Numerical simulation of energy consumption of artificial ground freezing applications subject to water seepage

Simulation numérique de la consummation d'énergie des applications pour la congélation artificielle du sol soumise au flux de l'eau souterraine

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ABSTRACT: The energy consumption of artificial ground freezing applications results from the necessary (consumed) refrigeration capacity. The determination of the refrigeration capacity requires the numerical modeling of heat transfer processes within the freeze pipe. Therefore a new module has been implemented in the Finite Difference Program SHEMAT. An approach for calculating the heat transfer processes and finally the refrigeration capacity is presented in this paper. Thus numerical simulations are carried out to determine the influence of groundwater flow on the refrigeration capacity and the related energy consumption for artifical ground freezing applications.

RÉSUMÉ : La consommation d'énergie des applications pour la congélation artificielle du sol résulte de la capacité nécessaire de réfrigération. La détermination de cette capacité de réfrigération exige un modèle numérique des processus de transfert de la chaleur dans le tuyau de congélation. C'est pourquoi un nouveau module a été implémenté dans le programme SHEMAT. Une approche pour calculer le transfert de la chaleur et finalement la capacité de réfrigération est présentée dans cet exposé. Ainsi, des simulations numérique ont été réalisées pour déterminer l'influence du flux des eaux souterraines sur la capacité de réfrigération et la consommation d'énergie qui en résulte des applications pour la congélation artificielle du sol.

KEYWORDS: artificial ground freezing, refrigeration capacity, heat transfer, Nusselt number

1 INTRODUCTION

The ground freezing method aims to provide artificially frozen soil which increases the strength of the ground and makes it impervious to water seepage. In recent years the artificial ground freezing method is more often used not only in tunneling but also in the construction of common basement stories. However, the regular application often fails as the energy consumption and its related costs are expected to be excessively high.

In the last few years several numerical simulations using the program SHEMAT (Simulator for Heat and Mass Transport) were carried out at the chair of Geotechnical Engineering of RWTH Aachen University. The Finite-Difference Program SHEMAT had been developed by a group of Prof. Clauser at the chair of Applied Geophysics of RWTH Aachen University for the simulation of geothermal processes in porous rocks (Clauser 2003). Mottaghy and Rath (2006) implemented a phase change model including latent heat effects due to freezing and thawing of subsurface fluids, as well as temperature dependent thermal properties. As part of the dissertation of Baier (2009), this work has been extended in terms of varying freezing curves and additional thermal and hydraulic ground properties and their temperature dependence. The numerical simulations carried out showed that significant reductions of the freezing time can be achieved by flow-adapted freeze pipe arrangements (Ziegler et al. 2009).

To increase the regular application of the artificial ground freezing method not only the freezing time but also the energy consumption has to be reduced. Therefore it is necessary to take into consideration the operating phase which essentially affects the total energy consumption of artificial ground freezing applications. The energy consumption can be estimated by determining the refrigeration capacity of artificial ground freezing applications first.

The aim of this paper is to highlight the determination of the refrigeration capacity by numerically modelling the heat transfer processes in the freeze-pipe itself and at the transition to the soil.

2 DETERMINATION OF REFRIGERATION CAPACITY

To ensure a realistic determination of the refrigeration capacity the freeze pipe has to be examined in detail. That implies the consideration of the so far neglected heat transfer within the freeze pipe itself.

2.1 *Basic heat transfer within freeze pipes*

The heat transport mechanisms, occuring during the ground freezing process, can be split into mechanisms in the soil, in the freeze pipes and in the transition of both. In the soil the heat transport consists of conduction and forced convection (advection) due to water seepage. These processes are not part of the current research and therefore not illustrated in detail. For a detailled explanation see e.g. Baier (2009).

In the following, the heat transfer processes in the freeze pipes are described in detail.

