COMPUTATIONAL MODEL AND EXPERIMENTAL VALIDATION OF THE ICE PLUG OBTURATION OF A HORIZONTAL PIPE

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The service lifespan of an installation, is decisively controlled by the components and equipment which degrade faster over time. The technique of forming controlled ice plugs is used for carrying out circuit maintenance or repair activities without shutting down the entire installation. In order to conduct the operation, the time and liquid nitrogen requirements must be established in advance through a reliable theoretical model; the model itself must be validated through experimental testing in a controlled environment. The paper describes a CFD − ANSYS Fluent computational model to reveal the time and quantity of liquid nitrogen required for this specific intervention.

Keywords: pipe freezing, liquid nitrogen, horizontal pipe, finite volume method, ANSYS Fluent.

1. Introduction

The technological pipe freezing process is continuously developed by specialized companies, considering the requirements of each individual application. The technique is used for maintenance in industrial installations, municipal pipelines, submarines, etc.

Forming the ice plug inside a pipeline requires using a specialized device and a specific technique for its nominal diameter and for the operating conditions of each individual application [1]. The collar shaped ice plugging device (fixed or removable) is made out of two half-clamps bolted together around the pipe (Fig. 1) [2].

The refrigerant draws heat from the device and pipe walls and from the liquid inside the pipe. The heat thus absorbed evaporates some of the liquid nitrogen and is transported outside the freezing device by the nitrogen gas. The heat and mass transfer determine the local pipe wall temperature reduction, favoring the

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deposition of ice in successive layers, until the pipe section is completely closed. From this point on, the ice formation continues along the pipe axis, towards the ends of the freezing jacket.

Fig. 1. Rigid freezing jacket with liquid nitrogen circulation

The freezing process is mainly influenced by the geometrical properties of the pipe (diameter, orientation), the geometrical properties of the freezing device and the water temperature; water temperature variation during the process characterizes the ice layers growth and implicitly the conductive and convective heat transfer coefficients at the two fluid/solid interfaces (between the water inside the pipe and the ice layer and between the pipe outer wall and the freezing agent). Maintenance operations cannot be conducted without an initial estimate regarding the required time and liquid nitrogen quantity. This paper proposes a CFD computational model using ANSYS Fluent 2019 R3 to simulate the process of forming an ice plug inside a horizontal 200 mm nominal diameter pipeline containing stationary water (the results will be validated by an experiment conducted in similar conditions) and a calculation model for the required mass of liquid nitrogen.

2. Computational models

The ANSYS Fluent Solver does not allow the direct implementation for the dimensionless values (Grashof, Prandtl, Rayleigh numbers, etc.). For the heat transfer analysis with Rayleigh numbers greater than 108, the User's guide [3] recommends implementing a two-step solver:

- The first stage will consist of a stationary analysis for calculating the reduced gravitational acceleration correspondent to the Rayleigh number, the fluid properties (Table 2), the pipe geometrical characteristics and the heat transfer potential to be used in a secondary non-stationary analysis as an approximation for the simulation performed with the real gravitational acceleration values. The water heat transfer coefficients in an enclosed space will be calculated using a dimensionless coefficient for each stage of the water temperature decrease towards 0°C;

- The second stage will be an ANSYS Fluent CFD simulation for the ice plug formation using the water heat transfer coefficients resulted from the first stage, and the values for density, specific heat, thermal conductivity (for ice), dynamic and kinematic viscosities and the thermal expansion coefficient from tables 2 and 3.

Material properties

The properties and parameters of the steel from which the test section pipe is made (steel with 0.1 % C concentration) are shown in Table 1 [4-5]. Water properties variation according to temperature are shown in Table 3 [5]. Ice properties variation according to temperature are shown in Table 2 [4-5]. Liquid nitrogen variation according to temperature are shown in Table 4 [6-7].

