

# THERMAL ANALYSIS OF A COOLING SYSTEM USING FORCED CONVECTION

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## ABSTRACT

The objective of this paper is the thermal analysis of a cooling system for a High Power Traveling Wave Tube Amplifier by using forced convection. This cooling system is an integrated part of an Electronic Warfare (EW) suite in The Royal Norwegian Air Force's EW training aircraft.

The analysis uses MSC/PATRAN for modeling and MSC/NASTRAN for the thermal analysis. Later versions of MSC/PATRAN has made it much easier for a design engineer to perform a preliminary thermal design than before. The paper will also show hand calculated values needed to run a thermal analysis with MSC/NASTRAN.

A comparison between analysis results and results from extensive laboratory testing of the actual device are made.

## INTRODUCTION

Due to a EW research program for the Royal Norwegian Air Force's DA20 Jet Falcon aircraft, several critical systems and components are developed by the Norwegian Defence Research Establishment (NDRE). A Power Amplifier is among these systems. The aircraft has two of these units, one in the front and one in the tail section. The main components of these units are the High Power Traveling Wave Tube Amplifier (HPTWTA) and a high voltage power supply.

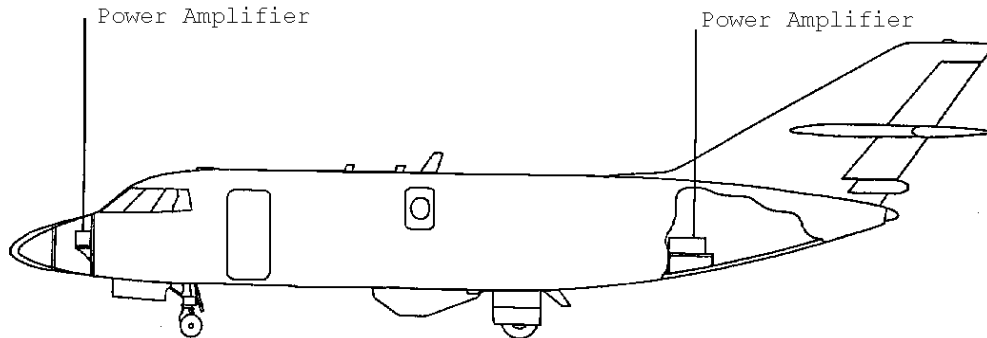


Figure 1. DA20 Jet Falcon, with two Power Amplifiers.

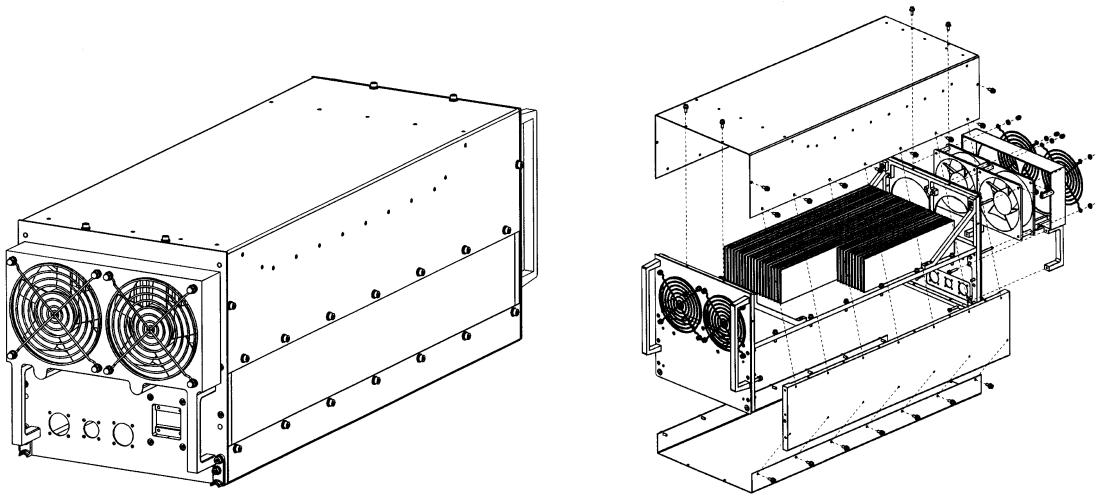


Figure 2. Power Amplifier.

