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Uniform Heating and Cooling of a Hollow Disc with Internal Water Channels

Karel Adamek
Dept. of Numerical Simulations, VUTS a.s, Liberec, Czech Republic.

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ABSTRACT

The paper deals with practical problem of uniform temperature of heated or cooled disc mass. It presents the influence of thermal insulation and of inlets/outlets positioning on temperature uniformity and on total energy consumption during working cycle. Together with essential balances of mass and energy there is used the method of flow numerical simulation. Results can be used for increasing of both productivity and thermal effectivity of observed equipment.

Keywords: Heating, Cooling, Flow Numerical Simulation, Energy Balance, Effectivity

1INTRODUCTION

1.1 Assignment

The flat and hollow steel disc of diameters 900/450 and thickness of 50 mm, with two spiral channels for heating or cooling water in the disc mass creates the heated/cooled part of thermal equipment. Water is flowing as counter flow, to get the most possible uniformity of the surface temperature. The working cycle consists from 30 minutes heating from 20°C to 70°C, next 30 minutes of temperature maintaining and last 30 minutes the cooling back. The aim is the definition of mass flow of hot and cold water, of heating and cooling input positioning and of thermal insulation influence, in view of the possible shortening the cycle time.

Large tables of used material constants and results of simple calculations of mass and energy balances are not presented here. Necessary values are available in any handbook of machinery or heat exchange [1]. The heat power Q (W) in the first medium 1 passes through the surface S(m^2) into the second medium 2 as follows

$$Q = m_1 \,.cp_1 \,.\, \Delta T_1 = \alpha \,.S \,.\Delta T_s = m_2 \,.\, cp_2 \,.\, \Delta T_2$$

where

1, 2	index of medium
Q (W)	exchanged heat (energy)
m (kg/s)	mass flow
cp (J/(kg.K))	specific thermal capacity
ΔT (K)	temperature difference between inlet and outlet
α (W/(m^2.K))	heat transfer coefficient
S (m^2)	heat exchange surface
ΔTs (K)	average temperature difference between 1 and 2

Using essential formula for calculation of total energy, the main parameters of the solved case are as follows:

average input of 2200 W for the heating time of 1800 s, total consumed energy of 3.96 MJ,
average flow of heating water for 90/70°C is 0.026 kg/s,
average flow of cooling water for 6/12°C is 0.088 kg/s – lower temperature difference requires higher flow, to get required heat power.

Calculated water mass is flowing through inner channel in the disc mass, created as two spirals in the counter flow - cross section of 10x11 mm, total length of 21.5 m. It defines the flow resistance, necessary for determination of operational point of used pump (its mass flow and pressure difference), after any handbook/textbook of fluid mechanics [2]. Calculated pressure resistances are 0.4 kPa for lower flow of heating water and higher value of 3.2 kPa for higher flow of cooling water, therefore neglected in comparison with the pressure loss of the whole circuit of heating/cooling water.

Check: The average heat exchange coefficients from water into wallare 123 W/(m^2.K) for heating and 306 W/(m^2.K) for cooling, calculated from the above designed area of heat exchange and temperature differences. After [1] etc. the real values for similar cases are usually in the range 400-1000 for water under natural convection and several thousands for forced flows; therefore, the above mentioned values are satisfactory for next procedure.

1.2 Checking numerical simulation

Preliminary calculation above does not consider the heat loss in the surroundings during heating period. It means that mentioned thermal input Q (W) must be really higher, using higher water flow or higher temperature difference. On contrary, during the cooling period, such heat loss helps for shortening of the cooling time.

The method of flow numerical simulation, using any standard commercial software [3] verifies results of calculations above – it gives more exact increasing/decreasing courses of the temperature in the time (heating/cooling of the disc). The informative model defines simply the uniform mass of the disc (kg) with internal heat source (W/m^3). The Fig, 1 to Fig. 4 below present surface temperatures for several solved cases.

Perfect thermal insulation between disc surface and surroundings (Fig. 1)
After 30 minutes of heating the constant surface temperature of 341 K = 68°C corresponds practically with the needed temperature of 70°C - see the Fig. 1. It should be to increase slightly the input theoretically in the ratio of temperatures difference at 2212 W – maybe the consequence of any rounding error.

All surfaces are not insulated, higher heat transfer (Fig. 2).
At all surfaces is defined the real higher heat transfer coefficient between wall and the stagnant air surroundings of α = 15 W/(m^2.K). The temperature of both frontal surfaces is 332.5 K = 59.5°C. To get needed temperature of 70°C it should be to increase the time of heating or of input of 16% approx.
Note: Higher water flow (kg/s) means higher flow velocity (m/s), also higher heat transfer coefficient α (W/(m^2.K)), typically as

$$\alpha = A + B\,.w$$

after [1] and also higher transferred heat power (W). Values of heat transfer coefficient for natural air flow along wall are typically in the range 7-15 W/(m^2.K) [1].

