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# COOLING EQUIPMENT WITH HEAT PIPES FOR POWER TRANSFORMERS RELIABILITY IMPROVEMENT

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***Abstract:*** *The increase of transformers' medium power per weight unit is a global problem that appears while using them. In order to increase this percent we have to find more efficient solutions to yield the heating from the transformer to the outside. A main interest area where the principle of cooling with heat pipes can be successfully applied is that of the medium and high power transformers. The use of heat pipes in the cooling circuit of medium and high power transformers has become a necessity. It can increase the liability of transformers, the reliability and maintainability of their cooling systems and, last but not least, it can decrease the power consumed in the auxiliary cooling circuit of the transformer. Changing the classical cooling batteries of 150 kW from the medium and high power transformers with cooling batteries with heat pipes was experimented as a sample with thermal dissipated power of 25 kW with aluminium tubes.*
*As a conclusion, the use of heat pipes to cool the oil from the medium and high power transformers has a range of important advantages such as: the improvement of the oil cooling process, good heat transfer, with possibilities to recover the energy that is currently dissipated in air, medium and high power transformer's clearance diagram dimensions' reduction .*

***Keywords****: medium and high power transformers, heat pipes, heat transfer, cooling system*

## 1. Introduction

The increasing of reliability of service for mean and high power transformers represents an important objective in enhancing the performances of electrical equipment intended to energy branch.
Mean and high power transformers achieved in Romania are fitted with two types of classical cooling batteries that is: a cooling battery having tubes made of brass and aluminium plates as paddles and the other cooling battery in a brazed design, aluminium and aluminium alloy honey comb type.
Currently the transformers are provided with the classical cooling batteries showing a series of disadvantages: large overall dimensions, reduced reliability and maintenance, increased costs, efficiency and technical performances under the level of the cooling systems with heat pipes. Keeping an eye on the disadvantages of the cooling batteries with brass tubes and aluminium or honeycomb flanges, we recommend the use of a much more efficient new cooling system having the possibility to recover the heat energy.
Starting from a series of disadvantages the classical cooling systems have, namely big overall dimensions of coolers, low reliability and maintainability, lack of liability, high costs etc., it was proposed the replacing of these ones with cooling batteries with heat pipes improving the oil cooling process, which have the possibility to recover the energy that, in classical version, is dissipated in air.

## 2. New cooling solution with heat pipes

The new cooling system, much more proficient, with possibilities of recuperation of thermal energy, uses for transformers cooling, batteries with heat pipes [1]. These ones are inserted in the cooling circuit, according to the following fundamental circuit (fig.1).

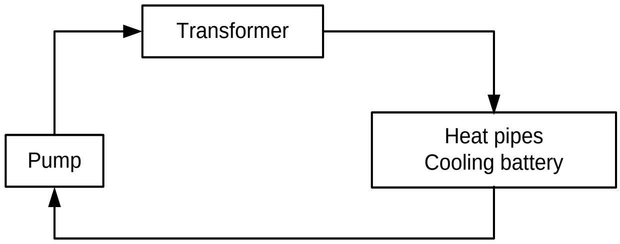


**Fig. 1.** Cooling solution

The cooling batteries with heat pipes are carried out in two constructive versions that is:
- cooling batteries with steel tubes provided with aluminium paddles;
- cooling batteries with aluminium tubes provided with aluminium paddles.

## 3. The heat pipe construction

From the view point of the heat quantity transferred by heat pipes, the theoretical difference between these versions is only 1 - 2 % that one with aluminium being better, but the experimental results obtained on tubes identical as concerns the dimensions (one of these was made of drawn pipe of steel, the other one of aluminium) have shown, paradoxically, a very high superiority for steel. It can be concluded that, the material of witch the tube is made is not restricted by considerations regarding thermal conductivity, such as it happens for the classical heat - exchanger. The heat pipe geometry, as essential element of cooling battery, is presented in fig. 2, where:

$d$ - outer diameter of container
$t$ - thickness of container wall
$p_a$ - paddle spacing
$t_a$ - paddle thickness
$L_v$ - evaporator length
$L_a$ - adiabatic part length
$L_c$ - condenser length
$L$ - heat pipe length

