Passive two-phase cooling of air circuit breakers in data center power distribution systems

A.J. Robinsonb,c,⁎, J. Colenbranderb, G. Byrneg, P. Burkea, J. McEvoya, R. Kempersc,d

a Anord-Mardix Ltd, Unit 17 Northlink Business Park, Coes Road East, Dundalk, Co Louth, Ireland
b Confluent Research Ltd., 6 The Avenue, Inse Bay, Laytown, Ireland
c Department of Mechanical and Manufacturing Engineering, Trinity College Dublin, Ireland
d Department of Mechanical Engineering, York University, Toronto, Canada

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A B S T R A C T

This work presents an experimental investigation of a two-phase looped thermsyphon design for cooling Air Circuit Breakers (ACBs) in switchgear used in data center power distribution systems. The FC72 charged thermsyphon is comprised of a bespoke evaporator that is fixed tightly to the copper adaptor of the ACB. Dielectric tubing connects the evaporator to a remote naturally aspirated and thin form factor condenser. Since both the working fluid and the connection tubing are dielectric, the thermsyphon provides electrical isolation between the ACB and the remote heat sink. The condenser is a thin serpentine channel with surface extensions and is cooled by natural convection and radiation to the ambient environment. Benchtop thermal performance tests were performed for increasing load power for two scenarios: for the stand-alone looped thermsyphon and for the case where copper bus bars are also fixed to the ACB adapter. For 60 W thermal power dissipation, a nominal operational condition, the thermal resistance of the looped thermsyphon system was determined to be 0.55 K/W without bus bars attached. With bus bars attached, this decreases to 0.42 K/W which results in a breaker to ambient temperature rise of about 25 K, which is significantly lower than operational limits of 85 K temperature rise. In-situ tests were then performed on a live ACB system carrying a 2000 A load and showed that the looped thermsyphon system was capable of decreasing the ambient temperature rise by 26 K, which is significant. The low thermal resistance of this passive cooling technology opens the opportunity for increased electrical service per breaker and/or a significant reduction in the volume of copper bus bars used with associated cost reduction of ACB technologies.

1. Introduction

Data centers are the heart of the information technology revolution, and they are powered by hundreds of TWhrs of electrical energy annually. Energy consumption by the data center industry is escalating as they manage, store and distribute ever increasing amounts of data. In any data center, system uptime is critical as outages can incur significant financial, productivity and reputational losses. Thus, the electrical power distribution system reliability is key to maintaining uninterrupted data center operation. To facilitate this, all data centers include switchgear to protect, isolate and control downstream electrical equipment. They consist of a combination of fuses, circuit breakers and disconnect switches that cut power to critical components when required. In some cases, circuit breakers are directly related to the power distribution reliability of data centers and thus key to data center reliability as a whole; they are mission critical infrastructure.

In data centers, Air Circuit Breaker (ACB) technology is used to provide protection at the low voltage (< 1000 VAC) nodes in the power distribution system where electrical current is in the range of hundreds to thousands of Amps [1]. ACBs are installed upstream and in series with the electrical equipment they are employed to protect and disengage (open circuit) when fault conditions arise due to anomalous electrical current. Hazardous electrical load conditions can arise from ground faults, short circuit faults, overload faults [2] or excessive temperatures. For the latter, heat generation by Ohmic heating along current carrying components as well as the electrical and thermal contact resistances across the breaker and other junctions causes temperature increases above ambient. In this way, the maximum current rating is dictated by the maximum continuous current that a breaker can carry before reaching prescribed temperature limits; above which causes damage to the ACB equipment. In particular, excessive temperatures can damage conductors, their insulation and springs which

⁎ Corresponding author.
E-mail address: AROBINS@tcd.ie (A.J. Robinson).

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can result in interrupted service, improper functioning or catastrophic failure [1]. This being the case, temperature rise and maximum temperature thresholds are strictly regulated. According to IEEE Standard C37.13 [3], the temperature rise limit above the ambient air surrounding the breaker enclosure for circuit breaker contacts, conducting joints, and other parts is 85 °C with a maximum total temperature limit of 125 °C.

