INVESTIGATION OF A RECIPOCATORY DRIVEN HEAT LOOP TO HIGH HEAT SINGLE PHASE LIQUID COOLING FOR TEMPERATURE UNIFORMITY

O.T. Popoola* and Y. Cao
Department of Mechanical and Materials Engineering, Florida International University, Miami, Florida, USA.

*Email of Corresponding Author: opopo002@fiu.edu

ABSTRACT
A bellows-type Reciprocating-Mechanism Driven Heat Loops (RMDHL) is a novel heat transfer device that could attain a high heat transfer rate through a reciprocating flow of the working fluid inside the heat transfer device. This paper investigates the possibility of applying the device for single phase liquid cooling for high performance computing. The objective of this paper is to apply the RMDHL to a liquid cooling system, experimentally and numerically, and compare its performance with a continuous cooling system. It proposes an efficiency term based on the temperature uniformity and overall performance, and compare both systems based on the proposed efficiency term. It was discovered that the RMDHL performed with an efficiency of more than 30%.

Keywords: Heat loop, reciprocating flow, cooling, heat flux.

INTRODUCTION
The most significant hindrances to the technological advances in power electronics, plasma-facing components, high heat-load optical components, laser diode arrays, X-ray medical devices, power electronics for hybrid vehicles as well as for toys and appliances, high-power computers, power generation, optoelectronic equipment, thermal management for the leading edges/nose caps of supersonic/hypersonic cruise vehicles, advanced gas turbines, and new directed energy based weapon systems has always been the issue of effective thermal management. Microelectronic chips may dissipate heat fluxes as high as 10 W through a 5 mm x 5 mm side (400 W/cm²) and heat fluxes over 1,000 W/cm² have been projected (Liu et al. 2014; Hetsroni et al. 2001). The failure rate of these systems themselves increases exponentially with temperature (DARPA - Microsystems Technology Office 2010). An effective thermal management system must find a solution to high heat flux as well as the non-uniform system temperature or heat flux distribution across the surfaces. Efforts are actively ongoing to develop better solutions to this problem.

An effective design options for a cooling high heat flux system is to use a two-phase pumped cooling loop to simultaneously satisfy the temperature uniformity and high heat flux requirements (Williams and Roux 2007). Cao and Gao (2003), Cao and Gao (2008) and Cao et al. (2013) conceived, designed and tested a novel solenoid and bellows-type Reciprocating-Mechanism Driven Heat Loops (RMDHL) or the reciprocating-flow heat loops, which attains a reciprocating flow of the working fluid inside the heat transfer device without requiring a reciprocating motion of the entire heat transfer device. The RMDHL includes a hollow loop having an interior flow passage, an amount of working fluid filled within the loop, and a reciprocating driver. The heat loop has an evaporator section, a condenser section, and a liquid reservoir. The reciprocating driver is integrated with a liquid reservoir and facilitates a reciprocating flow of the working fluid within the heat loop, so that liquid is supplied from the condenser section to the evaporator section and a high heat transfer rate from the evaporator section to the condenser section is achieved. Temperature uniformity is also attained when the air is evacuated from the loop and the working fluid hermetically sealed within the loop is under a substantially saturated condition. Experimental results for solenoid based heat loop indicated that the heat loop worked very well and could handle a heat flux of more than 300 W/cm² in the evaporator section. Additional studies are being carried out to extend the device to cool heat flux of up to 1000 W/cm² while simultaneously optimizing material and energy utilization.

With many choices of cooling approaches, and high number of process variables, it is essential that each cooling system be investigated on a case by case basis. It will also be ideal to determine a correlation that will provide a true representation of the efficiency of the system. Computer simulation has become an invaluable tool for greatly improving thermal management for particular applications (Williams and Roux 2007). This study aims to investigate the possibility of applying the RMDHL to possible liquid cooling systems and compare its performance to that the performance of a dynamic pumps driven heat loop (DPDHL) in a continuous cooling system. This experimental and numerical study looks into the heat transfer and fluid flow aspects of the (RMDHL) using the available commercial software. The main objective of the present study is to validate the numerical investigation with the experimental research and compare the performance of a RMDHL with the
conventional continues cooling loops in terms of
temperature, uniformity and heat removal from the
surface of the heat source. For the current study a 3D,
laminar flow setup of continuous and RMDHL cooling
loops with liquid water as the cooling fluid is provided.
The setup and its results will be the foundation for the
turbulent two phase model for various cooling fluids. It
is expected that the completed model will be used to
effectively evaluate the general concepts of RMDHL for
future development and optimization of a standard
industrial product.

