Air-Cooled Thermosyphon for Press-Pack Stack of Semiconductors

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Abstract. Medium voltage converters make extensive use of press-pack semiconductors in several products ranging from drives to rectifiers. Better cooling brings significant advantages: the possibility of drawing more power using the same semiconductors and thus effectively extending the converter operating range, a higher reliability, and cost savings on semiconductors for equal converter power. In this article, the experimental characterization of a baseplate to air thermosyphon for the double-side cooling of press pack thyristor modules in presented. Effects of refrigerant fluid filling, heat load, air temperature and air flow on the junction temperature were studied in both vertical and horizontal stack configuration. Vertical evaporator yields the best performance with 120°C junction temperature at 40°C air and 30W/cm² heat loss.

1. Introduction
Heat sinks are currently used in air cooled converters. An increase in the heat transfer rate would mean an increase in size and air velocity, which would in turn increase cost and noise, or switch to water cooling. Furthermore, the heat spreading from the semiconductor into the heat sink base is a limiting feature. Embedding heat pipes or a vapor chamber could help alleviating the problem, but at a considerable cost. Even then the heat transfer area would have to be significantly augmented to handle a power that is almost or more than double the maximum currently handled. This would call for a radical modification of the heat sink and of its manufacturing process.

Two-phase thermosyphon coolers are an intermediate solution between forced air and water coolers that cope with performance, reliability and maintenance-free operation. Thermosyphon are passive system based on the density difference between the vapour and liquid phases of the cooling medium. Early in 1988, Bordignon et al. [1] developed a two-phase thermosiphon for large thyristors cooling that enabled the full utilization of their current handling capability. Furthermore, this solution avoids the parallel connection of thyristors with increased performance and reliability. Kuwahara et al. [2] built a thermosyphon based cooling system for a traction inverter. The semiconductor devices are a press-pack stacked between evaporators connected to a common condenser with an intermediate liquid/vapor separator for 3kW cooling capacity for each power device. More recently, Derakhshanfar et al. [3] proposed an air-cooled two-phase thermosyphon solution for HVDC breaker cooling that can handle up to 5kW losses for each power module.

The low mass flow rate of the two-phase cooling medium obtained by natural convection usually limits their application to heat density up to 15-20W/cm². It is possible to develop two-phase natural
convection system that can reach higher heat loss density by using surface enhancements such as coating, micro-porous structure, mesh, wick... etc. The present study shows the thermal characterization of an air-cooled two-phase thermosyphon for press-pack stack of thyristors with vertical and horizontal evaporators.

2. Thermosyphon for press-pack of semiconductors
The working principle of a thermosyphon is explained in Figure 1. The thermosyphon is typically composed of the following four sections:
1. the evaporator – where the heat is applied and the liquid is partially evaporated,
2. the riser – where the vapour rises by natural convection,
3. the condenser – where the vapour condenses back into the liquid phase
4. the downcomer – where the liquid falls down by gravity to the bottom of the evaporator

Due to the influence of gravity, a thermosyphon can work only in a vertical position (or with a limited angle of inclination and degraded thermal performance). Furthermore, the condenser must be located above the evaporator for the fluid circulation to take place. Medium voltage converter requires the use of electrically insulating riser and downcomer, through which the fluid flows and that provides electrical isolation between evaporator and condenser.

A cost-optimized two-phase thermosyphon cooler was developed and manufactured using standard aluminium plate and bar condenser for the condenser and Nokoloc brazing process for the evaporator. The evaporator structure is made of aluminium multi-port extruded tubes brazed together to provide both a large heat transfer area and a good fin efficiency, and thus enabling good thermal spreading and bubble pumping. Figure 2 shows a sectional view of the evaporator internal structure. Internal liquid distributor and vapor collector were designed to minimize the pressure losses in the evaporator. The entire cooling system is shown Figure 2. The evaporator section is 18 x 134 mm with 1.5mm channel hydraulic diameter, the plate and bar condenser is 500 x 100 x 100mm with wavy fins and the vapor riser and liquid downcomer are 12mm internal diameter.
3. Experimental set-up
A schematic of the test facility is shown in Figure 3.