Table 1

Properties of the pipe and the steel from which it is made

Parameter	Unit	Value
External/internal diameters	mm	219/213
Density	kg/m^3	7850
Specific heat	J/kg·K	465
Thermal conductivity	W/m·K	59,313

Table 2

Ice properties variation with temperature

Parameter	Temperature	Density	Specific heat	Thermal conductivity
Unit	K	kg/m^3	J/kg·K	W/m·K
Value	273,15	916,2	2050	2,22
	268,15	917,5	2027	2,25
	263,15	918,9	2000	2,3
	258,15	919,4	1972	2,34
	253,15	919,4	1943	2,39
	248,15	919,6	1913	2,45
	243,15	920	1882	2,5
	238,15	920,4	1851	2,57
	233,15	920,8	1818	2,63
	223,15	921,6	1751	2,76
	213,15	922,4	1681	2,9
	203,15	923,3	1609	3,05
	193,15	924,1	1536	3,19
	183,15	924,9	1463	3,34
	173,15	925,7	1389	3,48

Table 3

Water properties variation with temperature

Temp.	Dens.	Specific heat	Thermal cond.	Dynamic viscosity	Kinematic viscosity	Thermal expansion coeff.
K	kg/m^3	J/kg·K	W/m·K	kg/m·s	m^2/s	1/K
273,15	999,9	4226	0,558	0,00179	$1.789 \cdot 10^{-6}$	$-0,7 \cdot 10^{-4}$
278,15	1000	4206	0,568	0,00146	$1.535 \cdot 10^{-6}$	$0,52 \cdot 10^{-5}$
283,15	999,7	4195	0,577	0,00127	$1.3 \cdot 10^{-6}$	$0,95 \cdot 10^{-4}$
288,15	999,1	4187	0,587	0,00111	$1.146 \cdot 10^{-6}$	$1,53 \cdot 10^{-4}$
293,15	998,2	4182	0,597	0,00098	$1.006 \cdot 10^{-6}$	$2,1 \cdot 10^{-4}$
298,15	997,1	4178	0,606	0,00087	$0.994 \cdot 10^{-6}$	$2,4 \cdot 10^{-4}$

Table 4

Liquid nitrogen properties variation with temperature

Parameter	Temperature	Density	Dynamic viscosity
Unit	K	kg/m^3	kg/m·s
Value	100	3,437	0,00000696
	120	2,858	0,00000824
	140	2,425	0,00000946
	160	2,104	0,00001065
	180	1,867	0,00001179
	200	1,689	0,0000129
	220	1,55	0,00001396
	240	1,433	0,000015
	260	1,328	0,00001599
	280	1,226	0,00001696
	300	1,126	0,0000179

Heat transfer coefficients

Due to the fact that the temperature difference between the pipe surface and the liquid nitrogen is large, the heat transfer at the pipe is achieved by turbulent film pool boiling. A correlation for the heat transfer coefficient in the case of steady turbulent film pool boiling is given by Miheev & Miheeva [8-9]:

$$\overline{\propto_{lN_2}} = 0{,}25 \cdot \sqrt[a]{\frac{\lambda"^2 \cdot c_p" \cdot g(\rho' \cdot \rho")}{\nu"}}, \tag{1}$$

where:

- $\overline{\propto_{lN_2}}$ – mean heat transfer coefficient for liquid nitrogen, $[\mathrm{W/m^2 \cdot K}]$;
- $\lambda"$ – vapour phase thermal conductivity, $\lambda" = 15{,}89 \cdot 10^{-3}\ \mathrm{W/m \cdot K}$;
- $c_p"$ – isobaric specific heat for the nitrogen vapour film, $c_p" = 1{,}044 \cdot 10^3\ \mathrm{J/kg}$;
- g – gravitational acceleration, $g = 9{,}81\ \mathrm{m/s^2}$;
- ρ' – density of the liquid phase, $\rho' = 804\ \mathrm{kg/m^3}$;
- $\rho"$ – density of the vapour phase, $\rho" = 1{,}975\ \mathrm{kg/m^3}$;
- $\nu"$ – vapour phase kinematic viscosity, $\nu" = 5{,}75 \cdot 10^{-6}\mathrm{m^2/s}$.

which gives

$$\overline{\propto_{lN_2}} = 178\mathrm{W/m^2 \cdot K}. \tag{2}$$

According to Newton's fundamental equation for thermal convection, the heat flux absorbed by the volume of liquid nitrogen inside the freezing device is

$$\dot{Q}" = \propto_{lN_2} \cdot \Delta T,$$

where ΔT is the heat transfer potential (the temperature between the water inside the pipe and the liquid nitrogen boiling temperature), $\Delta T = 211K$.