Considering the above, thermal-electrical design considerations must be considered to ensure the temperature rise does not exceed dangerous conditions. On one hand, it is desired to maximize the current load capacity of an ACB system to provide maximum electrical service to the downstream equipment. However, heat generation limits the current due to limiting temperatures, thus requiring specialized heat transfer equipment which increases cost and complexity. As a result, switchgear equipment designers must find the balance between increasing the current carrying capacity without compromising the cost, complexity and overall reliability with excessively complex thermal management hardware and systems.

Reliability considerations demand that ACBs are cooled passively. Problematically, the overall systems are contained within an enclosure which inhibits air flow and heats the enclosed air over that of the ambient. For typical IP65 NEMA 3R and similar enclosures, they are in fact sealed from the ambient environment which makes effective air cooling by natural convection particularly challenging. Whether sealed or vented, the poor cooling capacity of the air is overcome by limiting the electrical current, oversizing the current-carrying bus bars and/or providing additional cooling with attached heat sinks. These are depicted in Fig. 1 for a typical data center ACB system assembly. As shown in Fig. 1(b), both the attached heat sink and the oversized bus bar options have been implemented, with the former required for the middle breakers due to their position in a more aggressive thermal environment. For the latter, the oversizing of the bus bars is a simple and effective means of increasing the surface area for heat transfer as well as decreasing the electrical heat generation.

As one can imagine, a passive technology that can cool the ACBs more effectively than current practices will have the positive outcome of decreasing their operational temperature and thus increasing long term reliability. Also, improved cooling can facilitate increasing their rated electrical current capacity, and/or decreasing the volume of metal, in particular the copper used for the bus bars. These will decrease the size, weight and cost of the integrated ACB system.

Two-phase looped thermosyphons use evaporation and condensation within a closed system to transport heat from a concentrated source to a remote heat exchanger. The driving force for circulating the working fluid is gravity which makes them a passive component in the thermal management system. In the looped configuration, the working fluid has a singular direction of flow with gravity acting on the liquid phase that exits the condenser heat exchanger. A requirement of any looped thermosyphon system is that the condenser is positioned above the evaporator in order for sufficient hydrostatic head to be developed to circulate the working fluid. A significant design challenge of looped thermosyphon thermal hardware implementation for the present ACB cooling application is that the condenser must be cooled by natural convection and radiation making the system completely passive. Further to this, the thermosyphon must be capable of transporting the rated thermal power, with a factor of safety, with a sufficiently low thermal resistance that the temperature limit is not reached and dryout of the evaporator is avoided.

Critical reviews of heat pipes and thermosyphons can be found in Refs. [4-6] and the reader is referred to these works for details regarding the basics of operation, theoretical modelling, performance limitations and various applications to which they have and may be employed. Put simply, when deployed as part of a thermal management technology, the overall thermal resistance of a heat pipe / thermosyphon-type heat exchange system involves those primarily associated with the heat source-to-evaporator, the two-phase flow of the refrigerant in the evaporator and condenser, and the heat rejection from the condenser to the heat sink coolant.

A specific subset of thermosyphon heat exchange systems are those that use natural convection, with ambient air as the heat sink coolant. Natural convection air cooling combined with the gravity-driven two-phase flow within the thermosyphon makes the entire cooling solution of the evaporator is avoided.

Fig. 1. Air Circuit Breaker (a) view of partially assembled unit showing connection ‘toe’ and current-carrying copper bus bars, (b) view of assembled unit with oversized bus bars and copper extensions onto which heat sinks are fixed.
passive, which is critical for those applications in which long term reliability is of utmost importance [7]. In this scenario, it is crucial that the system is designed in such a way that the thermosyphon itself introduces a small thermal resistance compared with the natural convection heat sink. This minimizes the temperature difference between the heat source and the exposed air-side surfaces and maximizes the convective heat transfer coefficient and the effective radiative heat transfer coefficient, both being crucial in achieving as low as possible thermal resistance on the air-side heat exchanger.

A survey of the literature reveals that there is limited prior research on naturally aspirated looped thermosyphons. Samba et al. [8] experimentally investigated a n-pentane charged looped thermosyphon for passive cooling of telecommunications outdoor cabinets. Importantly, this work performed a like-for-like comparison between the looped thermosyphon system and that traditionally used and found that the former was twice as effective. Following this work, Chehade et al. [9] developed thermal and hydraulic network models of an air cooled naturally aspirated looped thermosyphon for cooling telecommunications cabinets. They performed experiments in order to provide empirical correlations for the evaporator and condenser heat transfer coefficients and were able to show adequate agreement with the model and the experimental performance. Subsequent to this they carried out a parametric study in order to elucidate the parameters which improve the cooling system performance.