It is composed of different elements:
- A test section equipped with the following instrumentation:
  - 2 circular heating elements (diameter 3 inches) equipped with 5 electrical cartridges (1000W heating power each)
  - 1 thermosyphon cooler
- 1 absolute pressure sensor and 4 differential pressure transducers
- Thermocouples at evaporator and condenser inlet and outlet manifolds
- 24 thermocouples (12 on each side) directly in contact with the evaporator baseplate as shown on Figure 3.
- 2 Eurotherm 16A 230V single-phase power controller
- An inlet air distribution cone equipped with 1 thermocouple (middle) and an air pressure measurement port
- An outlet air distribution cone equipped with 3 thermocouples (top, middle and bottom) and an air pressure measurement port
- A differential air pressure sensor

All signals are acquired through a National Instruments SCXI box connected to a laptop running Labiew. For temperature measurement a 1102 module with a cut off frequency at 2 Hz is used. The measuring devices and their accuracy are presented in Table 1.

| Instrument                  | Type                              | Range          | Uncertainty                  |
|-----------------------------|----------------------------------|----------------|------------------------------|
| Thermocouples               | Thermocoax, type K, shielded, class 2 | -40–1200 °C   | ± 0.1 K calibrated (all acquisition chain) |
| Fluid pressure sensor       | Omega PX 409                      | 0-500 psi      | ± 0.08 %                     |
| Diff. pressure sensor       | Omega PX 409                      | 0-2.5 psi      | ± 0.1 %                      |
| Data acquisition            | National Instruments, SCXI 1000, modules 1102, 1124, 1600, 1303, 1325 | 0-10 V         | ± 0.02 %                     |
| Air pressure drop           | Huba control                      | 0-500 Pa       | ± 2 %                        |
| Air mass flow rate          | ABB Topaz                        | 80-4000 kg/h   | ± 0.1 %                      |
| Power supply                | Eurotherm TE-16A                  | 230V, 16 A     | ± 1 %                        |
| Refrigerated bath           | Lauda, R207                       | -40-200 °C     | ± 0.2 K                      |
| Precision thermometer + Platinum Probes | Omega, DP97                      | -50-400 °C     | ± 0.04 K                     |

Table 1: Measuring device and accuracy

The test setup is inserted into a wind tunnel equipped with a centrifugal fan, a mass flowmeter, a pre-heater, and an air-water heat exchanger connected to a chiller in order to keep the air at a preset temperature. All the thermocouples were calibrated using an Omega DP97 precision thermometer with platinum probes to measure the reference temperature and a Lauda R207 chiller to control the temperature. The uncertainty of the calibrated thermocouples is ± 0.1 K (it is ± 2.5 K if not calibrated).

4. Data reduction

The fluid filling ratio is defined as the ratio of the volume of fluid filled in the system divided by the internal volume of the loop thermosiphon unit. The mass of fluid was measured with a precision of 10g leading to an uncertainty of 0.3% on the fluid filling ratio.

The total maximum thermal resistance is calculated as follows:

\[ R_{th} = \frac{T_{b,\text{max}} - T_{\text{air,in}}}{Q} \]  

(1)
where $T_{b,\text{max}}$ is the maximum baseplate temperature.