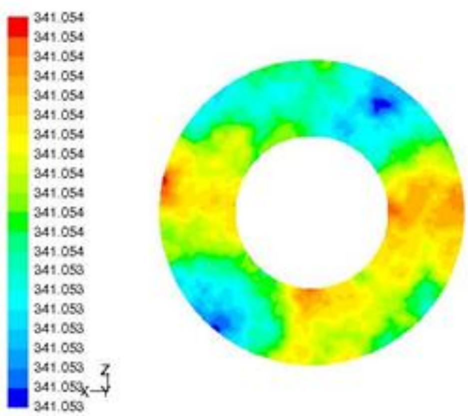

Fig. 1: Temperature – perfect thermal insulation

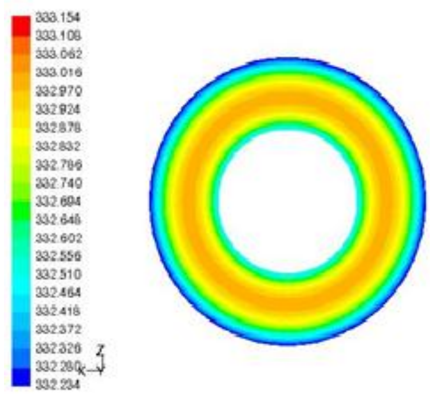

Fig. 2: Temperature with heat transfer coefficient of α = 15 W/(m^2.K).

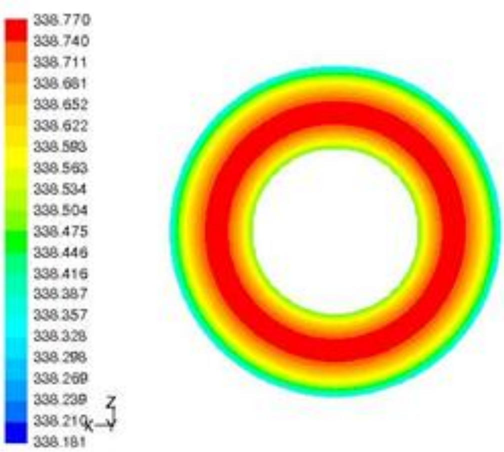

Fig. 3: Temperature of insulated bottom surface

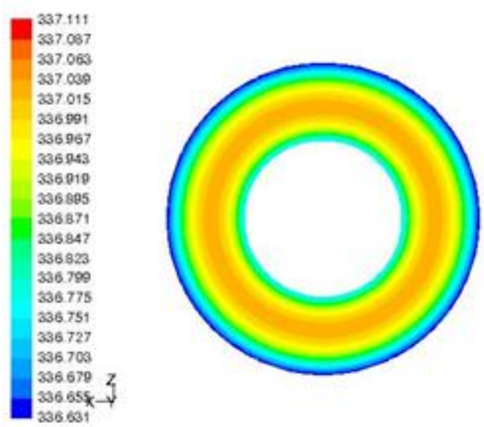

Fig. 4: Temperature of not insulated upper surface

Only the bottom surface is thermally insulated (Fig. 3 and Fig. 4)

On the upper and side surfaces the real lower heat transfer coefficient of $\alpha = 7$ W/(m^2.K) is defined between wall and stagnant surroundings, the bottom surface (base) is thermally insulated. The Fig. 3 presents the temperature of the bottom surface after 30 minutes of heating, 338.5 K = 65.5°C, approx. The upper surface in the Fig. 4 is a little cooler, 337 K = 64°C approx. To get needed temperature it should be slightly increase the time of heating or the input of 10% approx.

All surfaces are not insulated, lower heat transfer coefficient
At all surfaces is defined the heat transfer coefficient of $\alpha = 7$ W/(m^2.K) from disc surface into the stagnant surroundings. The temperature of both frontal surfaces of 337 K = 64°C is identical with the Fig. 4, not presented again. Compared with the Fig. 2 the temperature is higher, because lower heat transfer coefficient means better thermal insulation of observed surface.

In general, along peripheral surfaces the temperature is logically lower, due to higher heat loss in surroundings and lower power input from channels.

Note: In not insulated model can be neglected the temperature differences of some tenths of K, only, between the disc center and periphery. However, not insulated surface means higher energy consumption.

