Technical University "Gheorghe Asachi" of Iasi

Faculty of Civil Engineering and Building Services

HEAT TRANSFER
INSIDE THE COOLING CIRCUIT OF A
POWER TRANSFORMER

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Summary

- Objectives
- Description of thermoconvective model
- Steady state
- Experimental simulations
- Unsteady state

Numerical part

- Optimal spacing between vertical flat plates
- Selection criteria for distinguishing different convection regimes

Partie fondamentale

- Conclusions

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Objectives

- *Analysing* the fluid flow and heat transfer in steady and unsteady state
- *Optimising* the heat transfer and reducing the temperature of hot points in steady state
- *Validation* the assumptions made in the modeling of experimental tests
- *Calculate the optimal spacing* of a channel between two vertical flat plates in mixed convection
- *Study* different criteria to distinguish different convective regimes

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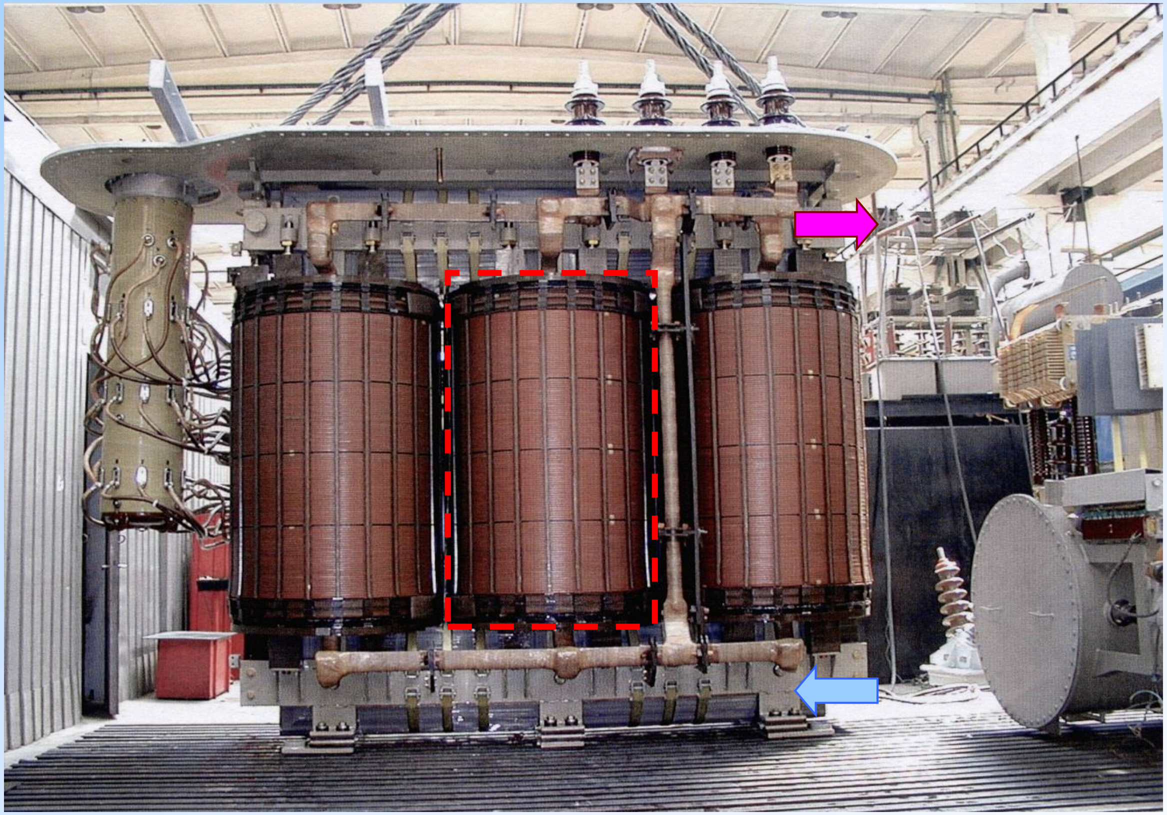
Description of electrical power transformer