The heat flux through the surface cooled by liquid nitrogen (at an initial working temperature of 15°C):

$$\dot{Q}" = 37549{,}28\ \mathrm{W/m^2} \tag{3}$$

The thermal transfer between the testing pipe and the environment will be neglected.

The heat transfer between the ice and the water in the interior of the pipe is carried out by turbulent free convection of the water. For the case of steady free convection in an enclosed space Isachenko, Osipova & Sukomel give the following correlation between the Grashof number and the ratio of the equivalent thermal conductivity (including both conduction and convection) to the fluid thermal conductivity [8] [10]:

$$\varepsilon = \frac{\lambda_{ech}}{\lambda} = 0{,}4 \cdot (\mathrm{Gr} \cdot \mathrm{Pr})^{0{,}2}, \tag{4}$$

$$\mathrm{Pr} = \frac{\nu \cdot \rho \cdot c_p}{\lambda}, \tag{5}$$

$$\mathrm{Gr} = \frac{g \cdot \beta \cdot \Delta T \cdot \delta^3}{\nu^2}, \tag{6}$$

where,

- ν – water kinematic viscosity for each temperature stage, $[\mathrm{m^2/s}]$;
- ρ – water density for each temperature stage, $[\mathrm{kg/m^3}]$;
- c_p – water specific heat for each temperature stage, $[\mathrm{J/kg \cdot K}]$;
- λ – water thermal conductivity for each temperature stage, $[\mathrm{W/m \cdot K}]$;
- g – gravitational acceleration, $[\mathrm{m/s^2}]$;
- β – water thermal expansion coefficient for each temperature stage, $[1/\mathrm{K}]$;
- ΔT – temperature difference between water and the ice layer for each temperature stage, [K];
- δ – ice layer length for each temperature stage, [m].

Considering that the water temperature decrease is linear in the area of influence from the freezing device (which will be confirmed experimentally on chapter 3), the mean heat transfer coefficient through convection for the water inside the pipe shall be calculated considering four stages that the water reaches before freezing: 15°C, 10°C, 5°C and 0,5°C. Results are shown in table 5.

Table 5

Equivalent thermal conductivities for different water-cooling stages

Water temp, [°C]	Pr	Gr	ε	λ_{ech}, [W/m·K]
15	8,17	$26{,}8\cdot10^6$	18,62	10,93
10	9,45	$8{,}61\cdot10^6$	15,28	8,82
5	11,37	$1{,}3\cdot10^6$	10,86	6,17
0,5	13,55	$1{,}67\cdot10^6$	7,47	4,17

The total amount of heat absorbed by the freezing device throughout the ice plug development process (Q_t) is equal to the heat transferred during each stage of the process in the freeze area plus the heat loss through the pipe wall and the water inside the pipe:

$$Q_t = Q_1 + Q_2 + Q_3 + Q_4$$

- Q_1 – necessary heat extracted from the water volume in order for it to reach 0°C, [J]:

$$Q_1 = m_a \cdot c_{p_a} \cdot (T_a - T_0)\ , \tag{7}$$

where m_a – mass of water under the influence from the freezing device, [kg]; c_{p_a} – water specific heat, $c_{p_a} = 4187$ J/kg·K; T_a – water temperature; $T_0 = 273{,}15$ K – water freezing temperature;

- Q_2 – necessary heat extracted from the water volume at 0°C in order for it to freeze, [J]:

$$Q_2 = m_a \cdot C_{LS_a}\ , \tag{8}$$

where C_{LS_a} – water latent heat of solidification, $C_{LS_a} = 332432$ J/kg;