Oliveira et al. [10] tested a water charged looped thermosyphon prototype for thermal management of aircraft components. A dual condenser concept was tested with one transferring heat to the fuselage and the other to the cabin air. Condenser cooling ranging from natural convection to forced convection and a broad range of operating temperatures was evaluated. For combined natural convection, they found that the operating temperature was kept below 80 °C for power inputs up to 500 W, and this improved for simulated flight conditions where forced air cooling and air conditioning systems were active.

Chen and Yang [11] studied the heat transfer performance of a looped thermosyphon system for cooling concentrating solar cells, with the aim of improving their conversion efficiency. They determined that acetone performed better than both water and ethanol. Noting that the largest thermal resistance of the system was associated with the air side heat exchanger, numerical simulations were carried out to optimise the natural convection heat sink. As alluded to earlier, this highlights the importance of the holistic nature of the looped thermosyphon system design where, if designed correctly, delegates the most significant portion of the thermal budget onto the air-side of the condenser heat exchanger.

With regard to what is available in the current literature, there is general deficiency of publications on the design and performance characteristics of end-to-end fully passive two-phase thermosyphon systems. The absence of moving components (pumps, fans etc) as part of the thermal management system is a unique design requirement that is a key requirement for critical infrastructure with long (5–20 year) service life requirement, as is the case for data center power distribution hardware.

The overarching aim of this research is to contribute new knowledge in the area of ACB thermal management for data center applications. Specifically, the work targets the design and performance testing of an entirely passive two-phase looped thermosyphon system. This end-to-end passive cooling system has the potential to replace the current practices of limiting supply current, oversizing bus bars and/or attaching finned heat exchangers. To the best of knowledge, this is the first attempt to disclose and discuss in detail an end-to-end fully passive two-phase cooling architecture for data center ACBs.

### 2. Looped thermosyphon design concept

The looped thermosyphon cooling system design concept integration into typical data center switchgear is illustrated in Fig. 2. The system comprises the ACB toe to which the evaporator section of the thermosyphon is connected. Heat is generated due to Ohmic heating in the breaker junction which is transferred through the copper toe to the evaporator. A portion of the heat is then transferred by conduction through the evaporator housing to the working fluid which evaporates. The latent heat absorbed by the vapour is transported upward via dielectric tubing to a naturally aspirated, air-cooled heat sink. Here the latent heat is released when the fluid condenses and transports across the condenser wall and onward by natural convection and radiation to the ambient surroundings. The heavier condensate is then forced by gravity downward through dielectric tubing back to the evaporator, thus feeding fresh liquid for evaporation. In this way the system is completely passive.

#### 2.1. The evaporator

In the present technology demonstrator, the evaporator, depicted in Fig. 3, is manufactured from a 90 mm × 60 mm × 10 mm solid block of copper. Two large transverse-through-holes are drilled off-centre in order to allow for bolts to pass through to facilitate clamping of the evaporator between the bus bars and the ACB toe. One key mechanical consideration for this design is that it must withstand 10.5 MPa of pressure without yielding or deforming significantly since this is required for connection of the current-carrying bus bars. The requirement of the large through holes, the mechanical strength requirement and the necessity to seal working fluid within a channelled volume to facilitate boiling heat transfer together informed the cross-drilled evaporator concept used here. As shown in Fig. 3, nine 6.25 mm wells were drilled, 5 on the vertical and 3 on the horizontal edge, to facilitate two phase flow within the passage. The cross-drilling allowed for sufficient wetted surface area within the evaporator to afford a low thermal resistance whilst ensuring mechanical strength. The wetted internal area of the evaporator was approximately 0.0086 m², which was calculated to be sufficient to provide a sufficiently low thermal resistance for the evaporator sizing during the initial design phase whilst leaving sufficient space for the large through holes and the cap fittings. On the bottom and top edges, one of the drill holes was used for the liquid return line and vapour exit line respectively. This was made possible by threaded compression fittings which were connected to the Teflon feed and return tubing. Here, Teflon was chosen for convenience due to its material compatibility with FC72 as well as its optical transparency, with the latter desirable for visualization of the flow exiting the evaporator and condenser as well as locating the liquid level in the down-comer line. In application, careful consideration should be made to ensure the dielectric tubing is chemically suited for long term operation. All other drill holes were closed and sealed with NPT threaded plugs.