The heat pipe achieves the transfer of a big amount of heat, at a constant temperature, on a relatively long distance between the source fluid (hot) and the destination fluid (cold) even under the influence of a small difference between the temperatures of the two fluids, without involving the consumption of an additional amount of energy. In order to obtain a thermal transfer as good as possible, the choice of the work fluid adequate to an imposed temperature range, the compatibility of the work fluid with the container material, and also the setting up of an optimum amount of fluid are necessary [2], [3].
The cooling battery with heat pipes, in module design, is achieved as an assembly of modules, the number of which is depending on the dissipated total power of the battery.
The cooling module is presented as an enclosed nest of tubes, bounded by a separation plate in oil box, air box respectively. The heat pipe is from physical point of view, a device operating

isothermally on its entire length, in a continuous cycle of removing a great quantity of heat, at a small difference of temperature between the tube ends [2], [3].
With a view of obtaining a maximum possible thermal transfer, it is required the choice of a working fluid corresponding to an imposed temperature range, the compatibility of the working fluid with container material (heat pipe), and not at last, the setting up of an optimum quantity of fluid[4].

The cooling module is formed by a tube beam being placed inside a case, separated by means of a separation plane in the box oil, air box respectively.

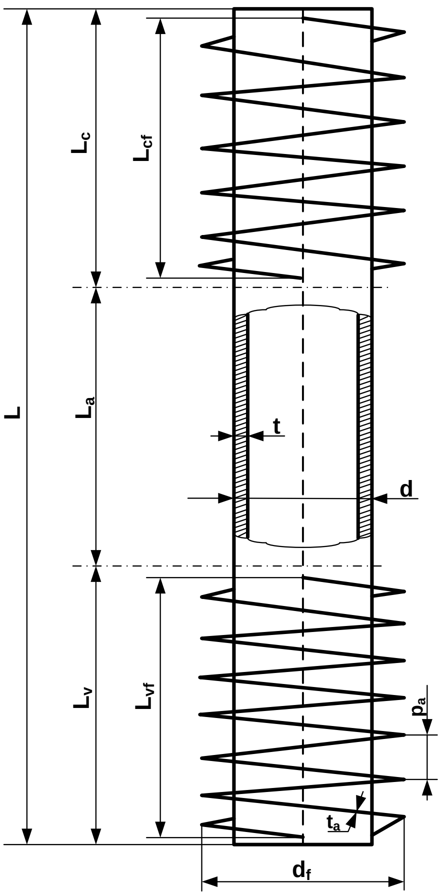


**Fig. 2.** The heat pipe geometry

With a view of obtaining a maximum possible thermal transfer, it is required the choice of a working fluid corresponding to an imposed temperature range, the compatibility of the working fluid with container material (heat pipe), and not at last, the setting up of an optimum quantity of fluid [4]. The cooling module is formed by a tube beam being placed inside a case, separated by means of a separation plane in the box oil, air box respectively.
In the arrangement of heat pipes it has started from a pre-dimensioning of tubes, from reasons of heat transmission and also from reasons of resistance against the internal pressure of the working fluid [4], [5].
By the manner of achieving the tube arrangement, the configuration of a heat exchanger in counter flow is desired, knowing that it is much more efficient the keeping of a temperature difference $\Delta T$, relative constant during the process, when the working fluid changes successively the heat in counter flow in "oil phase", carrying out the evaporation and then in "air phase", carrying out the condensation.