- Q_3 – necessary heat extracted from the ice in order for it to reach the temperature required for the ice plug formation (determined experimentally), [J]:

$$Q_3 = m_a \cdot c_{p_g} \cdot \left(T_0 - T_{fp}\right)\ , \tag{9}$$

where T_{fp} – temperature required for the ice plug formation (determined experimentally), $T_{fp} = 223$ K; c_{p_g} – ice specific heat at T_{fp}, $c_{p_g} = 1773{,}9$ J/kg·K;

- Q_4 – heat loss in the pipe wall and in the water around the ice plug, [J]:

$$Q_4 = Q_{pc} + Q_{pa}\ , \tag{10}$$

where Q_{pc} and Q_{pa} are the heat losses on the pipe wall and in the water around the ice plug, [J]:

$$Q_{pc} = 2 \cdot \frac{\lambda_c}{L_{c1}} \cdot \Delta T_{c1} \cdot \pi \cdot (r_e - r_i)^2 \cdot t\ , \tag{11}$$

$$Q_{pa} = 2 \cdot \frac{\lambda_{ech}}{\lambda_{c2}} \cdot \Delta T_{c2} \cdot \pi \cdot r_i^{\,2} \cdot t\ , \tag{12}$$

where:

- $L_{c1} = 0{,}3$ m and $L_{c2} = 0{,}1$ m are the pipe wall lengths corresponding to the $T(L)_{c1}$ and $T(L)_{c2}$ temperatures,

$$T(L)_{c1} = T_a + \frac{T_{mc} - T_a}{\cosh(m_1 \cdot L_{c1})} = 287{,}85\ \text{K}; \quad (13)$$

$$T(L)_{c2} = T_a + \frac{T_{mc} - T_a}{\cosh(m_2 \cdot L_{c2})} = 261{,}5\ \text{K}; \quad (14)$$

- T_{mc} – pipe medium temperature at the ends of the freezing device, $T_{mc} = 258{,}15$ K;
- $m_1 = \sqrt{\frac{2\pi\cdot(r_e-r_i)\cdot\alpha_a}{\pi\cdot(r_e-r_i)^2\cdot\lambda_c}} = 17{,}67$;
- $m_2 = \sqrt{\frac{2\pi\cdot r_i\cdot\alpha_a}{\pi\cdot r_i{}^2\cdot\lambda_c}} = 4{,}96$;
- $\Delta T_{c1} = T(L)_{c1} - T_{mc}$ – medium pipe temperature gradient between the end of the ice plug ($T(L)_{c1}$) and the zone under influence from the freezing device (T_{mc}), $\Delta T_{c1} = 29{,}7$ K;
- $\Delta T_{c2} = T(L)_{c2} - T_{mc}$ – medium water temperature gradient between the zone at L_{c2} distance ($T(L)_{c2}$) and the zone next to the ice plug (T_{mc}), $\Delta T_{c2} = 3{,}35$ K;
- r_e – pipe outer radius, [m];
- r_i – pipe inner radius, [m];
- t – time required for the ice plug primary formation, [s].

Which gives:

$$Q_{pc} = 8841{,}8\ \text{J} = 8{,}84\ \text{kJ}\,,$$
$$Q_{pa} = 61032{,}6\ \text{J} = 61{,}03\ \text{kJ}\,,$$
$$Q_1 = 236864{,}8\ \text{J} = 236{,}86\ \text{kJ}\,,$$
$$Q_2 = 1248081\ \text{J} = 1248{,}08\ \text{kJ}\,,$$
$$Q_3 = 332996{,}1\ \text{J} \cong 333\ \text{kJ}\,,$$
$$Q_4 = 69874{,}5\ \text{J} = 69{,}87\ \text{kJ}\,,$$
$$Q_t = 1887{,}82\ \text{kJ}\,. \quad (15)$$

The mass of liquid nitrogen required for the primary ice plug formation:

$$m = N \cdot 2 \cdot m_{a_{N_2}},$$

where N – number of liquid nitrogen moles required for the ice plug formation,

$$N = \frac{Q_t}{C_{LV_{N_2}}} = 2621{,}97\ \text{moles}\,,$$

where $C_{LV_{N_2}} = 0{,}72$ kJ/mol – liquid nitrogen latent heat of vaporization