2 NUMERICAL MODEL

The more detailed analysis of heating/cooling of disc mass by flowing water allows the more precise model after the Fig. 5. Very large table of used dimensions, material constants, thermal and hydraulic parameters [1, 2] is cut out. Using demineralized water for heating/cooling, the stainless steel of different thermal parameters (lower heat conduction coefficient) must be used instead standard steel, therefore the heat energy consumption is of 16% approx. higher than above [3].

2.1 Model description

This model corresponds to the reality. The aim is to simulate the temperature courses in time (disc surface, outputs of heating/cooling water).

Geometry: The width of spiral channels is 10 mm, the symmetry plane splits in half the channel height of 11 mm. In radial direction, the disc mass thickness of 10 mm is the same as adjoining channels of 10 mm in width. The layout of water channels are two spirals for counter flow of water, see the Fig. 5. Due to better meshing, the system of half coils is used. The main volume of the disc continues in frontal direction. For simplification, the air surroundings is not modelled, the heat loss is defined here as the boundary condition of constant coefficient of heat transfer on the disc surface.

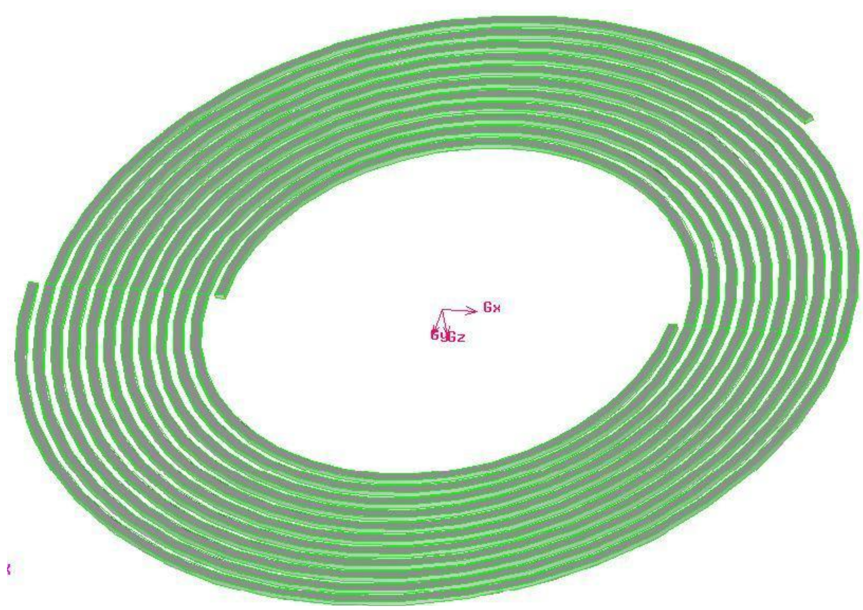

Fig. 5: Layout of two spiral water channels

Mesh: Because this is the case with heat transfer, the mesh elements should be both rectangular and small as possible. On the transition water-wall are created boundary layers of fine meshing. The base is the mesh on the bottom of individual coils and draws in channels in one side and in the disc mass on the other side. It is all together 800 thousands of element approx.

Initial and boundary conditions: Inlets simply at ends of spiral channels, without attached supply channels. In inlet cross sections are defined water velocity and temperature after the assignment in the Par. 1.1.
On the central surface of the half model there is defined the symmetry plane.

The adjoining air surroundings is not modelled, instead it, is used the recommended value of the heat transfer coefficient for spontaneous flowing as 7 and 15 $W/(m^2.K)$ respectively on the disc outer surface. For checking model is used the simulation with full thermal insulation, with zero heat transfer.

Solver: Used unsteady solver. At the start of simulation of heating period, the disc is cold (20°C) and water is warm (90°C) in the whole volume of the disc and water respectively. Analogically, at the start of the cooling period the disc is warm (70°C) and the water is cold (6°C).During the simulation are recorded courses of temperatures in time for outlet water and outer surface of the disc.

Turbulence model: For lower flow of water (Re = 870) the laminar model is used, for higher flow of water (Re = 2900) the turbulent models k-ε or k-ω are tested and used, too.

2.2 Results - heating

Typical thermal and velocity fields for turbulent model present next figures. For various values of heat transfer coefficient the character of fields is similar, therefore one of solved cases is presented, only.

