Design Calculations of the Limiting Characteristics of Heat Pipes for Cooling Active Phased Antenna Arrays

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Abstract—The article provides an algorithm for calculating the limiting characteristics of heat pipes for cooling active phased antenna arrays at a given saturation temperature. The maximum transmitted power is determined taking into account the limitations of the heat pipes operation by the capillary limit, by boiling (transition to film boiling, boiling limit), by the sonic limit at which the speed of steam reaches the speed of sound (sonic limit), by the entrainment of droplets liquid coolant from the surface of the wick with a counter flow of steam (entertainment limit) and viscous limit, which is realized at low temperatures (viscous limit).

It is shown that an increase in the thickness of the wick and its porosity may be necessary to increase the capillary limit of heat pipes, while an increase in the thickness of the wick increases the thermal resistance of the tube and, accordingly, can lead to overheating of the cooled elements. Based on the above algorithm, design calculations for two types of heat pipes have been carried out. The dependences of various limits of the heat pipe on the operating temperature are plotted.

Based on the above algorithm, calculations were performed for two types of heat pipes.

Keywords—Heat pipes, limiting characteristics, active phased antenna arrays, saturation temperature.

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1. Introduction

ACTIVE phased antenna arrays are widely used in modern ground-based, airborne radar systems. Cooling of active phased antenna arrays at a high heat flux density (more than 1000 kW/m^2) is especially important when moving to a higher frequency range, when the dimensions of the element base and, accordingly, modules decrease, and the heat release practically does not change [1],[2],[3],[4],[5],[6],[7],[8],[9],[10],[11],[12],[13],[14]. An increase in the efficiency of heat removal can be achieved, in particular, with the help of liquid cooling systems, consisting of heat pipes, in a strictly limited volume for their placement, which is the most important condition for the creation of modern antennas. Designing heat pipes traditionally comes down to calculating the maximum power transmitted by them at a given saturation temperature T_0. The saturation temperature is determined during the manufacture of heat pipes by the degree of filling with the coolant and the vacuum implemented in it and, in fact, is the operating temperature of the heat pipes. The maximum transmitted power is determined taking into account the limitations of the heat pipes operation by the capillary limit, by boiling (transition to film boiling, boiling limit), by the sonic limit at which the speed of steam reaches the speed of sound (sonic limit), by the entrainment of droplets liquid coolant from the surface of the wick with a counter flow of steam (entertainment limit) and viscous limit, which is realized at low temperatures (viscous limit). These techniques are described, for example, in [15],[16],[17],[18],[19],[20],[21],[22],[23] and were involved in the design of heat pipes of various types [24],[25],[26],[27],[28],[29],[30],[31],[32],[33],[34],[35],[36]. Initially, these techniques were applied to pipes with circular cross-section [37],[38],[39],[40],[41],[42],[43],[44],[45],[46],[47],[48],[49],[50],[51],[52],[53], but later they were also used to evaluate heat pipes of flat-oval and rectangular cross-section [54], [55], [56], [57], [58], [59], [60], [61], [62], [63], [64], [65], [66], [67], [68], [69], [70], [71], [72], [73], [74], [75], [76], [77], [78], [79], [80] by recalculating the area of the rectangular cross-section of the wick and the steam flow zone (parawire).

2. Algorithm for calculating the limiting characteristics

The calculations are reduced to calculating the limits of the heat pipes. The viscous limit of the transmitted power is calculated based on the ratio:

$$Q_{viscous} = \frac{d_v^2 h_{fg}}{64 \mu_v l_{eff}} \rho_v p_v A_v \quad (2)$$

where $l_{eff} = \frac{l_e}{2} + l_a + \frac{l_c}{2}$ - effective heat pipe length, h_{fg} - specific heat of vaporization, p_v - saturation pressure, ρ_v - vapor density and μ_v - dynamic vapor viscosity, $A_v = a_v b_v$ – cross-sectional area of rectangular steam pipe with width a_v and height b_v.

The maximum power transmitted without reaching the vapor velocity of the sound velocity is determined by the relation:

$$Q_{sonic} = A_v \rho_v h_{fg} \sqrt{\frac{\gamma_v R_v T_0}{2(\gamma_v + 1)}} \quad (2)$$

where γ_v — adiabatic exponent, R_v = 461 J/(kg K) – specific gas constant of water vapor.

The entrainment limitation is determined by the ratio:

$$Q_{entraiment} = A_v h_{fg} \sqrt{\frac{\rho_v \sigma}{2 r_h}}, \quad (3)$$

where σ – water surface tension coefficient, r_h — hydraulic radius of wick surface pores.