CONCEPTUAL DESIGN OF THE BELLOW-
TYPE RMDHL
Cao and Gao (2003), Cao and Gao (2008) and Cao et
al. (2013) conceived, designed and tested novel
solenoid and bellows-type Reciprocating-Mechanism
Driven Heat Loops (RMDHL) or the reciprocating-flow
heat loops, which attains a reciprocating flow of the
working fluid inside the heat transfer device without
requiring a reciprocating motion of the entire heat
transfer device. The RMDHL uses cooling loop to
simultaneously satisfy the temperature uniformity and
high heat flux requirements (Cao et al., 2013). All design
parameters remain as specified by (Cao et al., 2013). The
most important relation that describes the critical
requirement for the operation of the heat loop is given as,

\[ A_p S \geq A_e L_e + 2A_t L_t + A_c L_c \]

(1)

The length and average interior cross-sectional area of
the evaporator are denoted by \( L_e \) and \( A_e \), respectively, the
length and average interior cross-sectional area of the
connection tubing between the evaporator and the
condenser are \( L_t \) and \( A_t \), the length and interior cross-
sectional area of each condenser section are \( L_c \) and \( A_c \).
The piston cross-sectional area and reciprocating stroke
are \( A_p \) and \( S \), respectively.

From the geometry of the heat loop Fig 1b and 2b, and
consideration of all the components of the RMDHL
designed in equation (1), the critical \( S \) was calculated as
7 cm. A value of 7.62 cm was used for the reciprocating
driver.

PERFORMANCE EVALUATION OF THE
MODEL BELLOW-TYPE RMDHL
The prototype used for this investigation is shown in Fig
2. The experimental setup for the bellows-type RMDHL
demonstration model is similar to the setup shown in Fig.
1 (Cao et al., 2013). The setup includes electric heaters
for supplying heat to the cold plate of the heat loop, a DC
power supply to the reciprocating driver, a control circuit
board to control the reciprocating frequency of the
driver, a constant temperature circulator to maintain a
constant coolant inlet temperature of 10°C to the
condenser, which would also control the operating
temperature level of the heat loop, a data acquisition
system, and the fabricated heat loop.

Fig 1: Concept of a below-type reciprocating-mechanism
driven heat loop

Eight Omega flexible heaters (four on each side of the
cold plate) are clamped by two aluminum plates onto the
cold plate with insulation layers sandwiched between the
heaters and clamping plates. Each heater has a dimension
of 2 cm by 12 cm and could provide a heat input up to
200 W.
Twelve thermocouples were placed at different locations of the heat loop. Nine of them (numbered as No. 1 through No. 9) are placed on the top surface of the cold plate at different locations for the study of temperature uniformity on the cold plate. Figure 3b shows detailed locations of these thermocouples. One thermocouple is placed near the condenser and another two thermocouples are placed on two ends of the bellows pump to monitor the heat loop temperature distribution in addition to those on the cold plate. These three thermocouples are numbered, respectively, as T10, T11, and T12, as shown in Fig. 3a.

During the experimentation, the heat loss from the clamping plate to the ambient was amounted to be less than 1%. Therefore, the sources of the experimental uncertainty were primarily due to the instruments themselves. The scanning thermocouple thermometer has an accuracy of ±0.1% of reading ±0.4°C. The power meter has an accuracy of ± (1% reading + 5 digits). So the maximum uncertainty for the temperature and heat power measurements would be 1.0% and 2%, respectively.

**NUMERICAL ANALYSIS**

A Computational fluid Dynamic (CFD) code was employed to numerically simulate both the Dynamic Pump Driven Heat Loop (DPDHL) and the reciprocating loop. The CFD code yields numerical solutions of the equations. The first step of the model development was the generation of a 3D CAD model of the process by SolidWorks.

The condenser and evaporator loops are enclosed with solid wall. The two loops are separated from each other by the wall off the part body. The solid walls are aluminum and thermophysical properties for them are obtained from the fluent database. The working liquid in the both loops is water. The heat fluxes are applied on both side of the evaporator and the boundary conditions are presented in the Table 1.

\[ V = \pi n S \sin(2\pi nt) \]  

The same average velocity was applied to the DPDHL system but increased by a factor of 22/14.