a) in symmetry plane b) in disc frontal face

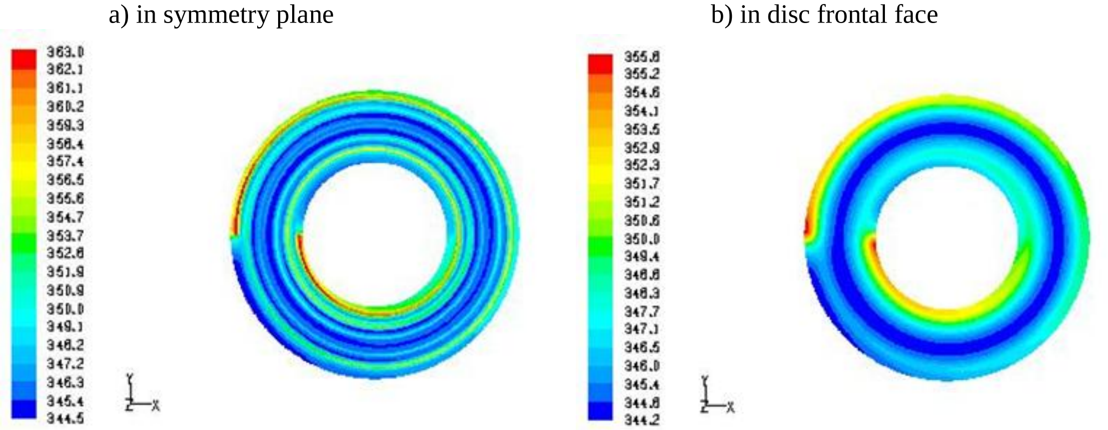

Fig. 6: Temperature fields - heating

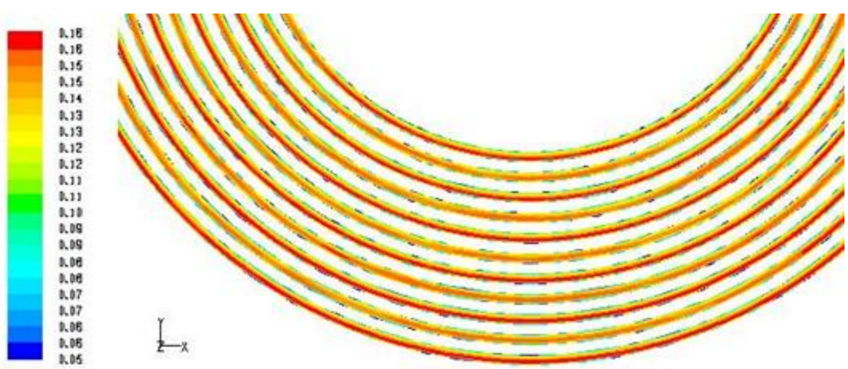

Fig. 7: Detail of velocity field in channels

The Fig. 6a presents temperature field on the symmetry plane of the model (in the middle of the disc volume). There are visible temperatures of individual channels and of mass between them. The inlet of warm water, situated at the disc periphery, means local increasing of heat loss. It should be suitable to insulate the disc and inlets to situate near to the middle of disc body.

Next Fig. 6b presents the temperature field at outer frontal plane of disc. It is analogous with previous figure, fine details are suppressed by radial heat conduction in thick material.

The Fig. 7 presents the detail of velocity field in channels. Due to relative large mesh elements in relative narrow channels the details are not well visible.

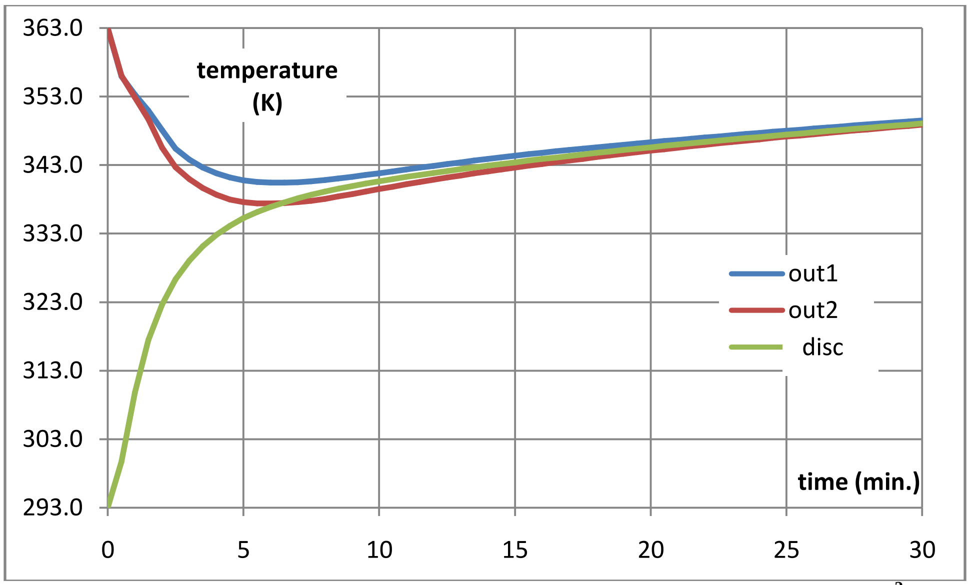

Fig. 8: Temperature course in time for thermally insulated surfaces, 0 W/(m^2.K)

