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The final publication is available at:

<https://doi.org/10.1007/s12273-015-0255-2>

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Three dimensional transient heat transfer and airflow in an indoor ice rink with radiant heat sources

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Three dimensional mixed convection in an ice rink heated by thermostatically controlled radiant heaters was simulated numerically using the standard k- ϵ model with wall functions. This large building was modelled under transient conditions by considering the real outdoors atmospheric conditions for a typical spring day in Montréal, Canada. Results indicate the usefulness of the CFD technique as a powerful tool which provides a detailed description of the flow and temperature fields as well as the heat fluxes into the ice. The most important results are:

- Heating is needed only during the night (from 22h to 7h) when the outdoors temperature is relatively low.
- The On/Off switching of the radiant panels influences the temperature profiles throughout the ice rink; even the ice surface temperature is affected although the view factor between these two surfaces is zero.
- The radiation heat flux towards the ice increases significantly when the radiant panels are turned On; this can influence ice quality.
- The resurfacing operation increases the ice temperature and, by convection, the temperature of the air immediately above the stands.
- The air near the ice surface (1 meter over the ice) is essentially stagnant and significant air velocities can only be found above the stands and near the ceiling above the ice.
- The volumetric flow rate and temperature of the air evacuated from the building vary significantly between the different outlets.

Keywords: CFD; radiation; convection; transient simulation; building

1. Introduction

Indoor ice rinks are large buildings without internal partitions with high refrigeration loads and high energy consumption for heating the spectator zone and the ventilation air. The flow structure within those large buildings is complex since they include zones where forced convection predominates and others where natural convection is dominant. Heat is transmitted to the ice sheet principally by convection from the air and by radiation from the other internal surfaces. Computational fluid dynamic (CFD) can provide an accurate prediction of air flow patterns and heat fluxes towards the ice and through the envelope in order to evaluate refrigeration and heating loads as well as the air quality for such buildings.

Jones & Whittle (1992) described the status and capabilities of CFD for building air flow prediction while Jian & Chen (1995) as well as Yang et al. (2000) used a CFD code to evaluate air quality in large ventilated enclosures. Yang et al. (2001) studied experimentally and numerically indoor air quality in an ice rink; they used the standard k- ϵ model (Launder & Spalding 1974) and indicated that the numerical results agree reasonably well with the corresponding experimental data for both steady-state and transient conditions. More recently Bellache et al. (2005) have carried out numerical simulations in 2D and steady state conditions using a CFD code which predicts velocity, temperature and absolute humidity distributions in an indoor ice rink with heating provided by the ventilation air. Their CFD code also calculates the heat fluxes toward the ice due to convection from the air, condensation of vapour and radiation from the walls and ceiling. This 2D CFD model was later improved by Bellache et al. (2006) by including transient phenomena, heat transfer through the ground and energy gains from lights as well as the effects of resurfacing and dissipation of pump work in the coolant pipes. Despite a fairly good qualitative agreement between some calculated and measured air temperatures and heat fluxes these 2D models cannot reproduce accurately the flow of the air within the ice rink which is necessarily three-dimensional since air inlets and outlets are not continuous along the length of the building. Furthermore, 2D models cannot take into consideration radiation exchanges between all the inside surfaces since they only treat a cross-section of the building.

Recently, the present authors (Omri & Galanis 2010) described and used a CFD model for three dimensional turbulent mixed convection in an ice rink heated by eight radiant heaters. They used the standard k- ϵ model (Launder & Spalding 1974) because of its simplicity and numerical robustness (Chen Q. 1995) and wall functions in order

to minimize the number of nodes (Omri & Galanis 2006 & 2009). Radiation was calculated using a discrete ordinate method (Michael 2003, Raithby 1999). The flow and temperature fields in this large building were first calculated under steady state conditions by considering or not the presence of radiant heaters and also by considering or not radiation between the internal surfaces. After those relatively basic cases, its transient behaviour (for 400 seconds) was evaluated for constant outdoors conditions by considering that the radiant heaters are thermostatically controlled. All the results were obtained by assuming that the flow rate and temperature of the ventilation air are constant. They indicate that in a large part of the ice rink the air is essentially stagnant and significant air velocities can only be found above the spectator stands and near the ceiling above the ice; turbulent kinetic energy is also localised in the same regions. It was also found that the radiant heat flux into the ice is significantly higher than the convective one. When the radiant heaters are turned on the temperatures of the inside surfaces of the building envelope and of the air increase; this creates significant fluctuations of the radiant flux reaching the ice and can damage its quality.

In the present work, the three-dimensional flow and temperature fields in the same ice rink are calculated numerically and analysed over a 24-hour period for time dependent outdoors conditions with variable flowrate and temperature of the ventilation air. The model used to describe the flow and temperature fields is the same as the one used in our previous paper (Omri & Galanis 2010). The main objectives of the present paper are to predict the airflow and temperature field during a typical day, to demonstrate the importance of surface-to-surface radiation heat exchanges, to calculate the heat fluxes towards the ice due to radiation and to determine the effects of the radiant heaters on the temperature of the ice surface.

2. Description and modelling of the problem

A schematic representation of the ice rink under consideration (which corresponds to an actual municipal one situated in Montréal, Canada) is shown in Figure 1. The overall dimensions of the building are $64 \times 9.36 \times 36.7$ (m³) and the spectator zone represents almost a tenth of its volume. Thirty identical air inlets (0.3×1.5 m²) are disposed in two lines above the stands; the first line contains sixteen equidistant inlets and the second one contains fourteen inlets and eight identical radiant heaters (0.3×1 m²). These radiant heaters are inclined towards the stands in order to avoid heating of the ice surface. The seven air outlets are situated behind the stands. The xOz plane coincides with the ice surface and the origin of the coordinate system is at its center.

An important point that must be addressed when applying a CFD technique in large buildings is the choice of the grid since a large number of nodes are necessary. In order to reduce their number we consider only half of the ice rink (32 m instead of 64 m) taking advantage of the inherent symmetry of the configuration about the plane $x = 0$.

The specific heat and the molecular transport properties of the air are assumed to be constant. However, its effective thermal conductivity and effective viscosity depend on the local flow conditions. The density of the air is assumed to be constant except in the gravity force where the assumption of a linear dependence on the temperature is introduced.