- $m_{a_{N_2}} = 14$ u – nitrogen atomic mass,

which gives

$$m = 73{,}4\ \text{kg}\,. \quad (16)$$

Calculating the thermal conductivity coefficients for the case of steady free convection in an enclosed space using the ε coefficient proposed by Isacenko requires using a mean value for the Rayleigh numbers (Ra = Gr · Pr) for each of the four stages that the water reaches before freezing. In order to implement the Rayleigh number value for the phase change calculus, the code requires that the gravitational acceleration should be recalculated considering the Rayleigh number, the liquid characteristics and the heat transfer potential.

$$g = \frac{\mathrm{Ra} \cdot \bar{\nu} \cdot \bar{\propto}}{\bar{\beta} \cdot \Delta T \cdot {d_i}^3}\ , \tag{17}$$

where:

- $\bar{\nu}$ – water kinematic viscosity for each stage of the process, $[\mathrm{m^2/s}]$;
- $\bar{\propto}$ – water medium convection coefficient, $[\mathrm{W/m^2 \cdot K}]$;
- $\bar{\beta}$ – water medium thermal expansion coefficient, $[1/\mathrm{K}]$;
- ΔT – medium temperature difference between water and pipe (determined experimentally), $\Delta T = 51{,}1$ K;
- d_i – pipe inner diameter, $d_i = 0{,}203$ m.

Considering the new values for the gravitational acceleration the CFD simulation requires that the Rayleigh and Grashof numbers, and the thermal conductivities for each stage of the water freezing process should also be recalculated using the relations (4), (5), and (6). The results are shown in table 6.

Table 6

Equivalent thermal conductivities for different water stages used in the CFD simulation

Water temperature [°C]	Ra	g [m/s²]	Gr	ε	λ_{ech} [W/m·K]
15	$7{,}92 \cdot 10^7$	0,41	$1{,}11 \cdot 10^6$	9,85	5,78
10			$3{,}56 \cdot 10^6$	8,08	4,66
5			$5{,}38 \cdot 10^6$	5,75	3,26
0,5			$6{,}93 \cdot 10^6$	1	0,56

3. Results and experimental validation

CFD modelling the heat transfer during the process of freezing a horizontal 200 mm diameter pipe containing stationary water at 15°C

The models implemented in the CFD calculation program are: Multiphase − Volume of Fluid applied for the two phases of the working agent (liquid and solid − ice); Energy; Viscous-laminar; Solidification & Melting.

Geometric model

The geometry for the chosen model was implemented using the Design Modeler environment proper to the ANSYS calculation code. A 6-meter pipe section divided into three sections was modelled in 3D; the length of the pipe was chosen in accordance with that of the experimental test section, in order to have a similar water temperature gradient during the CFD simulation.

Experimental CFD installation geometrical characteristics:

- Pipe inner/outer diameters: $D_i = 203$ mm; $D_e = 219$ mm;
- Pipe length: $L = 6$ m;

- Heat transfer zone length: l [mm], in the center of the pipe.

The ice plug was primarily formed one hour into the 3D simulation; the outer pipe wall temperatures at both sides of the freezing device dropped below 0°C, measuring −4,1°C on the upper section of the pipe (Th3 and Th7) and −47,7°C on the lower half of the pipe (Th4 and Th8). The ice layers measured ~ 310 mm on the upper inner pipe wall and ~ 510 mm on the lower inner pipe wall, Fig. 2.

