Front Cover Thermodynamics

Tobias Rybka, Hanno Holzhüter, Georg Schneider and Holger Blume

***Abstract*—Deicing of an automotive sensor for autonomous driving is necessary to provide a clear sight. The heat generated by a sensor itself typically is not sufficient for defrosting ice accumulated on the outer surface at low temperatures and high wind speeds. This results in a need for integrating an active heating element in the system. Finding the right heating layout can be a complex problem that is shaped by the requirement to maintain the optical quality during heating and to achieve a high heating performance in parallel. Every design must be verified to meet this goal and the verification is usually achieved by a sophisticated simulation or experiment. The aim of this study is to ease the verification process by providing a description of the heat flow in a plane front cover design with different heating layouts and finding a (semi-)analytical expression for the surface temperature as a function of the heating power, ambient temperature and wind speed. Only steady conditions and temperatures are considered. The model accuracy is demonstrated for the Ibeo NEXT LiDAR sensor. In addition, a numerical FEM analysis is performed to verify the results in a 3D model.**

***Index Terms*—sensor front cover heating; deicing; all weather sensing; automotive LiDAR**

I. Scope

Deicing of a LiDAR sensor is necessary to provide a clear sight. The heat generated by a sensor itself is typically not sufficient for defrosting ice accumulated on the outer surface (see appendix A.1). That's true in particular at low temperatures and high wind speeds resulting in a need for integrating an active heating element in the system. Finding the right heating layout can be a complex problem that is shaped by the requirement to maintain the optical quality during heating and to achieve a high heating performance in parallel. Which heating performance is required depends on use cases. For deicing the system must be designed in such a way that the available heating power is sufficient to heat up the surface temperature above $0°C$ for a specified range of ambient temperatures and wind speeds. Every design must be verified to meet this goal and the verification is usually achieved by a sophisticated simulation or experiment. The aim of this document is to ease the verification process by providing a description of the heat flow in a plane front cover with different heating layouts and finding a (semi-)analytical expression for the surface temperature. Only steady conditions and temperatures are considered. The model accuracy is demonstrated for the NEXT LiDAR sensor. In addition, a Matlab script based on finite element analysis is provided to solve the heat transfer problem for a 3-D model of the NEXT LiDAR front cover (see appendix A.3).

H. Holzhüter is with Ibeo AS, Merkurring 60-62, 22143 Hamburg, Germany (e-mail: hanno.holzhueter@ibeo-as.com) and the Institute of Microelectronic Systems (IMS) at Leibniz University Hannover, Appelstraße 4, 30167 Hanover, Germany

T. Rybka and G. Schneider are with Ibeo AS, Merkurring 60-62, 22143 Hamburg, Germany

H. Blume is the Head of Architectures and Systems Group at the Institute of Microelectronic Systems (IMS) at Leibniz University Hannover, Appelstraße 4, 30167 Hanover, Germany

In general, the surface temperature depends on the following influences:

Environmental influences

- Ambient temperature T_o
- Relative wind speed v
- Precipitation on the surface

Design and control parameters

- Heating power P or temperature of the heating wire/plate T_{hs}
- Front cover design (thickness, distance between wires, ...)
- Material (composition) of the front cover
- Internal heat transfer from the LiDAR sensor electronics to the surface
- Mounting position (on the car)

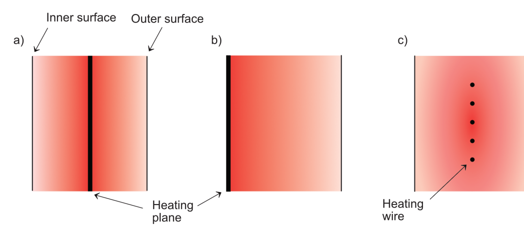

Fig. 1: Side view on three front covers with different heating architectures where a) a heating plate is buried in the mid plane b) a heating plate is attached to the inner surface and c) a row of equally spaced parallel heating wires are buried in the mid plane.