## PROBLEM DEFINITION

A large portion of the power in the HPTWTA dissipates as heat into a relative small area. The HPTWTA is sensitive to high temperatures, it will be permanently damaged if the temperature exceeds a given level. A proper cooling system had to be developed to ensure safe operation. Several options of cooling systems were discussed. A forced air system was chosen

over a liquid system because of lower weight, easier maintenance and no risk for leaks into sensitive electronics and other aircraft installations.

The maximum ambient air temperature ( inside the compartments where the units are stored ) for the Power Amplifier is set to  $55^{\circ}\text{C}$ , i.e. when the aircraft is flying a low altitude mission on a hot day. The normal ambient air temperature should be less than  $45^{\circ}\text{C}$ . Due to the nature of this problem it was decided to use MSC/NASTRAN for preliminary thermal design of the cooling system.

## **THERMAL ANALYSIS**

### **Solid model definition**

The solid model is created by using MSC/PATRAN. It consists of several boxes put together to form the cooling system, which in this case is a horizontal plate with several vertical fins. The amount of boxes gives good user control when adding boundary condition on the surfaces of the geometry in MSC/PATRAN.

Two side sections have also been included in the model. At least one of the sections may have impact on the result, due to the short distance to the HPTWTA. The HPTWTA and side walls have been placed with a small distance to the horizontal plate. This has been done because of the need to simulate the thermal contact resistance. This is vital in order to get realistic results from the thermal analysis.

### **Thermal model definition**

MSC/PATRAN version 7.5 has a sound graphical user interface in order to implement thermal boundary conditions for MSC/NASTRAN. This has made it much easier to create convection boundary conditions than in early versions of MSC/PATRAN. For coupled advection / forced convection MSC/PATRAN uses a closest approach algorithm to connect the surface elements with the flow tube elements.

The Power Amplifier's air inlets have been designed so that 80% of the air stream will flow through the fins and 20% of the air will flow along the side walls, in the area of the HPTWTA and the high voltage power supply, see figure 4.

Following thermal boundary conditions are used:

- Forced convection through the fins, 31 ducts/flow tubes are needed.
- Forced convection inside the side walls, the lower area of the Power Amplifier, 2 flow tubes are needed.
- Free convection on the side walls (external).
- Coupled convection between cooling plate and side walls to simulate thermal contact resistance.
- Coupled convection between cooling plate and HPTWTA to simulate thermal contact resistance.
- No radiation boundary condition has been applied to this model.