The CFD package used for this work was Fluent 6.3.26. This code solves all considered equations under the form of a balance between convection, diffusion and source terms:

$$\frac{\partial(\rho\phi)}{\partial t} + \frac{\partial}{\partial x_j}(\rho\phi_i) = \frac{\partial}{\partial x_j}(\Gamma_\phi \frac{\partial\phi}{\partial x_j}) + S_\phi \quad (1)$$

where ϕ , Γ_ϕ and S_ϕ are respectively the considered variable (velocity, temperature,...), the corresponding diffusion coefficient and the source terms.

For radiation calculations, the air is considered radiatively not participating and the discrete ordinates (DO) radiation model is used (all surfaces are grey and diffuse). The choice of this radiation model, rather than the simpler surface-to-surface model, is due to the fact that the latter does not accept the symmetry condition used in the calculations for the reason explained earlier. To avoid numerical difficulties due to the relatively small size of the radiant heaters their heat flux was redistributed over the inclined surface shown in yellow in figure 1. The order of quadrature used in radiation calculations is 8x2x2 with increasing number of pixels (theta pixels and phi pixels) from 1x1 to 3x3.

The computational domain includes the solid walls where the velocity is zero and energy conservation is expressed by the conduction equation. The thermophysical properties of the solids constituting the envelope (walls, ceiling and the concrete slab under the ice) are considered to be uniform and constant. The emissivity of the ceiling and ice surface is 0.9 and 0.97 respectively. The following boundary conditions are applied:

- The velocity components are zero on all inside wall surfaces of the ice rink.
- The outside surfaces of the south ($x=32$ m) and west ($z=14.95$ m) walls as well as that of the ceiling exchange heat by convection and radiation with the ambient air; the exterior temperature profile and the corresponding solar temperature are indicated in figure 2; these are the conditions for a typical spring day in Montréal, Canada.

- The temperature under the concrete slab below the ice is 264 K while that of the outside surface of the walls behind the stands is 291 K.

- At the air inlets the velocity and temperature correspond to the real conditions in the considered ice rink, i.e. there is a permanent ventilation flow of 4.274 m³/s which increases to 10.384 m³/s during the resurfacing operations. As seen in figure 3 resurfacing takes place eight times during the day and each one lasts 12 minutes.

The ventilation air is 100% fresh air. It is treated by an HVAC system which heats it up to 288K during cold periods and cools it down to 296K during warm periods; when the outdoors air temperature is between these two values, the temperature at the air inlets is identical to the outdoors one. Figure 2 shows the temperature of the ventilation air entering the ice rink for the typical day under consideration.

- The load due to each resurfacing operation was introduced as a constant heat flux at the ice surface equal to 314 W/m²; this value was calculated using the following equation from the ASHRAE Refrigeration-Handbook (2002):

$$Q_{\text{resurfacing}} = \frac{m_{\text{water}}}{t} (4.18(T_{\text{water}} - 273.15) + 334 + 2.04(273.15 - T_{\text{ice}})) \quad (2)$$

Where, $m_{\text{water}} = 500 \text{ kg}$, $T_{\text{water}} = 313.15 \text{ K}$, $T_{\text{ice}} = 267.15 \text{ K}$ and $t = 12 \text{ min}$

- Turbulent kinetic energy and dissipation rate at the air inlets are deduced from the following equations:

$$k = 1.5(V_{\text{in}} I)^2 \quad \varepsilon = 0.09^{0.75} k^{1.5} / (0.07 D_h) \quad (3)$$

Where, $I = 0.16 \text{Re}_{D_h}^{-1/8}$ is the turbulence intensity, Re_{D_h} is the Reynolds number and D_h is the hydraulic diameter of the inlets.

- The form of the stands is simplified by using an inclined wall (see Figure 1); the temperature of its back surface is fixed at 291 K.

- When the radiant heaters are used, their total power is 8x22 kW. Simulations have been carried out with these heaters thermostatically controlled by the temperature of a control zone which includes the inclined surface of the stands and the air within 1m from this surface: the heaters are turned On if this control temperature is ≤ 286 K and turned Off if it is ≥ 288 K.

3. Numerical solution

The coupled elliptic partial differential equations describing the flow field are discretized with the finite volume method. Second order central discretization is used, except for the convection terms where the third order MUSCL (Monotone Upstream-Centered Schemes for Conservation Laws) is applied. The pressure and velocity are coupled using the SIMPLEC algorithm.

The ability of the CFD code to model complex flow problems has been established in our previous studies by comparison with published data. Thus, natural convection in a cavity with differentially heated vertical walls and conducting horizontal walls was studied first (Omri & Galanis, 2007). The results indicate that such a flow field can be satisfactorily modelled by considering a grid of only 30x30 nodes. Secondly, an isothermal flow in a ventilated cavity was analysed (Omri & Galanis, 2006 and 2009). The most important result of that study is that the ventilation jet can be modelled by considering a few uniformly distributed nodes in the transverse direction (in the ice rink this is the x direction) while the mesh in the perpendicular plane must be considerably finer and judiciously distributed (in the present application this is the yOz plane). Also natural convection in non partitioned and partitioned cavities with and without radiation was studied (Omri, Galanis & Abid

2008c) to establish the degree of success of turbulence models in the simple case and the difficulty in predicting quantitatively the second one.

Preliminary studies of the flow field under consideration have been published by the present authors (Omri & Galanis 2008a, 2008b and 2010) who initially treated it as a time independent and isothermal problem. Grids with different steps in the x direction and various refinements near the inlets/outlets were considered. It was concluded that an x-step equal to 0.25m is sufficient with more refinement near the walls. The real case including convection, conduction in the solids and radiation was calculated using an x-step equal to 0.25m with more refinement near the walls, the inlets and the outlets resulting in a total number of nodes equal to 2.05×10^6 (see Figure 4). These results indicate that the structure of the flow is totally different from the isothermal case since warm air (in the zone of spectators) comes into contact with the back of the radiant heaters, becomes even warmer and moves parallel to the ceiling in the z direction. The velocity of the warm air near the ceiling reaches a maximum of 0.3m/s exceeding that of the isothermal case. Close to the ice surface the velocities are negligible since this zone is protected by vertical partitions of height 2.4m (see Figure 1). This indicates that convective heat transfer between the ice and air is not very important compared to radiation heat transfer. Also, the fact that the air is essentially stagnant up to approximately 1.5m above the ice surface indicates that after each resurfacing operation fresh air should be injected to evacuate pollutants. Similarly, important turbulent kinetic energy is localised above the stands, near the ceiling and close to the west vertical wall ($z=14.95$ m). In the entire zone over the ice, up to a height of 6m, the turbulent kinetic energy is not important.