II. Generic Thermodynamic Model

In this section, a generic analytical expression for the surface temperature of a (LiDAR) sensor is derived as a function of relative wind speed, ambient temperature and the heating power applied to the heating element. The expression can be applied to planar front covers in thermodynamic equilibrium.

Many thermodynamic problems encountered in practice are two- or three-dimensional and involve rather complicated geometries for which no simple solutions exist. The front cover of the NEXT sensor shown in Fig. 8 is such an example. One-dimensional geometries have exact solutions. Figures 1a and 1b show the side view of two heating geometries of which simple analytical solutions can be derived. The geometry in Fig. 1c is a two-dimensional problem and requires a more elaborate calculation. Nonetheless, the solution is identical to the solution of the geometry in Fig. 1a when introducing a shape factor. This shape factor must be adjusted accordingly for more complex heating architectures such as the NEXT sensors front cover and can be found only numerically or experimentally.

In the following, first the basic underlying thermodynamic expressions are introduced. Secondly, the surface temperature of the heating plate architecture shown in Fig. 1a is derived. Based on this solution, a general equation for more complex, planar geometries is derived by introducing the shape factor.

A. Thermodynamic Model

Figure 2 depicts a sketch of a wall with a temperature difference between the inner and outer side. The heat transferred per unit time, $\dot{Q}$, through a solid wall of thickness d and area A depends on the temperature difference $T_i - T_o$ between the inner side and the outer side of the wall:

$$\frac{\dot{Q}}{A} = \frac{1}{R_{tot}} \cdot (T_i - T_o) \tag{1}$$

with R_{tot} being the total thermal resistance of the wall. The higher the temperature difference and the lower the thermal resistance, the higher is the heat lost through the area per unit time.

The heat is transferred by conduction through the wall and by convection of air and radiation from the inside to the wall and also from the wall to the outside. Heat transferred by radiation only needs to be considered if the temperatures in the system are very high and only natural convection occurs (no wind). The purpose of this heat transfer analysis is mainly to determine whether a chosen heating design is sufficient to deice the surface or not. For this we need to know the maximum rate of heat loss from the surface, which is determined by considering the heat loss from the surface under worst conditions for an extended period of time, that is, during steady operation under worst conditions. If the heating layout is good enough to deice the outer surface under most demanding conditions, i.e. at high winds and low temperatures, it is good enough for all conditions. At high wind speeds, the radiative heat transfer can be neglected. But even for low wind conditions, the radiative heat transfer rate is lower than the heat transferred by natural convection. Additionally, a layer of ice adds a layer of thermal resistance and lowers the necessary heat transfer rate to melt the ice. In this model, by omitting an additional layer of ice, the solution is more conservative. Nonetheless, the additional layer can be easily added to the model.

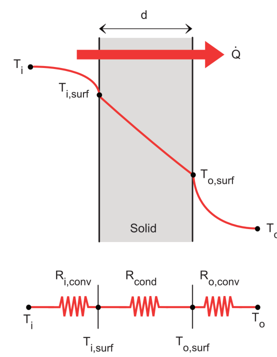

Fig. 2: The upper parts shows the side view sketch of a solid with a qualitative temperature profile (red line) and heat flow (red arrow). The lower part shows an analogous electrical circuit.

The total thermal resistance of the wall is a sum of the thermal resistance for convection at the inner surface, R_i^{conv}, and outer surface, R_o^{conv}, and of the resistance of conduction:

$$R_{tot} = R_i^{conv} + R^{cond} + R_o^{conv} \tag{2}$$

The thermal resistance for conduction, R^{cond}, depends on the thermal conductivity λ, the area A and thickness of the wall d:

$$R^{cond} = \frac{d}{\lambda} \tag{3}$$

The thermal resistance for convection is usually expressed by the heat transfer coefficient h^{conv}, which is reciprocal to the thermal resistance

$$R^{conv} = \frac{1}{h^{conv}} \tag{4}$$

It depends on the wind speed and wind direction and therefore on the overall geometry of the system. Their relationship is described in chapter II-C.