## 4. The assessment of heat loss by free convection in the vaporization area of the experimental model

The case through which the oil circulates in the vaporization area of thermal tubes could not be insulated , so arising heat loss by free convection to the environment. It will see further on that the cooling effect of this heat loss on the oil is very little and it can be neglected.
The heat loss can be assessed by means of the method presented further on.
For the free convection in open spaces the criterial equation by means of which it calculates the coefficient of heat transfer is:

$$Nu = 0.135\,(Gr \cdot Pr)^{1/3} \qquad (1)$$

Nu – the Nusselt invariant

$$Nu = \frac{\alpha \cdot d}{\lambda} \qquad (2)$$

- $\alpha$ - the coefficient of heat transfer (W / $m^2 \cdot$ K);
- d - the equivalent diameter (m);
- $\lambda$ - the coefficient of thermal conductivity of the air (W / m · K) at average temperature

$$t_m = \frac{t_p + t_{amb}}{2} \qquad (3)$$

$t_p$ – the temperature of the wall that delivers the heat;
$t_{amb}$ – the environment temperature.

Gr – the Grashoff invariant

$$Gr = \frac{g \cdot d^3 \cdot \beta}{\nu^2}\Delta t \qquad (4)$$

g – the gravitational acceleration (m / $s^2$);
v – the coefficient of cinematic viscosity of the air at the average temperature $t_m$ ($m^2$/s);
β – the extension coefficient ($degree^{-1}$);

$$\beta = \frac{1}{T_m} \qquad (5)$$

$T_m$ – the average temperature $t_m$ calculated in K;
Δt – the difference of temperature,
Pr – Prandtl criterion.
After the calculation of the $\alpha$ coefficient , it is possible to determine the heat loss by means of the formula:

$$Q_p = \alpha \cdot (t_p - t_{amb}) \cdot A \qquad [w] \qquad (6)$$

where :
A – is the lateral surface of the oil bath that delivers the heat to the environment.

Performing the calculations by means of the above described method it is possible to see that for a temperature of the oil bath wall of $75^0$C and a temperature of the environment of $30^0$C, the heat loss by free convection is insignificant, a negligible heat quantity in comparison with the functional performance of the experimental model.

## 5. Experimental results

In the figure **3** it is presented a partial view of the test bench together with the experimental model mounted in vertical position is presented.
The main parts of the test bench are:

- oil vessel containing 1500 l of hydraulic oil;
- electrical resistances for oil heating;
- cylindrical vessel of 350 l for measuring the oil weight rate;
- system for measuring the oil weight rate;
- gear pump for oil circulation;
- electric panels for control and regulation;
- mercury thermometers for measuring the oil temperature at the input and the output from the cooler with thermal tubes;
- different supports, connecting ducts, regulating cocks for oil flow.

The experimental model is mounted on the bench in vertical position. The connections on the oil side are from flexible rubber tube.
Within the experiment performed on a heat pipe, in close connection with the main geometrical parameters previously, specified, there are the following experimental variables:
- mean temperature difference $\Delta T$ [°C]
- weight of working fluid from thermal tube Wf [g]
- hot fluid speed (oil) V1 [m/s]
- cold fluid speed (air) V2 [m/s]

The mean temperature difference between the hot fluid and the cold fluid ($\Delta T$) is one of the most important performance parameters of heat pipes.
The increase of mean temperature difference has a strong stimulate effect on the energy quantity transferred in the time unit through the surface unit of heat pipe. In order to replace the classical cooling batteries of 150 kW from mean and high power transformers with cooling batteries with heat pipes, a cooling module having aluminium tubes with dissipated power of 25 kw and overall dimensions 700 x 888 x 1034 mm was experimented [5], [6]. The arrangement of the tubes within this module was done on 2 lines (22 tubes on a line) the separation plate to circumscribe to condensation and vaporisation zones being arranged such as:
- evaporator length 201 mm;
- condenser length 750 mm.

These 2 lengths are in a ratio considered as being optimal, between the length of condenser, respective evaporator, the other dimensions related to the geometry of tube and tube arrangements being presented below:
- tube length $L = 993$ mm;
- outer diameter of container $d = 16$ mm;
- pipe thickness $t = 1$ mm;
- paddle thickness $t_A = 0{,}3$ mm;
- paddle length $L_A = 836$ mm;
- paddle width $l_A = 48$ mm.