In general freeze pipes are coaxial pipes that consist of an inner pipe of polyethylene where the refrigerant moves downwards and an outer pipe of steel where the refrigerant moves upwards. In most cases a calcium chloride brine is used as refrigerant. The heat extraction of the surrounding soil warms the refrigerant in the outer pipe moving upwards. Therefore the refrigeration capacity can be determined from the temperature difference of inlet and outlet refrigerant temperature, the pump rate Q and the volumetric heat capacity of the refrigerant c_v (see Eq. 1).

$$P = c_V \cdot Q \cdot (T_{inlet} - T_{outlet}) \quad (1)$$

For a realistic determination of the outlet temperature the ongoing heat transfer processes ($\dot{Q}_s / \dot{Q}_e$) within the freeze pipe have to be considered (see Figure 1).

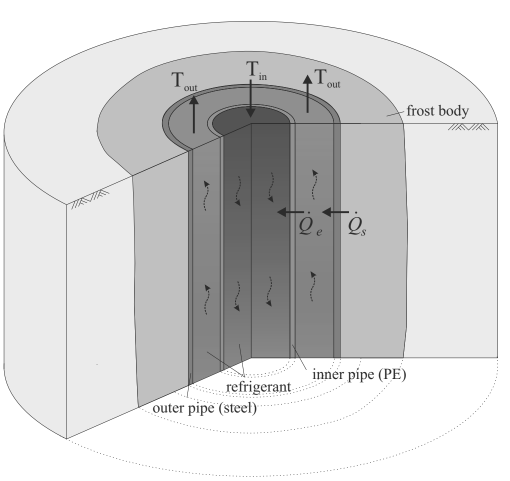

Figure 1. Freeze pipe in detail with occurring heat flow.

The heat flow $\dot{Q}_s$ between the soil and the outer freeze pipe comprises conductive heat flow through the outer freeze pipe and convective heat flow due to the flowing refrigerant. The heat flow $\dot{Q}_e$ between the down- and upstream via the inner freeze pipe can be divided into two mechanisms. On the one hand, conductive heat flow through the inner freeze pipe and on the other hand convective heat flow both inside and outside the inner freeze pipe. In case of a flowing refrigerant the vertical heat transfer is dominated by advection which is already considered in the horizontal heat flow. Therefore the vertical conductive heat flow within the refrigerant is neglected.

The heat transfer due to conduction can be determined by using Fourier's law. The conductive heat flow for a coaxial freeze pipe for n conductive layers with the thermal conductivity of the pipe material λ_i [W/(mK)] results in:

$$\dot{Q}_{conduction} = \sum_{i=1}^{n} 2 \cdot \pi \cdot \lambda_i \cdot \frac{\Delta T_i}{ln(r_{out,i}/r_{in,i})} \tag{2}$$

The convective heat flow depends on the heat transfer coefficient α_i [W/(m²K)]:

$$\dot{Q}_{convection} = \sum_{i=1}^{n} 2 \cdot \pi \cdot r_i \cdot \alpha_i \cdot \Delta T \tag{3}$$

The heat transfer coefficient depends on the freeze pipe geometry and the flow and material properties of the refrigerant. As a function of the Nusselt number Nu the heat transfer coefficient is defined as (Baehr and Stephan 2006):

$$\alpha = \frac{Nu \cdot \lambda_F}{d_{hydraulic}} \tag{4}$$

with the thermal conductivity of the refrigerant λ_F and the hydraulic freeze pipe diameter $d_{hydraulic}$. The hydraulic diameter of the inner freeze pipe corresponds to its inner diameter (see Figure 2).

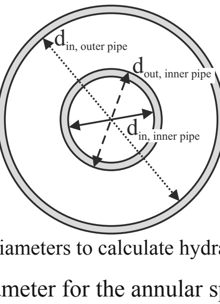

Figure 2. Freeze pipe diameters to calculate hydraulic diameter.