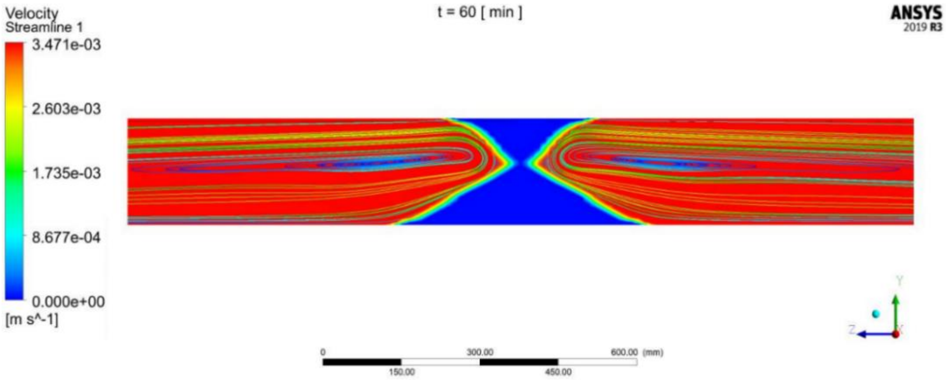

Fig. 2. Primary ice plug formation and water convection currents, t = 60 min

The simulation continued for 34 minutes. The outer pipe wall temperatures measured −49,2°C on the upper section of the pipe (Th3 and Th7) and −85,7°C on the lower half of the pipe (Th4 and Th8). The ice layers measured ~ 410 mm and ~ 614 mm on the upper and lower inner pipe wall, Fig. 3.

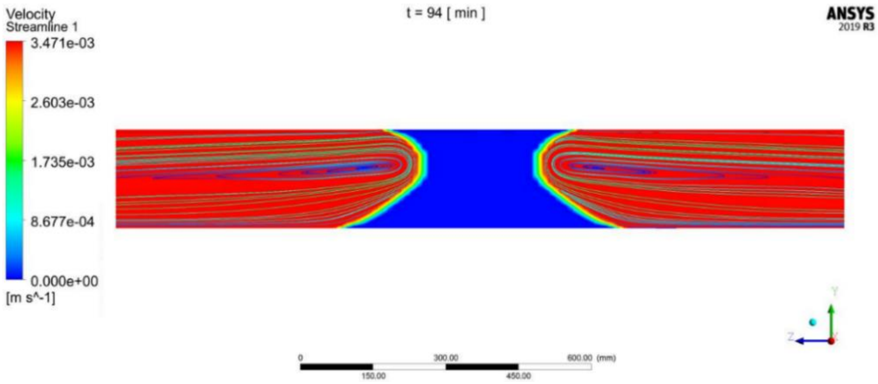

Fig. 3. Ice plug and convection currents at the end of the simulation, t = 94 min

Experimental research

The experiment aimed to form an ice plug inside a horizontal pipe with a 200 mm nominal diameter containing stationary demineralized water. The freezing device is composed of two subassemblies (8 and 2) bolted together forming a collar with an annular compartment (4) around the outside of the pipe (1), figure 4 [10]. The liquid nitrogen compartment 4 is thermally insulated on the outside by the vacuum compartments (5) and (3). The supply with liquid nitrogen is made through the nozzle (6); nozzle (7) is a nitrogen vapour discharge. The thermocouples placement on the pipe outer wall is presented in figure 5. A video surveillance device was used to monitor the ice layers formation inside the pipe [11].

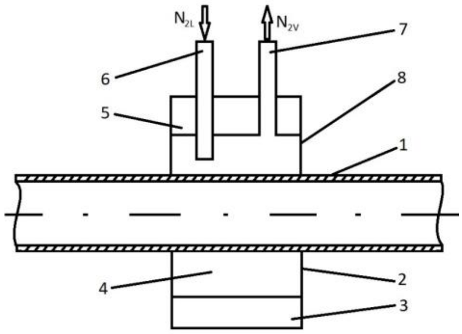

Fig. 4. Freezing device components

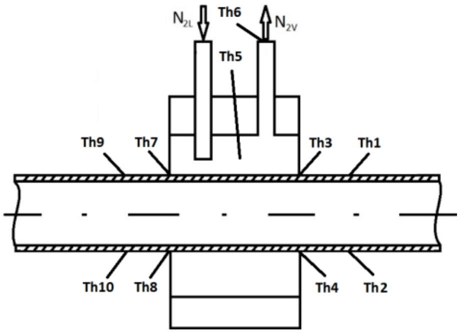

Fig. 5. Thermocouples placement

77 minutes after the start of the test the ice plug had formed primarily, figure 6. The outer pipe wall temperatures measured −44,8°C and −47,3°C on the upper section of the pipe upstream and downstream (Th3 and Th7) and −64,4°C and −67,8 on the lower half of the pipe upstream and downstream (Th4 and Th8). The water temperature measured 13,1°C.