As a work fluid for heat pipes it was used acetone.
The experiments performed together with TRANSTERM BRASOV on a module with the dissipated thermal power of 25 kW, with aluminium tubes and using acetone as working fluid for the heat pipe, has shown the increasing of thermal performance, in the same time with oil speed increasing. The measurements were done on the oil side, where the necessary thermodynamic parameters could be measured and calculated with accuracy. By means of hydraulic networks used it was varied the oil flow, respectively were then into account three values of this one, for which measurements and calculations of other parameters were performed [5], [6].
The mounting of thermal tubes in the case on the oil side was done so that to assure a single passing through the horizontal spaces between the fins.

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So it resulted the following free section for oil passing:

$S_{oil}$ = 0.078 x 0.194 – [2 x 0.016 (0.194 – 64 x 0.0003) + (0.076 x 0.0003 x 64)] = 8.0792 x $10^{-3}$ $m^2$

The mounting of thermal tubes on the air side was done in the case put at the disposal by the beneficiary. It resulted the following free section for air passing:
$S_{air}$ = 0.840 x 0.740 - [22 x 0.016 (0.740 – 246 x 0.0003) + (0.840 x 0.0003 x 246)] = 0.3251 $m^2$
The calculation of the functional performance of the experimental model of oil cooler with thermal tubes was done on the oil side where it was possible to measure and calculate with precision the necessary thermodynamic parameters.
It presents further on the main used symbols and the main calculation parameters.

- $t'_1$ - the oil temperature at the input in the vaporization area ($^0$C);
- $t''_1$ - the oil temperature at the output from the vaporization area ($^0$C);
- $t'_2$ - the air temperature at the input in the condensation area ($^0$C);
- $t''_2$ - the air temperature at the output from the condensation area ($^0$C);
- $t_{TT}$ - the average working temperature of thermal tubes ($^0$C);
- $V_1$ - volumetric flow rate of oil ($m^3$ / s);
- $v_1$ - the oil rate in the free section of vaporization area (m / s);
- ρ - the oil density at the average temperature from the vaporization area (Kg / $m^3$);
- $c_1$ - the specific heat of oil at the average temperature from the vaporization area and at the constant pressure (J / Kg.K);
- Q - the heat transfer (the functional performance) carried out by the experimental model with thermal tubes (W);
- Δ t – the temperature difference that activates the heat transfer process in the experimental model with thermal tubes ($^0$C)

$$t_2 = \begin{aligned} \Delta t &= t_1 - t_2 \\ t_1 &= \frac{t_1^{'} + t_1^{''}}{2} \\ t_2 &= \frac{t_2^{'} + t_2^{''}}{2} \end{aligned} \qquad (7)$$