The cross-drilling also preserved the structural integrity of the evaporator. Fig. 4 shows a finite element simulation of the stress...
distribution on the face of the evaporator under 10.5 MPa loading. The highest stress is ~ 2-fold lower than the yield strength of copper (~640 MPa) and the maximum deflection was under 10 µm.

2.2. The condenser

One main design considerations for the condenser is that it must have a thin form factor in order that, for scenarios where it is housed outside of the main breaker housing, it does not add significantly to the size of the cabinet. Further to this, the desired condenser should have a sufficient vertical length to provide adequate hydrostatic head. Other considerations are that the condenser should be light weight and low cost as well as have sufficient surface area to ensure a low enough overall thermal resistance.

With these in mind, a simple serpentine tube structure was chosen, as depicted in Fig. 5 which is oriented vertically and above the evaporator. A $d_{\text{coil}} = 4.75$ mm outer diameter aluminium tube is bent into the shape shown forming an overall condenser footprint of $l \times w = 450$ mm $\times$ 250 mm. To increase the surface area to volume ratio, 1.4 mm diameter bars are braised to the main tubing in a vertical orientation and act as fins for improved heat transfer. The overall height of the fins was $h_{\text{fin}} = 255$ mm and had a nominal spacing of $S_{\text{fin}} = 7.0$ mm. The outer surfaces are anodized to increase emissivity and thus increase the radiative heat transfer. The overall surface area of the condenser is about 0.058 m$^2$. Compression fittings are fitted to the inlet and outlet in order to connect with the dielectric tubing. A thermal image of the operating condenser is given in Fig. 8.

2.3. The working fluid

The working fluid must be dielectric since it is in direct contact with a live electrical component at the evaporator. Also, the maximum operating temperature of the system is 125 °C as this is the cut-off temperature of the breaker. With this in mind, a working fluid with a critical temperature above 125 °C must be used. Furthermore, the system
pressure must not be excessively above ambient from leak and mechanical failure perspectives, especially considering that non-metallic i.e. non-electrically conductive, connection tubing must be used to electrically isolate the evaporator and condenser. At the same time, the critical pressure must not be so high that at the target operating temperature range the system pressure is significantly below ambient, since this can negatively influence the ingress of air into the system. Thus, the choice of working fluid is informed by its electrical insulation properties together with thermodynamic properties that result in moderate (~20 year) service life and was thus chosen as the working fluid. Although FC72 has a relatively high GWP (~9000), only very small amounts are required (~50 mL) it is in a closed system with a long (~20 year) service life and was thus chosen as the working fluid as it is a non-toxic and low boiling point dielectric fluid with the desired operating pressure range. Further to this and has been researched extensively with a proven track record for electronics cooling applications. Some relevant properties are listed in Table 1.

3. Experimental facility and data reduction

The assembled looped thermosyphon is shown in the photographs of Fig. 6. In this benchtop configuration the evaporator sits upon an insulating block and the condenser is positioned about a meter above it. Below the condenser, the vacuum filling port is visible. The inset figure shows the evaporator depicted in Fig. 3 connected to the copper ACB toe mock up, including the connection bolts which were torqued to 70 Nm. The applied pressure of 10.5 MPa is more than sufficient to ensure a negligible thermal contact resistance of the dry-contact between the toe and the evaporator. Also shown are the pressure transducer, heater, relevant thermocouples and fill level of the working fluid. Not shown in this configuration are the two copper bus bars which are fixed to the outer sides of the toe. The bus bars, which more closely emulate the scenario within an ACB system, are 15 cm wide, 1.0 cm thick and 100 cm high, and are visible in the left thermal image in Fig. 8. For the test configuration without the bus bars, the evaporator section was insulated in order to reduce heat loss to the environment to negligible levels. In order to be consistent with its real-life application, all tests were performed at steady state.