Equation (1) has the form of Ohm's law. The thermal resistance corresponds to electrical resistance, temperature difference to voltage, and the heat transfer rate to electric current. Thus the thermal condition of a system in the steady state can be described by applying the thermal resistance concept in analogous manner to electrical circuit problems. The analogous electric circuit of the heating plate architecture is shown on the lower part of Fig. 2.

Fundamental knowledge on thermodynamics and on how to solve heat transfer problems can be found in the standard literature [1], [2].

B. Heating Plate

Figure 3 depicts a sketch of a heating plate buried in the midplane of a solid. The solid is typically glass or plastic such as polycarbonate. The outside ambient temperature is T_o and the temperature in the inside of the sensor is T_i. Heat transfer takes place from the heating source, i.e. the heated plate in the midplane, to both the outside and to the inside of the front cover as illustrated by the red arrows in Fig. 3.

In this chapter, the outer surface temperature of the heating plate architecture is derived as a function of wind speed, ambient temperature and heating source temperature or heating power by applying the thermal resistance concept introduced in chapter II-A. First, the outer surface temperature is derived as a function of outer ambient temperature T_o, heating source temperature T_{hs} and wind speed ν:

$$T_o^{surf} = T_o^{surf}(T_o, T_{hs}, \nu) \tag{5}$$

The surface temperature can be determined by those three parameters, because in the NEXT system the electronics board controls the temperature of the heating source T_{hs}. The heating source temperature can be derived as a function of the heating power P, outer ambient temperature, inner temperature T_i and wind speed:

$$T_{hs} = T_{hs}(P, T_o, T_i, \nu) = T_{hs}(D, U, R_{20}, T_o, T_i, \nu) \tag{6}$$

If the heating source is a electrical resistor, the heating power can be expressed as a function of the supply voltage V, the applied electrical duty cycle D of the pulse width modulation, and the electrical resistance of the heating wire at a specific temperature, for example, R_{20} at a temperature of $20°C$.

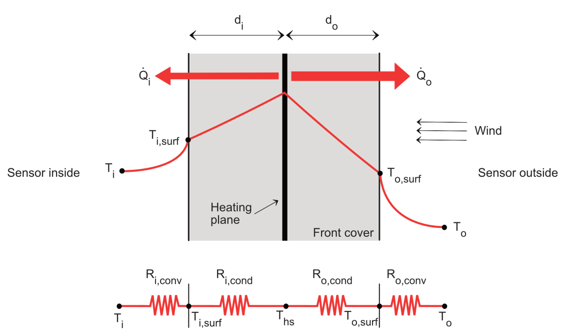

Fig. 3: Side view sketch of a front cover with a heating plate buried in the mid plane

The relationship of equation (5) can be found by applying the thermal resistance concept. The analogous electrical circuit is shown in the lower part of Fig. 3. In steady state conditions, the heat transfer rate from the heated plate to the outside is constant. Therefore, the heat transferred by conduction from the heating plate to the outer surface is exactly equal to the convection heat transfer rate from the outer surface to the ambient:

$$\begin{aligned} \frac{\dot{Q}_o^{cond}}{\cancel{A_h}} &= \frac{1}{R_o^{cond}} \cdot (T_{hs} - T_o^{surf}) \\ &= \frac{\dot{Q}_o^{conv}}{\cancel{A_h}} = \frac{1}{R_o^{conv}} \cdot (T_o^{surf} - T_o) \end{aligned} \tag{7}$$

The conduction heat transfer rate per unit area $\dot{Q}_o^{cond}/A_h$ is proportional to the difference between the heating plate temperature and the outer surface temperature, where $R_o^{cond} = d_o/(\lambda A_h)$ is the thermal resistance of the material expressed in $m^2 \cdot K/\mathrm{W}$. The convection heat transfer rate per unit area $\dot{Q}_o^{conv}/A_h$ is proportional to the temperature difference between the outer surface temperature and the ambient temperature. Comparing both rates leads to an expression for the surface temperature T_o^{surf}. It depends on the ambient temperature T_o, the heating plate temperature T_{hs}, the material parameters and the convective thermal resistance which is determined by the wind condition:

$$\begin{aligned} T_o^{surf}(T_o, T_{hs}, \nu) &= \frac{R_o^{conv}(\nu)}{R_o(\nu)} \cdot T_{hs} \\ &\quad + (1 - \frac{R_o^{conv}(\nu)}{R_o(\nu)}) \cdot T_o \end{aligned} \tag{8}$$

Here, $R_o(\nu) = R_o^{cond} + R_o^{conv}(\nu)$. The relationship between R_o^{conv} and the wind speed ν is described in chapter II-C.

