up to 15÷20%.

However, under some set of conditions use of HTP could cause a negative effect, when the auxiliary thermal resistance (internal HTP thermal resistance) is higher than overall external thermal resistance drop (due to, for instance, an external heat transfer surface improvement). Hence, some parameters (k_c, S_c, F_p, etc.) designates the circumstances when the HTP application is irrational.

4.5 Cooling of electronics with thermosyphon collectors

Cooling of electronics is one of well-known applications of heat pipes and thermosyphons. However, one of the most important problems of such a usage is a lack of ability to gain a uniformity of the temperature fields inside the electronic units, blocks, and sets, especially in the entire electronic system only by using of conventional heat pipe types.

To solve a mentioned problem, it was suggested to apply a new type of heat pipes and thermosyphons, named "collector heat pipes" and "collector thermosyphons" (CTS), correspondingly [20, 21]. Basic concept of design is integration of internal volumes of separate heat pipes or thermosyphons into the mutual volume with a formation of a shared heat removal zone. Simultaneously, both geometrical parameters and internal two-phase mass transfer identified for each level of the accumulation design ensure basic technological regulations, when the thermosyphon collectors represent as effective heat transfer links both essential construction elements, where the tools, nodes, and blocks of electronic systems could be positioned. Consequently, typical engineering solutions could be classified into:

1. Collector thermosyphons of the assemble level named as “primary” (Fig.8 a).
2. Collector thermosyphons of the block or system level named as “secondary”(Fig.8 b).

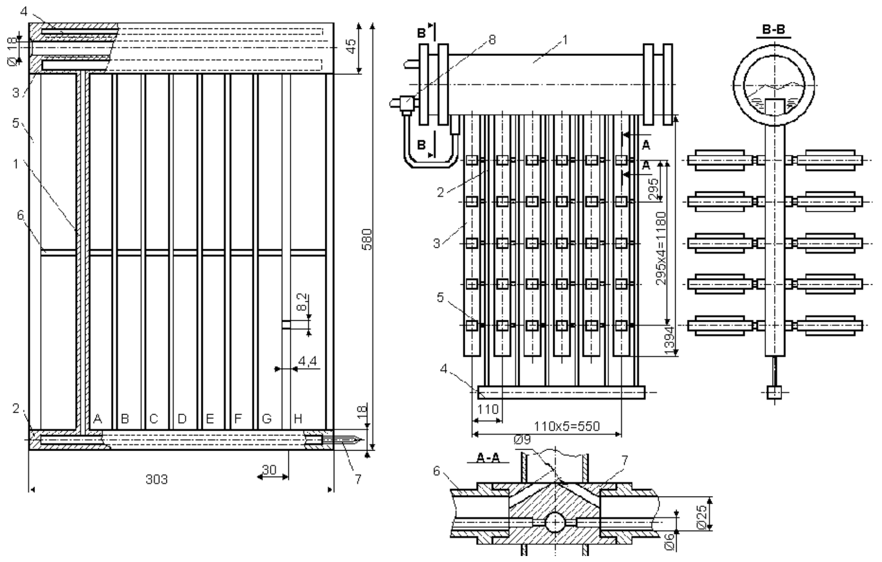

Figure 8. a) "Primary" collector thermosyphon (assemble level):
1 – evaporative channels ("A", "B", "C", "D", "E", "F", "G", "H"); 2 –lower collector; 3 –upper collector; 4 – finned condensation surface (condenser); 5, 6 – frame elements; 7 – filling tube.
b) "Secondary" collector thermosyphon, (panel level):
1 –shell and tube condenser; 2 – movable down tube; 3 – lifted up tube; 4 –lower collector; 5, 6 – evaporative channels; 7 –joint of the evaporative channels with tubes 2 and 3; 8 – regulator of condensate level; 9 –plates for placing of electronics.

Collecting node of each CTS represents a condenser, from where the heat output is transferred to other CTS, or to others heat removal elements, or to the environment. In such a case, a high quality of a “heat joint” between the heat-reject surfaces and the corresponding heat loads of CTS is extremely important. Using of cylindrical (piping) joint surfaces is appropriate for capable cooling systems of electronics on the basis of primary and secondary CTS, as well.