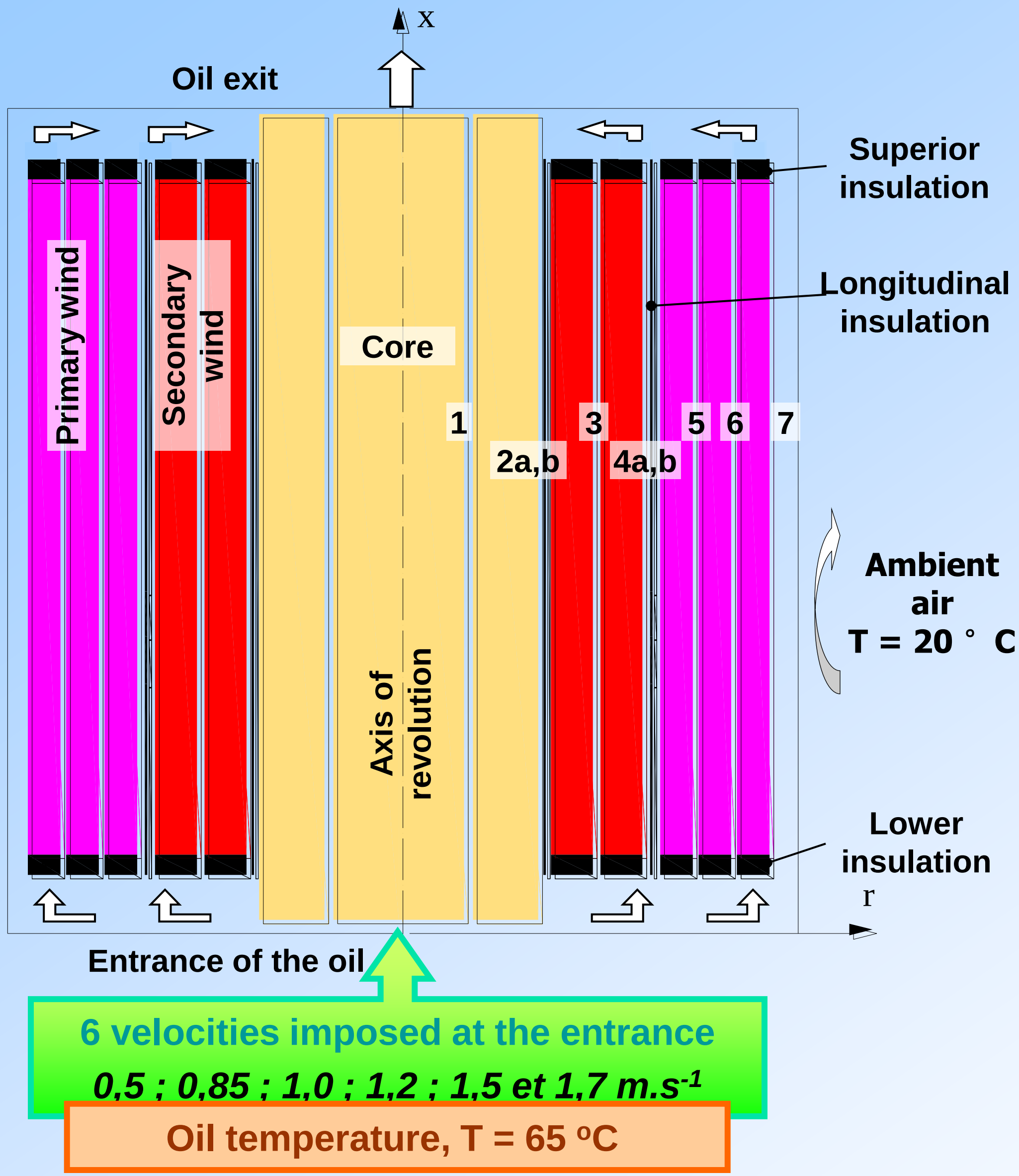
Three-phase electric power transformer of 40 MVA made by

S.C. ELECTROPUTERE CRAIOVA



- **Apparent power: S = 40 MVA ;**
- **Primary and secondary rated voltage: U_1 = 110 kV, U_2 = 20,5 kV**
- **Primary and secondary rated current: I_1 = 209,95 A, I_2 = 650,41 A**

