BASIC RESEARCH OF THERMAL TRANSFER SIMULATIONS

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Abstract

The article summarizes basic research into thermal transfer simulations. Problems of thermal influence in mechanical systems are solved there. The first steps focus on matching simple heat-transfer samples with CAE software. Simple cases are performed in a real environment. Thermal values are measured. Cases are also solved using CAE software tools. Solutions are compared. CAE solutions are matched to real values. CAE results are verified or refuted. There are many differences between the options in the solvers, there are steady states and transient-run possibilities, etc. Software tools like Nastran, etc. need many coefficients to solve the problem. This procedure is able to identify specific conditions, fits the solver to the specific sample and performs CAE simulations to get real, verified results. For example, passive radiators heated by an induction heater are used for real tests. Temperature fields are measured by thermal camera and structural deformations by measuring displacement. These values are used in simulations and solved by finite elements method. Simulations are performed in Siemens NX10 software, supported by solvers Nastran, MAYA, and NX Multiphysics. Results are compared and matched in the simulation to acquire a more precise solution in the following steps. All these steps are processed to get characteristics of thermal transfer simulation which will be useful in difficult examples of simulation machines, machine tools etc.

Keywords: heat load; thermal flow simulation; FEM

1. Verifying of thermal load

A simple simulation of the heat load was performed in the first part of the research. A small iron cylinder was heated by a high frequency induction heater. The cylinder was heated from 25°C to nearly 250°C as you can see on the Fig. 1. The heat load was turned on for 60 seconds, and then it was cooled by radiation and natural convection. The temperature of the cylinder was logged by a thermocouple. The temperature was monitored in relation to time. Thermal fields were monitored by a thermal camera. The thermal camera was used to verify the optimum homogeneous warming. Four measurements were made. Three measurements of a cylinder with diameter 25 mm and 30 mm high are performed. Effective emissivity was guaranteed by paint with guaranteed emissivity ($\epsilon = 0.95$).

This effect was simulated. Derivation of the measured curve provided the loading characteristic. The same loading characteristic is used in the simulation. That obtains two views, measured and simulated, as you can see in the Fig. 1. The results of the experiments are shown in the graphs Figs. 2.

Heat load was measured in the experiment. Measuring determined heat load 219 W. For equivalent simulation was estimated heat load 220 W. Then was computed convection heat exchange. Measured exchange was 16.3 W and simulated heat exchange less, about 14 W. Cooling losses were calculated by derivation of curves in Fig. 2.

Fig. 1 Temperature fields, heat loading – measured(left) and simulated(right)

Fig. 2 Heat loading graph – measured (left) and simulated (right)

2. Verification of thermal cooling capacity of passive cooler

The experiment with a passive cooler takes a more precise look at the thermal effects. Thermal effects like free convection and radiation are very difficult to simulate precisely. Results are particularly sensitive to the mesh.[6] Results are also sensitive to the other coefficients. Therefore, it is necessary to perform many experiments to obtain the real coefficients and to acquire information about the basic thermal activity. The heat transfer is generally expressed by conduction and convection. The major balance equation which describes this problem is represented below (1), where ρ means density, c_p heat capacity, T temperature, t time, k heat conductivity, Q heat source. [10]

$$\rho \cdot c_p \cdot \frac{\partial T}{\partial t} + \nabla(-k \cdot \nabla T) = Q \quad (1)$$

The heat flux is computed by general convection equation, the convective heat flux is given by following equation (2). Where q means heat flux, h heat transfer coefficient, T_2 outer temperature, T_1 temperature of material [10]:

$$q = h \cdot (T_1 - T_2) \quad (2)$$

The essence of the topic is the harmonization of thermal simulation and real measurement.

A high frequency induction-heating device heats the iron test samples (S235). The main disadvantage of this source is that we do not know the precise thermal input into the sample. It is crucial to find the heating parameters.

2.1. Experiment

An experiment was performed for verification. Experiment was consisted of this setup:

- heat load from high frequency heating device(1)
- cooling by free convection (2)
- thermocouple measuring(3)
- thermal camera monitoring(4)

The experimental setup is presented in the Fig.3.