A number of such elements could be varied up to several hundreds, and connecting the matching number of electronic units on the set (container) level. Here, internal problem of reaching the optimum heat routine conditions depends on the precise arrangement of two-phase heat and mass transfer processes within the developed natural circulation loop, when an external problem is related as to the law values of the “heat joint” thermal resistances both to the internal heat resistances’ reduced in the primary CTS.

Research tests discovered this application of CTS designs made it possible to increase the heat density of internal volumes of sets, containers, etc. in tens times, i.e. to minimize the dimensions of electronic devices; to reduce the temperature drops in 1.5 –2 times, and to achieve the high uniformity level of temperature fields as for the primary CTS both for the secondary one, i.e. to improve the reliability of electronics significantly.

4.6 Thermosyphons for soil freezing. Heat pipe snow melting & deicing systems

The soil freezing is a well-known thermosyphon’s application field with a quite fruitful expertise, developed both theoretically and experimentally, and worthily referred. This heat pipe technology successfully developed in the cold climate regions (Russia, Northern China, Alaska, etc.).

The snow melting & deicing systems have been emerged only during last decades in Japan. Data on peculiarities of this heat pipe technology were presented by Adachi, Sugihara *et al.* [22]. The authors have pointed out that electric heating use in the heat pipe snow melting and deicing systems affected a range of problems such as dry-up of the underground water, ground subsidence, etc. As a result, using of the low-potential heat source was identified as more reasonable. Authors suggested utilizing of thermal spring waters with the application of long corrugated type heat pipe. The flow chart of this system is shown in Fig. 9.

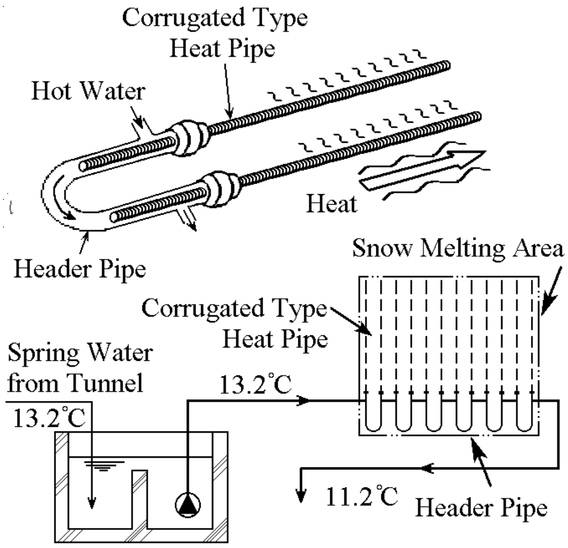

Figure 9. Flow chart of snow melting system with tunnel spring water as a heat source.

The encouraging data on heat pipe snow melting and deicing systems utilized a geothermal heat, solar energy, drainage heat and waste heat from the steam generator as the heat sources were presented in the same paper. The running costs of the snow melting system that uses the geothermal heat as the heat source were very low, but the capital costs were extremely high. Thus, upcoming focus of this system improvement is reducing of the initial costs. The authors noticed that quite a sufficient part of high quality energy was unused in the surroundings.

4.7 Heat pipe technology for cold storage design

The accumulation of cold has a valuable meaning for the acting periodically food processing technologies such as milk industry with precise temperature control to ensure the chilled milk storage at 0 ^{o}C, various chemical technologies with crystallization processes when phase transient temperature smaller than ambient temperatures, etc. Thermal storage systems provide sufficient energy savings in air conditioning performance that was determined by the following settings:

- air conditioning load varies with the season;
- difference in the daily variations of the air conditioning load was large;
- air conditioning time zone of the building was diversified.

Thermal storage system made it possible to reduce a capacity of refrigerating machines, to cut the peak of the cooling load, and to utilize the midnight power consumption, which was cheaper than in day time by operating the refrigerating machine for a long time a day. On the other hand, a large amount of chilled water was required as the heat carrier for daily cooling. Place needed for setting of the water storage tank, and manufacturing expenses represented obstacles avoiding the wide use of the thermal storage systems. One of the possible solutions of such kind of problem is ice thermal storage system, which utilizes the latent heat of ice melting, considered by authors of [23÷25].