In the present study for the DPDHL loop the following assumptions are made:

(i) Both fluid flow and heat transfer are in steady-state and two dimensional.
(ii) Fluid is in single phase and flow and laminar.
(iii) Properties of both fluid and heat sink materials are temperature independent.
(iv) All the surfaces of heat sink exposed to the surroundings are assumed to be insulated except the walls of evaporator where constant heat flux simulating the heat generation from different components.

![Fig 2: (a) assembly diagram of RMDHL (b) Physical dimensions of cold plate (a) Geometry of the RMDH loop](image-url)
Fig 2: (c) Geometry of the RMDH loop used for numerical model (d) Grid distribution for the RMDH loop

Fig. 3: Experimental setup: (a) Overall arrangement and (b) Thermocouple locations on the cold plate (size in mm)

Table 1. Dimensions, geometries and boundary conditions for the present cooling loop

<table>
<thead>
<tr>
<th>Boundary conditions</th>
<th>Values and their ranges.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet velocity of the evaporator</td>
<td>varying stroke (cm) 7.62, 8.62, 9.62, 10.62</td>
</tr>
<tr>
<td></td>
<td>frequency (n) 0.14, 0.28, 0.42, 1</td>
</tr>
<tr>
<td>Inlet velocity of the condenser</td>
<td>1.223 m/s</td>
</tr>
<tr>
<td>Inlet temperature of the condenser</td>
<td>283 K</td>
</tr>
<tr>
<td>Solid wall thickness</td>
<td>0.0001 m</td>
</tr>
<tr>
<td>Heat (Q)</td>
<td>Heat (W)</td>
</tr>
<tr>
<td></td>
<td>551</td>
</tr>
<tr>
<td></td>
<td>500</td>
</tr>
<tr>
<td></td>
<td>455</td>
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<tr>
<td></td>
<td>400</td>
</tr>
</tbody>
</table>

**Governing Equations**

The geometry and thermal boundary conditions for the RMDHL is a bit more complex than the DPDHL. DPDHL will have permanent inflow and outflow regions, where one can easily define the inlet velocity and temperature boundary conditions. RMDHL flows require interchange between the inflow and out-flow boundaries during a cycle. For most applications, it is difficult to
determine the inflow/outflow boundary conditions, since fluid particles exiting the flow domain during a part of the cycle are fed back into the domain, later in the cycle. The DPDHL which represents the case of forced convection heat transfer in channels and tubes is well understood. Several results in analytical and empirical relations for the Nusselt number variations in terms of the flow parameters are available (Faghri et al., 2010).

The displacement of the piston $x_p(t)$ with time (De-Jongh and Rijs 2004) is

$$x_p(t) = - \frac{1}{2} S - \frac{1}{2} S \cos (\omega t)$$  

(3)

where, $\omega$ is the pump frequency in radians per second; stroke frequency $n$ (in strokes or cycles per second) follows from:

$$n = \frac{\omega}{(2\pi)}$$  

(4)

The average flow $q_{av}$ is equal to the product of stroke volume and pump frequency (Jongh and Rijs, 2004):

$$q_{av} = \frac{\omega}{(2 \pi) S A_p}$$  

(5)

Based on above assumptions, the governing equations for fluid and energy transport are

Fluid flow:

$$\nabla \cdot \vec{V} = 0$$  

(6)

$$\rho \left( \frac{\vec{V} \cdot \nabla V}{\rho} \right) = -\nabla p + \mu \nabla^2 \vec{V}$$  

(7)

Energy in fluid flow

$$\rho c_v \left( \frac{\vec{V} \cdot \nabla T}{\rho} \right) = k \nabla^2 T$$  

(8)

Energy in heat sink solid part

$$k (\nabla^2 T, n) = 0$$  

(9)

The boundary conditions for these equations are specified in the previous table. Based on the operating conditions described above, the boundary conditions for the governing equations are given as:

Inlet:

$$V = V_{in}, \quad T = T_{in}$$  

(10)

Outlet:

$$P = P_{out}, \quad \frac{\partial T}{\partial n} = 0$$  

(11)

Fluid–solid interface:

$$\vec{V} = 0, \quad T = T_{s}, \quad -k_n \frac{\partial T}{\partial n} = -k_n \frac{\partial T}{\partial n}$$  

(12)

At the top wall:

$$q_w = -k_n \frac{\partial T}{\partial n}$$  

(13)

In Eq. (9), $V_{in}$ and $T_{in}$ are the fluid inlet pressure and temperature, respectively; $P_{out}$ is the pressure at the outlet, $n$ is the direction normal to the wall or the outlet plane, and $q_w$ is the heat flux applied at the top wall of the heat sink.