Even though classic wall functions are associated to the adopted two equation model, the first node is situated within a very small distance from the solid walls. The

position used here for the beginning of the log layer is $y^* \approx 20$, since the wall functions implemented in Fluent are for a flat plate boundary layer and do not reproduce the behaviour of a natural convection boundary layer (that is why Holling & Herwig (2005) proposed new wall functions for natural convection).

In order to eliminate the effect of the more-or-less arbitrary initial conditions, the calculations were performed for a period of 48 hours using the temperature profiles of figure 2 for both 24-hour periods. The timestep used for the calculation was 20 sec and the presented results are those of the second 24-hour period.

4. Results

Figure 5 shows that the temperature of radiant heaters during the day under consideration ranges from 288 K (thermostat Off) to 547 K (thermostat On). During this typical day the radiant heaters were ON 22 times for a total of 21435 sec (about six hours i.e. a quarter of the day). From 9h (118800 sec) to approximately 21h (162000 sec) the heaters are continuously off due to the fact that the outdoor temperature is warm (greater than 288K) and there is no need to heat the ventilation air. The longest period where the heaters are On occurs when the outdoor temperature is minimum (around 5h or 104400 sec). However, ice rinks are not in use 24 hours a day; if we consider that the ice rink is in use from 7h to 22h the heaters will be switched On for a total period of only 1 hour and 11 minutes during this typical spring day.

Figure 6 shows the temperature profile of the control zone during the 24 hours under consideration. As already mentioned the control zone is the volume constituted by the inclined surface (stands) and the air within 1 meter above this surface. It can be seen that during the first and last parts of the day, i.e. when the heaters are On, the

temperature of the control zone oscillates between the high and low limits imposed by the thermostat. From approximately 9h (118800 sec) to 21h (162000 sec) the evolution of the control zone temperature is similar to that of the ambient temperature (see Figure 2) since during this period the ventilation air is neither heated nor cooled. The short duration temperature spikes of approximately 1 K occurring during this period are due to the increase of the flowrate of the ventilation warm air during the resurfacing operations (see Figure 3). Obviously, this calculated temperature would have been higher if the presence of spectators was accounted for.

Figure 7 shows the ceiling temperature as a function of the time. One curve is the average temperature of the entire ceiling surface while the second is the average temperature of the central line of the ceiling ($x=0$, $y=9.36\text{m}$). Compared to the area weighted average temperature, the temperature of the central line is slightly higher since it is not close to the cooler end walls at $x = \pm 32\text{m}$. These temperature profiles are qualitatively similar to that of the control zone temperature (see Figure 6). This can be explained by the importance of radiation exchanges between the ceiling and other internal surfaces of the building envelope. However the ceiling temperature is always slightly lower than that of the control zone which is heated by the radiant heaters; the amplitude of the overnight fluctuations is larger while the spikes due to the resurfacing operations are not as important as those in the control zone (see Figure 6).

The most important variable in the ice rink is the ice surface temperature which determines the quality of the ice. Figure 8-a shows three temperature profiles for the day under consideration: the first (green) is the area weighted average temperature of the entire ice surface, the second (red) is an area weighted temperature of a $2 \times 2 \text{ m}^2$ surface situated in the corner of the ice far from the radiant heaters ($x=28$

m and $z=12$ m) and the third is the volume weighted temperature of the ice. The three profiles are not very different which indicates a fairly homogeneous distribution of the temperature within the ice. This fact can be explained by the small thickness of the ice (2.5cm). During the early hours (before 9h or 118800 sec) the influence of the radiant heaters is manifested by small temperature fluctuations (amplitude less than 0.25 K); they are due to radiation exchanges from the heaters to the stands, from the stands to the ceiling and from the ceiling to the ice. During each resurfacing operation the ice surface temperature increases rapidly but does not exceed 273K; thus melting of the ice does not take place. It is interesting to note that after each of the first three resurfacing operations, which take place at 2h intervals, the ice temperature decreases and reaches a fairly low value consistent with the earlier tendency and the outdoor temperature. Later, when the resurfacing is done every hour, there is not sufficient time to sufficiently lower the ice temperature. Figure 8-b is a zoom of the same figure encompassing the first two resurfacing operations which shows more clearly the differences between the three temperature profiles. The surface temperature in the right side corner is always slightly colder than the area weighted average temperature; this is logical since the right side corner is far from the radiant heaters. During the pick due to resurfacing the volume weighted temperature is about 1K colder than the other two temperatures while just before the resurfacing the volume weighted temperature is the highest. This is due to the thermal inertia of the ice.

Figure 9 shows the principal components of the air velocity one meter above the ice at $x=0$, $z=0$. As seen in the figure the x component is negligible and the dominant one is the velocity in the z direction. However, even this component is not significant since it doesn't exceed 0.1 m/s and also changes direction; this was experimentally verified by Ouzzane et al. (2006) who indicated that the velocity over

the ice is negligible. Therefore the principal mode of heat transfer between the ice and its surroundings is radiation as pointed out in our earlier article (Omri & Galanis 2010).

Figure 10 shows the area weighted average net radiation heat flux towards the entire ice surface and also on one line ($x=0$, $y=0$, z) during the day under consideration. The magnitude of these fluxes is similar to those published in different sources. Their evolution with time follows qualitatively the corresponding variations of the ceiling temperature (see Figure 7). During the first five hours (up to 104400 sec) these heat fluxes vary between 55 W/m^2 and 71 W/m^2 reflecting the influence of the intermittent operation of the radiant heaters. They then increase, reach a maximum and finally decrease due to the corresponding variation of the outdoors conditions (see Figure 2) and their effect on the ceiling temperature (see Figure 7). The effect of resurfacing operations becomes less pronounced in late afternoon and early evening as the outdoors temperature and that of the ventilation air decrease (see Figure 2). In late evening, the heat fluxes vary between 44 W/m^2 and 69 W/m^2 due to the combined effects of the radiant heaters and the resurfacing operations. Despite these large fluctuations of the net radiation heat fluxes, the ice temperature remains below the freezing point (see Figure 8a) provided the refrigeration system can maintain the temperature under the concrete slab below the ice at the assumed value (264 K).