Concept of heat pipe application for cold storage systems [23] is based on the following issues:

1. The HP use in the cold storage tank assigns filling of the tank's internal volume with the maximum ice concentration in comparison to the conventional position of the evaporator at the tank internal surfaces.
2. The HP with small external diameters allocates a low thermal resistance between phase surface and HP freezing surface.
3. The above mentioned advantages yield decrease of the tank volume and sufficient energy savings.

Other ways of improvement of cold storage performance with the heat pipe technology offered in [24, 25]. Authors suggested HP use for the energy savings in ice formation process. This is a typically periodical process that comprises the following stages: ice block formation, ice thin layer melting near the HP outer surface, and ice block separation from HP with submerging in the water volume. Ice formation was studied both experimentally and analytically. Test procedure, mathematical model and its numerical analysis were discussed in references [24, 25]. Authors concluded that heat pipe technology application for the periodically acting ice generators improved the ice formation with minimum energy losses and low costs, and made available a compact cold storage tank design, i.e. moderates material and fuel losses considerably.

Heat pipe technology application for solid sorption systems was studied during last years by Prof. Vasilliev's research team. Some significant results were presented in papers [26, 27].

4.8 Heat pipe technology in the air conditioning systems

The humidity control has always been one of the main concerns of modern buildings that gave rise to need of air conditioning (AC) systems with both dehumidifying and cooling capabilities. If the dehumidifying ability of the designed AC system would be less than needed, the humidity will rise. The entering air subcooling usually solves this problem with humidity reduction, before the necessary temperature level. It leads to the extra energy consumption.

Therefore, warning of reduction of the air humidity by the corresponding way became clear. Low humidity could be hardly achieved by using such an auxiliary system as liquid or solid type dehumidification methods. The existed experience shows that this way is connected with the considerable complications. The application of heat pipe heat exchanger (HPHE) has advantages of easier installation, lower costs, less maintenance and much less energy consumption.

The scheme of the HPHE application in AC is given in Fig. 10. Khattar [28, 29] has tested the dehumidification improvement of an AC unit with the HPHE. The tests were performed with return air at various relative humidity values of 35% ÷ 55%, and HPHE effectiveness of 0.47. HPHE positive effect on AC system parameters has been confirmed. It was discovered also that HPHE has very large potential to be used in low humidity places where further reduction of humidity is required. This evaluates HPHE as very perspective devices for improvement of AC systems.

The next HP application is connected with the AC system as for office apartment both for office equipment air-cooling. The AC individual system scheme is presented in Figure 11. The important problem appeared, when these systems were built. It was connected with a necessity of high heat transfer coefficient values in two-phase thermosyphon system, when the heat fluxes were low. This problem could be solved by the application of refrigerants as working fluids in HP simultaneously with the arrangement of boiling in narrow slits of the heat inputs.

The modeling and test [30] proved positive results for such systems. It has been confirmed that individual cooling systems function was satisfactory for all operating conditions.

The illustrative testing of the HPHE application as the base for AC systems control was presented in [31].

The principal idea coincided with this one given in Figure 10. The final results were presented in Figure 12 a, b. It is seen that the above-mentioned HPHE application gave the considerable reheat recovery in the AC systems.

The other idea of HPHE use as regenerative dehumidifier was suggested by Dinh in patents [32-34] and obtained the wide and popular consideration in the manufacturing of AC units [35, 36]. The similar approach to the dehumidification problem in AC with HPHE has been adopted among other suppliers, for example "Greenheck" [36].

The modern tendency in the AC energy consumption diminishing is connected with the successful solving of dehumidification problem. The HPHE application helps to resolve this query, but only partially.

That is why, recent years research activity of professionals was concentrated into the advances of the dehumidification technology, and particularly of the desiccant systems as one of the most capable solutions, which is based on the water absorption and sorption processes application. The last ones are required a considerably low-temperature heat input. The absorption process, as it is known, is accompanied with the heat emission.