A wireframe schematic of the experimental facility is given in Fig. 7. The thermosyphon is filled with 50 mL of working fluid by evacuating the system and drawing in the FC72 from a graduated cylinder. As discussed by Agostini et al. [12], the fill ratio must be carefully considered since too little fluid can result in early dryout of the evaporator whereas too much can result in flooding of the condenser. Here, the working fluid volume was chosen so that, when operating, there was sufficient liquid in the system that a churn-type flow was observed at the outlet of the evaporator. This ensures that there is full wetting of the evaporator walls. This liquid level in the down-comer was approximately 10 cm above the top of the evaporator, ensuring that the condenser was not flooded. This required in the region of 30–60 mL of working fluid, and it was found that the overall performance was quite insensitive to fill volumes in this range. This is consistent with the work of Tong et al. [13] who determined that, if the end-to-end system is designed well, the heat transfer performance of looped two phase thermosyphons will not be very sensitive to fill ratio because the refrigerant-side thermal resistance should only account for a small portion of the total thermal resistance of the integrated system.

As Fig. 7 depicts, the mock up ACB toe is heated by a flexible silicon heater rated to 500 W. The heater is powered by an EA-PS 8360-10 T DC power supply, which has inbuilt current and voltage readouts as well as output power. The uncertainty on the electrical power to the heater was better than ±1%.

Type-T thermocouples were positioned at strategic locations at the evaporator section. These included a heater thermocouple which was positioned in the toe mock up base at the spot where the breaker would be in real-life application. Two thermocouples were soldered into grooves which were machined into the outer surface of the evaporator block. A final thermocouple was placed in the ambient air in the vicinity of the condenser. All temperature measurements were taken from Fluke 54 T high precision meters with a calibrated uncertainty of ±0.3 °C. The system nominal pressure was measured by an Omegadyne PX01C1-05AST pressure meter fixed to the inlet port of the evaporator. The pressure meter allowed the monitoring of the system operating pressure as well as the fluid saturation temperature. The saturation temperature was determined by using the system pressure. The accuracy of the pressure sensor was ± 0.05% FS which, for FC72, translates to approximately ± 0.1 °C on the saturation temperature estimation. Finally, a FLIR thermal infrared camera was used for general thermal imaging of the system and system components, as depicted in Fig. 8.

The primary performance indicators for the looped thermosyphon are the heater temperature, $T_{\text{Toe}}$ which is measured directly by a thermocouple, and the overall toe-to-air thermal resistance. The thermal resistance was calculated as,

$$R_{\text{Tot}} = \frac{T_{\text{Toe}} - T_{\text{e}}}{Q}$$  \hspace{1cm} (1)

where $Q=IV$ is the input power to the heater as measured by the DC power supply where $I$ is the supplied current and $V$ is the applied voltage across the heater.

From a design perspective, knowledge of the component thermal resistances is important as they ultimately define the overall thermal performance. To achieve this, the heater-to-evaporator (including contact resistance) is approximated as,

$$R_{\text{Tot}} = \frac{T_{\text{Toe}} - T_{\text{e}}}{Q}$$  \hspace{1cm} (2)

where $T_{\text{e}}$ is the average temperature of the evaporator surface, as measured by the two embedded thermocouples. The next thermal resistance in the network is that of the evaporator which is calculated as,

$$R_{\text{e}} = \frac{T_{\text{e}} - T_{\text{sat}}}{Q}$$  \hspace{1cm} (3)

The working fluid saturation temperature is determined from the measured saturation pressure. In a like manner, the overall thermal resistance of the condenser, including condensation, wall and air-side resistances, can be approximated as,

$$R_{\text{c}} = \frac{T_{\text{sat}} - T_{\text{e}}}{Q}$$  \hspace{1cm} (4)

where it is straight forward to show that the air-side thermal resistance is dominant.

The uncertainty of derived parameters was calculated using the method of Kline and McClintock [14]. At The nominal power of 60 W the uncertainties on $R_{\text{e}}$, $R_{\text{c}}$ and $R_{\text{Tot}}$ are 6%, 2% and 7% respectively.

Table 1 Room temperature properties of FC72.