The relationship in equation (6) also can be found by referring to the circuit model in the lower part of Fig. 3. One possible way is to derive the heat transfer from the heating plate to the outer ambient $(T_{hs} - T_0)/R_o$. The heat transfer rate is a sum of two terms. The first term is the heat transfer occurs through two contributions. The first originates from heat transferred from the warmer inside to the outside. The transfer rate is equal to $(T_i - T_o)/R_{tot}$, where $R_{tot} = R_o + R_i$ with $R_i = R_i^{cond} + R_i^{conv}$. The second from the part of the heat that is transferred from the heating plate to the outside. The total heat emitted by the heating source is equal to the electrical power. Only a fraction is transmitted to the outer ambient. When R_{tot} is the total thermal resistance between the inside and the outside, only the fraction of R_i/R_{tot} is transferred to the outside. Therefore, expression for the total heat transfer between the heating plate to the outside amounts to

$$\begin{aligned} \frac{T_{hs} - T_0}{R_o} &= \frac{1}{R_{tot}} \cdot (T_i - T_o) + \frac{R_i}{R_{tot}} \cdot \frac{P}{A} \\ &= \frac{T_o^{surf} - T_o}{R_o^{conv}} \end{aligned} \tag{9}$$

The equation can be solved for T_{hs}:

$$\begin{aligned} T_{hs}(P, T_o, T_i, \nu) &= T_o + \frac{R_o(\nu)}{R_{tot}(\nu)} \cdot (T_i - T_o) \\ &\quad + \frac{R_o(\nu) R_i}{R_{tot}(\nu)} \cdot \frac{P}{A_h} \end{aligned} \tag{10}$$

The heat transfer from the heating plate to the outside is the same as the heat transfer through the outer surface, which is reflected in the right expression in equation (8). This leads to the following expression for the surface temperature

$$\begin{aligned} T_o^{surf}(P, T_o, T_i, \nu) &= T_o + \frac{R_o^{conv}(\nu)}{R_{tot}(\nu)} \cdot (T_i - T_o) \\ &\quad + \frac{R_o^{conv}(\nu) R_i}{R_{tot}(\nu)} \cdot \frac{P}{A_h} \end{aligned} \tag{11}$$

One of the key parameters for quantifying the heating performance is the so-called heat resistance β (not to mixed up with the thermal resistance R). It is a figure stating by how much the temperature increases at a certain position in the system with the heating power brought into the system. Important positions are the heating source and the surface temperature:

$$\beta_{hs} = \frac{\partial T_{hs}}{\partial P} = \frac{R_o R_i}{R_{tot} A_h} \tag{12}$$

$$\beta_o^{surf} = \frac{\partial T_o^{surf}}{\partial P} = \frac{R_o^{conv} R_i}{R_{tot} A_h} \tag{13}$$

Both coefficients decrease with increasing wind speed. Finally, depending on the LiDAR sensor design the inner temperature T_i may increase when heating up the front cover. Then, it must be considered that the inner temperature is a function of the heating power.

The goal of this analysis is to figure out which heating plate temperature (or heating power) is sufficient to heat up the outer surface to +5°C. The required temperature depends on the outer heat transfer coefficient, so on the wind speed, and on the ambient temperature. The temperature value can be plotted in a two-dimensional graph as shown in Fig. 4 for the following material and design configuration parameters.