Figure 11 shows the temperature profiles of the air at the four outlets (see figure 1). The letters “t” and “b” indicate outlets near the top and the bottom respectively; the number “1” indicates the outlets closest to the symmetry plane ($x=0$) and the number “2” indicates the outlets furthest away. Naturally all these profiles reflect the temperature of the incoming ventilation air and the effects of heating and resurfacing. During the early part of the day, each of these temperatures fluctuates

around an essentially constant average value; the amplitude of the fluctuations is highest for the outlets near the symmetry plane and lowest for the outlets far from this plane. During the middle of the day the average value of these four temperatures increase and then decrease following the profile of the incoming air (see Figure 2); during this period, the resurfacing operations cause fluctuations whose amplitude is highest for outlet “b1” and quite small for the three other outlets. Clearly, the warmest air leaves the ice rink through the two highest outlets. The maximum temperature corresponds to the outlet situated in the middle top of the stands (outlet t1). This situation leads to considerable energy waste but at the same time it is necessary to preserve air quality in the ice rink. Measures to recover the energy in the outgoing air should be considered. Due to stratification the air leaving through the bottom outlets is cooler.

Figure 12 shows the volumetric flow rates through the four outlets during the day. It is clear that most of the air leaves by the two outlets situated in the top of the stands where the air is warmer. Between midnight and about 9h (118800sec) the four profiles are relatively constant. Between 9h (118800 sec) and 20h (158400sec) when the exterior temperature is warm (see figure 2), the flow rates in the two top outlets (outlet-t1 and outlet-t2) behave differently from those in the two bottom outlets (outlet-b1 and outlet-b2): the former increase while the latter decrease. This can be explained by the difference in temperature between the two sets of outlets (see figure 11); the air in the top row is warmer and therefore its velocity increases. During the resurfacing operations, all flow rates increase but the relative order of magnitude is maintained.

5. Conclusion

Transient turbulent mixed convection in an ice rink was analysed numerically with the standard $k-\epsilon$ model associated to wall functions. The calculations were performed with real exterior atmospheric conditions for a typical spring day. If the ice rink is used continuously during this 24 hour period heating must be provided during approximately 25% of the time; but if it is not in use during the night (from 22h to 7h) only 1 hour of heating is necessary. The presence of thermostatically controlled radiant heaters influences all parameters in the building, especially the temperature of the stands, the ceiling and most importantly that of the ice. Furthermore, this heating method influences the radiant heat fluxes into the ice which fluctuate with large amplitude when the radiant heaters are switched On and Off. The air at 1m above the ice is essentially stagnant indicating that radiation is the principal heat transfer mode. The results show significant differences between the volumetric flow rates and temperatures of the air evacuated through the outlets of the building. The overall results illustrate the usefulness of the CFD technique as a powerful tool which provides a detailed description of the air flow, the temperature field.

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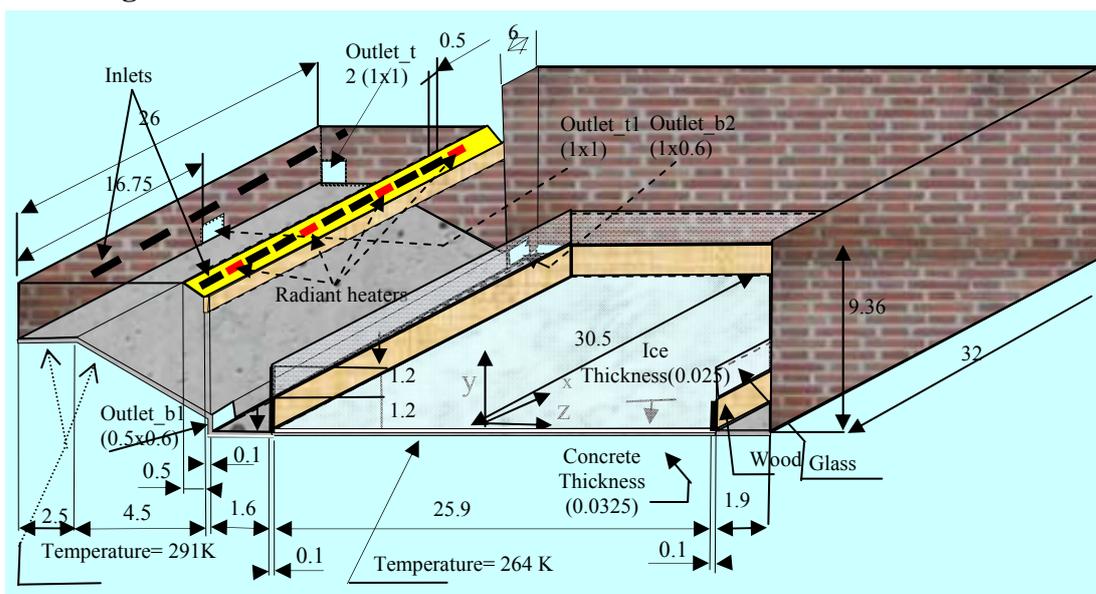


Figure 1. Schematic representation of the ice rink (units in meters).