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Critical Temperature</td>
<td>449 K</td>
</tr>
<tr>
<td>Critical Pressure</td>
<td>1.83 × 10^6 Pa</td>
</tr>
<tr>
<td>Boiling point</td>
<td>56 °C</td>
</tr>
<tr>
<td>Kinematic viscosity</td>
<td>0.38 centistokes</td>
</tr>
<tr>
<td>Absolute viscosity</td>
<td>0.64 centipoise</td>
</tr>
<tr>
<td>Liquid Density</td>
<td>1680 kg/m^3</td>
</tr>
<tr>
<td>Liquid specific heat</td>
<td>1100 J kg^-1°C^-1</td>
</tr>
<tr>
<td>Liquid Thermal Conductivity</td>
<td>0.097 W m^-1°C^-1</td>
</tr>
<tr>
<td>Dielectric Strength</td>
<td>38 KV, 0.1° gap</td>
</tr>
<tr>
<td>Dielectric Constant</td>
<td>1.75</td>
</tr>
<tr>
<td>Electrical Resistivity</td>
<td>1.0 × 10^15 ohm cm</td>
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</tbody>
</table>
4. Results and discussion

In this section the thermal performance is first discussed for the thermosyphon as a stand-alone system in the bench-top configuration. Subsequent to this the inclusion of the bus bars is discussed in order to gain some insight into their influence on the overall performance. Finally, tests were performed in live data center switchgear.

4.1. Thermal performance without bus bars

Fig. 9 shows the thermal performance of the stand-alone looped thermosyphon system in terms of the toe-to-air thermal resistance as well as the toe-to-air temperature rise above ambient for increasing applied power levels. The toe-to-air temperature difference is shown to increase with increasing power. With regard to the thermal resistance it is clear that there is a non-linear behaviour with increasing power, whereby the thermal resistance tends to decrease with increasing thermal power being transported. This is rather common and expected behaviour in thermosyphon systems and is generally due to the improved evaporator heat transfer coefficient as the boiling becomes more rigorous with increasing heat flux [7,15,16]. This is depicted in Fig. 10 where the overall thermal resistance as well as those associated with the individual components in the thermal network are shown. Here it is clear that the most significant change in thermal resistance is that associated with the evaporator. It is noticed however that there is a small decrease in the condenser thermal resistance. Considering that natural convection and radiation are the dominant thermal resistances in this region, this decrease is due to the non-linear relationship between the condenser-to-air temperature difference and the convective and radiation heat transfer coefficients, whereby a larger temperature difference induces stronger buoyancy driven flow and radiation heat transfer and thus lower the overall thermal resistance.
The thermal resistance breakdown in Fig. 10 shows that regardless of power, the dominant thermal resistance is that associated with the condenser. This is of course the purpose of the looped heat pipe design since its function is to extract heat from a concentrated heat source and transport it to the air-side heat exchanger with as low as is feasible a temperature difference. In this way the condenser is maintained at a high relative temperature which, for buoyancy and radiation driven heat transfer, is of utmost importance. As importantly, the thermal resistance of the evaporator, albeit not negligible, is relatively small considering the small effective surface area for heat transfer across the toe-evaporator interface. There is a thermal resistance associated with conduction through the copper toe as well as the contact resistance between the toe and evaporator and this is shown to be small compared with the other resistances in the thermal network.

4.2. Thermal performance with bus bars

Fig. 11 shows the primary thermal performance metrics with the bus bars connected to the toe. For comparison, the case without the toes is also included in this figure. As it is illustrated, the inclusion of the bus bars results in an escalating improvement in the thermal resistance with increasing input power. This is of course due to the fact that the bus bars themselves act as a heat sink and provide an additional path for heat flow to the air which is thermally in parallel to the looped thermosyphon. At low powers their influence is small, though becomes progressively more significant at the heat input increases. The bus bars are ostensibly fins, in this case of semi-infinite length since they are long enough to be at ambient temperature at their tip. As the thermal power, and subsequent toe temperature increases, heat transfer to the fins increases both due to the increased base temperature and the increase in the natural convection and radiation heat transfer coefficients. For an infinite fin, the thermal resistance decreases as $R \sim h^{-1/2}$, where $h$ is the effective combined convective-radiative heat transfer coefficient which increases non-linearly with temperature. Thus, as the total thermal power is increased, the overall system thermal resistance decreases due to the lower bus bar thermal resistance combined, to a smaller extent, with the looped thermosyphon thermal resistance decrease. This is illustrated in Fig. 12 where the proportion of heat transported to the air via the thermosyphon and that by the bus bars is approximated for different input powers. The bus bar contribution was estimated by approximating the thermosyphon thermal resistance from the measured toe-to-ambient temperature difference and then calculating the associated thermal power transferred through it. Noting that the thermosyphon and bus bars are thermally in parallel, the bus bar thermal power is then estimated and the resistance calculated from knowledge of the measured temperatures. As it is shown, the relative proportion of the input thermal power transported by the bus bars continually increases with increased input power.