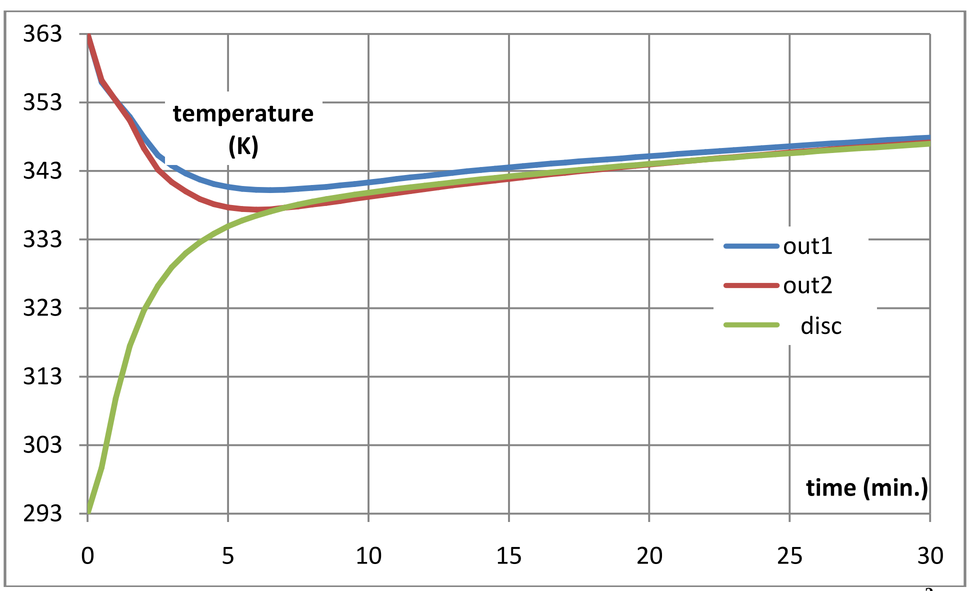

Fig. 9: Temperature course in time for surface heat transfer coefficient of 7 W/(m^2.K)

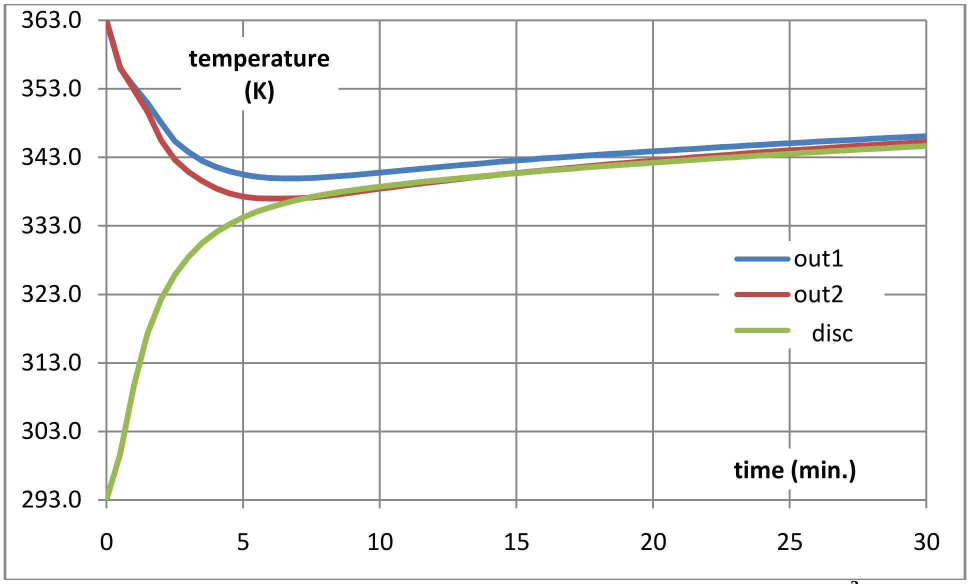

Fig. 10: Temperature course in time for surface heat transfer coefficient of 15 W/(m^2.K), turbulent flow

