Pressure Drop of Impingement Air Cooled Plate Fin Heat Sinks

The performance of impingement air cooled plate fin heat sinks differs significantly from that of parallel flow plate fin heat sinks. A simple impingement flow pressure drop model based on developing laminar flow in rectangular channels is proposed. The model is developed from simple momentum balance and utilizes fundamental solutions from fluid dynamics to predict its constitutive components. To test the validity of the model, experimental measurements of pressure drop are performed with heat sinks of various impingement inlet widths, fin spacings, fin heights, and airflow velocities. The accuracy of the pressure drop model was found to be within 20% of the experimental data taken on four heat sinks and other experimental data from the published literature at channel Reynolds numbers less than 1200. The simple model is suitable for impingement air cooled plate fin heat sinks parametric design studies. [DOI: 10.1115/1.2721094]

Keywords: impingement flow, heat sink, pressure drop, plate fin

1 Introduction

The heat dissipated in electronic components is increasing with advances in the performance of modern computers. Therefore, thermal management in the electronics environment is becoming increasingly difficult due to high heat load and dimensional constraints. Impingement air cooling with heat sinks is one attractive solution to these problems.

Nottage [1] suggested that the heat sink fin and channel may be thought of as a type of heat exchanger in which the hot fluid stream is replaced with the solid fin. The flowstream direction relative to heat flow direction plays a significant role in determining the heat transfer effectiveness of a fin-fluid arrangement. Three basic one-dimensional heat exchanger flow arrangements are counterflow, crossflow, and parallelflow. The counterflow arrangement has the greatest potential to achieve high effectiveness. This requires an airflow direction normal to the heat sink base. Since the impingement airflow in a heat sink is intermediate between counterflow and crossflow, its thermal performance is expected to exceed that of a crossflow heat sink.

The present work is focused on the impingement flow plate fin geometry. The research objectives are to develop a robust model for predicting pressure drop of plate fin heat sinks for impingement cooling. To test the validity of the model, experimental measurements of pressure drop are performed with heat sinks of various dimensions and flow velocities.

2 Literature Review

Culham and Muzychka [2] proposed a heat sink model in parallel flow using the apparent friction factor model developed by Muzychka and Yovanovich [3]. The friction model is asymptotic between a developing and fully developed flow. Muzychka and Yovanovich [3] validated the model with most of the developing flow data and found the estimation error was within ±12% for a wide range of duct shapes but within ±3% for the rectangular channel. Copeland [4] suggested using a laminar flow pressure drop model for parallel flow in isothermal rectangular channels to model the heat sink. The friction factor data for developing laminar flow were taken from Shah and London [5] and fitted to an equation of the Churchill–Usagi form.

Although there has been a wide range of research reporting on impingement air cooling, there have been few studies specifically on impingement cooling with heat sinks. The geometry of a heat sink in impingement flow is shown schematically in Fig. 1. In this flow arrangement the air enters at the top and exits out the sides, i.e., top inlet side exit (TISE). Biskeborn et al. [6] reported experimental results for a TISE design using unique “serpentine” square pin fins. Sparrow et al. [7] performed heat transfer experiments on an isothermal TISE type single channel passage. A novel laminar-flow heat sink with two sets of triangular or trapezoidal shaped fins on the two inclined faces of a base has been reported by Hilbert et al. [8]. The impingement air divides into two streams which flow between the two sets of fins in a direction that is transverse to the direction of heat transport in the fins. This design is efficient because the downward flow increases the air speed near the base of the fins where the fin temperatures are highest. By having the cool air enter at the center of the heat sink and exit at the sides, the length of the fins in the flow direction is reduced so that frictional pressure drop is decreased. Sathe et al. [9] conducted a numerical and experimental study of a TISE plate fin heat sink that was notched in the center to reduce flow stagnation. It was found that pressure drop is reduced by cutting the fins in the central impingement zone without losing the heat transfer. Copeland [10] performed theoretical, experimental, and numerical analyses on a manifold microchannel heat sink with multiple top inlets alternated with top outlets. At a given pumping power, increasing the number of inlet/outlet channels requires an increase in the volume flow rate, but permits higher flow velocity, and produces significantly lower pressure drop. Kang and Holahan [11] developed a one-dimensional pressure drop model of impingement air cooled plate fin heat sinks to understand how the heat sink performance depends on the different geometry variables. Holahan et al. [12] modeled the impingement flow field in the channel between the fins as a Hele–Shaw flow. Kondo et al. [13] reported on a semi-empirical development of a pressure drop prediction for impingement cooling of heat sinks with plate fins. The flow region is divided into five parts. Each part is modeled by different pressure drop models. These predictions agree with the experimental data within ±30%. Dividing the heat sink into parts requires a large number of equations and makes the model very complicated. Sathe et al. [14] presented a computational analysis for three-dimensional flow and heat transfer in the IBM 4381 heat sink. Biber [15] carried out a numerical study to determine the pressure drop of a single isothermal channel with variable width impingement flow. Biber numerically studied many different com-
3 Theoretical Analysis