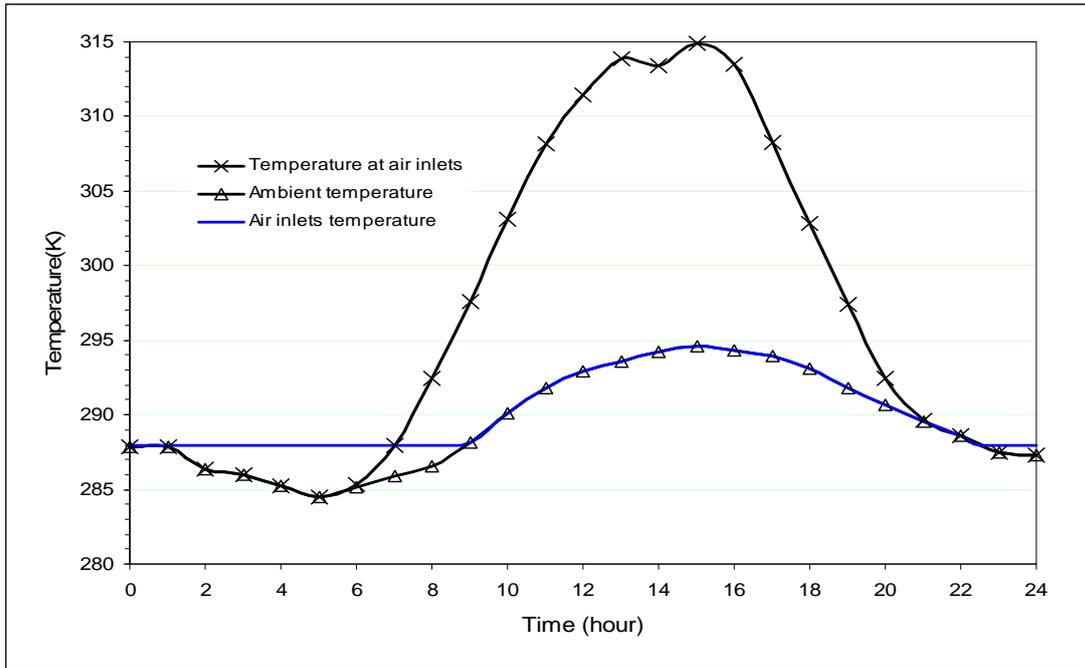


Figure 2. Outdoors and ventilation air temperature profiles

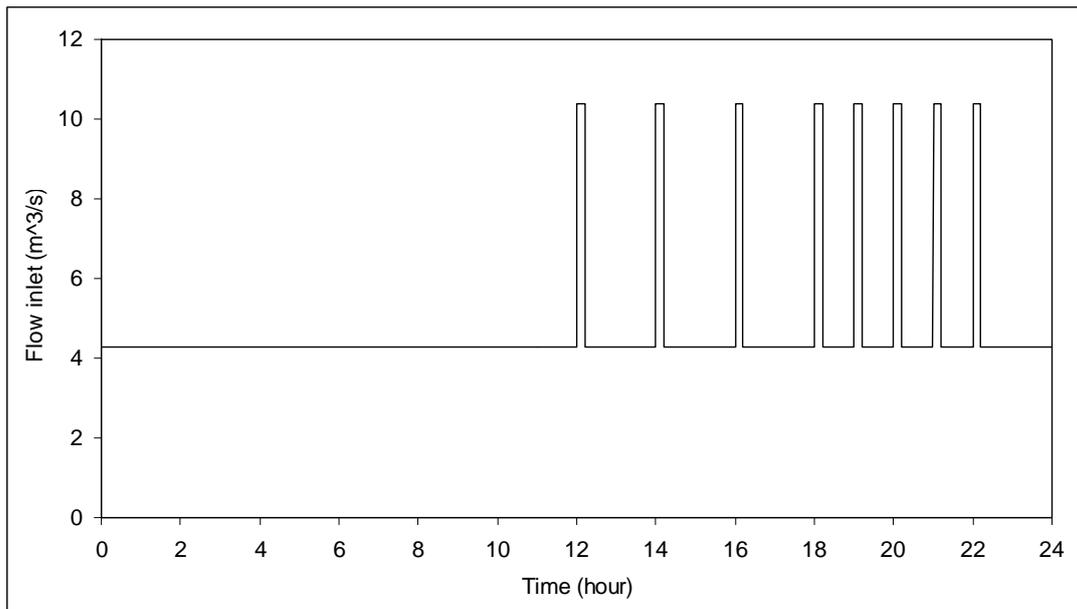


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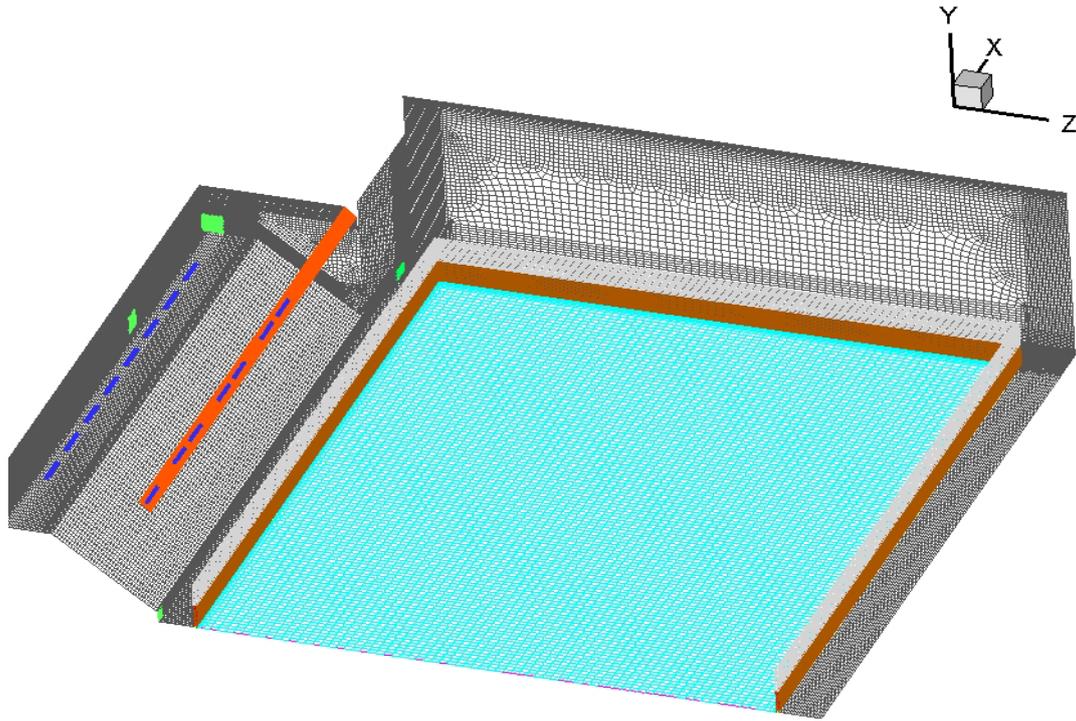


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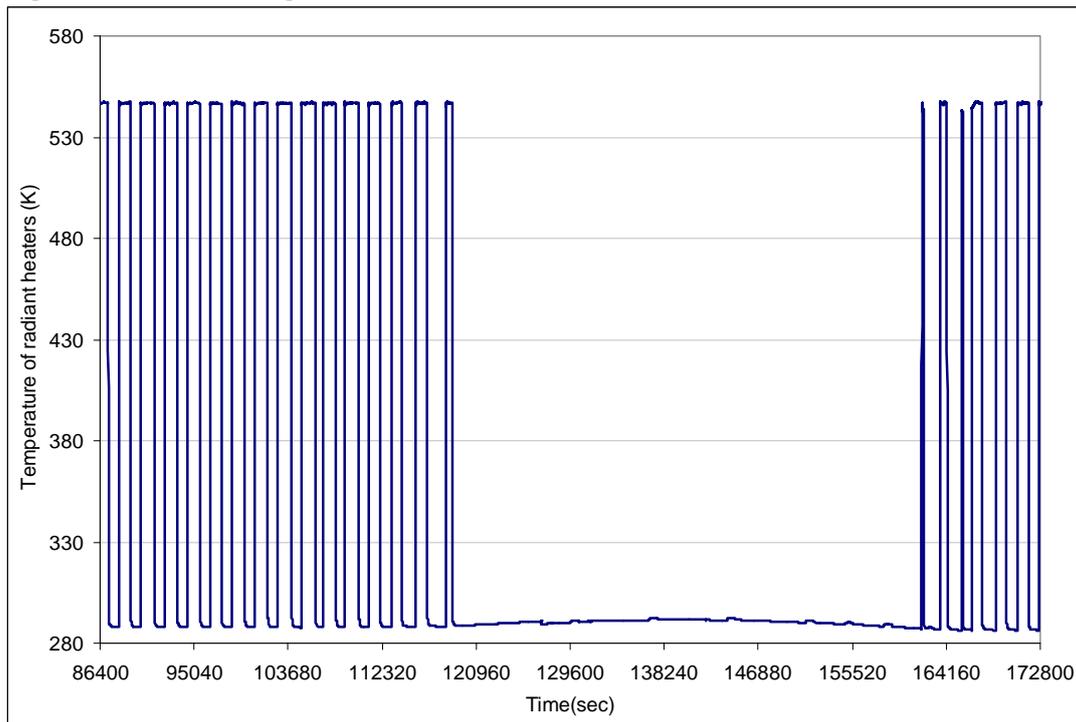