As mentioned earlier, the approximate maximum thermal power that is generated at the breaker contact is $\sim 60$ W and the maximum operating temperature is 125 °C, corresponding to a maximum temperature rise of 85 °C for a 40 °C ambient. Fig. 11 illustrates that the looped thermosyphon system is more than sufficient to achieve this level of cooling with a temperature rise of about 25 °C at the design power. From a practical perspective, this allows for significant reduction of the volume of copper currently being used for the current carrying bus bars without exceeding the design maximum temperature. Alternatively, the maximum current rating can be increased thus improving the electrical service of the ACB system.

Fig. 13 depicts the temperature drop associated with the nodes of the thermal network of the thermosyphon for varying heat input. Again, from a practical perspective, it is clear that the main temperature drop occurs across the condenser with comparatively minor decreases across the toe and evaporator sections. The temperature drop across the condenser could be reduced even further by increasing the overall size of the condenser, though there would be a diminishing gain since the fluid temperature within the condenser channel continuously decreases from top to bottom as the working fluid condensed and subsequently subcools, as shown in Fig. 8. Thus the local heat flux decreases owing to a decreasing condenser-to-air temperature difference and to a lesser extent so does the natural convection and radiation heat transfer coefficients due to its dependency on the wall to ambient temperature differential.

4.3. In situ testing in data center switchgear

Subsequent to bench testing, a test was performed in situ at a data center switchgear facility to provide preliminary proof-of-concept verification of the efficacy of the looped thermosyphon in real world conditions. The main components of the facility are shown in Fig. 14, which shows the ACB unit under test, the current-carrying bus bars and the transformer. The ACB unit under test was constructed with two identical ACB cabinets, left and right, that were connected in parallel such that each unit was isolated from one another yet carried the same electrical load during testing.

Fig. 15 (left) shows the rear view of the ACB unit prior to the installation of the looped thermosyphon. The unit is shown with the panel removed to illustrate the layout of the ACB toes and bus bars within the enclosure. Each ACB consisted of six breakers, 3 top and 3 bottom. For the test, the left cabinet, here termed the control console, was cooled in a conventional manner, with natural convection and radiation acting to transfer heat from to the ambient surroundings in the enclosure. The right console was fitted with one thermosyphon cooling loop and is here termed the experimental cabinet. In the figure, the left control console is shown with the dielectric isolation barrier between the top and bottom three toes, whereas the right experimental console in Fig. 15 (left) is shown prior to installation of the isolation barrier in order to better show the layout of the 6 toes. Fig. 15 (right) is a photograph of the right experimental console with the looped two phase system.
The thermosyphon and the isolation barrier were installed. The thermosyphon was installed on the middle toe of the top row as this position invariably runs hottest in operation. The lower door panel is subsequently fitted prior to testing.

The unit was instrumented with several thermocouples to monitor key temperatures during operation. The most relevant were the temperature of the twelve toes. The test was performed by passing 2000 A through each of control and experimental consoles and sufficient time was allowed for each to reach steady state, which took several hours.

The key result is shown in Fig. 16, which plots the temperature rise over ambient of the ACB toes of each unit under steady state operation. The leftmost plots, which illustrate the operation of the top and bottom toes of the control console, show operating differential temperatures over ambient in the range of 48 K–70 K, which are within the established operating criterion. As is typical, the top row of toes on the control console (LTL, LTM, LTM) run hotter than the bottom row (LBL, LBM, LBR) as they are susceptible to the heated air rising from latter combined with the deleterious effect of the flow obstruction caused by the dielectric barrier which separates the two rows. Also, it is noted that the middle toes (LTM and LBM) run hotter than their neighbours, which is also typical as they have less access, both for convection and radiation, to the ambient within the enclosure. Ultimately, the top middle toe of the control console (LMT) runs hottest, with a temperature rise over ambient of 70 K.