Fig. 3 Experimental measuring, thermocouple measuring(left), thermal camera (right)

2.2. Heat loading

Heat load has to be determined and the result is used for the heating of the passive cooler in thermal simulation. This choice is based on the integration of the thermal field in time dependent states. Heating cycles can be seen in Table. 1.

Cycle	Heat load [s]	Cooling [s]	Total time [s]
1	120	180	300
2	90	210	300
3	90	510	600

Table 1. Heating cycles

Fig. 4 heat load of passive cooler – measuring by thermal camera and simulation

2.3. Integration area

Figure 4 shows the integration area. Temperatures were measured in this area. Energy states are computed by their integration. The results are the energy states in three periods. It takes the heat load by their differences. Graphs of the temperatures are shown in Fig 5. Values, which are represented by graph, are approximated values, because heat conductivity is neglected. It is neglected because of the fast process during 20 s. These results are the input values for the simulation and they will be refined by iterations. Figure 4 also represents graphic result of temperature fields of the first iteration. In Fig.4 can be compared thermal influence of heat source computed by FEM and measured.

Fig. 5 Measured temperatures in Integration Area, Temperatures dependent on position, three graphs for three times

Average heat load was estimated to be 347 W by difference of energetic states defined by graphs in Fig5. In these graph can be seen energetic states of measured samples. It is represented in two measuring times 20s and 30s. Integration was performed in excel file by discretization of a steel part.

2.4. Simulation of heat loading and verification

Simulations are performed in Siemens NX10 with a palette of solvers. Three options of heat loading states were used there. Simulations and measurements present the states of thermal behaviour during heating and cooling time. States are solved by a pallet of the provided solvers. Solved states are shown in Table 2.

2.5. DATA for comparison

All data for procession are given bellow. Data are represented by four sets:

A) NX Thermal NASTRAN

The simplest way is to use NX Nastran solver. Environment provides easy option of boundary condition.

B) Solver NX Thermal/Flow – Thermal solution

More sophisticated solver is Thermal/Flow solver included in NX. It supports more options to specify real states. However, in this case can be used only thermal solver. Convection can be defined merely by coefficients.

C) Solver NX Thermal/Flow – Coupled Thermal/Flow solution

Last performed option is full thermal/flow solution. This option respects many properties of heat and flow effects. [5] You can see result in the Fig. 10.

D) Measuring

Measuring provides experimental data, which are used to verification of computed results.

Four points on the body were verified. Three points are situated on the aluminum cooler (see Fig. 4) and one is situated on the heated iron sample. The evaluation is presented below. For heating in the first iteration was used heating schema as shows Table 2.

Cycle	Heat load [W]	Heat load timing [s]	Cooling [s]	Total time [s]
1	350	120	180	300
2	350	90	210	300
3	350	90	510	600

Table. 1 Heat load cycle

2.6. Values before solver settings (default set)

Solver	C1- max temp [°C]	C2- max temp [°C]	C3- max temp [°C]
Nastran	109.1	82.06	82.6
Thermal-Flow: Thermal	130.4	105.1	105.2
Thermal-Flow: Coupled	127	98.2	98.05
Measuring	154.17	97.5	104.7

Table. 2 Maximal temperatures

Solver	C1- finish temp [°C]	C2- finish temp [°C]	C3- finish temp [°C]
Nastran	88.03	79.22	66.37
Thermal-Flow: Thermal	113	90.1	66.4
Thermal-Flow: Coupled	106.8	76.6	49.38
Measuring	151.01	95.6	73.9

Table. 3 Final temperatures

Fig.6 Measured(left) and simulated(left default set) temperatures dependent on time

2.7- Comparison

Tables 3 and 4 present measured temperatures. As can be seen in Fig. 6, differences in the default simulation with perfect contact are substantial. The large difference in maximum temperature is obviously caused by a more powerful heat load. A useful parameter is the temperature of point 4 on the iron sample. Points 1, 2 and 3 show the effects of face contact between the iron sample and the aluminium cooler. The heat transfer coefficient is a characteristic quantity of convection [4]. Values of point 4 before solver settings are given in the Tables 5 and 6. Measured points can be seen in the Fig. 4 made by thermal camera. Simulated temperature fields are presented in the Fig. 4 on the right side.