Geometry of the studied model



Assumptions

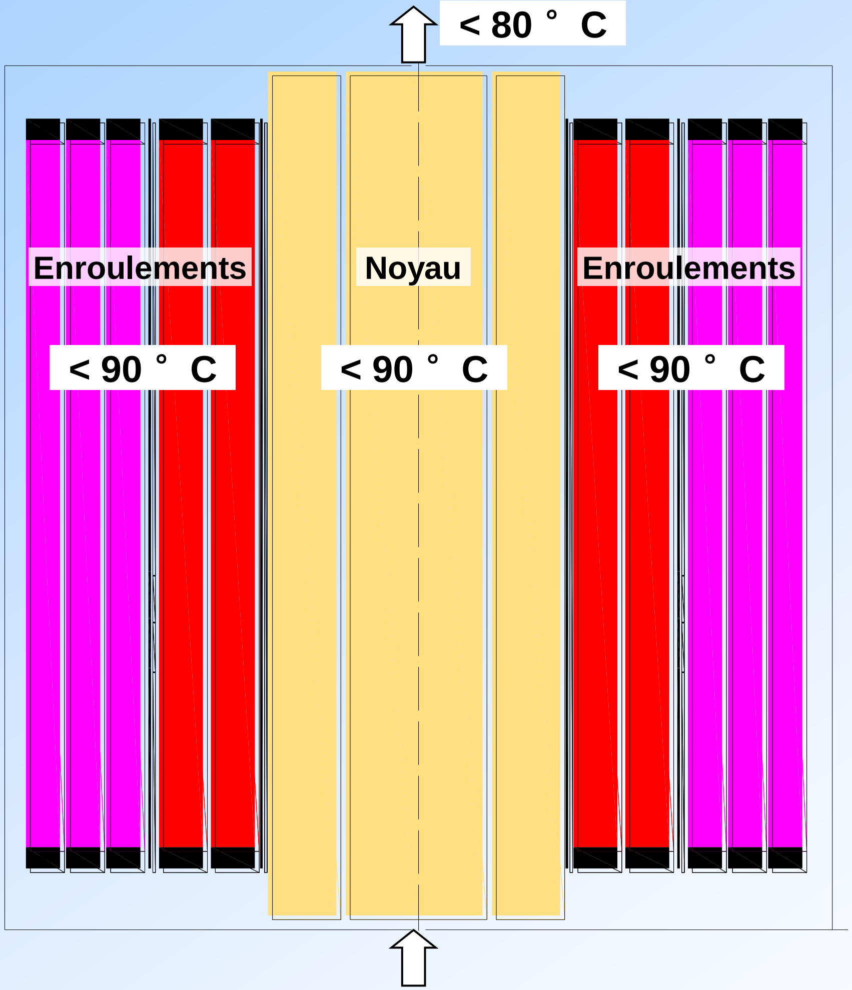
- **2D-axisymmetric problem**
- ***Mixed convection in steady and unsteady state***
- **Ascending and laminar flow**
- ***Uniform velocities*** **imposed on the input of the transformer**
- **Convective exchange with ambient air**
- ***Flux densities*** **or** ***volumic source*** **imposed uniform**
- **Dependent thermophysical properties of the oil temperature**

Limits of heating

Steady state

Hot point temperature

< 98 ° C



Transient state

Hot point temprature

< 140 ° C

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Solved equations

✓ **Continuity equation :**

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho U)}{\partial x}+\frac{1}{r}\frac{\partial(r\rho V)}{\partial r}=0$$

✓ **Equations of momentum:**

$$\frac{\partial(\rho U)}{\partial t}+U\frac{\partial(\rho U)}{\partial x}+V\frac{\partial(\rho U)}{\partial r}=\rho g_x-\frac{\partial p}{\partial r}+\frac{\partial}{\partial x}\left(\mu\frac{\partial U}{\partial x}\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(r\mu\frac{\partial U}{\partial r}\right)$$

$$\frac{\partial(\rho V)}{\partial t}+U\frac{\partial(\rho V)}{\partial x}+V\frac{\partial(\rho V)}{\partial r}=-\frac{\partial p}{\partial r}+\frac{\partial}{\partial x}\left(\mu\frac{\partial V}{\partial x}\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(r\mu\frac{\partial V}{\partial r}\right)$$