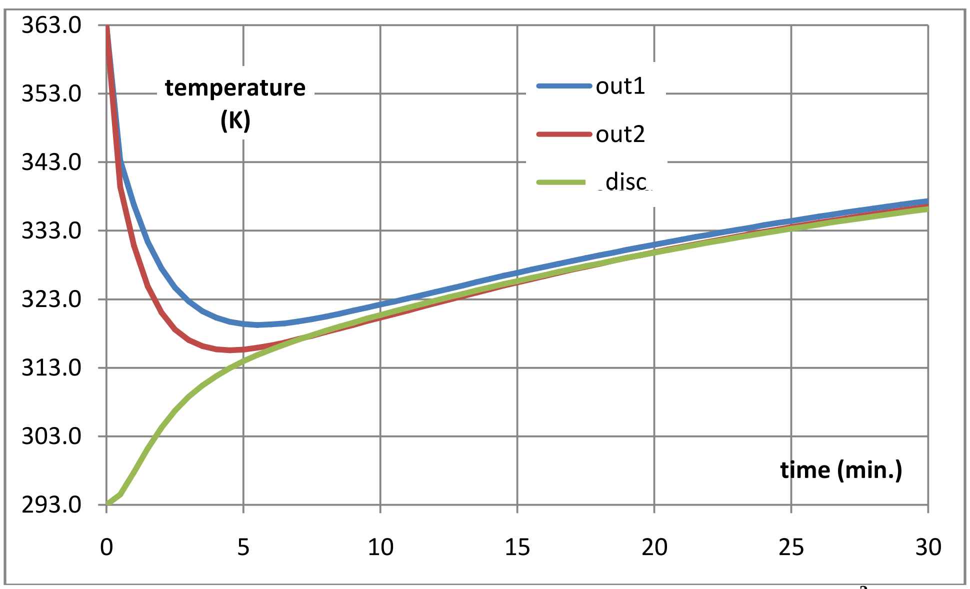

Fig. 11: Temperature course in time for surface heat transfer coefficient of 15 W/(m^2.K), laminar flow

Graphs in Fig. 8 to Fig. 10 present courses of main temperatures in time (water outlets of spiralsout1 and out2**,** together with frontal face of disc), for different values of heat transfer coefficient on the disc surface of 0 - 7 - 15 W/(m^2.K) and as turbulent model.

At the start of simulation (time = 0) the disc is cold (20°C) and the inlet water is warm (90°C). At the beginning are visible intense changes of both temperatures - water temperature decreasing and disc temperature increasing. During the time of 7 minutes, both temperatures become very close. Subsequently, due to the continuing warm water supply, the water temperature slightly increases again and the disc heating continues. Generally, it is visible that the required average temperature of disc surface of 70°C (343 K) is reached after the time of 13 – 17 – 22 minutes, depending on defined value of heat transfer coefficient. Therefore, the requirement of heating time of 30 minutes is fulfilled reliably.

Outlet water temperatures in individual spirals are slightly different; probably it is the consequence of slightly different both total lengths and positions towards disc periphery. Therefore some slightly different mesh could have an effect, too.

Check: For the same case with laminar flow (Re = 870) the needed final temperature is not reached in the needed time. The reason is the low coefficient of heat transfer for laminar flow, the elementary mixing in the flow does not exist. Maybe, the turbulent flowing is really created by flow disturbing at the inlet in the spiral entry.

2.3 Results - cooling

For cooling water temperature of 6°/12°C and starting temperature of the disc mass of 70°C analogous simulations of the disc cooling are carried out for two values of heat transfer coefficient of 0 and 15 W/(m^2.K), only. For lower temperature difference between inlet and outlet, the flow of cooling water is higher than in the heating phase above. Higher velocity means higher value of heat exchange coefficient in general [1]; therefore, the cooling time back on the starting temperature is shorter than the warming time. Also, the thermal loss here helps to the disc cooling, in contrary to cases of heating.

For illustration, the Fig. 12 presents the temperature field in the central symmetry plane and on the frontal disc face. Inlets of both channels are locally cooler, peripheral areas are locally overheated, due to local concentration of mass to be cooled. The character of temperature fields is identical for different values of heat transfer coefficient, therefore without next details.