The pressure drop model for the impingement flow plate fin sink will be based on correlations for laminar duct flows, which are essentially one dimensional. We need only study one half of the heat sink since the flow field and pressure fields on the other half are a mirror image due to symmetry. One half of the impingement cooling heat sink channel is considered as two connected rectangular channels; one is vertical and the other is horizontal. Their effective lengths are \( L_{\text{eff1}} \) and \( L_{\text{eff2}} \), as illustrated in Fig. 2. This consideration is justifiable if one imagines a typical streamline, for example near the middle of the impingement slot. This streamline length is better approximated by the L-shaped path of height 0.5\( H \) and length 0.5\( L \)–0.25\( s \) after a 90 deg turn.

Summing all of the frictional and dynamic losses, the total pressure drop model function is given in terms of Bernoulli’s equation

\[
\Delta P = \left[ K_e + K_{90} + 4 f_{\text{app}} \frac{L_{\text{eff1}}}{D_b} \right] \frac{4H^2}{s^2} + 4 f_{\text{app}} \frac{L_{\text{eff2}}}{D_b} + K_e \frac{1}{2} \rho V_{\text{ch2}}^2
\]  

Impingement flow modeling in a plate fin heat sink is essentially a simultaneously developing hydraulic and thermal boundary layer problem in rectangular ducts. The flow may become fully developed if the heat sink channel is sufficiently long in the flow direction or with small fin spacing, however, this is very unlikely for electronic cooling application heat sinks. The apparent friction factor, \( f_{\text{app}} \), for a rectangular channel may be computed using a form of the model developed by Muzychka and Yovanovich [3] for developing laminar flow

\[
f_{\text{app}} = \left( 3.44 \frac{H}{L} \right)^2 + \left( \frac{f Re D_b}{H} \right)^{1/2}
\]

They validated the model with most of the available developing flow data and found the estimation error was within ±3% for the rectangular channel.

For the inlet and exit pressure losses for a heat sink, Kays and London [20] provide loss coefficients in the form \( \Delta P = K \rho V^2/2 \) as a function of the ratio of free-flow area to frontal area \( \sigma = b/(b+s) \). The graphs for laminar flow in Ref. [20] have been curve fit here for laminar flow

\[
K_e = 0.4 \left( 1 - \sigma^2 \right)^2 + 0.4
\]

\[
K_e = (1 - \sigma)^2 - 0.4 \sigma
\]
The experimental total air pressure drop for impingement flow can be found in terms of Bernoulli’s equation

$$\Delta P = P_{\text{inlet}} - P_{\text{outlet}} + \frac{1}{2}\rho(V_{\text{inlet}}^2 - V_{\text{outlet}}^2)$$

Tests were conducted for four heat sink geometries for impingement flow. Heat sink pressure drop data were taken for different flow rate conditions and different impingement inlet widths. For each heat sink, the experimental measurements were carried out at seven different velocities in the plenum chamber, 0.4 m/s, 0.5 m/s, 0.6 m/s, 0.7 m/s, 0.8 m/s, 0.9 m/s, and 1.0 m/s, and six different impingement inlet widths, 5%L, 10%L, 25%L, 50%L, 75%L, and 100%L, respectively. The details of the heat sinks used for the tests are summarized in Table 1.

The uncertainty analysis for the test data was conducted using the root sum square method described in Moffat [21] and Holman [22]. The uncertainties in pressure drop measurement were 19% (for minimum \(\Delta P=1.28\) Pa) and 0.7% (for maximum \(\Delta P=60.18\) Pa), respectively. Further details on uncertainty analysis and experimental data can be found in Duan [23].