Therefore the problem of the profitable energy savings in the desiccant systems is notably connected with the improvement of the heat regeneration process between the supply and exhaust airflow.

It was shown in papers [13, 37, and 38], that the HPHE application in these conditions provides significant energy savings.

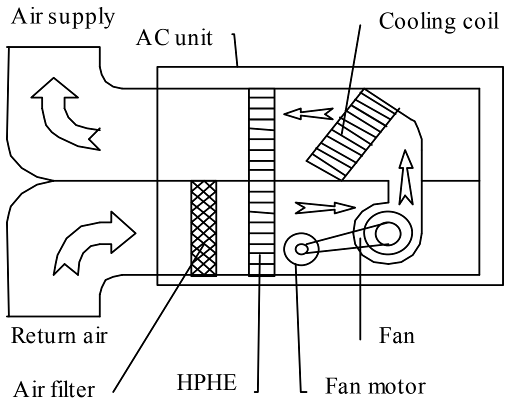

Figure 10. Schematic of the HPHE application in air conditioner

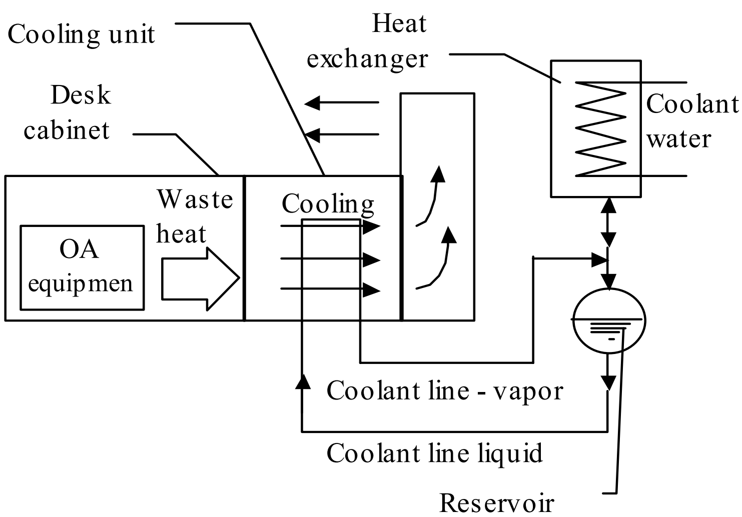

Figure 11. Schematic of individual office apartment cooling system

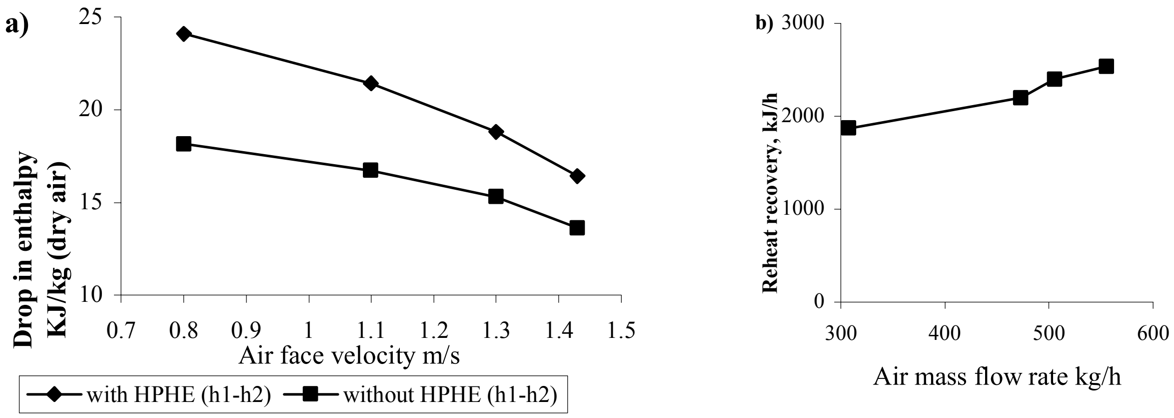

Figure 12. a) Enthalpy change for the air passing through the cooling section *vs.* air face velocity. b) Reheat energy recovery by the condenser of the HPHE *vs.* air mass flow rate.