2.8. Values before solver settings (default set) POINT 4

Solver	C1- max temp [°C]	C2- max temp [°C]	C3- max temp [°C]
Nastran	265.9	222	222
Thermal-Flow: Thermal	252.7	219.3	219.7
Thermal-Flow: Coupled	248.8	214.7	176.9
Measuring	-----	307	325.1

Table. 4 Maximal temperatures

Solver	C1- finish temp [°C]	C2- finish temp [°C]	C3- finish temp [°C]
Nastran	117.39	86.4	86.4
Thermal-Flow: Thermal	116.4	92.15	67.5
Thermal-Flow: Coupled	109.5	78.85	50.46
Measuring	-----	131	84.4

Table. 5 Final temperature

3. Solution match

Solvers provide a lot of options for improving the results of the simulations. As can be seen, the differences are more than 20%. Generally, good accuracy of thermal simulations is about 10–15%.

Main factors in thermal simulations are below [6]:

- face contact between objects (thermal resistance)
- effective emissivity – radiation
- roughness of the walls
- mesh quality
- flow effects – turbulent/laminar flow, etc.

3.1- Match of solvers

Values provided by three solution setups were verified by the measuring. Solution setups/solvers are given below:

- NX Thermal NASTRAN;
- NX Thermal/Flow – Thermal solution;
- NX Thermal/Flow coupled – This is a coupled thermal and flow solution.

Computation was processed with these boundary conditions:

- contact resistance 0.75 C/W [3]; this temperature change is known as the thermal contact resistance [1, 9, 8]
- heat load 375 W.

Table 7 shows experiment input values in three cycles. Figure 7 shows graphic comparison of cycle 2. Graph shows measured and simulated point 4. Measuring was performed by thermocouple. Graph in the Fig.7 represents temperatures dependent on time and influence of contact resistance. Table 8 and table 9 show results of the second iteration of simulation.

3.2. Heat load – Iteration 2

Cycle	Heat load [W]	Heat load time [s]	Cooling [s]	Total time [s]
1	375	120	180	300
2	375	90	210	300
3	375	90	510	600

Table. 6 Heat load - iteration 2

3.3. Correlated values of point 1

Solver	C1- max temp [°C]	C2- max temp [°C]	C3- max temp [°C]
Thermal-Flow: Thermal	120.78	101.8	109
Thermal-Flow: Coupled	118	91.75	95.9
Measuring	154.17	97.5	104.7

Table. 7 Corellated values

Solver	C1- finish temp [°C]	C2- finish temp [°C]	C3- finish temp [°C]
Thermal-Flow: Thermal	117.18	97.67	76.7
Thermal-Flow: Coupled	112	81.35	56.2
Measuring	151.01	95.6	73.9

Table. 8 Correlated values

3.4. Graphic comparison of cycle 2, point 4 – simulation, measuring – tmc (thermocouple)

Fig.7 Results for variation of contact resistance [C/W]

3.5. Achieved result

Achieved results can be seen below. You can see influence of a face contact and temperature fields. Figure 9 represents matched solution of the simulation and thermal camera measuring, which verify the simulation. Figure 8 shows achieved solution of thermal simulation in graph. There can be seen differences of temperatures caused by surface contact. Graph shows temperature of point dependent on time. Measured temperature fields are shown in Fig. 9 on the left side. Similar temperature fields are obtained by simulation in the Fig. 9 on the right side. Temperature fields are similar, small differences could be neglected.