The limitation on the capillary limit, which is often the most significant, since it determines the operation of heat pipes in the temperature ranges most important for technical devices, is determined by the ratio:

$$Q_{capillary} = \frac{\left(2\sigma / r_{eff}\right) - \rho_l g l_{eff} \sin\theta}{\dfrac{32\mu_v l_{eff}}{d_v^2 A_v \rho_v h_{fg}} + \dfrac{\mu_l l_{eff}}{K A_w h_{fg} \rho_l}} \quad (4)$$

where θ - the angle of inclination of the heat pipe relative to the horizon, μ_l, ρ_l - viscosity and density of water, $A_w = a_w b_w$ – cross-sectional area of a wick layer of width aw and thickness bw, r_{eff} - effective pore radius of the wick, K- wick permeability calculated by the Kozeny-Karman formula:

$$K = \frac{d^2 \varepsilon^3}{122\left(1-\varepsilon\right)^2}, \quad (5)$$

where $d_{eff} = 2 r_{eff}$ – effective pore diameter, ε - wick porosity.

Refined calculation formulas for steam pressure losses in pipes with a steam pipeline with a rectangular cross section were proposed in the works [45],[46],[47],[48]:

$$\Delta P_v = \frac{48 \mu_v l_{eff}}{\rho_v a_v b_v D^2 h_{fg} \left(1+s\right)^2 \left(1 - 0.63 s \tanh \dfrac{\pi}{2s}\right)} Q_{capillary} \quad (6)$$

where $s = b_v / a_v$ – steam line size ratio, D=2avbv/(av+bv) – hydraulic diameter of parapipe.

The boiling limit is determined by the ratio:

$$Q_{boiling} = \frac{A_v k_{eff} T_0}{b_w h_{fg} \rho_v} \left(\frac{2\sigma}{R_n} - \frac{\sigma}{2 r_{eff}} \right), \quad (7)$$

where $R_n = 2 \cdot 10^{-6}\,\mathrm{m}$ – critical radius of the vapor bubble nucleus; k_{eff} - effective coefficient of thermal conductivity of a saturated wick, determined through the thermal conductivity of the frame (copper) k_s and the thermal conductivity of water k_l from the ratio:

$$k_{eff} = \frac{\beta - \varepsilon}{\beta + \varepsilon} k_l \ ; \ \beta = \left(1 + \frac{k_s}{k_l}\right) \Big/ \left(1 + \frac{k_s}{k_l}\right)$$

Based on relations (1) - (5), the limits of operation of heat pipes are determined. For a given temperature, all these limits are calculated and the smallest of them determines the maximum power transmitted by the pipe, and the mechanism that prevents a further increase in the transmitted power.

From a design point of view, it is also important to be able to estimate the heating temperature in the area of heat sources (this temperature is limited by the requirements for equipment - transistors, amplifiers, etc.). The approximate value of the temperature difference between the condensation zone and the evaporation zone can be found through the total thermal resistance of the heat pipe R_{ov} and the given power of the heat sources Q according to the formula: $\Delta T = R_{ov} Q$.

Accordingly, the temperature in the region of the evaporator zone can be estimated by the formula:

$$T_e = T_0 + \Delta T / 2 = T_0 + R_{ov} Q / 2, \quad (8)$$

The total thermal resistance of a flat heat pipe is determined on the assumption of zero effective thermal resistance of the steam pipeline zone (which is negligible compared to the thermal resistance of the wall and wick) from the ratio:

$$R_{ov} = 2 R_{e,c} + \frac{1}{\dfrac{1}{2\left(R_s + R_w\right)} + \dfrac{1}{R_{w,s}}} \quad (9)$$

$R_{e,c} = h_{e,c}/k_{e,c}$ contact thermal resistance realized between the source (s) of cooling and the surface of the heat pipe, calculated as the ratio of the thickness of the thermal paste layer $h_{e,c}$ its thermal conductivity $k_{e,c}$; $R_s = b_s/k_s$ – thermal resistance of the heat pipe wall in the direction of its thickness, $R_w = b_w/k_{eff}$ – thermal resistance of the heat pipe wick in the direction of thickness; $R_{w,s} = L / (A_w k_{eff} + A_s k_s)$ – total thermal resistance of the wall and layer of the heat pipe wick in the direction of its length L, $A_s = a_s b_s$ – cross-sectional area of the heat pipe wall with width a_s and thickness b_s.