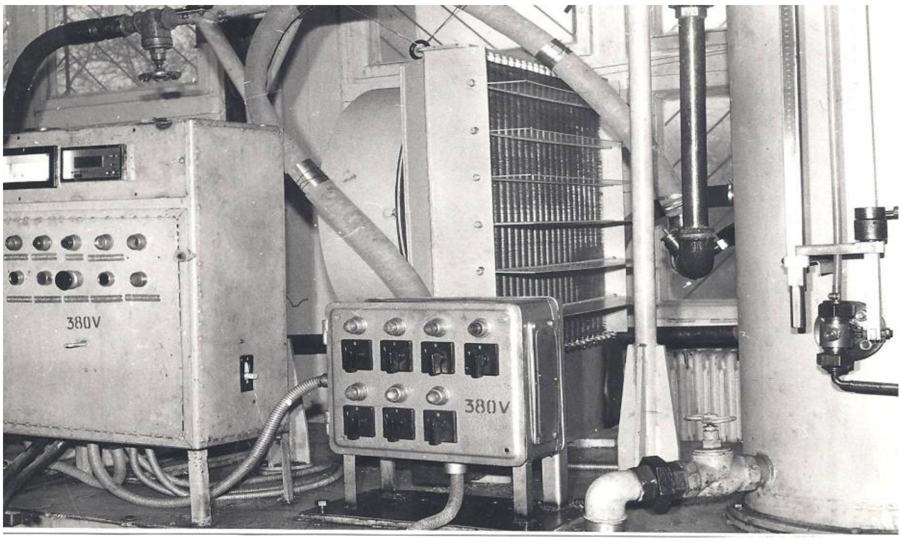


**Fig. 3.** Partial view of the test bench with the experimental model mounted in vertical position

**TABLE 1**

| Nr. Crt. | $t_{1'}$ [°C] | $t_{1''}$ [°C] | $t_{2'}$ [°C] | $t_{2''}$ [°C] | $t_{TT}$ [°C] | $V_1 \times 10^3$ [m/s] | $v_1$ [m/s] | $\rho_1$ [kg/m$^3$] | $c_1$ [j/kg*K] | Q [W] | Δt [°C] |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1. | 76,30 | 73,30 | 20,97 | 28,92 | 39 | 2,880 | 0,355 | 846 | 1980 | 14,432 | 49,85 |
| 2. | 76,50 | 71,70 | 21,20 | 29,15 | 38 | 1,703 | 0,211 | 847 | 1979 | 13,702 | 48,92 |
| 3. | 76,80 | 74,30 | 21,70 | 29,75 | 39 | 4,035 | 0,499 | 846 | 1985 | 16,940 | 49,82 |
| 4. | 80,10 | 76,90 | 22,10 | 30,48 | 41 | 2,880 | 0,355 | 844 | 2016 | 15,681 | 52,18 |
| 5. | 80,60 | 75,60 | 22,50 | 30,93 | 40 | 1,703 | 0,211 | 844 | 2013 | 14,466 | 51,38 |
| 6. | 80,7, | 78,10 | 24,50 | 33,45 | 40 | 4,035 | 0,499 | 843 | 2024 | 17,900 | 50,42 |
| 7. | 83,80 | 80,60 | 23,17 | 31,88 | 42 | 2,880 | 0,355 | 845 | 2030 | 15,809 | 53,79 |
| 8. | 83,70 | 78,40 | 23,15 | 31,86 | 41 | 1,703 | 0,211 | 842 | 2032 | 15,442 | 53,54 |
| 9. | 82,30 | 79,60 | 25,10 | 34,95 | 41 | 4,035 | 0,499 | 842 | 2032 | 18,639 | 50,92 |
| 10. | 83,80 | 80,50 | 24,17 | 32,88 | 42 | 2,880 | 0,355 | 842 | 2039 | 16,316 | 53,62 |
| 11. | 83,60 | 78,10 | 22,15 | 30,86 | 41 | 1,703 | 0,211 | 842 | 2031 | 16,017 | 54,34 |
| 12. | 82,90 | 80,20 | 23,20 | 31,91 | 41 | 4,035 | 0,499 | 842 | 2034 | 18,658 | 53,99 |
| 13. | 86,30 | 82,80 | 25,30 | 34,30 | 43 | 2,880 | 0,355 | 840 | 2053 | 17,383 | 54,75 |
| 14. | 86,20 | 80,60 | 24,30 | 33,05 | 42 | 1,703 | 0,211 | 841 | 2046 | 16,409 | 54,72 |
| 15. | 86,50 | 83,70 | 25,30 | 34,35 | 42 | 4,035 | 0,499 | 840 | 2056 | 19,512 | 55,32 |
| 16. | 91,80 | 88,00 | 26,25 | 36,77 | 44 | 2,880 | 0,355 | 837 | 2084 | 19,089 | 58,39 |
| 17. | 91,50 | 85,80 | 27,40 | 37,82 | 45 | 1,703 | 0,211 | 838 | 2077 | 16,895 | 56,04 |
| 18. | 92,60 | 89,60 | 28,10 | 38,25 | 44 | 4,035 | 0,499 | 836 | 2091 | 21,160 | 57,92 |