Heating Plate Architecture	
Parameter	Value
d_i	0 mm
d_o	4.5 mm
Front Cover Thickness	4.5 mm
Front Cover Width	54 mm
Front Cover Length	88 mm
Front Cover Area	47 cm²
Thermal Conductivity	0.7 W/m-K

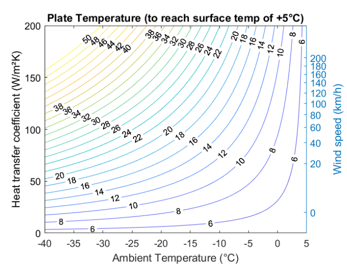

Fig. 4: Plate temperature (°C) required to reach an outer surface temperature of +5°C.

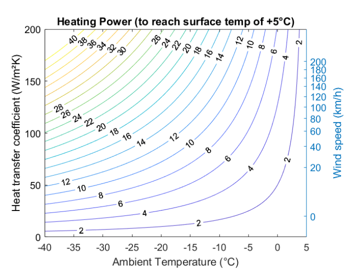

Fig. 5: Heating power (W) required to reach a outer surface temperature of +5°C.

a) Voltage and Duty Cycle: Instead of expressing the heating source temperature as a function of electrical power, sometimes it is more convenient to express the surface temperature as a function of the applied voltage and the applied electrical duty cycle D of the pulse width modulation. Therefore, the power can be replaced by the following expression

$$P = \frac{D \cdot U^2}{R_{el}(T_{hs})} \tag{14}$$

with the supply voltage V, the applied electrical duty cycle D of the pulse width modulation, and the electrical resistance as a function of temperature $R_{el}(T_{hs})$. However, as the electrical resistance of the heating element depends on its temperature, also the electrical power P depends indirectly on the heating source temperature. The resistance depends on the temperature as follows:

$$R_{el} = R_{el,20} \cdot (1 + \alpha \cdot (T_{hs} - T_{20})) \tag{15}$$

Here, α is the temperature coefficient of electrical resistance and $R_{el,20}$ the electrical resistance at $T_{20} = 20°C$. Inserting equation (12) into equation (10) leads to a quadratic equation in the heating source temperature,

$$0 = a \cdot T_{hs}^2 + b \cdot T_{hs} + c \tag{16}$$

with a solution leading to the expression of the temperature as a function of

$$T_{hs}(D, U, R_{el,20}, T_o, T_i, \nu) = \frac{-b + \sqrt{b^2 - 4ac}}{2a} \tag{17}$$

with the coefficients

$$\begin{aligned} a &= R_{el,20} \cdot \alpha \\ b &= R_{el,20} \cdot \left(1 - \alpha \cdot (T_o + T_{20} + \frac{R_o}{R_{tot}}(T_i - T_o))\right) \\ c &= (\alpha \cdot T_{20} - 1) \cdot \left(R_{el,20} T_o + \frac{R_{el,20} R_o}{R_{tot}}(T_i - T_o)\right) \\ &\quad - \frac{R_o R_i}{R_{tot} A_h} \cdot D \cdot U^2 \end{aligned} \tag{18}$$

- Shape Factor
- NEXT Makrolon Picture
- NEXT Glass Picture
- NEXT Glass Formula Plot

C. Relationship between Heat Transfer Coefficient and Wind Speed

The thermal resistance for convection is typically defined as the inverse of the heat transfer coefficient $h_o^{conv} = 1/R_o^{conv}$ expressed in $\mathrm{W}/m^2 \cdot K$. The following table lists typical values of the heat transfer coefficient for different type of convection [1]:

Type of convection	h, $\mathrm{W/m^2{\cdot}K}$
Natural convection of gases	2 - 25
Forced convection of gases	10 - 250

The relationship between the heat transfer coefficient and the wind speed depends on the exact inflow and is therefore not easy to be calculated for the real system. The heat transfer coefficient for a low speed flow of air over a surface is about $h_o^{conv} = 20\ \mathrm{W/m^2{\cdot}K}$. Low speed means speeds of around $1\mathrm{km/h}$. This value gives well matching simulation results when comparing to experimental data recorded with the NEXT LiDAR sensor front cover. For higher wind speeds the following relation is the best fit to a series of data points shown in Fig. 6:

$$h_o^{conv} = h_0 + a \cdot \sqrt{v} \tag{19}$$

with $a = 20\ \mathrm{W} \cdot \mathrm{s}^{1/2}/(\mathrm{K} \cdot \mathrm{m}^{5/2})$ and the wind speed v is given in m/s. Figure 6 shows experimental data points in blue with a fitting curve given by equation (21).