a) in symmetry plane

b) in frontal face of the disc

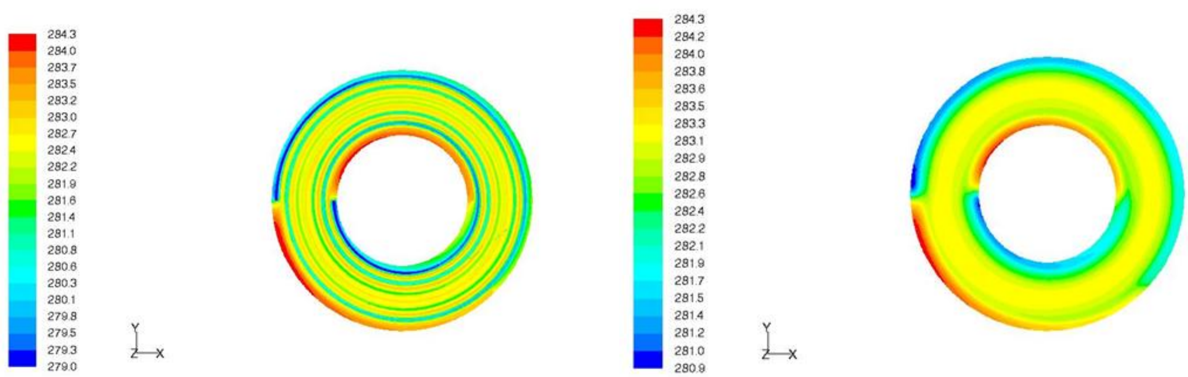

Fig. 12: Temperature fields – cooling

Graphs in Fig. 13 and Fig. 14 present courses of main temperatures in time - water outlets out1 and out2 and frontal face of disc, for two values of heat loss in the surroundings, defined as 0 and 15 W/(m^2.K) . At the start of simulation (time = 0) the disc is warm (70°C) and the inlet water is cold (6°C). At the beginning is visible intense changes of both temperatures - water temperature increasing and disc temperature is strongly decreasing. During the short time of 7 minutes the disc mass temperature decreases at the needed value of 20°C. Influence of different heat transfer coefficients on cooling intensity is hardly visible, therefore it could be said that the main influence is here the heat transfer from the disc mass into the flow of cooling water.

Next Fig. 15 presents the influence of half velocity of cooling water. The character of observed temperature courses is similar, but the cooling on the final temperature of 20°C needs more time, 15-16 minutes approx., because at lower flow velocity, the heat transfer coefficient is lower, too.

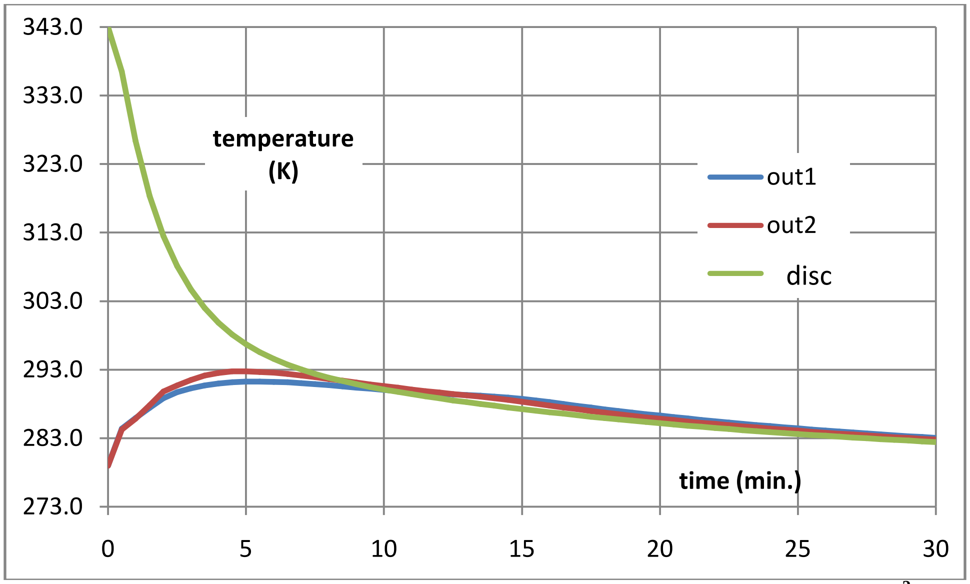

Fig. 13: Temperature course in time for thermally insulated surfaces (0 W/(m^2.K)