Indirect evaporative cooling (IDEC) systems represent a significant advance in the use of heat pipe technology. In operation, air passing through the exhaust site of the IDEC cooler is sprayed with water. Evaporation occurred on the fins of the heat pipe, raising the moisture content of only the exhaust air. The heat of vaporization needed to evaporate the water on the exhaust side came from the supply side in surfaces, cooling the incoming air. The result is a cycle in which supply air was constantly being dry-cooled, thereby reducing the size of, or something eliminating, and the need for mechanical AC equipment. In either case IDEC systems permit achieving significant reductions in electric power consumption. For instance, a conventional air-cooled mechanical system utilizing a compressor consumed 1.45 kW per ton of cooling capacity. Thus, under the most adverse conditions, the IDEC system could cut power consumption by 20%. Or when wet bulb temperatures are favorable, power costs could be slashed by as much as 50%. Additional savings could often be achieved during colder periods through the recovery of heat in exhaust air. Actual power cost reductions would vary with local power prices and demand penalty charges. There are three basic ways in which IDEC systems could be utilized (Fig. 13):

- The first is the typical shop or manufacturing area in which ventilation air was to be cooled about 18-27 °C, without increasing of the moisture. In this situation, an IDEC system was installed in the area's ventilating ducts to lower the temperature of the incoming air supply.
  Again, the same system could be utilized in the heating season to recover normally wasted exhaust air heat. The compactness and simplicity of the system assured, that disturbance of existing ventilating ducts was minimal, thereby helping to keep first cost low. Operating costs were equally low since water supply and small pump motor were all that's required.
- The second commonly used configuration combines an IDEC system with a conventional air conditioning or refrigeration system. In such an arrangement, building exhaust air was frequently at 24- 25 °C with humidity at 50%. The lower relative humidity of the air coming out of the building greatly increased the ability of the exhaust airstreams to evaporative water. Thus by spraying the exhaust side of the IDEC system, substantial make up air cooling was achieved when the exhaust air temperature was lower than ambient air temperature. Make up air precooling could be sufficient to reduce mechanical refrigeration load.
- The third IDEC system arrangement is commonly utilized in drier climates locations where average humidity is low. This system incorporated an IDEC module with a direct evaporative cooler (swamp cooler) to cool the space as well as to add moisture.

Mathur [12] grouped evaporative cooling systems that might be used with thermosyphon loop heat exchangers, in next three categories:

- Media pads, utilized a wetted media, through which the exhaust air passed, producing evaporative cooling.
- Spray type, which was a hybrid air washer/cooler that used a spray system to wet the media elements.
- Non-spray type, where water was introduced from the top through a header or trough and flown down by gravity to the sump at the bottom on the unit.

Author simulated performance of a thermosyphon loop heat exchanger with media pad evaporative cooling. The analysis concluded that by using evaporative cooling, precooling of the air was increased, resulting in lower cooling operation and equipment costs.