Figure 5. Temperature of the radiant heaters

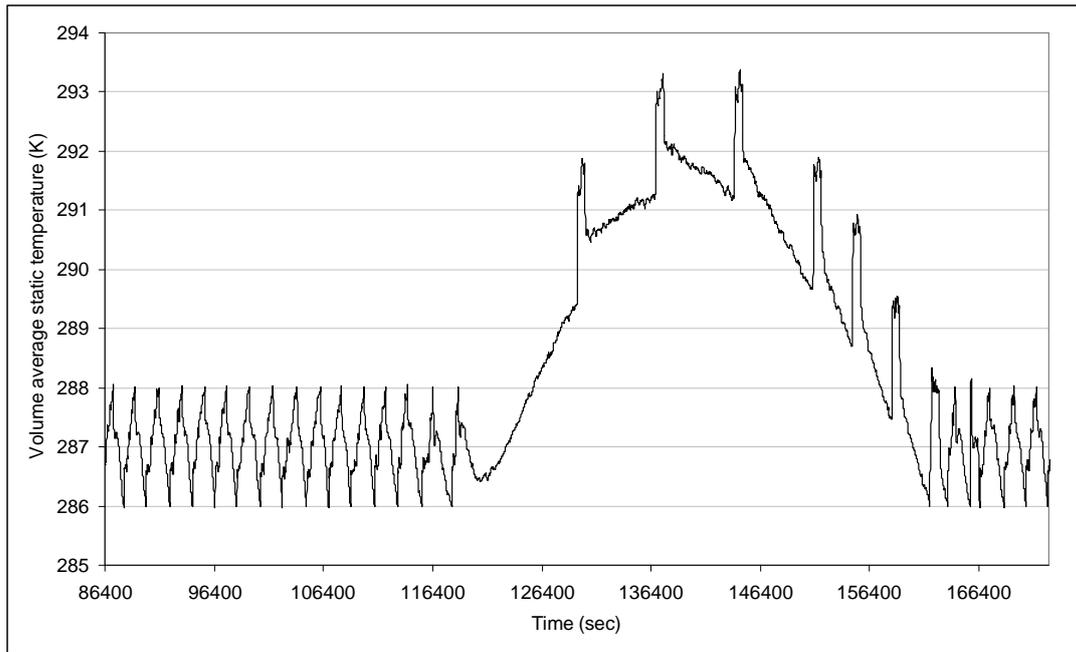


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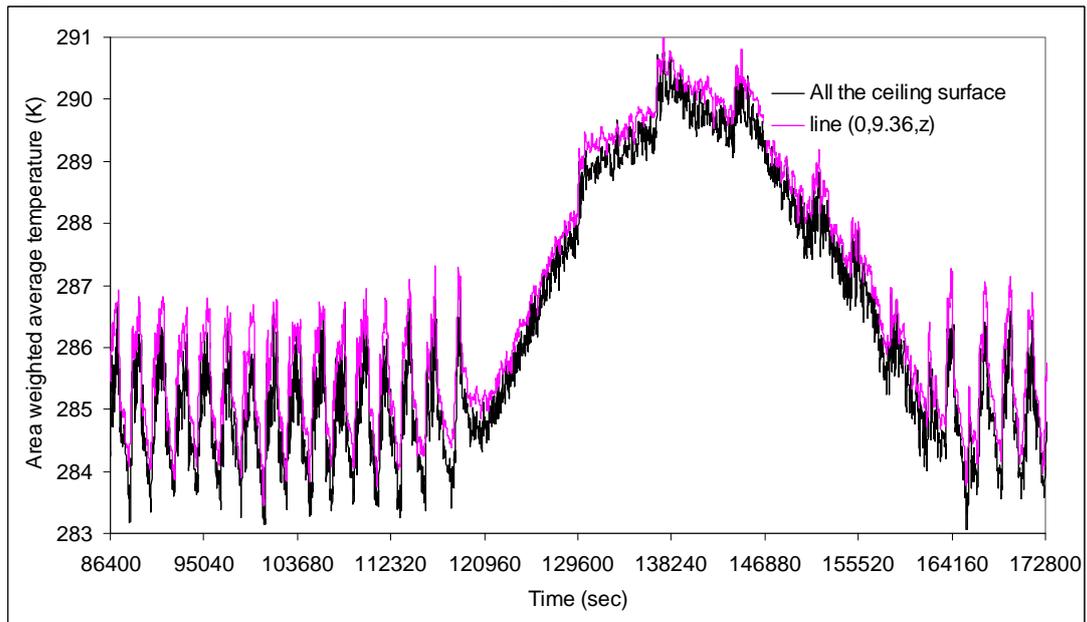