5 Results and Discussion

The model is validated with the experimental data taken on four heat sinks and other experimental data from the published literature. Figures 4–7 show the measured and model predicted air pressure drop of Heat Sinks #1–#4 for different impingement inlet widths. The highest Reynolds number in the experimental data was 1270, which is in the laminar regime. The differences between predictions and test results increase slightly with increasing flow rate.

Figure 8 shows the comparison between the Saini and Webb [17] experimental data and the analytical model predictions of total pressure drop. These test data are consistently lower than the predictions. Overall, the trend is very good. Figure 9 demonstrates the comparison between the Holahan et al. [12] experimental data and the analytical model predictions of total pressure drop. The experimental data and predictions are in excellent agreement.

It was found that all experimental data errors are within ±20% with a rms error of 11.5%. Although the pressure drop prediction algorithm is based on a very simple model, it succeeds in representing the trends of the experimental values fairly well. The agreement is quite satisfying in view of the simplicity of the model. Given the uncertainties of pressure drop measurements, the model is reasonably well validated.

6 Conclusion

This paper investigated pressure drop of impingement air cooled plate fin heat sinks for a variety of impingement inlet

<table>
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<th>Table 1 Geometry of the heat sinks used in the experiments</th>
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widths, fin spacings, and fin heights. The analytic model is developed for the low Reynolds number laminar flow in the interfin channels of impingement flow plate fin heat sinks, since the expected practical operating range of this type of high-performance heat sink would typically produce flows in the range of \( \text{Re} < 1200 \). The accuracy range of the analytical model was established by comparison with experimental measurements of four actual heat sinks and other published experimental data.

The analytical pressure drop model predictions agree with experimentally measured values within ±20% and 11.5% rms over a Reynolds number range \( 300 < \text{Re} < 1200 \). The pressure drop increases with a decrease in impingement inlet width for the same flow rate. The pressure drop model developed may be suitable for impingement air cooled plate fin heat sinks parametric design studies.

**Acknowledgment**

The authors acknowledge the support of the Natural Sciences and Engineering Research Council of Canada (NSERC), and R-Theta Inc., for providing heat sinks for the present study.

**Nomenclature**

- \( b \) = fin spacing, m
- \( D_h \) = hydraulic diameter, m
- \( f, f_{app} \) = friction factor and apparent friction factor, respectively
- \( H \) = fin height, m
- \( K_c, K_e \) = contraction and expansion loss coefficient, respectively
- \( K_{90} \) = loss coefficient due to 90 deg bend in airflow
- \( L \) = length of heat sink base, m

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**Fig. 6** Pressure drop comparison for Heat Sink #3

**Fig. 7** Pressure drop comparison for Heat Sink #4

**Fig. 8** Pressure drop comparison for Saini and Webb [17] test data

**Fig. 9** Pressure drop comparison for Holahan et al. [12] test data
\[ L' = \text{dimensionless flow length} \]
\[ L_{\text{eff}} = \text{effective length, m} \]
\[ N_f = \text{number of fins} \]
\[ P_{\text{inlet}} = \text{static pressure of heat sink impingement inlet, Pa} \]
\[ P_{\text{outlet}} = \text{static pressure of heat sink outlet, Pa} \]
\[ \Delta P = \text{pressure drop, Pa} \]
\[ Re_{D_h} = \text{channel Reynolds number, } = D_h V_{ch} \rho / \nu \]
\[ s = \text{impingement inlet width, m} \]
\[ t = \text{fin thickness, m} \]
\[ t_b = \text{base plate thickness, m} \]
\[ V_{ch} = \text{channel average velocity, m/s} \]
\[ V_{d} = \text{velocity in the plenum chamber, m/s} \]
\[ V_{\text{inlet}} = \text{heat sink impingement inlet velocity, m/s} \]
\[ V_{\text{outlet}} = \text{heat sink outlet velocity, m/s} \]
\[ W = \text{width of heat sink base, m} \]

Greek symbols

\[ \rho = \text{density of air, kg/m}^3 \]
\[ \sigma = \text{fraction of frontal free flow area} \]

Subscripts

\[ 1 = \text{based upon vertical channel} \]
\[ 2 = \text{based upon horizontal channel} \]
\[ \text{ch} = \text{channel} \]

References