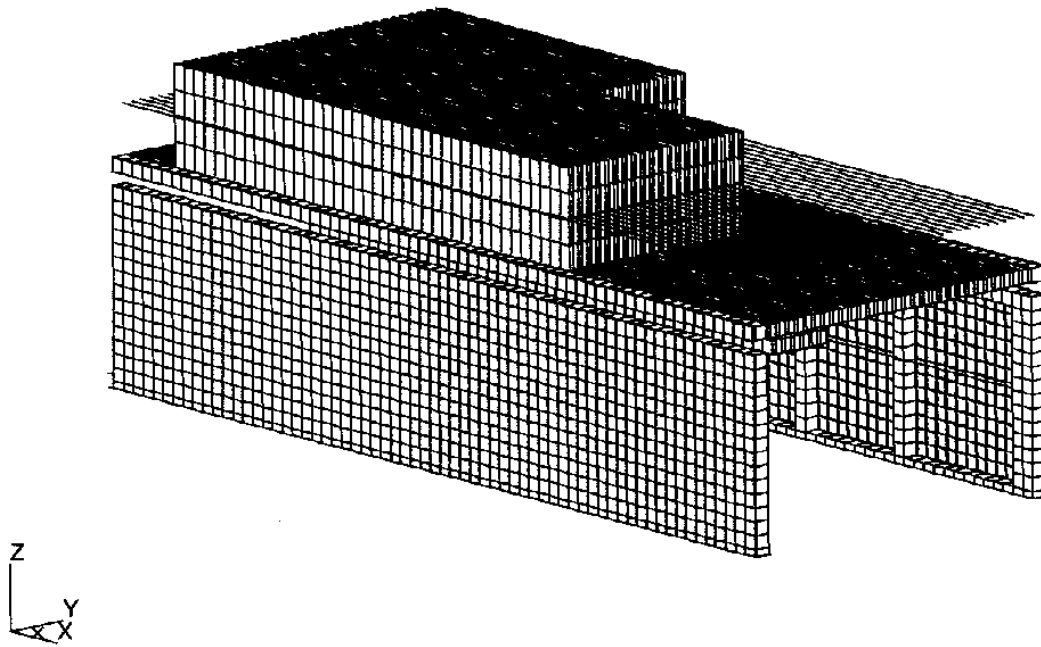


Figure 3. Finite element model of cooling plate/system.

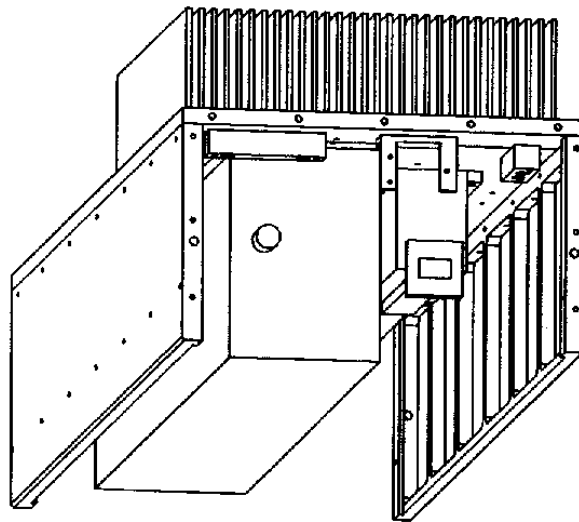


Figure 4. Entrance area for the two flow-tubes along the side walls.

## Material properties for cooling plate

Aluminum is used in the model, because of its low density and high conductivity coefficient  $k$ .

	<i>Air through fins at <math>T_{film} : 68.5 \text{ }^\circ\text{C}</math></i>
Thermal Conductivity, k	190.0

Table 1. Thermal conductivity  $k$  for aluminum

The following table show a simplified input in MSC/PATRAN. It is very easy to create or modify material properties or any of the boundary conditions in MSC/PATRAN version 7.5.

<i>MSC/PATRAN application:</i> ◆ <b>Materials</b> <b>Action:</b> Create <b>Object:</b> Isotropic <b>Method:</b> Manual Input	<i>Material name :</i> Aluminum
Thermal Conductivity, k	190.0

Table 2. Define the material property for aluminum

## Hand calculated values for forced convection

When performing a thermal analysis, air properties for the flow tubes needs to be found. The values for the air properties is temperature dependent. The temperature, often called the film temperature  $T_{film}$ , can be found due to the relationship:

$$T_{film} = \frac{T_{wall} + T_{ambient}}{2}, \text{ Reference 2} \quad (1)$$

Were the wall temperature  $T_{wall}$  is the average temperature on the fins. This temperature has to be assumed, and the ambient temperature  $T_{ambient}$  is the operating temperature.

Mass flow rate  $m$  need to be calculated. It is assumed that the fans are working at 75% of its capacity. Since 80 % of the air is forced through the fins and 20% of the air forced along the side walls, the estimated flow rate will be 0.116 kg/s and 0.030 kg/s respectively. When divided with the number of ducts and side walls it will become 0.0037 kg/s and 0.015 kg/s respectively.