The Geometry and grid distribution for RMDHL is as same as DPDHLHL. To simulate a closed RMDHL loop the following setup is used:

For RMDHL loop:

a) Instead of a closed loop, an open loop is simulated.
b) A UDF is used to generate the sinusoidal inlet velocity for the evaporator loop.
c) Using a UDF, the inlet temperature of the loop is set as the average outlet temperature of the loop and the backflow temperature of the outlet is set to the average temperature of the inlet boundary to mimic the temperature and velocity in distribution of a closed loop.

The CFD numerical algorithm utilizes Pressure Implicit with Splitting of Operators (PISO) formulation for the solution of the Navier-Stokes and heat transport equations. Gradient discretization was Green-Gauss Node based, pressure discretization was Second order while the momentum and energy discretization were Second order upwind. The turbulent Kinetic Energy and turbulent dissipation were First order upwind using the $k-\epsilon$ turbulence model. Transient formulation was second order Implicit. Appropriate under relaxations were used to improve the numerical stability for all governing equations. The criteria for convergence was set at $10^{-4}$ for the continuity, $x, y$ and $z$ momentum, turbulent kinetic energy and turbulence dissipation rate. While a convergence criterion of $10^{-8}$ was set for the DPDHL and $10^{-6}$ for the RMDHL.

**RESULTS AND DISCUSSION**

Details of the numerical solution scheme are presented in Table 2. The resulting computational mesh is presented in Fig. 2d. In order to establish computational accuracy, grid independence studies are always necessary and were equally performed in this work. The mesh independence studies were conducted for two more grids with coarse and fine meshes settings. A variation of about 7% (max) with respect to temperature variation on the flux surface. The results of the computational study were compared with the experimental work and in general, good agreement with experimental data see Fig 4. Hence the numerical solution was used as the basis for the result analysis.

For the RMDHL, the temperature profile is more uniform than the temperature profile for the DPDHL. Table 3 and Table 5 show that not only is the mean temperature for the RMDHL lower than the mean temperature for the DPDHL, the standard deviation around the mean is about 1.5 °C for the RMDHL and about 5°C for the DPDHL. The maximum temperature
occurs at the midpoint of the walls while the maximum temperature for the DPDHL is at the exit side of the cold plate. In terms of performance; thus, the RHMDL is more effective in cooling the plate than the DPDHL by up to 5°C. As shown in Table 5, for a heat transfer rate of 550 W, 500 W, 450 W, 400 W, the DPDHL produced an average hot plate temperature of 331.71, 327.77, 323.18, and 318.19 °C, respectively; while the RMDHL produced an average of 328.09, 327.31, 324.37 and 313.19 °C, respectively. The basis for determining the average hot plate temperature is provided in Equation (16). The reason for the better performance is that RMHDL is that rate of heat loss to the condenser is higher for the RMDHL than for the DPDHL. Fig 5 and Fig 6 shows the temperature distribution across the evaporator plate for RHMDL at 550W and 450W and also the DPDHL for 550W and 450W. The colder sections represent the portion of the evaporator in contact with the condenser. The reciprocating loop maintains a higher on the condenser side than the continuous loop and this represents a higher heat removal rate from evaporator by the reciprocatory loop.

Table 2: Details of the Numerical solution variables

<table>
<thead>
<tr>
<th>Solver parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solution Method</td>
<td>PISO</td>
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<tr>
<td>Time step (s)</td>
<td>0.5</td>
</tr>
<tr>
<td>Relaxation factor momentum</td>
<td>0.4</td>
</tr>
<tr>
<td>Relaxation factor pressure</td>
<td>0.3</td>
</tr>
<tr>
<td>Relaxation factor energy</td>
<td>0.9</td>
</tr>
</tbody>
</table>

Table 3: Standard deviation of temperature across cross sections of hot plate

<table>
<thead>
<tr>
<th></th>
<th>400W</th>
<th>550W</th>
<th>400W</th>
<th>550W</th>
</tr>
</thead>
<tbody>
<tr>
<td>Section A-A</td>
<td>2.96</td>
<td>4.09</td>
<td>1.67</td>
<td>1.38</td>
</tr>
<tr>
<td>Section B-B</td>
<td>2.86</td>
<td>3.96</td>
<td>0.86</td>
<td>0.86</td>
</tr>
<tr>
<td>Section B-B</td>
<td>3.49</td>
<td>4.74</td>
<td>1.99</td>
<td>1.51</td>
</tr>
</tbody>
</table>

Fig 4: Comparison of numerical and experimental Temperature distributions over the cold plate at different heat inputs for a coolant inlet temperature of 10°C (a) 550 W (b) 400 W.