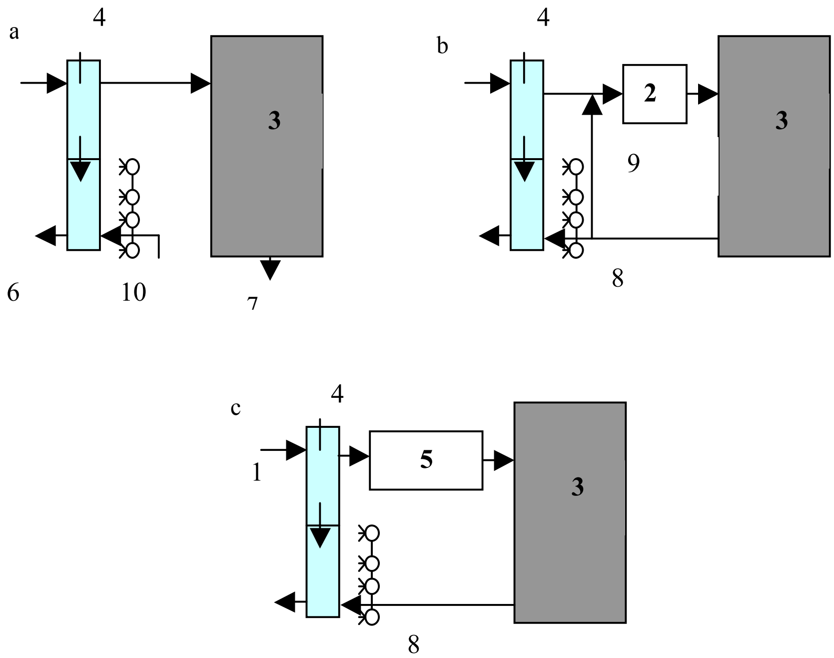

Figure 13. Indirect evaporative cooling. a) For ventilation using ambient air; b) coupled with standard air conditioning; c) coupled with evaporative cooling.
1- make up air, 2 – a/c, 3 – area to be cooled, 4- heat pipe heat exchanger, 5- evaporative cooler, 6- exhausted air, 7 –exhaust, 8- return air, 9- recirculated air, 10 – ambient air

Hill and Lau [39] reported test results that were aimed to evaluate the use of heat pipe heat exchangers (HPHE) in conjunction with conventional AC. HPHE applied to the single-path air conditioners provided better moisture control than dual-path configurations. It was obvious at several of the locations that match moisture control in supermarkets. The success in maintaining low indoor dew-point temperature of the total latent load was a result of infiltration. The main evaporator treated infiltration loads. The application of a HPHE on single-path systems was adequate for treating outside air latent loads and more effective at addressing infiltration latent loads than the dual-path designs. Also, the HPHE on single-path systems affected the total cooling capacity rather than just that associated with the outside air system. Overall moisture control increased as a result. Test results allowed estimation of air-conditioning efficiency and the effect of the HPHE on cooling efficiency at a number of sites. These tests clearly proved that operation with the HPHE improved the latent efficiency. All tested sites showed a considerable decrease in refrigeration energy use with reduced indoor dew-point temperature. Many of the sites were effectively controlled to indoor dew-point temperatures below conventional values through the application of HPHE.

Authors concluded that HPHE should be most successful on parallel-configured dual-path systems.

5. CONCLUSIONS

- Refrigeration is one of the most attractive fields for the successful application of the heat pipe theory [40, 41].
- Heat pipe technology provides considerable improvements in engineering, environmental, technological, and economical performance of refrigeration, cooling and air conditioning systems.
- A number of new applied fields of heat pipe technology are connected with using of loop heat pipes.
- Different ways of HPHE application for the solution of the dehumidification problem in air conditioning should contribute significantly with decreasing of power consumption, i.e. increasing of energy savings, and improvement of equipment performance and reliability.

NOMENCLATURE

a – degree of dissociation
F_v – density of driving forces, N/m^3
k_r – specific energy expenditures mass equivalent, kg/W
Q_0 – evaporator cold output, W
Q_g – vapor generator total power, W
L – half-length between high and low-pressure zones, m
ΔP– pressure drop, Pa
r – radius, m
T – absolute temperature, K;
ΔT– temperature difference, K;
x – concentration of the dissolved substance

Greek Symbols

μ = (1 / COP)
δ_r – radiator thickness, m;
ρ – density, kg/m3;
σ_0 – Stephan-Boltsman constant, W/K;
ω – frequency of rotation, 1/s

AC – air conditioning;
ADA - absorption-diffusion apparatus;
ARM - absorption refrigerating machine;
CTS - collector thermosyphon;
HP - heat pipe;
HPHE - heat pipe heat exchanger;
HTP - heat transfer panel;
IDEC - indirect evaporative cooling;
LHP - loop heat pipe;
RHP - refrigerating heat pipe;
TER - thermoelectric refrigerator;
VCRM - vapor - compression refrigerating machine;
VERM - vapor - ejector refrigerating machine;
VERHP - vapor - ejector heat recovery unit.

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