Because of the forced convection boundary condition the convection coefficient  $h$  has to be determined, from the relationship:

$$h = \frac{k}{d_h} Nu, \text{ see Reference 2} \quad (2)$$

were  $Nu$  is the Nusselt number,  $k$  is thermal conductivity coefficient and  $d_h$  is the hydraulic diameter for the fins:

$$d_h = \frac{4s_d h_d}{(s_d + 2h_d)} \quad (3)$$

In the case of the side wall, hydraulic diameter  $d_h$  is equal to the high of the side wall. Furthermore Nusselt number is related to Reynolds number  $Re$  and Prandtl number  $Pr$

$$Nu = 0.023 Re^{0.8} Pr^{0.4}, \text{ see Reference 1} \quad (4)$$

for a tube Reynolds number becomes:

$$Re = \frac{ud_h}{\mathbf{n}} = \frac{ud_h \mathbf{r}}{\mathbf{m}} \quad (5)$$

and Prandtl number:

$$Pr = \frac{C_p \mathbf{m}}{k} \quad (6)$$

Values of  $C_p$ ,  $k$ ,  $\mathbf{m}$ , and  $\mathbf{r}$  at temperature  $T_{film}$  are found in Heat Transfer, see Reference 1. It is necessary to interpolate in order to get the correct values. In this analysis there are two different  $T_{film}$  temperatures, one for the 31 flow tubes through the fins or ducts and one for the 2 flow tubes along the side walls.

The velocity  $u$  is found by dividing the volume flow rate (  $m^3/s$  ) with the air inlet area (  $m^2$  ). The air velocity through the fins and along the side walls are 10.2 m/s and 1.5 m/s respectively.

These velocities are based on the assumed flow rate. In can be very difficult to calculate the flow rate exact. This paper presents input values and results for two thermal simulations, one is based on the assumed velocity and the other one is based one the measured velocity.

In the tables below, values that are different are written on this format:

<assumed value> / <measured value> .

	Air through fins at $T_{film} : 68.5 \text{ }^\circ\text{C}$ assumed / measured	Air along side walls at $T_{film} : 57.5 \text{ }^\circ\text{C}$ assumed / measured
Thermal Conductivity, k	0.0292	0.0284
Specific heat, Cp	1007.6	1007.7
Density, $\rho$	1.028	1.061
Dynamic Viscosity, $\mu$	2.036E -5	1.995E -5
Hydraulic Diameter, $d_h$	0.011	0.14
Velocity, u	10.11 / 8.22	1.34 / 1.09
Reynolds number, Re	4.93E3 / 4.01E3	9.99E3 / 8.13E3
Prandtl number, Pr	0.703	0.708
Nusselt number, Nu	17.98 / 15.24	31.74 / 26.91
Convection Coefficient, $h$	54.33 / 46.05	6.43 / 5.46

Table 3. This table show all the values that are needed for the two  $T_{film}$  temperatures in order to calculate the convection coefficient,  $h$ .

The following four tables show examples of values MSC/PATRAN uses to create material air properties, flow tube, initial starting temperature and advection.

MSC/PATRAN application: ◆ <b>Materials</b> <b>Action:</b> Create <b>Object:</b> Isotropic <b>Method:</b> Manual Input	Material name : Air_through_the_fins  at $T_{film} : 68.5 \text{ }^\circ\text{C}$	Material name : Air_along_iside_walls  at $T_{film} : 57.5 \text{ }^\circ\text{C}$
Thermal Conductivity, k	0.0292	0.0284
Specific heat, Cp	1007.6	1007.7
Density, $\rho$	1.028	1.061
Dynamic Viscosity, $\mu$	2.036E-5	1.995E-5

Table 4. Apply the material properties.

MSC/PATRAN application: ◆ <b>Properties</b> <b>Action:</b> Create <b>Object:</b> 1D <b>Type:</b> Flow Tube	Property set name : Flow_tube_fins  at $T_{film} : 68.5 \text{ }^\circ\text{C}$	Property set name: Flow_tube_iside_walls  at $T_{film} : 57.5 \text{ }^\circ\text{C}$
Diameter at Node 1, $d_h$	0.01	0.14

Table 5. Apply the flow tubes.