Fig 5: Contour of varying temperature across evaporator for varying heat transfer rate
Another emphasis of this paper is the temperature uniformity over the cold plate. Fig 5 and Fig 6 show the temperature continues loop for cold plate at a condenser inlet velocity of 1.23 m/s, heat transfer rate varying from 400W to 550W and inlet temperature of the condenser 283 K. The figures clearly show that the uniformity of the temperature profiles of the RMDHL is much higher than the uniformity of the DPDHL loop. As seen in Fig 6, there are two temperature gradients. One gradient along the evaporator and another gradient across the evaporator. The gradient along the evaporator is common to both the DPDHL and the RMDHL and is as a result of pressure drop along the piping in the evaporator. The effect of this gradient can be reduced if a different piping configuration is adopted.

The empirical variation of the temperature is investigated at 3 cross-sections. The cross sections are shown in Fig. 7, section A-A at one end, Section B-B at the center, and section C-C at the other end. Temperatures at the centerline was compared for both reciprocating and DPDHL flow at two heat transfer rate, 400 and 550 W and compared statistically. Table 4 presents the details of the analysis. The plate temperature is minimum at the inlet of the evaporator and gradually increases toward the outlet of the outlet for the DPDHL flow loop. Another statistic indicator that may be used to gage the temperature uniformity over the cold plate is the standard deviation, defined (Cao et al. 2013) as,

$$\sigma_T = \sqrt{\frac{1}{N-1} \sum_{i=1}^{N} (T_i - T_m)^2}$$

(14)
For this purpose, the steady-state minimum temperature, maximum temperature, and the temperature difference, at different power input for water coolant inlet temperatures of 10°C is shown in Table 4. Although the maximum temperature differences may vary with the power input, they are generally around 6°C for the RMDHL and 20°C for the DPDHL loop. The standard deviation for the RMDHL is less than half of the value for the DPDHL (Table 4).

The standards deviation of the RMDHL changes only slightly with increasing rate of heat transfer but for the DPDHL it increases significantly with increasing rate of heat transfer. It should be pointed out that all of the data presented herein are based on a low driver reciprocating frequency of 0.14/s for a maximum stroke. The figures show that not only is the mean temperature for the RMDHL lower than the mean temperature for the DPDHL, the standard deviation around the mean is about 1.5°C for the RMDHL and about 4°C for the DPDHL.

Table 4: Temperature variation across various cross sectional areas of the hot plate

<table>
<thead>
<tr>
<th>Q (W)</th>
<th>400</th>
<th>550</th>
<th>400 RMDHL</th>
<th>550 RMDHL</th>
</tr>
</thead>
<tbody>
<tr>
<td>x (m)</td>
<td>T&lt;sub&gt;min&lt;/sub&gt;</td>
<td>T&lt;sub&gt;max&lt;/sub&gt;</td>
<td>ΔT</td>
<td>T&lt;sub&gt;min&lt;/sub&gt;</td>
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<tr>
<td>0.01</td>
<td>305.90</td>
<td>321.80</td>
<td>15.93</td>
<td>314.70</td>
</tr>
<tr>
<td>0.03</td>
<td>305.90</td>
<td>321.20</td>
<td>15.33</td>
<td>314.80</td>
</tr>
<tr>
<td>0.04</td>
<td>306.00</td>
<td>320.60</td>
<td>14.64</td>
<td>314.90</td>
</tr>
<tr>
<td>0.06</td>
<td>306.00</td>
<td>320.40</td>
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<tr>
<td>0.08</td>
<td>306.10</td>
<td>320.70</td>
<td>14.53</td>
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</tr>
<tr>
<td>0.10</td>
<td>306.70</td>
<td>321.10</td>
<td>14.39</td>
<td>315.50</td>
</tr>
<tr>
<td>0.12</td>
<td>306.80</td>
<td>322.00</td>
<td>15.15</td>
<td>316.10</td>
</tr>
<tr>
<td>0.14</td>
<td>311.90</td>
<td>322.40</td>
<td>10.58</td>
<td>323.10</td>
</tr>
</tbody>
</table>