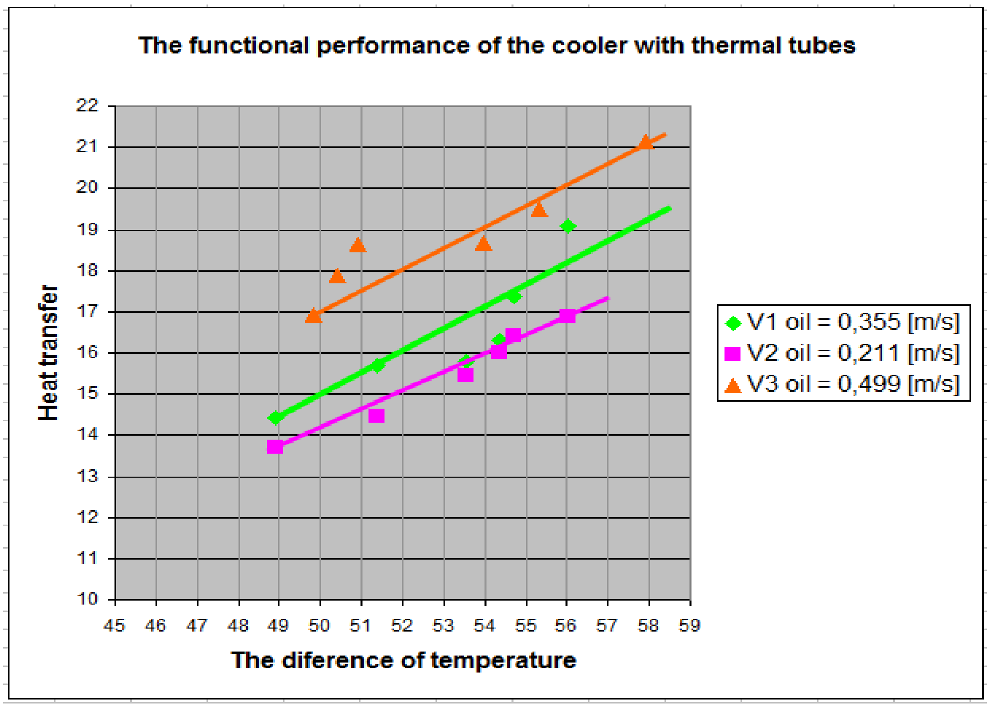


**Fig. 4.** The variation of dissipated heat, depending on temperature difference

The quantity of heat transported by the experimental model with thermal tubes in a certain working condition, determined by the oil and air temperatures, is calculated by means of the following formula:

$$Q = m \cdot c_1 \cdot (t_1^{'} - t_1^{''}) \qquad (8)$$

where:
m is the oil weight rate ($m = \rho \cdot V_1$).
The experimental results are presented in Table 1 where the dependence between heat transfer (dissipated power) Q and mean temperature difference $\Delta T$ was set up, for few speeds of oil.
The mean temperature difference $\Delta T$ is given by the relation:

$$\Delta T = (T_1^{'} + T_1^{''}) / 2 - (T_2^{'} + T_2^{''}) / 2 \qquad (9)$$

where:
$T_1^{'}$ = oil temperature at cooler inlet;
$T_1^{''}$ = oil temperature at cooler outlet;
$T_2^{'}$ = oil temperature at cooler inlet;
$T_2^{''}$ = oil temperature at cooler outlet.
In figure 2 the variation of the dissipated heat is presented, depending on the temperature difference for three speeds of the oil.
The experimental model was tested at three different rates of the oil in the free section of the vaporization area. The influence of this rate on the functional performance can be very well observed in the figure 4.
At the rate of 0.499 m/s, the dispersion of the points indicates the instability of the operation of the thermal tubes as a result of the approach to the drive limit.
It appreciates that the optimum rate of the oil is situated in the interval 0.3 – 0.4 m/s.
As the experimental model was built with the fan on the upsetting, it was not possible to do measurements on the air side. However, using an anemometer with helix connected to a thermocouple it was possible to do estimative measurements, but it was not possible to use them at the calculation of the functional performance. For instance, the temperatures $t'_2$ and $t''_2$ of the air are calculated as arithmetic average of 30 – 50 measurements in different points of the sucking and upsetting sections. The rate measurements in the same points indicate an estimative air flow rate of 6.700 $m^3$ / h at the temperature of $25^0$C.