MSC/PATRAN application: ◆ <b>Load/Boundary/Cond.</b> <b>Action:</b> Create <b>Object:</b> Temp (Thermal) <b>Type:</b> Nodal	New set name : Inlet_t_Flow_tube_fins  at $T_{film} : 68.5 \text{ }^\circ\text{C}$	New set name: Inlet_t_Flow_tube_iside_walls  at $T_{film} : 57.5 \text{ }^\circ\text{C}$
Boundary Temperature, $T_{ambient}$	55	55

Table 6. Apply the Inlet Temperature.

MSC/PATRAN application: ◆ <b>Load/Boundary/Cond.</b> <b>Action:</b> Create <b>Object:</b> Convection <b>Type:</b> Element Uniform <b>Option:</b> Coupled Advect.	New set name : Conv_Flow_tube_fins  at $T_{film} : 68.5 \text{ }^\circ\text{C}$ assumed / measured	New set name: Conv_Flow_tube_iside_walls  at $T_{film} : 57.5 \text{ }^\circ\text{C}$ assumed / measured
Convection Coefficient, $h$	54.33 / 46.05	6.43 / 5.46
Mass Flow Rate, $m$	0.0037 / 0.0031	0.015 / 0.0122

Table 7. Apply Coupled Advection.

### Hand calculated values for free convection.

In this model only the outside of the side walls will have free convection boundary condition, and the convection coefficient  $h$  is assumed to be several times smaller than forced convection through the fins.

The convection coefficient can be found using Eq. (2) :

$$h = \frac{k}{d_h} Nu$$

were the Nusselt number can be written as:

$$Nu = C(Gr Pr)^m \quad (7)$$

Prandtl number  $Pr$  is found from Eq. (6)

$Gr$  is called the Grashof number and can be found from this relationship:

$$Gr = \frac{g b d_h^3 (T_{wall} - T_{ambient})}{\nu^2} \quad (8)$$



The product of Grashof and Prandtl is called the Rayleigh number  $Ra$  and is vital in order to find the values for  $C$  and  $m$ , see Reference 1. When the Rayleigh number is between  $10E4$  -  $10E9$  for vertical plates  $C$  and  $m$  become 0.59 and 0.25 respectively.

Eq. (2) transformed into MSC/NASTRAN format will become

$$h = \frac{k}{d_h} C \frac{g \beta d_h^3 \text{Pr}^m}{\nu^2} (T - T_{Ambient})^m, \text{ see Reference 1} \quad (9)$$

	Air outside side walls at $T_{film} : 57.5 \text{ }^\circ\text{C}$
Thermal Conductivity, $k$	0.0284
Specific heat, $C_p$	1007.7
Acceleration of grav., $g$	9.81
Dynamic Viscosity, $\mu$	1.995E -5
Hydraulic Diameter, $d_h$	0.14
Volume Coefficient of expansion	0.003
Constant, $C$	0.59
Constant, $m$	0.25
Reynolds number, $Re$	5.27E3
Prandtl number, $Pr$	0.693
Grashof, $Gr$	1.15E6
Rayleigh number, $Ra$	8.0E5
Convection Coefficient, $h$	2.68

Table 8. This table show all the values that are needed for the  $T_{film}$  temperature in order to calculate the convection coefficient,  $h$ .

The following table show MSC/PATRAN input to create free convection on the model.

MSC/PATRAN application: ◆ <b>Load/Boundary/Conditions.</b> <b>Action:</b> Create <b>Object:</b> Convection <b>Type:</b> Element Uniform <b>Option:</b> To Ambient	New set name: Conv _outside_walls  at $T_{film} : 57.5 \text{ }^\circ\text{C}$
Convection Coefficient, $h$	2.68
Ambient temperature, $T_{ambient}$	55.0

Table 9. Apply free convection to the side walls.