The hydraulic diameter for the annular space is:

$$d_{hydraulic} = d_{in,outer pipe} - d_{out,inner pipe} \tag{5}$$

The Nusselt number Nu depends on the flow type – laminar or turbulent. To differ between the two flow types the Reynolds number Re can be used, which depends on the refrigerant flow velocity v_F, the kinematic viscosity ν and the hydraulic diameter (VDI Heat Atlas 2010).

$$Re = \frac{v \cdot d_{hydraulic}}{\nu_F} \tag{6}$$

In the literature (Gnielinski 1995, VDI Heat Atlas 2010) it is generally mentioned that a full developed turbulent fluid flow in a pipe exists for $Re > 10^4$. For $Re < 2300$ a laminar fluid flow occurs. In the range of $2300 < Re < 10^4$ the transition from laminar to turbulent flow takes place. Furthermore, for all flow types a distinction has to be made between a modified fluid flow in the inlet area and a thermic and hydrodynamic fully developed fluid flow behind this area (VDI Heat Atlas 2010). Due to the spatial separation of the mechanical refrigeration plant and the inlet area of the freeze pipes, it can be assumed that the fluid flow reaching the inlet area is already fully developed.

Moreover, the VDI Heat Atlas (2010) indicates the differentiation between flow inside a pipe and in a concentric annular gap. In this paper only the Nusselt numbers for the inner freeze pipe are outlined. The equations for the calculation in the annular gap can be found in VDI Heat Atlas (2010).

In general the Nusselt number depends on the Reynolds number Re, the Prandtl number Pr, the inner pipe diameter d_i and the length of the pipe l. Thus the Nusselt number in case of a laminar flow and a constant heat flux density along the freeze pipe can be calculated with:

$$Nu_{m,q} = \left[4{,}364^3 + 0{,}6^3 + \left(1{,}953 \left(Re\,Pr \frac{d_i}{l} \right)^{1/3} - 0{,}6 \right)^3 \right]^{1/3} \tag{7}$$

In case of a turbulent flow, there is no need for a differentiation between the boundary conditions "constant wall temperature" and "constant heat flux density" since the Nusselt numbers are nearly equal. Thus the Nusselt number for turbulent flow is defined as:

$$Nu_m = \frac{(\xi/8)\,Re\,Pr}{1 + 12{,}7\sqrt{\xi/8}\left(Pr^{2/3} - 1\right)} \left[1 + \left(\frac{d_i}{l} \right)^{2/3} \right] \tag{8}$$

with:

$$\xi = \left(1{,}8\,log_{10} Re - 1{,}5\right)^{-2} \tag{9}$$

According to Gnielinski (1995) the following interpolation function for the transition region between laminar and fully turbulent flow should be used:

$$Nu_m = (1 - \gamma) \cdot Nu_{m,L,2300} + \gamma \cdot Nu_{m,T,10^4} \tag{10}$$

with:

$$\gamma = \frac{Re - 2300}{10^4 - 2300} \quad and \quad 0 \le \gamma \le 1$$

Besides the Reynolds number the Prandtl number Pr is needed for the calculation of the Nusselt number. The Prandtl number characterizes the material properties of the refrigerant (kinematic viscosity ν, thermal conductivity λ_F and volumetric heat capacity c_F) (Baehr and Stephan 2006):

$$Pr = \frac{\nu \cdot c_F}{\lambda_F} \tag{11}$$

2.2 *Numerical modeling*

For a realistic calculation of all heat transfer mechanisms described afore the use of numerical methods becomes necessary. To avoid a very fine discretization which causes long simulation times a seperate module "freezrefcap" for the calculation of the heat transfer processes within the freeze pipe has been developed in cooperation with Geophysica

Beratungsgesellschaft mbH. This module is based on a Finite Difference formulation for simulating borehole heat exchanger developed by Mottaghy and Dijkshoorn (2012) and has been modified for the simulation of artificial ground freezing applications.

Adapted from the Kelvin line source theory the freeze pipes are modeled as line sources and the horizontal heat transfer is determined, using the concept of thermal resistances (Hellström 1991). To realize the coupling of the module "freezrefcap" with SHEMAT the soil temperature calculated in SHEMAT T_{Soil} is passed to the new module. In turn, a cooling generation $\dot{Q}_s$ returns to SHEMAT (see Figure 3).

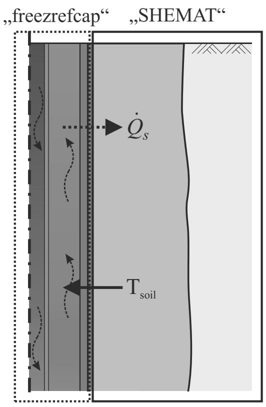

Figure 3. Coupling of "freezrefcap" module and SHEMAT.