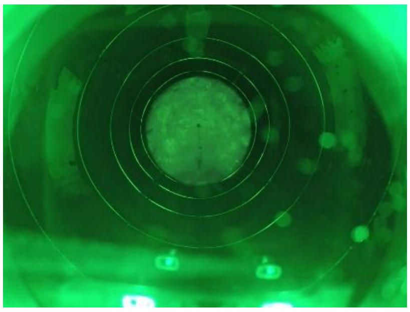

Fig. 6. Primary formed ice plug

After 17 minutes the pressure-drop on the ice plug reached 0,2 bar. Subsequently, the pipe sections upstream and downstream of the test section were dismantled in parallel with the emptying of the installation and their removal. To measure the amount of liquid nitrogen consumed during the freezing process, the Dewar vessel was weighed before the start of the test and at the end; the amount of liquid nitrogen consumed for the application was 116 kg.

Results and comparative analysis

During the experimental application, the water temperature decreased by 2,2°C (14,9°C at the beginning and 12,7°C at the end of the experiment), with a mean temperature gradient of 0,023°C/min; during the CFD simulation, the water temperature decreased by 5,1°C (15°C at the beginning and 9,9°C at the end) with a mean temperature gradient of 0,054°C/min, Fig. 7.

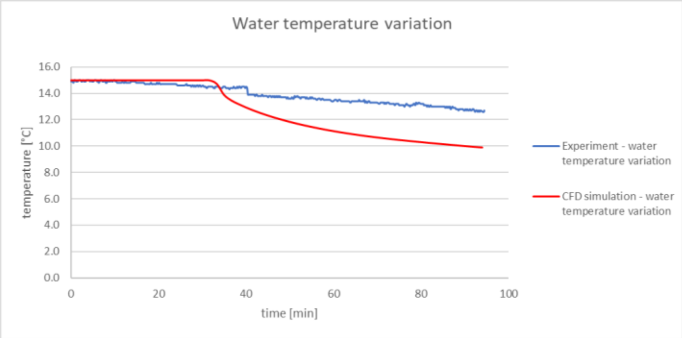

Fig. 7. Water temperature variation during the two tests

Temperatures measured on the outer pipe wall near the freezing device had a different variation during the two tests (caused mainly by the liquid nitrogen flow inside the freezing device from the nozzle mounted on the upper clamp) but reached similar values after 94 minutes: –50°C and –80°C on its upper (Th3 and Th7) and lower (Th4 and Th8) sides, Fig. 8.

The ice plug was primarily formed after 60 minutes on the Ansys Fluent simulation and after 94 minutes during the experimental application. The duration difference of about 28 % was caused by two factors: the experiment required an additional time to fill the freezing device with liquid nitrogen; the CFD model neglects the heat loss from the freezing device and from the outer pipe wall.

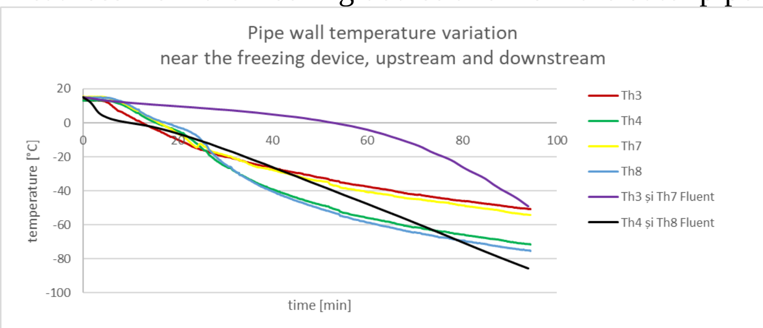

Fig. 8. Pipe wall temperature variation near the freezing device during the two tests

The ice layers length evolution is presented in figures 9 and 10.