**Coupling convection between the cooling plate and side walls and coupling convection between cooling plate and HPTWTA.**

The side walls and HPTWTA are mechanically joined to the cooling plate. A temperature drop is expected between these components. This is due to the contact resistance between components that are only mechanically joined. In this analysis this can be simulated by using Coupled Convection in MSC/PATRAN.

When using thermal compound between the cooling plate and HPTWTA, the contact coefficient becomes 10,000.0 W/m\*\*2 C. When there is a dry connection the value is set to 6,300.0 W/m \*\*2 C. The value for contact coefficient is highly dependent on the surface roughness and pressure. When aluminum components are joined together with steel screws, the pressure will increase with increasing temperature. Values presented in Table 10 are found in Reference 3.

MSC/PATRAN application: ◆ <b>Load/Boundary cond.</b> <b>Action:</b> Create <b>Object:</b> Convection <b>Type:</b> Element Uniform <b>Option:</b> Coupled Convect.	New set name : Coup_Conv_Cool_HPTWTA	New set name: Coup_Conv_Cool_sidewalls
Convection Coefficient, <i>h</i>	10, 000.0	6,300.0

Table 10. Apply contact coefficient to the model.

**Applied heat flux on the HPTWTA**

Approximately 800 watts is dissipated as heat from the HPTWTA, this value is converted into heat flux, *q* (W/m\*\*2). Heat flux is applied on the top surface of the HPTWTA. The area is 0.0058 m\*\*2. The MSC/PATRAN input is shown in Table 11.

MSC/PATRAN application: ◆ <b>Load/Boundary/Conditions.</b> <b>Action:</b> Create <b>Object:</b> Applied Heat <b>Type:</b> Element Uniform <b>Option:</b> Normal Fluxes	New set name: Heat_Flux_HPTWTA
Heat Flux, <i>q</i>	138888.0

Table 11. Apply heat flux on HPTWTA.

## **DISCUSSIONS**

The Finite Element model consist of 9867 CHEXA solid elements, 13309 CHBDYG and 2015 CHDYP elements. The thermal finite model is analyzed with SOL 153, steady state thermal analysis.

### **Comparison between thermal simulations and test results.**

Four Power Amplifiers, from S/N02 to S/N05, are tested in an environmental chamber, before the units are installed in the aircraft. The tests have been run at 55°C.

Fourteen thermal sensors are used to collect temperature data. Two of the thermal sensors have been placed on the flange of the HPTWTA and two on the cooling plate, close to the flange. This have been done in order to measure the maximum temperature and the temperature drop due to thermal resistance between HPTWTA and the cooling plate. The others have been placed i.e. at air inlet, air outlet and outside on the side walls.

The location of each thermal sensor is measured and the accuracy has been within 1- 4 mm, but for unit S/N03, the thermal sensors placed on the flange of HPTWTA is 12 mm offset from the ideal location. The temperature difference due to this can be seen from thermal sensor output at location 1 and 3, see Table 12.

To keep the environmental chamber at a constant temperature, which has a volume of 1.0 m<sup>3</sup>, it is provided with a big fan. This will create a turbulent air flow around the Power Amplifier during testing, this is not accounted for when applying the convection coefficient on the side walls in the thermal model. Particularly for thermal sensor at location 10 , which is placed outside on the side walls, see Table 12. Another contribution to this 10-12 degree difference could come from using incorrect contact coefficient value. A third contribution could be that radiation is not accounted for in the thermal model.