Thus, for a given length L, the effective thermal resistance of the heat pipe is, in fact, determined by the properties of the wick and the ratio of the thicknesses of the wick, the body wall and the height of the parawire zone. These parameters, as mentioned above, are also determined from the design calculation when determining the limitations of the heat pipe operation in terms of capillary, sound, etc. limits.

In fact, there are two opposite effects:

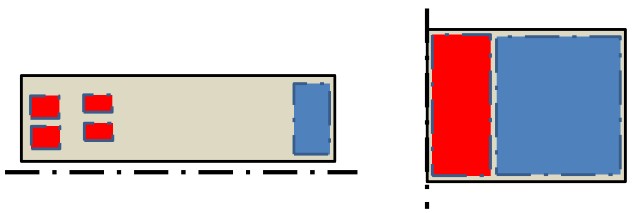
1. Increasing the thickness of the wick and its porosity may be necessary to increase the capillary limit of heat pipes.

2. At the same time, an increase in the thickness of the wick

increases the thermal resistance of the tube and, accordingly, can lead to overheating of the cooled elements.

As a result, it becomes necessary to simultaneously increase the thickness of the wick, the thickness of the walls, the height of the steam line zone (which decreases with increasing thickness of the wick), and, as a result, the total thickness of the heat pipe increases, which may be unacceptable. Also, by increasing the thickness of the wick, other limits, for example, on boiling, can "work". Preliminary calculations should make it possible to assess the fundamental possibility of using a heat pipe in a given design of active phased antenna arrays and select the optimal parameters of the heat pipe [12], [13].

The design will be carried out for two types of heat pipes (Fig. 1). The integral heat dissipation in the first case (Fig. 1a) can be up to 50 W, in the second - up to 80 W. In the second variant (Fig. 1b), the design must be performed at a power of 40 W, taking into account the symmetry of the heat pipes in the direction of heat transfer.



a) b)

Fig. 1 Designed options for heat pipes for housings operating, respectively, for heat transfer and heat distribution.

The main task of the calculation is to assess the required thickness of heat pipes and the parameters of the wick (thickness, porosity, pore size) to ensure their performance under conditions of an inclination of 55 degrees to the horizon. The maximum allowable thickness of heat pipes for option 1 (Fig. 1a) is H = 3.5 mm, and for option 2 (Fig. 1b) the maximum thickness is H = 2 mm. This thickness, respectively, is the sum of two thicknesses of the copper walls b_s = 0.4 mm, the thickness of the wick, which we will initially take equal to b_s = 0.4 mm, and the height of the steam line zone, which we will set initially equal to $b_v = H - 2\ b_s - b_w$ = 2.3 mm or 0.8 mm. The pipe width is $a_w = a_s$ = 70 mm in the first version and $a_w = a_s$ = 44 mm in the second. Taking into account the presence of "columns" of the wick inside the pipe, providing the pipe collapse stiffness, the effective width of the para-wire zone is less than the external overall dimension and is $a_v = 0.6\ a_w$. The length of the evaporator zone is l_e = 100 mm (1 option) and l_e = 43 mm (2 option, taking into account symmetry). Condenser zone length l_c = 60 mm for both options. The length of the adiabatic pipe section in the first version is $l_a = L - l_e - l_c$ = 350 – 100 – 60 = 190 mm. In the second version, there is no adiabatic section, and the pipe, in fact, works as a heat-distributing base. The design will be carried out for the most difficult case of the pipe operation with its inclination of 55 degrees relative to the horizon, with the location of the evaporator zone above the condenser zone.

We will consider four options for the operating temperature of the pipe (saturation temperature) T_0 = {40, 50, 60, 70} ºC, for which we will determine the limits of work by the transmitted power of heat release. For both pipe options, we will assume a uniform heat supply in the evaporator area (generally speaking, there are several discrete sources in this area, a refined calculation for which is presented in the next section of the report). In the calculations, we will take into account the dependence of some characteristics of steam and water on the operating temperature. First, the vapor saturation pressure for the indicated temperatures is: p_v = {7400, 12300, 19900, 31100} Pa. Vapor density ρ_v = {0.051, 0.083, 0.13, 0.198} kg/m^3, specific heat of vaporization h_{fg} = {2406, 2382, 2358, 2333} kJ/kg, surface tension of water σ = {0.069, 0.067, 0.066, 0.064} N/m, water viscosity μ_l = {0.00065, 0. 00055, 0. 00047, 0. 0004} Pa s. Steam viscosity μ_v = 0.000011 Pa s, vapor adiabatic exponent γ_v = 1.3, density of water ρ_l = 985 kg/m^3. Effective pore radius in the volume and on the surface of the wick r_{eff} = 0.05 mm and porosity ε = *0.6*. Thermal conductivity of water k_l = 0.65 W/(m K) and copper k_s = 380 W/(m K). The thickness and thermal conductivity of the thermal paste required for evaluating the contact thermal resistance are taken equal to $h_{e,c}$ = 0.1 mm and $k_{e,c}$ = 0.2 W/(m K).