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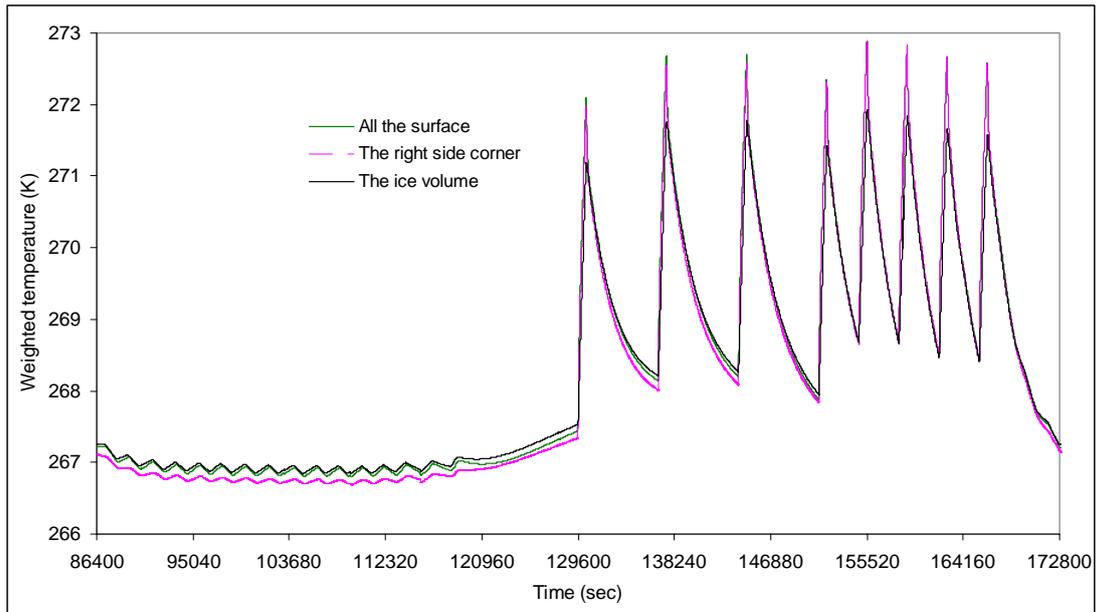


Figure 8-a. Ice temperature as a function of time

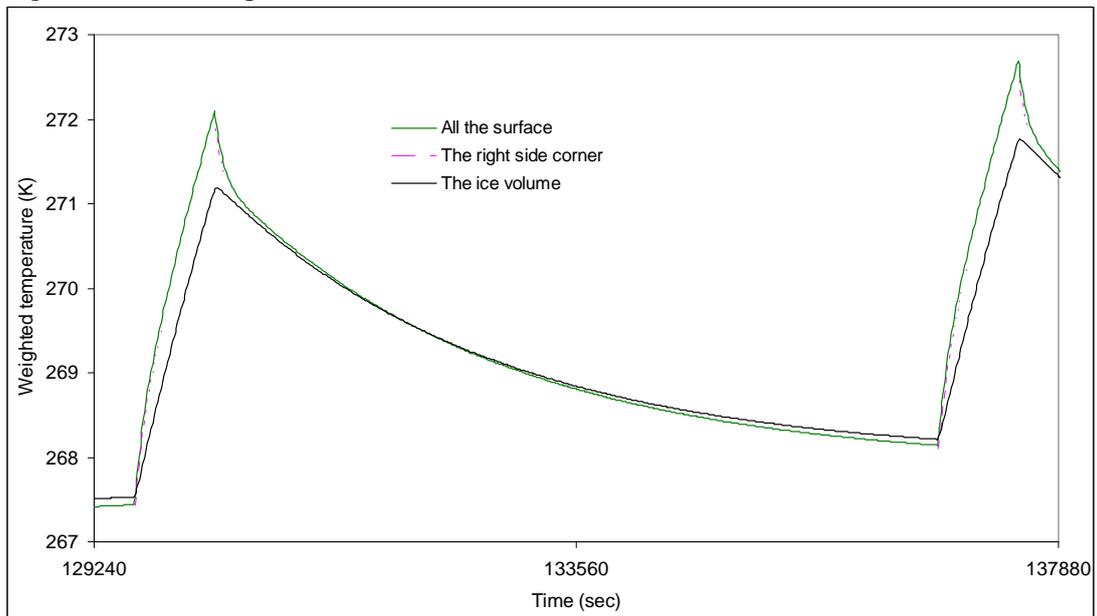


Figure 8-b. Ice temperature during resurfacing operations

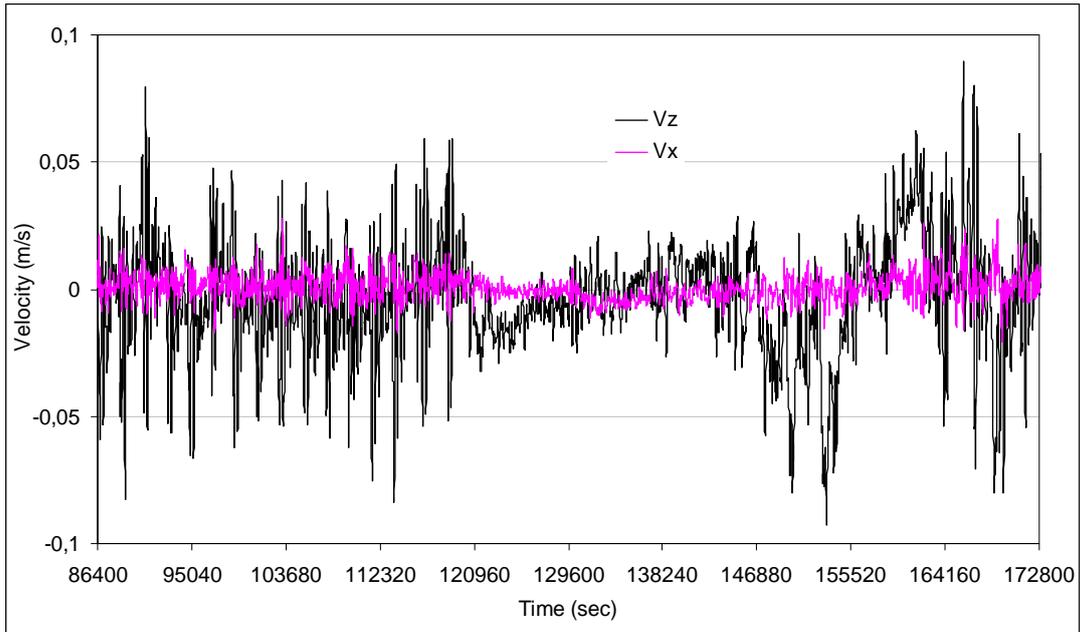


Figure 9. Velocity components one meter above the ice ($x=0, z=0$)