LOCATION OF THERMAL SENSORS	S/N02 °C	S/N03 °C	S/N04 °C	S/N05 °C	Simulation assumed velocity °C	Simulation measured velocity °C
<b>(1) Flange TWTA1</b>	<b>104.3</b>	<b>97.3</b>	<b>105.0</b>	<b>100.8</b>	<b>105.3</b>	<b>108.9</b>
<b>(2) Cooling plate TWTA1</b>	<b>83.0</b>	<b>87.9</b>	<b>85.5</b>	<b>84.4</b>	<b>86.9</b>	<b>90.7</b>
<b>(3) Flange TWTA2</b>	<b>102.4</b>	<b>97.2</b>	<b>104.0</b>	<b>100.7</b>	<b>105.3</b>	<b>109.1</b>
<b>(4) Cooling plate TWTA2</b>	<b>87.0</b>	<b>87.4</b>	<b>88.1</b>	<b>89.4</b>	<b>87.9</b>	<b>91.8</b>
(5) Cooling plate	77.3	76.9	80.9	79.4	81.4	84.8
(6) Cooling plate	70.1	73.3	72.0	74.0	75.8	80.7
(7)	73.8	72.3	76.5	75.4	N/A	N/A
(8) Cooling plate	59.3	59.9	57.2	59.4	64.4	66.7
(9) Cooling plate	62.2	61.0	62.3	62.2	64.2	66.5
(10) On side walls 1	67.9	64.8	67.0	64.3	76.6	78.8
(11) Air inlet	56.1	56.4	54.0	55.1	55.0	55.0
(12) On side walls 2	57.7	57.4	56.3	57.7	59.0	60.4
(13) Cooling plate	62.4	62.3	N/A	63.8	63.8	66.1
<b>(14) Air outlet</b>	<b>65.2</b>	<b>65.3</b>	<b>65.4</b>	<b>67.1</b>	<b>66.0</b>	<b>68.0</b>

Table 12.

Comparison of results, the first column from left contain the location number and name, column two to five contain test results from S/N02 to S/N05 and column six and seven contains the MSC/NASTRAN results.

## CONCLUSIONS

This paper presents the thermal analysis of a cooling plate. The paper can serve as an example how to input hand calculated values needed to run forced convection with MSC/NASTRAN. Using MSC/NASTRAN and MSC/PATRAN have been of great importance in order to reduce the development cost for the cooling system.

The accuracy of the result depends on the assumptions that are made. Particularly the flow rate and the contact coefficient are of great importance in order to achieve good results. The difference in the two thermal simulation results could indicate that an increased contact coefficient value is needed.

Even though this paper show a difference in assumed and measured air velocity, the result from the analysis, with assumed air velocity, is used to build and test a cooling system with great success.

When comparing analysis and test results, it is important to fully understand the test environment in order to understand what can contribute to temperature difference.

This paper shows that there is good correlation between thermal analysis and test results.

## ACKNOWLEDGMENT

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### TRADEMARKS

NASTRAN is a registered trademark of NASA. MSC/NASTRAN is an enhanced, proprietary version developed and maintained by MacNeal Schwendler Corporation.  
MSC/NASTRAN and MSC/PATRAN is a registered trademark of the MacNeal Schwendler Corporation.

### ABBREVIATIONS LIST

EW, Electronic Warfare  
HPTWTA, High Power Traveling Wave Tube Amplifier  
NDRE, Norwegian Defence Research Establishment

### LIST OF SYMBOLS

$C_p$	Specific heat, J / kg °C
$d_h$	Hydraulic diameter, m
$g$	Acceleration of gravity, m / s**2
$k$	Thermal conductivity, W / m °C
$h$	Heat transfer coefficient, convection heat coefficient W / m**2 °C
$h_d$	The height of the fin, m
$h_i$	Contact coefficient, W / m**2 °C
$\dot{m}$	Mass rate of flow, kg / s
$q$	Heat flux, J / s
$Q$	Heat, J
$s_d$	The inner distance between two fins, m
$T$	Temperature, °C
$T_{ambient}$	Ambient temperature, °C
$T_{film}$	Film temperature, °C
$T_{wall}$	Wall temperature, °C

<b>u</b>	Velocity, m / s
$V_f$	Volum rate of flow, m <sup>**3</sup> / s
<b>b</b>	Volum coefficient of expansion, 1 / K
<b>r</b>	Density, kg/m <sup>**3</sup>
<b>m</b>	Dynamic viscosity, kg / m s
<b>u</b>	Kinematic viscosity, m <sup>**2</sup> / s
Gr	Grashof number
Nu	Nusselt number
Pr	Prandtl number
Ra	Raleigh number

## REFERENCE LIST GUIDE

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