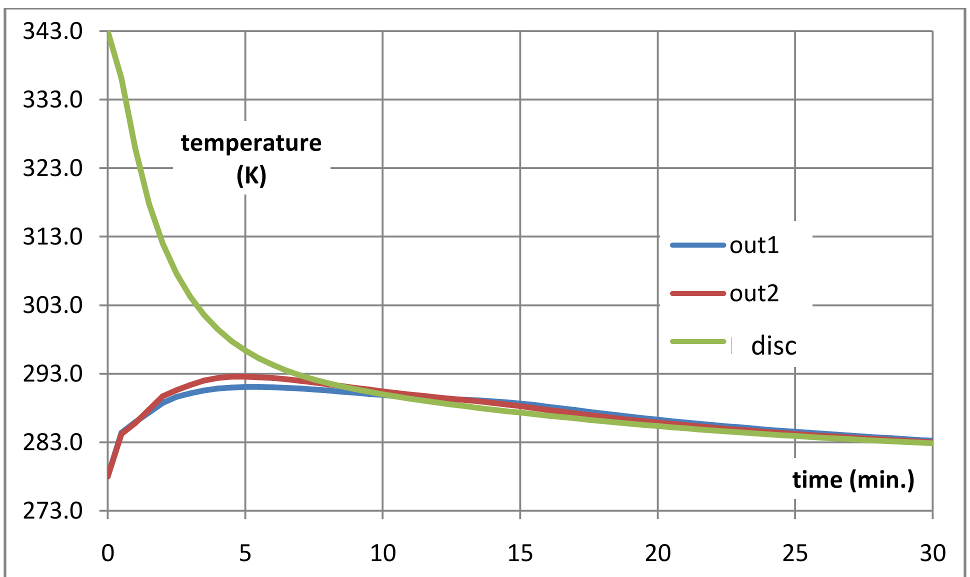

Fig. 14: Temperature course in time for surface heat transfer coefficient of 15 W/(m^2.K)

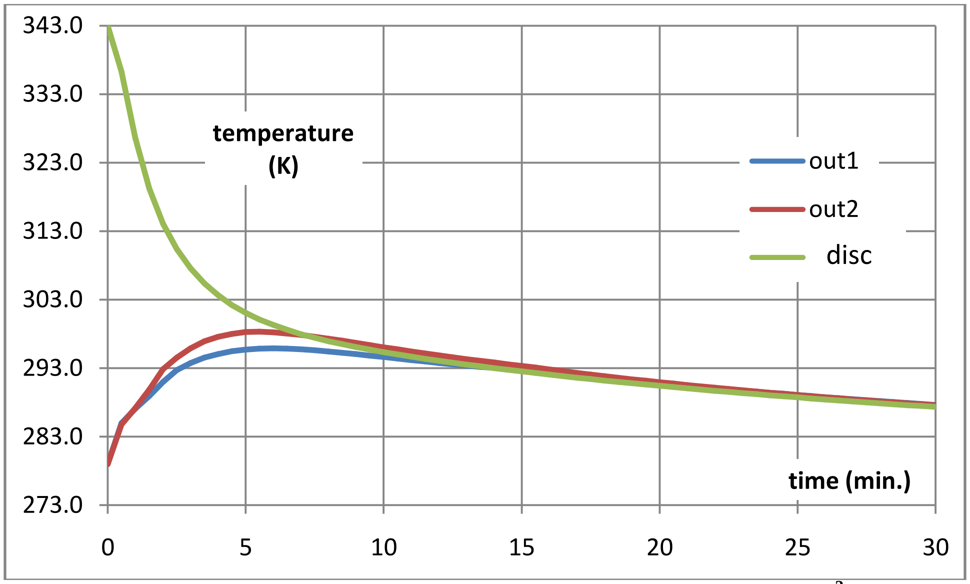

Fig. 15: Temperature course in time for surface heat transfer coefficient of 15 W/(m^2.K), half flow velocity

3 CONCLUSION

Table 1 presents values of average temperatures of cross section and outputs at the end, after 30 minutes of heating or cooling of disc mass.

Heating – with increased heat transfer coefficient (increased thermal loss) the final temperatures of water and disc are slightly decreasing and also the time of heating at needed final temperature. The power input, necessary for heating and for maintaining of reached temperature, increases, too. On contrary, during the cooling, the thermal loss shorts the time of cooling.

Table 1: Summary results of disc heating/cooling

case	mass flow	heat transf. coeff.	water out1	water out2	disc surf.	thermal output	time
	g/s	W/(m^2.K)	°C	°C	°C	W	Minutes.
Heat.	13,73	0	74,4	75,8	76,0	812	13
		7	74,1	74,8	74,0	908	17
		15	73,0	72,2	71,6	1007	22
Heat lam.	13,73	15	64,2	63,4	63,1	1516	>30
Cool.	45,02	0	10,0	9,7	9,3	761	7
		15	10,1	9,9	9,7	792	7
	22,50	15	14,5	14,5	14,2	810	15

Recommendation for energy saving:

- It should be suitable to insulate all disc surfaces during the heating period and peripheral surfaces during maintaining period. The heat loss in surroundings shorts the time of cooling period.
- Water inlets should be located away from the disc periphery – high inlet temperature just at the disc periphery means higher heat loss in the surroundings and lower heat conduction in the inner mass of the disc.
- Estimating the total efficiency of observed equipment, it is necessary to take into account that one unit of cold costs 3 times more approx. than one unit of heat.
- Different used models (boundary layers, turbulence models, meshing quality etc.) can give different results, the checking measuring is necessary.

- Higher water flow kg/s means higher flow velocity m/s, also higher heat transfer coefficient W/(m^2.K)and higher transferred heat power W, too.

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[3] Software Fluent in Actual Version