4. Results

Experimental data are obtained by fixing the hot plate temperature to 250ºC and varying the voltage supply to the fan. This leads to different flow rates and, therefore, to different average flow velocities in the rectangular tube v_∞.
Results for both hydraulic and thermal variables without bypass are shown in Fig. 3.

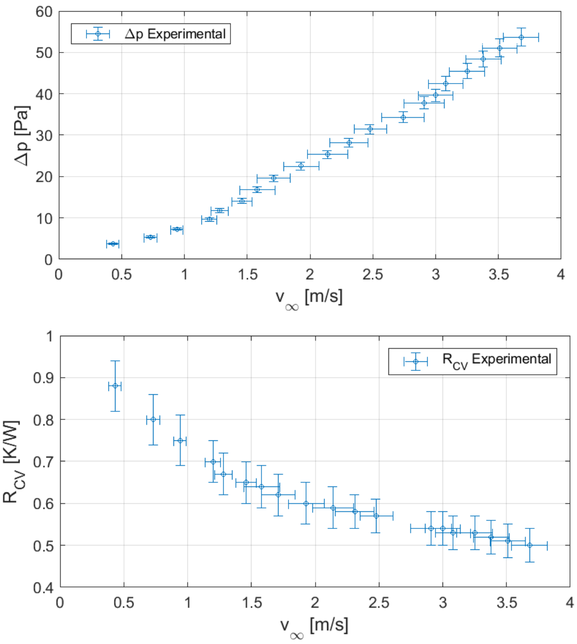

Fig. 3. Pressure drop through the heat sink (top) and thermal convective resistance (bottom) as a function of the air average velocity within the rectangular tube. No bypass case.

The air flow pressure drop through the heat sink behaves almost linearly as a function of the mean velocity at the rectangular tube. This is caused by the laminar regime observed in the flow between the channels of the dissipater, with Re < 1800 for the maximum flow rate available in our experiments. On the other hand, the thermal convective resistance values are similar to those reported by plate-fin heat sink manufacturers.
The comparison with numerical models is shown in Fig. 4 where the relative differences (%) between the modelled data and the experimental ones are reported. For the pressure drop, Lindstedt and Karvinen [6] and Kays and London [5] are the equations that provide a better fitting with the observations, especially in the high velocity range (with differences lower than 15% and 30%, respectively). With respect to the thermal model, the Lindstedt and Karvinen one is the best one with very accurate predictions in a broad range of flow rates (differences lower than 7% for most of the cases being tested). We point out that Kays and London [5] do not provide a thermal model.
For the bypass case, the experimental data is shown in Fig. 5. Note that the pressure drop values were quite similar to those obtained in the non-bypass set up. This is due to the different heat sink type employed in the bypass test (see Section 2 and Fig. 2).
In comparison with the heat sink employed in the non-bypass case, the thermal convective resistance of the heat sink for the bypass case is smaller, especially at large flow rates. The flow in between the channels of the heat sink is also laminar.

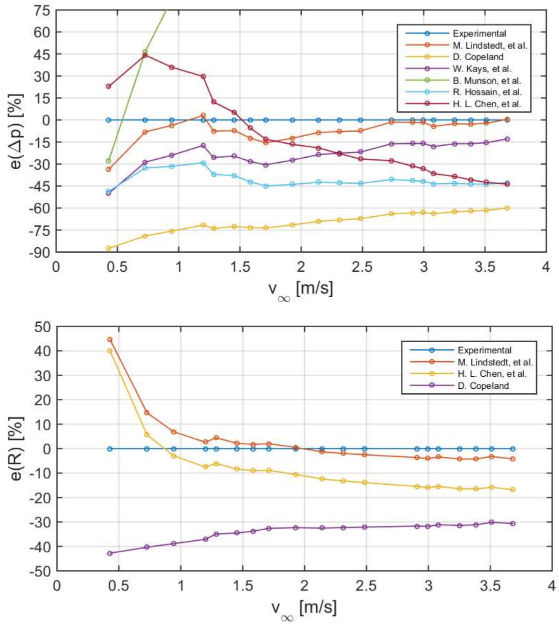

Fig. 4. Relative differences between modelled and experimental data of pressure drop through the heat sink (top) and thermal convective resistance (bottom) as a function of the air average velocity within the rectangular tube. No bypass case.

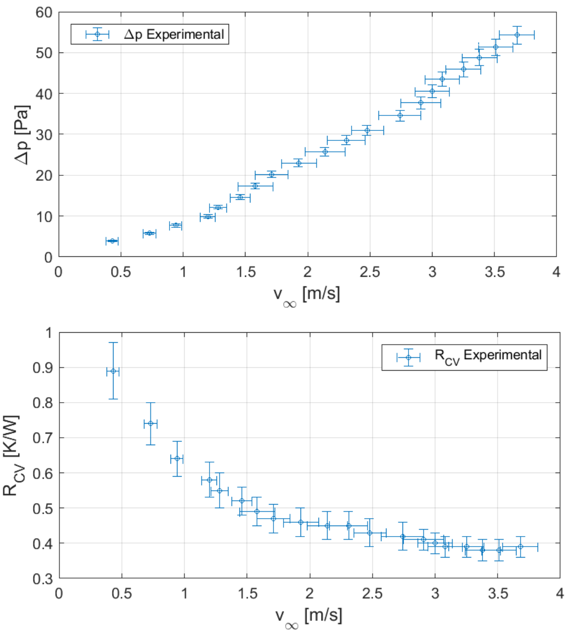

Fig. 5. Pressure drop through the heat sink (top) and thermal convective resistance (bottom) as a function of the air average velocity within the rectangular tube. Bypass case.