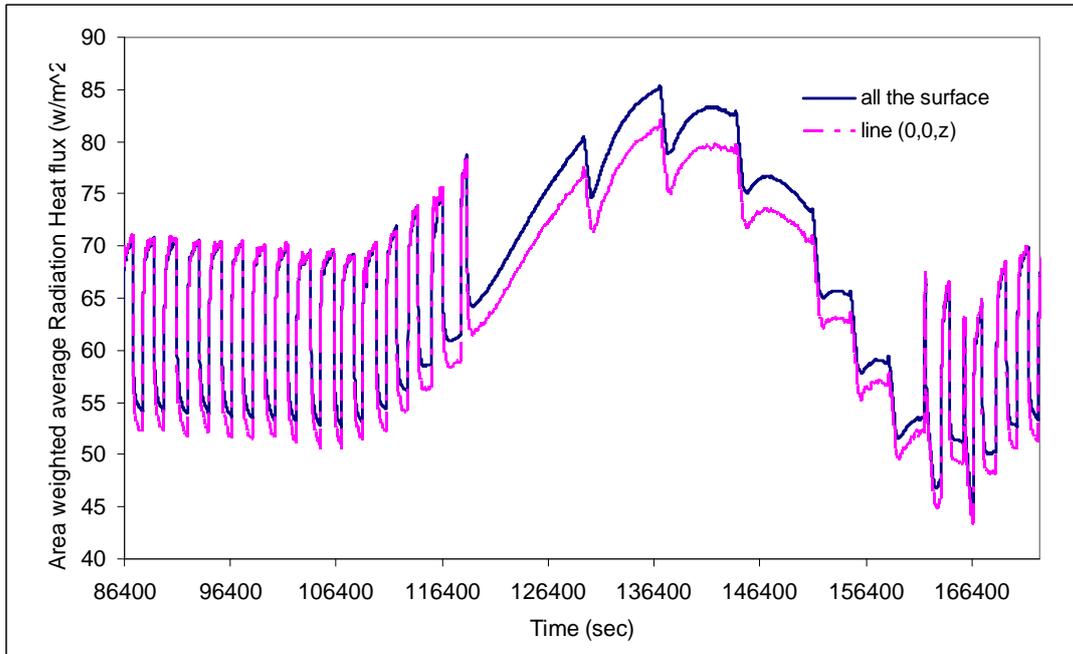


Figure 10. Net radiation heat flux into the ice

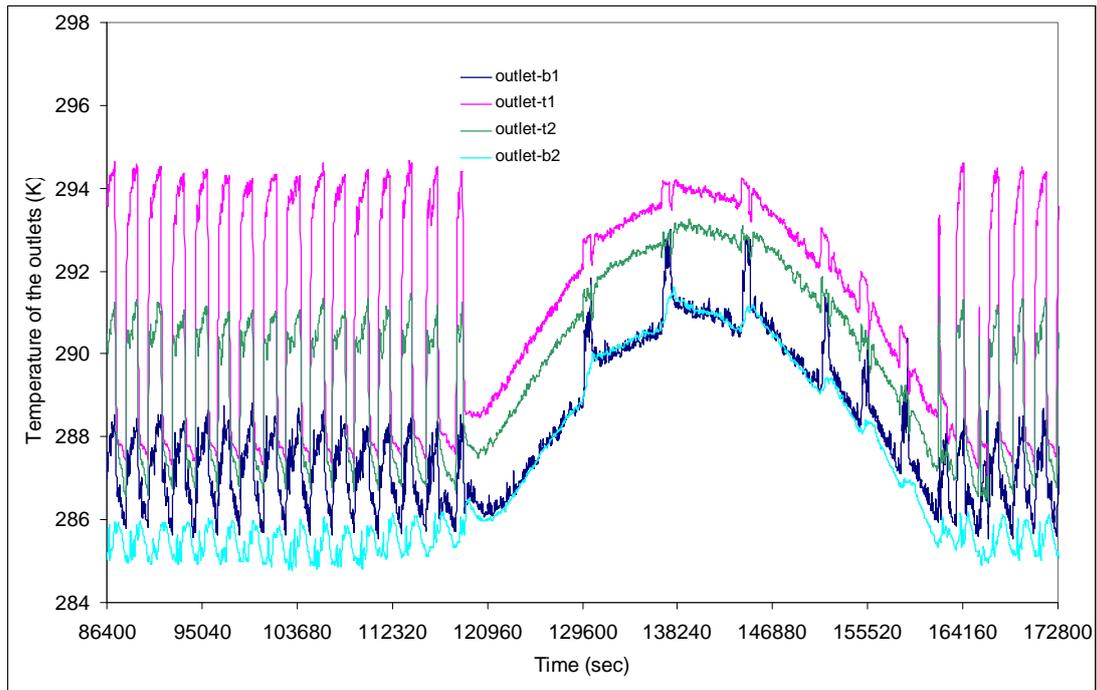


Figure 11. Air temperature at the outlets

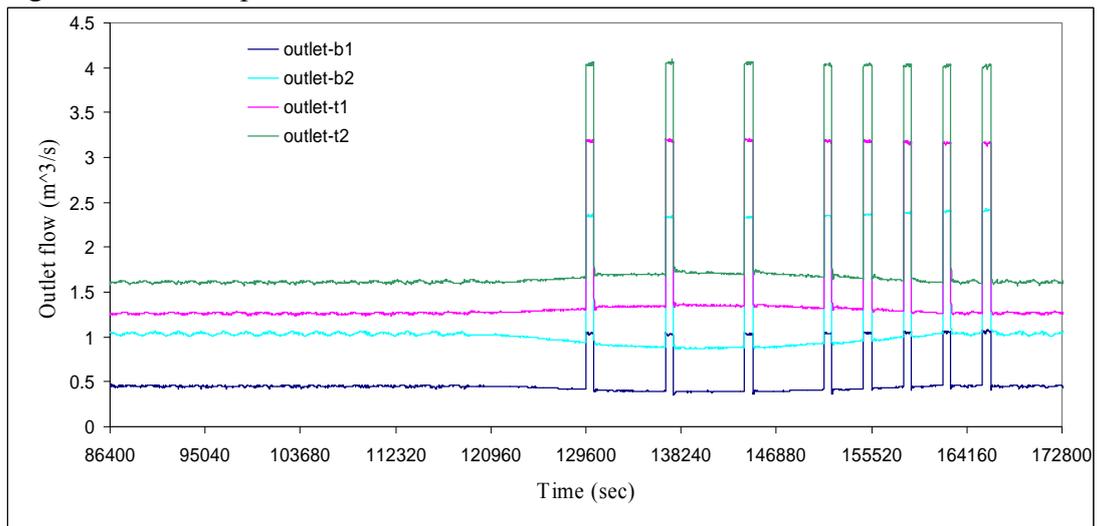


Figure 12. Volumetric flow rates through the four outlets

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