Fig. 8 Achieved solution, heat resistance of face contact 0.75C/W

Fig. 9 Measuring(left) and simulation(right) of temperature fields in time 300s

4. Cooling capacity simulation

This part of the article shows how to use the acquired data. Two methods for computing the theoretical cooling capacity were performed. Heat loading process was designed with a heat load of 20 W and 50 W. The cooling process was designed with a free and forced convection cooling. Final computation is processed in two ways. The first computation was processed with Nastran simple thermal simulation. Second computation was processed in Thermal/Flow environment for two conditions - free and force convection. If the fluid is made to move by the action of a pump, a fan, or a blower, we have a case of forced convection. [2, 7]

Results of a flow simulation are represented in Fig. 10 and table 9. Results of thermal/flow environment are more detailed. Computation is able to analyse fluid temperatures, flow directions and temperature of parts.

4.1. Cooling capacity

Solver	Heat load [W]	Heat load timing [s]	Max temperature [°C]
Thermal-flow - Steady state	20	Until steady state	86.33
Thermal-flow - Transient	50	600	143.8
Thermal-Flow Forced convection – Steady state	20	Until steady state	74.81
Thermal-Flow Forced convection – Transient	50	600	143.85

Table. 9 Cooling capacity

Fig. 10 Simulation of convection – air flow temperatures: forced(left), free(right) [1]

5. Conclusion

Vital issues in complex thermal transfer simulations are the unknown thermal transfer coefficients and the unknown precise definition of the environment condition. These facts cause that the results are not very precise. This problem is solved by a couple of measuring and simulation. Difficult transfer cases are supported by simulation and simply measured cases, which can help us to find the boundary conditions and to identify the coefficients. This article shows how to identify conditions and refine the results. Simply case of thermal transfer was simulated and measured. Results are refined by the iterations. Refined simulation provided information about transfer coefficients in the case. These coefficients are used in complex simulation, which is proofed. The goal of this paper is verified coefficients and also verified simulation in part 3. Low deviation of results of different solvers was also proved. It is possible to get results of simulations with low deviation as can be seen in part 4. Next research will show how to simulate thermal transfer in bearings, transmissions or other important nodes in machines.

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7. References

- [1] Bergman, T. L. & Frank P. Incropera. (2011). Fundamentals of heat and mass transfer, John Wiley & sons, ISBN 9780470501979, Hoboken
- [2] Baskharone, E. A.(2012) Thermal science: essentials of thermodynamics, fluid mechanics, and heat transfer, McGraw-Hill, ISBN 978-0-07-177234-1, New York
- [3] Brown, M. E. (1988) Introduction to thermal analysis: techniques and applications, Chapman and Hall, London
- [4] Balmer, R. T.(2011) Thermodynamic tables to accompany modern engineering thermodynamics, Elsevier, ISBN 978-0-12-385038-6, Amsterdam
- [5] Anaratone ,D. (2010) Engineering heat transfer, Springer, ISBN9783642039324, New York
- [6] Goncharov, P. , Artamonov, I. , Khalitov, T.(2014) Engineering Analysis With NX Advanced Simulation. Lulu Publishing Services, ISBN9781483417325
- [7] Stephan, P. (2006) VDI-Wärmeatlas, Berechnungsunterlagen für Druckverlust, Wärme- und Stoffübertragung. Springer-Verlag, ISBN 3540255044, Berlin, Heidelberg
- [8] Alexandru, T.G., Mantea T.A., Pupaza C., Velicu S, Heat transfer simulation for thermal management of electronic components, Proceedings in Manufacturing Systems, Vol. 11, No. 1 (2016), pp. 15–26
- [9] Yovanovich, M. (2005). Four Decades of Research on Thermal Contact, Gap, and Joint Resistance in Microelectronics, IEEE Transactions on components and packaging technologies, vol. 28, NO. 2, June 2005, pp. 182-206, 10.1109/TCAPT.2005.848483
- [10] Gerlich, V., Zálešák, M.(2010) Experimental validation of heat transfer model, Annals of DAAAM and Proceedings of the International DAAAM Symposium, 2010, pp. 969-970. ISSN 1726-9679, Zadar