3. Calculation results for the first variant of the heat pipe

The results of the design calculation (the dependence of the various limits of the heat pipe on the operating temperature) for the first version of the pipe are shown in Fig. 2. For the selected pipe parameters, the capillary limit and the boiling limit can be decisive, which determine the possibility of liquid returning to the evaporation zone due to capillary forces in the wick. Accepting these limits can substantially improve the quality of calculations. All other limits turn out to be higher and are not realized. The use of the calculation formula (5) does not significantly clarify the value of the capillary limit. The values of the capillary limit and the boiling point for cases of horizontal and inclined pipe arrangement are shown in Fig. 3a. In the horizontal position, the chosen structure of the pipe makes it possible to transmit the specified 50 W without reaching the limit (in this case, boiling) at an operating temperature of about 50°C, although in an inclined position the capillary limit limits the operation of the pipe to a power of 25 W (Fig. 3b). To increase the capillary limit of the heat pipe, it is proposed to increase the thickness of the wick layer by a factor of 1.5 to b_w = 0.6 mm and to decrease the effective pore size by half to r_{eff} = 0.025 mm.

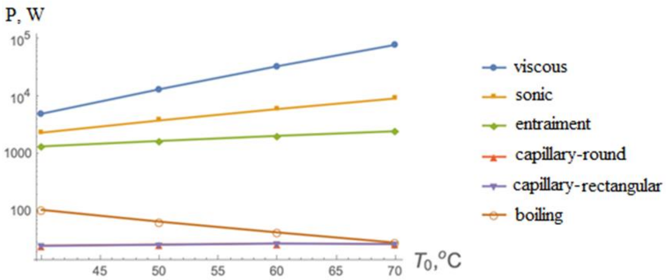


Fig. 2 Dependence of the calculated limits of pipe operation on the set saturation temperature.

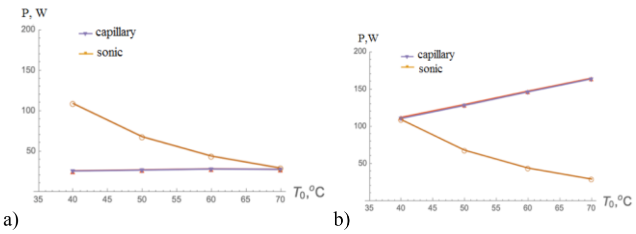


Fig. 3 Boiling limits for heat pipe and capillary, depending on the operating temperature for the location of the pipe at an angle (a) and horizontally (b).

In this case, according to the calculation, it is possible to achieve a transfer of 50 W when the pipe is tilted by 55 degrees at a saturation temperature of 45°C, which corresponds to a saturation pressure inside the tube of ~ 10 kPa (Fig. 4). In this case, the maximum heating in the region of heat supply, determined by expression (9), is about 50°C. This value is acceptable, however, it is underestimated and requires clarification, since heat is supplied locally at small areas, and the design calculation sets the average flow uniformly supplied to the entire evaporator zone. Therefore, the heat conductivity results in equal temperature in the region of heating.

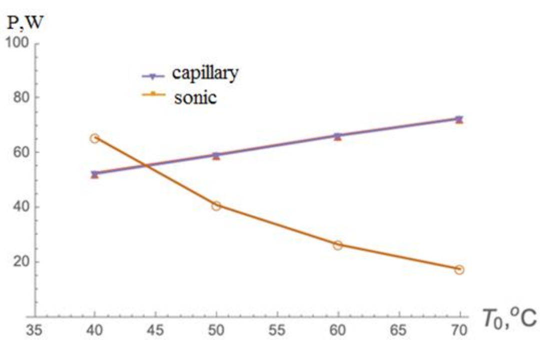


Fig. 4 Capillary limit and boiling limit of the heat pipe, depending on the operating temperature for the case of an inclined pipe arrangement and changed wick parameters.