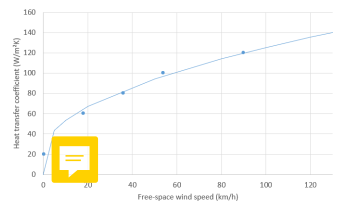

Fig. 6: Estimated heat transfer coefficient for the front cover outer surface of the NEXT 1T LiDAR sensor (in the B1 development phase) as a function of the wind speed. The plot shows the best fit (blue curve) to the series of data points from experiment and simulation (red dots).

D. Heating Wire Architecture

The shape factor is defined to be [3]

$$P = \lambda \cdot S \cdot \Delta T \tag{20}$$

For a wall, the shape factor is then

$$S = \frac{A}{d} \tag{21}$$

For complex geometries it must be found. For heating wire embedded in the front cover the following formula can be used

$$S = \frac{N \cdot 2pi \cdot L}{\log\left(\frac{4w}{\pi \cdot z} \cdot sinh(2\pi \cdot d/w)\right)} = 0.6\mathrm{m} \tag{22}$$

with $N = 14$ the number of wires, L the width of the heating area, z the diameter of the heating wires, w the wire pitch and d the thickness between the wire plane and the surface.

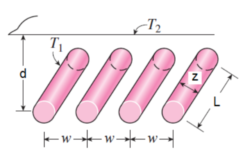

Fig. 7: ddd

III. PDE Simulation

IV. Front Cover Heating of the NEXT LiDAR Sensor

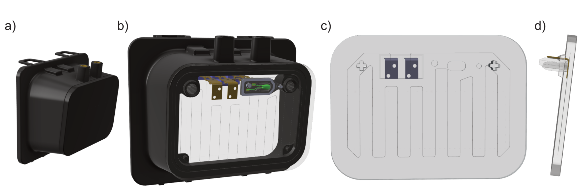

Fig. 8: ddd

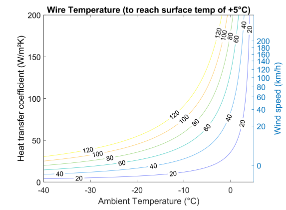

Fig. 9: Side view sketch of a front cover with a heating plate buried in the mid plane

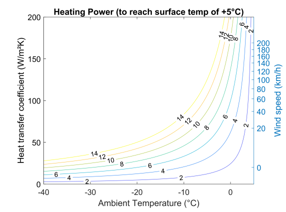

Fig. 10: Side view sketch of a front cover with a heating plate buried in the mid plane

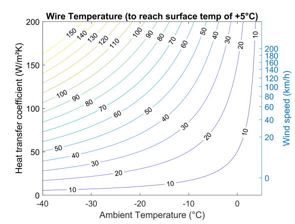

Fig. 11: Side view sketch of a front cover with a heating plate buried in the mid plane

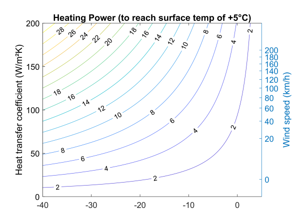

Fig. 12: Side view sketch of a front cover with a heating plate buried in the mid plane