When comparing the left control console to that of the right experimental console, it is noted that the bottom rows of toes show very similar behaviour, as do the outer toes of the top row of each. This verifies the efficacy of using the control console as a comparison module for the experimental one. This being established, it is correct to compare the performance of the hot spot toe on the control console (LMT) with that of the partnered toe in the experimental console (RMT), to which the looped thermosyphon system has been installed.

Comparing toe LMT with toe RMT it is clear that the passive cooling system offers a significantly lower thermal resistance pathway compared with that not fitted with the system, with it operating at 26 K lower than its counterpart in the control console. This is significant considering that an approximate 40% increase in current would be required for RMT to reach the temperature rise of LMT.

5. Conclusions

A completely passive two-phase looped thermosyphon system charged with FC-72 has been designed and tested for the purpose of cooling ACBs in data center switchgear. The end-to-end passive system provides improved cooling capacity compared with what is currently installed in ACB switchgear, which relies on oversizing current-carrying bus bars and/or attaching finned heat sinks. Tests were performed on a benchtop thermal test vehicle as well as in live equipment. The main findings of the study are outlined below:

- The stand-alone looped thermosyphon system (without bus bars) operated steadily with overall thermal resistances ranging between 0.67 and 0.53 K/W for input powers between 20 and 100 W. The dominant thermal resistance was the air-cooled and naturally aspirated condenser which had a thermal resistance of ~0.4 K/W which was not very sensitive to input power. The toe connector and evaporator, including contact resistances, had much lower, though not negligible, thermal resistances.

- With the attached bus bars the overall thermal resistance decreased over that of the without-bus-bar configuration to 0.66–0.25 K/W over the same input power range. This is due to the bus bars acting as surface extensions that act thermally in parallel with the thermosyphon system.
• The relative contribution of the bus bars increases with increasing input power as their fin efficiency increases with increasing base temperature. At the design power of ~60 W the bus bars transport about one-third of the total applied power.
• At the design power the temperature rise above ambient was 25 °C. This is much below the allowable 85 °C which opens the possible opportunity for reducing the volume of copper used for the copper bars and/or increasing the power i.e. by increasing the rated current in a live ACB system. In fact, the system was tested successfully up to 290 W and the toe-to-ambient temperature rise was measured to be 72 °C illustrating a significant overhead in the thermal budget.
• Live performance tests were undertaken on two full ACB systems, typical of those used in data center power distribution systems. One cabinet acted as a control case and one acted as the experimental case, with one toe in the latter fitted with the looped thermosyphon system. The live tests show a significant drop in toe operating temperature when the passive cooling system is installed.

Overall, the outlook of using passive two phase looped thermosyphons for ACB cooling in data center switchgear is quite promising. In Fig. 15. Rear view of ACB unit (left) pre-installation of looped thermosyphon, and (right) showing thermosyphon installed on top middle toe of the ACB unit.

Fig. 16. Steady state toe temperatures rise over ambient. L = Left, R = Right, M = Middle, T = Top, B = Bottom eg. RTM = Right Cabinet, Top Row, Middle toe.
benchtop tests it performed steadily and without signs of failure up to ~300 W of input power with an above ambient temperature rise comfortably below established limits, even at over 4-times the design power. Initial live tests in a switchgear test facility showed superior cooling capability of the thermosyphon system, which proves the concept that this technology can facilitate; lower operating temperatures, enabling improved long-term reliability; higher electricity throughput per breaker, allowing higher service density of ACB switchgear, and/or; a significant reduction in the volume of copper used in their manufacture, enabling reduced cost. Future work will involve engineering and testing of the passive system to cool multiple toes simultaneously.

CRediT authorship contribution statement

A.J. Robinson: Conceptualization, Formal analysis, Investigation, Writing - original draft, Visualization. J. Colenbrander: Conceptualization, Resources, Writing - original draft, Visualization. G. Byrne: Methodology, Investigation, Resources. P. Burke: Methodology, Supervision, Project administration, Funding acquisition. J. McEvoy: Conceptualization, Methodology, Writing - review & editing, Project administration, Funding acquisition.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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