The numerical model includes the afore-mentioned heat transfer conditions for the calculation of the thermal resistances. For the numerical model the inner freeze pipe and the annular space are divided into grid cells that are connected via thermal resistances (see Figure 4). Assuming a steady state solution the temperature of all grid cells is calculated considering the heat balance equation in every time step. Due to the high flow rate inside the freeze pipes and the comparatively short freeze pipe length this simplified steady state solution is a good approximation.

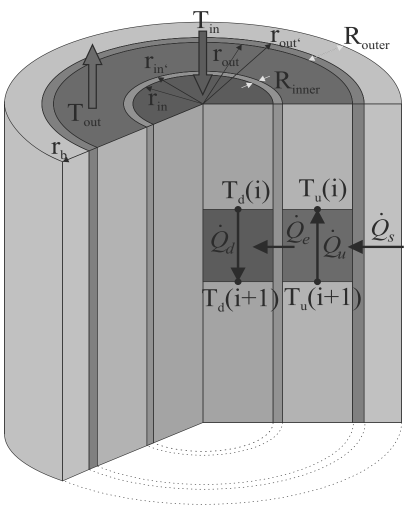

Figure 4. Calculation basis of module "freezrefcap" according to Mottaghy and Dijkshoorn (2012).

The module "freezrefcap" offers the opportunity to activate and deactivate freeze pipes, which provides the basis for the simulation of different modes during the operating phase. The distinction between "flow" and "no flow" case requires different calculations of thermal resistances and the temperature distribution inside the freeze pipe. In this paper only the "flow" case is described.

For the determination of the heat flow $\dot{Q}_s$ between the soil and the outer pipe and $\dot{Q}_e$ between the inner and the outer pipe the temperatures calculated in the previous time step (t-1) are used (see Eq. 12 and Eq. 13). The thermal resistances R_{inner} and R_{outer} are also calculated considering the results of the previous time step. Because of the flowing refrigerant the different grid layers i need to be taken into account.

$$\dot{Q}_s = \frac{T_{soil}(i) - T_u^{t-1}(i)}{R_{outer}} \tag{12}$$

For the determination of the heat flow between the outer and the inner pipe adjacent temperatures in the downstream and in the upstream are used (see Eq. 13).

$$\dot{Q}_e = \frac{T_u^{t-1}(i+1) - T_d^{t-1}(i)}{R_{inner}} \tag{13}$$

The temperature in the downstream $T_d^t(i+1)$ of the actual time step t is determined based on the downstream temperature $T_d^{(t-1)}(i)$ of the overlying grid cell i for the previous time step (t-1) because of the flowing refrigerant.

$$T_d^t(i+1) = T_d^{t-1}(i) + \frac{\dot{Q}_e}{q_F \cdot c_F} \tag{14}$$

$$T_u^t(i) = T_u^{t-1}(i+1) + \frac{-\dot{Q}_e + \dot{Q}_s}{q_F \cdot c_F} \tag{15}$$

c_F indicates the volumetric heat capacity of the refrigerant and q_F the flow rate.

Besides the flow rate the inlet temperature or the refrigeration capacity can be choosen as input parameters. Furthermore the simulation of different refrigerants requires just a simple implementation of the temperature dependent fluid parameters.

3 NUMERICAL SIMULATION

Former numerical simulations at the Chair of Geotechnical engineering showed that groundwater flow has an important influence on the freezing process. The results outlined that the freezing time increases disproportionately and the frost body development decreases with an increasing flow velocity.

To further investigate the influence of groundwater flow on the refrigeration capacity a numerical simulation of a simplified model with only one freeze pipe has been carried out using the module "freezrefcap". Because of a missing module validation against measured data from laboratory model tests only the qualitative influence of groundwater flow is outlined. For this example a freeze pipe with an outer diameter of 10 cm, an inner diameter of 5 cm and a length of 9.5 m has been chosen. The inner pipe was assumed to consist of polyethylene and the outer pipe of steel. As refrigerant a 29 % $CaCl_2$ brine has been chosen. The results of the numerical simulation are displayed in Figure 5.