Although the Lindsted and Karvinen model [6] does not consider the possibility of a bypass, and in view of its good behaviour observed in the non-bypass case, we have generalized this model in order to include the bypass cases. It has been done by assuming equal pressure drop between points 2 and 5 in Fig. 2 when calculated from the flow through the bypass and from the flow through the heat sink. This condition allows us to determine the fraction of the incoming flow that moves through the heat sink. The comparison of the models with measurements for the bypass case is shown in Fig. 6. The Lindstedt and Karvinen [6] still provides the best accurate prediction, with deviations less than 10% in most of the cases.

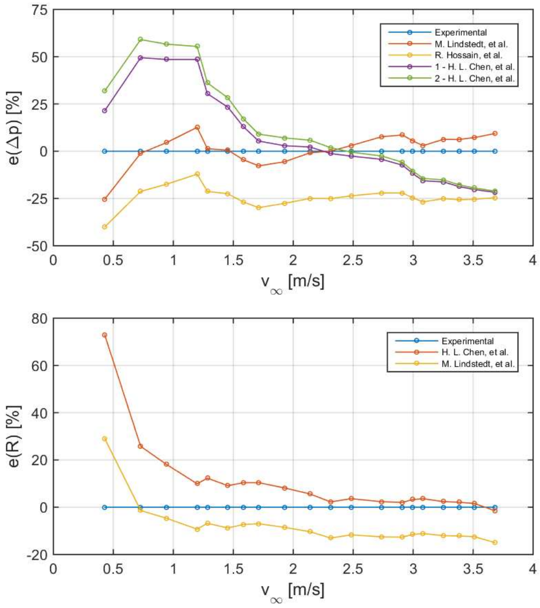

Fig. 6. Relative differences between modelled and experimental data of the pressure drop through the heat sink (top) and thermal convective resistance (bottom) as a function of the air average velocity within the rectangular tube. Bypass case.

Finally, we plot the experimental data of pressure loss and thermal convective resistance for both non-bypass (Fig. 7) and bypass (Fig. 8) cases adding the results of Lindstedt and Karvinen model [2]. The agreement is remarkable for both hydraulic and thermal variables although it diminishes at low air velocities, especially for the thermal convective resistance value.

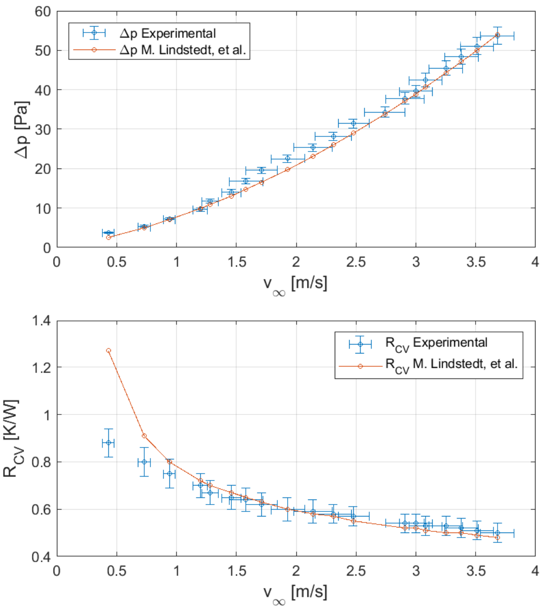

Fig. 7. Experimental data and predictions of the Lindstedt and Karvinen model [6] for the pressure drop through the heat sink (top) and thermal convective resistance (bottom) as a function of the air average velocity within the rectangular tube.

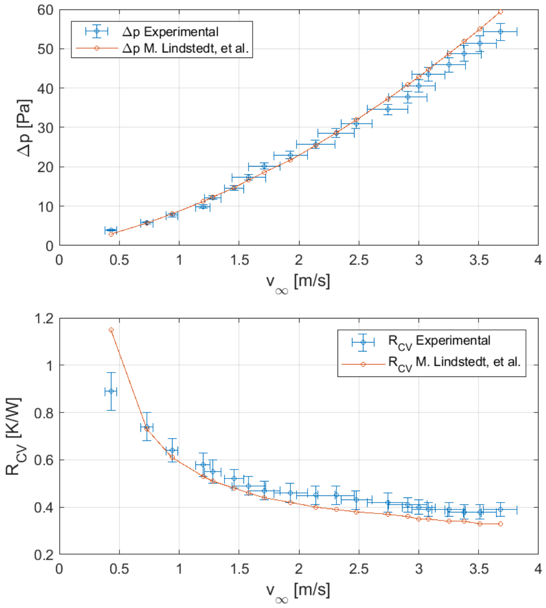

Fig. 8. Experimental data and predictions of the Lindstedt and Karvinen model [6] for the pressure drop through the heat sink (top) and thermal convective resistance (bottom) as a function of the air average velocity within the rectangular tube.

5. Conclusion

Several hydraulic and thermal models of plate fin heat sinks under forced convection have been analysed when applied in a thermoelectric power generation system. Model predictions have been compared with experimental data with and without bypass flow. The Lindstedt and Karvinen [2] model is the most accurate, with discrepancies below 12% in the majority of the values evaluated. It is recommended for the design of plate-fin heat sinks in thermoelectric power generation.

Acknowledgement

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