## 4. Conclusions

As a result of experiments there were drawn the following conclusions:
- at speed of 0,439 m/s, the spreading of the points on the diagram from Fig. 2 denote is the unsteadiness of heat pipes operation, because of the approaching to drive limit;
- it is estimated that the optimal speed of the oil is the range 0.3 - 0.4 m/s.

In order to improve the functional performance of cooling module, ammonia will be adopted as work fluid, estimating that functional performance of module will increase with, at least 35%.
The structure of pipes with paddles should be composed of steel pipes with aluminium paddles (the length of steel pipes will be 2400 mm) estimating that the functional performance will increase with about 10%.
In conclusion, the use of the heat pipes for cooling the oil from mean and high power transformer has a range of important advantages against the classical solutions currently used:
- higher heat transfer referred to volume and weight, due to the use of pipes with paddles, both on the air and the oil side;
- high reliability because every heat pipe of the packet is an independent heat exchanger. Accidental failure of a number of heat pipes does not lead to the playing out of operation of the entire cooler;
- reduced working expenses due to the use of fans with much higher powers;
- possibility of achieving a module design, so being obtained high power batteries;
- low energy consumption at the beneficiary, by decreasing the power consumed by the cooling subsidiary circuit of the transformer.

From the resulted diagrams, it is estimated that the optimum speed of the oil is between 0 . 3 - 0 . 4 m/s.
To improve the functional performances of the cooling module, acetone will be replaced by ammonia, hoping for an increase of the capacity of heat delivery by 35 %.
As a conclusion, the cooling of the mean and high power transformers with heat pipes batteries has many advantages confronted by the classical solution : a higher heat transfer capacity, high reliability and reduced energy consumption.

**There are two possibilities to improve the functional performance of the experimental model:**
**1.** The thermal tubes must be manufactured with ammonia as working fluid.
It appreciates that the functional performance will increase with at least 35%.
**2.** The structure of ducts with fins must be composed by steel ducts with aluminium fins. In accordance with the experimental results previous to the present contract, it appreciates that the functional performance will increase with approximately 10%.
Following the tests it was found that the experimental module has a very good availability of thermal transfer both on the oil side and the air side, being limited by the maximum value of 21 KW only by the used working fluid. Considering that initially the experimental module was designed for AMMONIA and knowing from many experimental data the difference of relative characteristics between the ACETONE and the AMMONIA, it is possible to consider as certain an increasing of at least 35% of the performance. This would make the model transfers, in the absolutely same conditions, a quantity of heat of over 28 KW.
If in addition it adopts the solution to change the aluminium ducts with steel ducts (having ammonia as working fluid), then the thermal performance will still increase with at least 10%, giving the possibility to the experimental model to transfer a heat quantity of at least 30 KW. Because the carried out experimental model had a height of 1 m, a half of the height of a cooling module which will equip the batteries intended for the medium power transformers, it is possible to conclude that an entire module can dissipate a heat quantity of at least 60 KW.
Thus using three modules in parallel, submitted to a cooling by means of only 2 fans VART 630, it will be possible to easily reach the performance of 150 KW, required by the cooling battery taken as comparison element in the study previously performed by us.
In comparison with the existing variant of battery, the advantages of the solution with modules with thermal tubes are certain and multiple. We shall only enumerate some of them:
- the replacement of the 2 fans VATP – 800 SV with fans VATP 630 much lighter, cheaper and more silent and with an installed power incomparably lower.
- the reliability incomparably higher of the batteries with thermal tubes in comparison with any of the classical variants used at present (practically the batteries with thermal tubes cannot break down because even the accidental fracturing of a tube does not affect at all the thermal performance of this one), in contrast with the classical variants where the fracturing leads to the oil loss and placing out of operation of the battery.
- the execution cost of the active part, system of thermal tubes with fins of the battery, composed of elements with modules, is much lower than the same active elements of the batteries used at present .
- the possibility to carry out the cooling batteries with powers comprised between 25 and 200 KW by connecting independent modules of 25 or 50 KW.
Therefore it observes the possibilities of internal typing with important effects regarding the decreasing of execution costs, but a real increasing also of the quality of these ones.

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