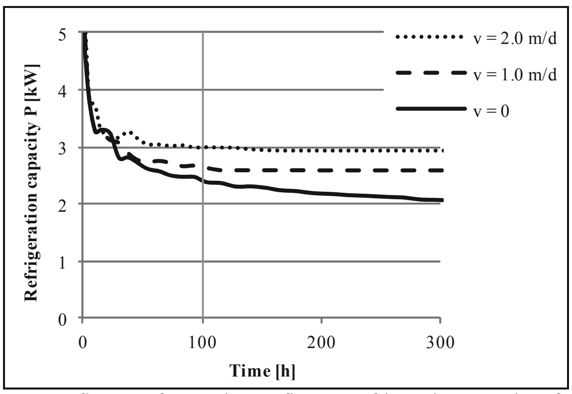

Figure 5. Influence of groundwater flow on refrigeration capacity of one freeze pipe.

It is obvious that an increase in flow velocity causes not only a reduced frost body development but also an increased refrigeration capacity. The reason for this increase is the additional convective heat flow caused by the groundwater flow. The flow velocity influences the artificial ground freezing application twice, because the increased refrigeration capacity has to be hold up for a longer time period to freeze the required frost body contour and during the operating phase.

Figure 5 displays that the refrigeration capacity with groundwater flow decreases with time due to an increasing frost body. However, the refrigeration capacity for a flow velocity of 1.0 m/d and 2.0 m/d proceeds constant. This implies an stagnating frost body growth, a steady state. Such a steady state indicates a thermal equlibrium of the heat supplied by the groundwater flow and extracted by the freeze pipes.

Monitoring points in the soil around the freeze pipe also indicating a stagnation in temperature course confirm this assumption (see Figure 6).

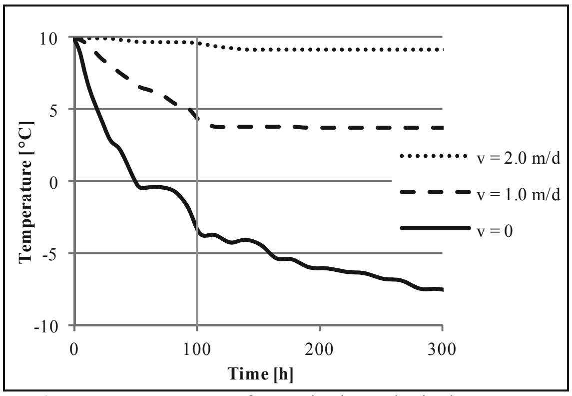

Figure 6. Temperature course of a monitoring point in the upstream.

Comparing the refrigeration capacity after 100 hours, when frost body still grows for all flow velocities, it becomes clear that the refrigeration capacity increases about 10 % for a flow velocity of 1.0 m/d and even about 25 % for a flow velocity of 2.0 m/d

4 CONCLUSION

An approach for the realistic determination of the refrigeration capacity by calculating the heat transfer processes within a freeze pipe was presented. By separating the “freezrefcap” module from SHEMAT and defining only two necessary interfaces for the coupling a very fine discretization and long computing times as a consequence can be avoided. The module offers the opportunity to calculate the outlet temperature and as a result the refrigeration capacity by entering the inlet temperature and the flow rate of the refrigerant. Thus the influence of different refrigerants on the refrigeration capacity can be estimated by numerical simulations.

The aim of further research is to validate the “freezrefcap” module by simulating a laboratory model test influenced by groundwater flow. Thus quantitative statements on the outlet temperature and the refrigeration capacity can be given for various artificial ground freezing applications subject to water seepage.

At this point qualitative statements already indicate that the refrigeration capacity increases disproportionally with an increasing flow velocity. In the further research process the influence of the operating phase on the total refrigeration capacity and the related energy consumption is determined. The aim of the research project is the simulation and optimization of artificial ground freezing applications regarding both time and energy aspects already in the design phase.

5 ACKNOWLEDGEMENTS

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