The condenser thermal resistance is calculated as follows:

$$R_{\text{th,c}} = \frac{T_{\text{c, e}} - T_{\text{air,in}}}{Q}$$

(2)

The evaporator thermal resistance is calculated as follows:

$$R_{\text{th,e}} = \frac{T_{\text{b, max}} - T_{\text{c, e}}}{Q}$$

(3)

The closing relation is given by:

$$R_h = R_{\text{th,e}} + R_{\text{th,c}}$$

(4)

The energy balance, calculated as the relative difference between the electrical power injected in the baseplate and the heat extracted by the air stream, was kept below 10%.

$$\Delta Q = \frac{UI - \dot{m}_{\text{air}} C_p \text{air} (T_{\text{air,out}} - T_{\text{air,in}})}{UI}$$

(5)

5. Test results

All measurements were performed at steady state, i.e. when all the over-temperatures (difference between measured temperature and ambient air temperature) are stable with time. Tests were performed with both vertical and horizontal evaporator orientations as shown on Figure 4:

5.1. Effect of fluid filling ratio

Figure 5 shows the variation of the maximum thermal resistances at 2000W heat losses (1000W per side) for fluid fillings between 25 and 85%.
At low fluid filling, the amount of vapor at the exit of the evaporator is higher, there is less liquid to wet the evaporator wall constantly so that the evaporator performance decreases. At high fluid filling there is more liquid in the thermosyphon so that the condenser is partially flooded. As a consequence the saturation temperature inside the thermosyphon has to increase in order to allow the condenser to condense all the vapour over a shorter flow length and the condenser performance is decreasing. Performance of the thermosyphon is less affected by the fluid filling with vertical evaporator than with horizontal evaporator. The minimum value of fluid filling to avoid dryout is ca. 42%, the maximum fluid filling to avoid condenser flooding is ca. 71%.

The temperature difference on the evaporator baseplate is strongly affected by the fluid filling at high heat loads: the more filling, the more uniform cooling due to a better channel rewetting. Similar observation can be made between top and bottom evaporator baseplate: the highest filling minimizes the difference between top and bottom. Temperature difference is reduced by half in vertical orientation: the difference between both side of the evaporator is almost negligible (below 4°C) even if the liquid/vapour flow maldistribution are amplified when increasing the heat load.

71% was chosen as optimum fluid filling for the following tests as it is a good compromise between cooling performance and temperature uniformity.

5.2. Effect of heat load
Tests were performed with 700m³/h air flow and 40°C air inlet temperature. Figure 6 shows the thermal resistances for heat loads up to 3500W.
With horizontal evaporator, the evaporation thermal resistance is first decreasing at low heat load before to remain constant and start increasing at higher heat loads. This is explained by the fact that with sufficient channel wetting the boiling heat transfer coefficient is an increasing function of the heat load as the number of activated nucleation sites is increasing. When channel dry-out starts, the evaporator performance decreases. In vertical orientation, no channel dry-out is observed and the evaporator thermal resistance is decreasing monotonically with increasing heat load.

The difference of performance between horizontal and vertical evaporator is probably due to the liquid maldistribution between the top and bottom channels and the bubbles rising from bottom to top which leads to less efficient channel rewetting on the top side of the evaporator. As the consequence, the temperature uniformity on the baseplate is higher in horizontal orientation as shown on Figure 7.
Having the mini-channels in the vertical direction helps the bubble to rise to the vapor collector and riser and at the same time, the bubbles push the liquid to the top of the channel which helps rewetting of the potential dry patches.

Figure 8 shows the performance map of the air-cooled two-phase thermosyphon cooler with vertical and horizontal evaporator.

![Figure 8: Temperature rise as a function of heat density](image)

Vertical evaporator can dissipate 31.4 W/cm² and 36.6 W/cm² with respectively 43.3°C and 51.1°C temperature rise, whereas only 20.9 W/cm² and 26.2 W/cm² could be dissipated with horizontal evaporator. This corresponds to ca. 33% more losses for the same temperature rise.

6. Conclusions
A high performance air-cooled insulated thermosyphon was designed and characterized in an extensive experimental campaign with evaporator in vertical and horizontal orientation. Vertical evaporator yields the best performance with 120°C junction temperature at 40°C air and 30W/cm² heat loss. The vertical evaporator is the preferred one with lower hot spot temperature and better temperature uniformity on the baseplate. Performance of horizontal evaporator is suspected to be limited by vapor/liquid mal-distribution leading to channel dry-out.

References
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