V. Appendix

A. Passive Front Cover Heating

To defrost ice accumulated on the outer surface of the LiDAR sensor without an active heating element, warm air need to be blown over the inner surface of the front cover, in the same way as an automobile windshield is defrosted. The air within a closed sensor is only weakly circulating though, with a wind speed of less than 1 km/h. Consider a front cover with a thickness of $d = 4\mathrm{mm}$ and thermal conductivity of $k = 1\ \mathrm{W/m \cdot K}$. The outside ambient temperature is $T_{amb} = -5°\mathrm{C}$ and the convection heat transfer coefficient is $h_{out} = 25\ \mathrm{W/mK}$, which corresponds to a front cover exposed to a wind speed of about 1 km/h. Also consider a sensor with a front cover area of 45 cm^2 and an average power consumption of 5 W. As shown in the following, the outer front cover surface is not heated up to the melting point. Even when increasing the power consumption and using a thinner front cover, the passive heating is not sufficient to cause the accumulated ice to begin melting if the surface is exposed to slightly higher wind speeds or lower ambient temperatures.

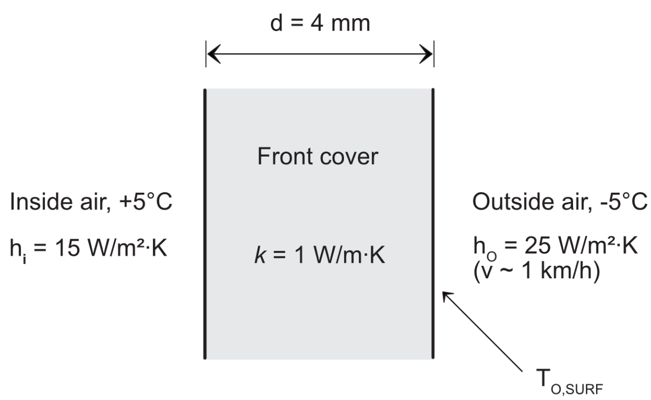

Fig. 13: Side view sketch of a front cover

The outer surface temperature is derived from the heat transfer rate per unit surface area, $\dot{Q}/A$ and the thermal resistance of the front cover. The heat transfer rate per unit surface area corresponds to the power dissipated through the front cover area and can be estimated by weighting the power consumption of the sensor with the fraction of the surface area of the front cover to the surface area of the sensor. A sensor with a size of roughly 11 cm x 10 cm x 8.5 cm has a surface area of 580 cm. Assuming a front cover area of 45 cm^2 and an uniform dissipation loss through the sensor surface, then the sensor looses about 8 $\%$ of its power through the front cover:

$$P = 8\% \cdot 5\mathrm{W} = 0.4\mathrm{W} \tag{23}$$

This corresponds to a heat transfer rate per unit surface of 90 $\mathrm{W/m}$. The temperature on the outer surface is then given by considering the heat transfer from the outer surface to the ambient air

$$T_{o,surf} = T_o + R_{o,conv} \cdot \frac{\dot{Q}}{A} = +5°\mathrm{C} - 3.4°\mathrm{C} = -1.4°\mathrm{C} \tag{24}$$

with the convection resistance $R_{o,conv} = 1/h_{o,conv} = 0.04\ \mathrm{m \cdot K/W}$. The temperature of the air inside the sensor can be derived in a similar manner,

$$T_i = T_o + R_{tot} \cdot \frac{\dot{Q}}{A} = -5°\mathrm{C} + 10°\mathrm{C} = +5°\mathrm{C} \tag{25}$$

by considering the heat transfer through the front cover with a total resistance $R_{tot} = 1/h_{i,conv} + d/k + 1/h_{o,conv} = 0.11\mathrm{m \cdot K/W}$.

B. Heating Plate

C. 3-D Thermal Simulation

References

[1] Y. Cengel, *Heat Transfer: A Practical Approach*. SEM, 2002.

[2] Y. Cengel and A. Ghajar, *Heat and Mass Transfer: Fundamentals and Application*. McGraw-Hill Education, 2014.

[3] E. Hahne and U. Grigull, "Formfaktor und formwiderstand der stationären mehrdimensionalen wärmeleitung," *International Journal of Heat and Mass Transfer*, vol. 18, no. 6, pp. 751–767, 1975. [Online]. Available: https://www.sciencedirect.com/science/article/pii/0017931075902057