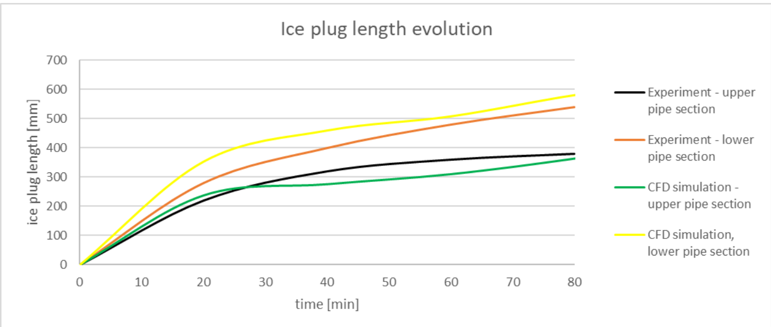

Fig. 9. Ice plug length evolution during the two tests

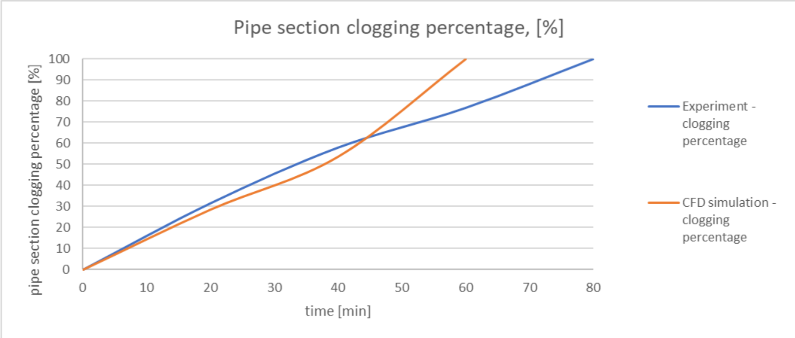

Fig. 10. Pipe section closing percentage during the two tests

4. Conclusions

The experimental results confirm the data obtained theoretically through the CFD simulation. Therefore, the method can be implemented on technology developments for horizontal pipes of larger diameters using water as the working agent.

REFERENCES

[1]. *T. Gyöngyöși, Ș. Valeca, L. Constantin*, Dispozitiv de înghețare pentru conducta Dn 200 mm. Proiect de execuție model experimental (200 mm nominal diameter pipe freezing device. Experimental model execution project), ICN Pitești, R.I. 7138/2005.
[2]. *Huntingdon Fusion Techniques*, http://www.huntingdonfusion.com. [Accessed on 5.11.2019].
[3]. *ANSYS*, ANSYS FLUENT User's Guide, Canonsburg, 2013.
[4]. *K. Raznjevic*, Tabele și diagrame termodinamice (Handbook of thermodynamic tables and charts), București: Editura Tehnică, 1978.
[5]. *Engineering Toolbox* www.engineeringtoolbox.com. [Accessed on 15.03.2020].
[6]. *Microelectronics Heat Transfer Laboratory*, University of Waterloo, www.mhtl.uwaterloo.ca/old/onlinetools. [Accessed on 7.04.2020].
[7]. *Dortmund Data Bank*. http://www.ddbst.com [Accessed on 15.04.2020].
[8]. *H. Tinoco*, Cryogenic pipe freezing − a theoretical model, Swedish State Power Board, Aelvkarleby, Vasteras, 1988.
[9]. *M. A. Miheev, I. M. Miheeva*, Principles of Heat Transfer (in Russian), Moscow: Energiya, 1977.
[10]. *V. P. Isachenko, V. A. Osipova, A. S. Sukomel*, Heat transfer, Mir Publishers, Moscow, 1980.
[11]. *B. Corbescu, T. Gyöngyöși, D. Puiu, R. A Chihaia, V. N. Panaitescu*, Vizualizare secvențială a procesului de formare a dopului de gheață în secțiunea de testare a unei conducte de diametru mare, Sequential view of the ice plug formation process in the test section of a large diameter pipe, Ştiinţă şi inginerie, ISSN 2067-7138, Editura AGIR București, 2018. (in Romanian)