Fig 7: Volume rendering of cross section of hot plate showing cross section for temperature.
It should be pointed out that all of the data presented herein are based on a low driver reciprocating frequency of 0.14/s for a maximum stroke. The figures show that not only is the mean temperature for the RMDHL lower than the average temperature for the DPDHL cooling loop, the standard deviation around the mean is about 1.5 °C for the RMDHL and about 4 °C for the DPDHL. Fig. 9 reveals that although the variation in hot plate temperature is different for the RHMDL and the DPDHL, the variation in the range of the temperature is the same. The range is highest at the entrance; it begins to drop towards the outlet of the fluid.

\[ \text{COP}_{cPT} = \frac{T_{cp}}{T_{hf} - T_{cp}} \]  

(15)

\[ \text{COP}_{hPT} = \frac{T_{hf}}{T_{hp} - T_{hf}} \]  

(16)

\[ \eta_T = \frac{\text{COP}_{hPT}}{\text{COP}_{cPT}} \]  

(17)

\[ T_{hp} = \text{average temperature of hot plate (ATHP)}; \quad T_{cp} = \text{average temperature of cold plate (ATCP)} \]

\[ T_{hf} = \text{average temperature of hot fluid (ATHF)}; \quad \eta_T = \text{efficiency} \]

\[ T_i = \frac{\sum_{i=1}^{n} T_i}{n} \]  

(18)

In a bid to determine the efficiency of both systems, this work also proposed an expression for the Coefficient of Performance (COP) based on temperature. The requirement of an adequate coefficient in predicting the system is that the coefficient must satisfy the second law of thermodynamics (Moran and Shapiro, 2004) and should be dependent on operating conditions, especially average plate temperature and relative temperature between sink and system. The coefficient is proposed in Equations 15 - 17. The COP terms is similar to the term used for determination of the COP for Carnot vapor refrigeration cycle. For the basis for selecting the cold reservoir and the hot reservoir is defined in eq 18. The summary of the calculations is presented in Table 5. Table 5 shows the value of the COP and the resulting efficiency is similar irrespective of the changing heat flux, and the various temperature parameters (Fig 8) used in computing the terms on the hot plate. The trend follows variations in similar terms in the literature (Moran and Shapiro 2004; Mcquiston et al. 2006). Ultimately the efficiencies show that the value of the efficiency of the RMDHL is about 30% higher that those attained by the DPDHL flow system (Fig 9b).

**CONCLUSION**

A numerical model for the simulation of the RMDHL has been successfully built and validated by experimental data. The results indicate that the cooling efficiency of the RMDHL is meaningfully higher than that of the DPDHL with a reduced maximum temperature under the same heat input and cooling conditions. The results show that the temperature increases with increase of heat flux on the walls or decrease of the flow rate. It was also shown that the temperature profiles are more uniform for an RMDHL loop compared to a DPDHL loop both in terms of temperature uniformity and efficiency.
Table 5: Average temperature, standard deviation and efficiency of cooling systems

<table>
<thead>
<tr>
<th>Cooling system</th>
<th>Parameter</th>
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<th>500.00</th>
<th>455.00</th>
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<tr>
<td></td>
<td>Heat transfer rate Q[W]</td>
<td></td>
<td></td>
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</tr>
<tr>
<td>DPDHL</td>
<td>ATHP (K)</td>
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<td>327.77</td>
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<tr>
<td></td>
<td>ATCP (K)</td>
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<td>289.67</td>
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<tr>
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<td>COPₚₜ (K)</td>
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<td>23.01</td>
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<tr>
<td></td>
<td>COPₚₜ (K)</td>
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<td>11.85</td>
<td>13.21</td>
<td>14.77</td>
</tr>
<tr>
<td></td>
<td>η ()</td>
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<td>0.51</td>
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</tr>
<tr>
<td></td>
<td>σₚₜ (K)</td>
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<td>3.54</td>
<td>3.17</td>
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<td>RMDHL</td>
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<td>327.31</td>
<td>324.37</td>
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</tr>
<tr>
<td></td>
<td>ATCP (K)</td>
<td>295.03</td>
<td>294.84</td>
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</tr>
<tr>
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<td>311.64</td>
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<td>COPₚₜ (K)</td>
<td>15.95</td>
<td>16.45</td>
<td>17.62</td>
<td>22.12</td>
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<td></td>
<td>η ()</td>
<td>0.74</td>
<td>0.76</td>
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<td>σₚₜ (K)</td>
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<td>1.34</td>
<td>1.51</td>
<td>1.62</td>
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REFERENCES