4. Calculation results for the second variant of the heat pipe

The calculation of the operating limits for the second version of the pipe with the initially selected parameters is presented in Fig. 5 and Fig. 6.

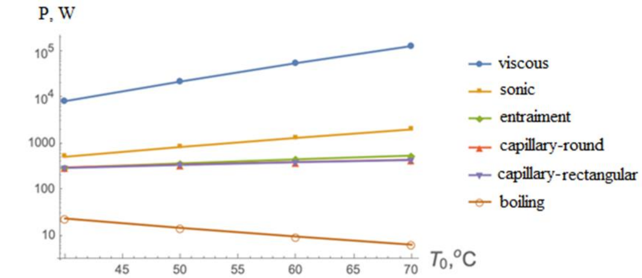


Fig. 5 Dependence of the calculated operating limits of the heat pipe for the second option on the set saturation temperature.

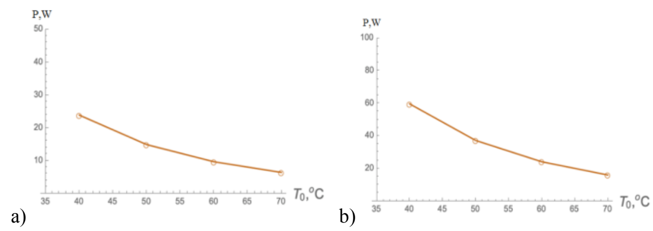


Fig. 6 The boiling limit of the heat pipe for the initial values of the parameters (a) and for the optimized version (b).

Here, it also turns out to be not essential to use a refined solution for a rectangular section of the para-conduit channel and also, it is necessary to change the parameters of the heat pipe to increase the limits of its operation. In this case, the operating limit of the heat pipe is determined by the film boiling phenomenon, and it does not depend on the pipe orientation (see formula (7)). The capillary limit turns out to be not so important, since the distance to which heat is required to be removed in the second version of the heat pipe is much less than in the first version. To increase the boiling point, it is necessary to increase the section of the para-wire and/or decrease the thickness of the wick, or change the type of heat carrier. Since the total thickness of the heat pipe cannot be increased, it is necessary to increase the section of the parawire by reducing the thickness of the wick, which will lead to a decrease in the capillary limit (however, there is a margin and it is possible to do this). The calculation for the optimized heat pipe structure is shown in Fig. 6b. Here, the thickness of the wick layer is halved to 0.2 mm, and the height of the steam line zone is increased, respectively, by 0.2 mm to 1 mm. In this case, it is possible to ensure the operability of the heat pipe in the case of a power transfer of up to 40 W at a pipe saturation temperature of about 45°C. In this case, the capillary limit of the pipe decreases, but remains at a sufficiently high level (it is not shown in Fig. 6b as it exceeds 100W). The temperature of the heating element, estimated on the basis of (9), is, in this case, about 57°C.

5. Conclusion

As a result of this study, the maximum transmitted power through the heat pipes is determined taking into account the limitations of the heat pipes operation by the capillary limit, by boiling (transition to film boiling, boiling limit), by the sonic limit at which the speed of steam reaches the speed of sound (sonic limit), by the entrainment of droplets liquid coolant from the surface of the wick with a counter flow of steam (entertainment limit) and viscous limit, which is realized at low temperatures (viscous limit).

1. An algorithm has been developed for calculating the limiting characteristics of heat pipes for cooling active phased antenna arrays at a given saturation temperature. The maximum transmitted power is determined taking into account the limitations of the heat pipes operation by the capillary limit, by boiling (transition to film boiling, boiling limit), by the sonic limit at which the speed of steam reaches the speed of sound (sonic limit), by the entrainment of droplets liquid coolant from the surface of the wick with a counterflow of steam (entertainment limit) and viscous limitation, which is realized at low temperatures (viscous limit).

2. It is shown that an increase in the thickness of the wick and its porosity may be necessary to increase the capillary limit of heat pipes, while an increase in the thickness of the wick increases the thermal resistance of the tube and, accordingly, can lead to overheating of the cooled elements.

3. Based on the above algorithm, design calculations for two types of heat pipes have been carried out. The dependences of various limits of the heat pipe on the operating temperature are plotted.

4. An optimal version of the pipe parameters is proposed to overcome the boiling limitation, while maintaining the capillary limit at a sufficiently high level.

The results of this study can give further prospective for calculation of characteristics of antenna heat pipes.

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