✓ **Energy equation:**

$$\frac{\partial(\rho T)}{\partial t}+U\frac{\partial(C_p\rho T)}{\partial x}+V\frac{\partial(C_p\rho T)}{\partial r}=\frac{\partial}{\partial x}\left(\lambda\frac{\partial T}{\partial x}\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(r\lambda\frac{\partial T}{\partial r}\right)$$

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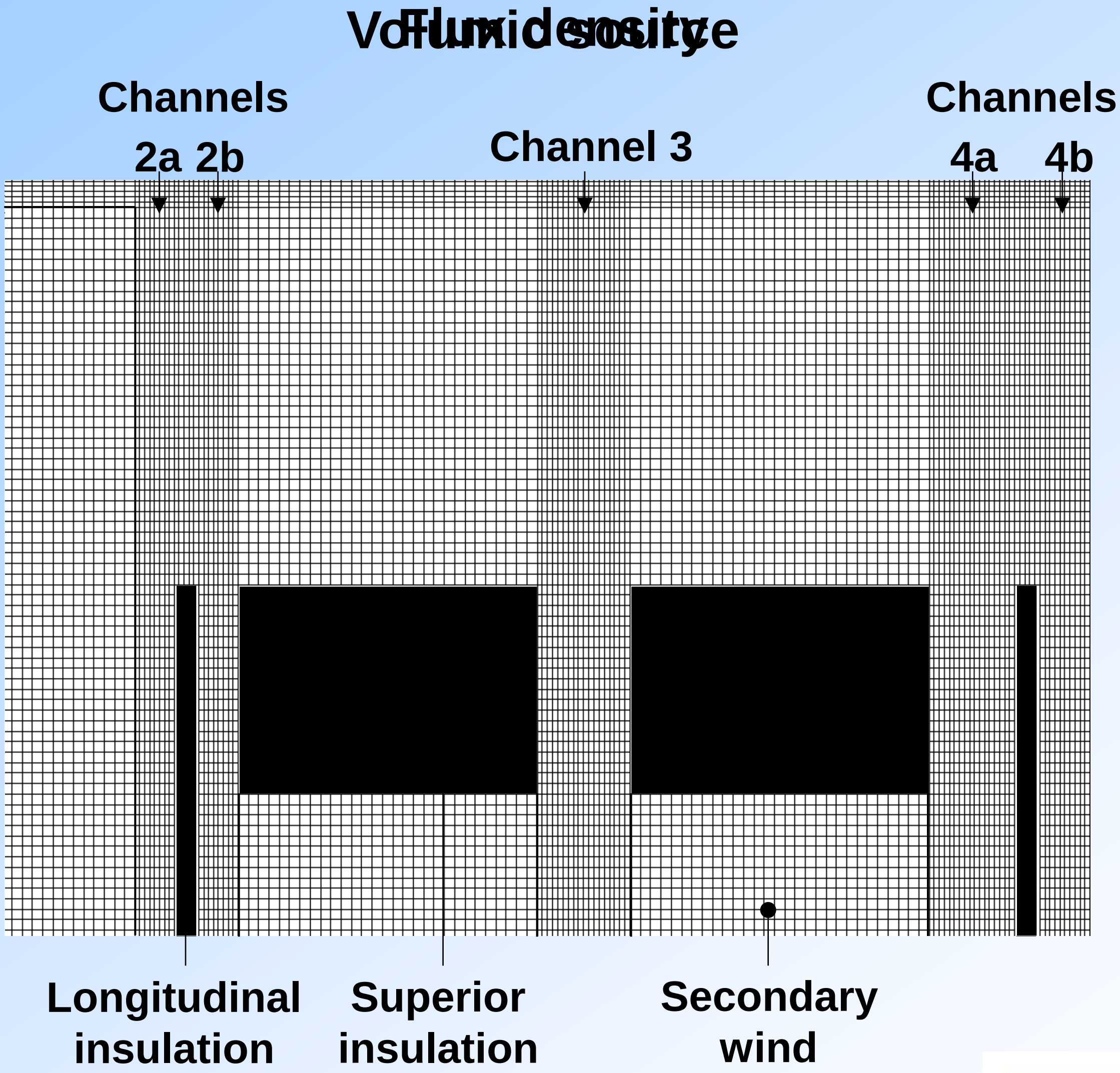
Discretization of the computational domain



Non-uniform structured grid (781 × 418 cells)

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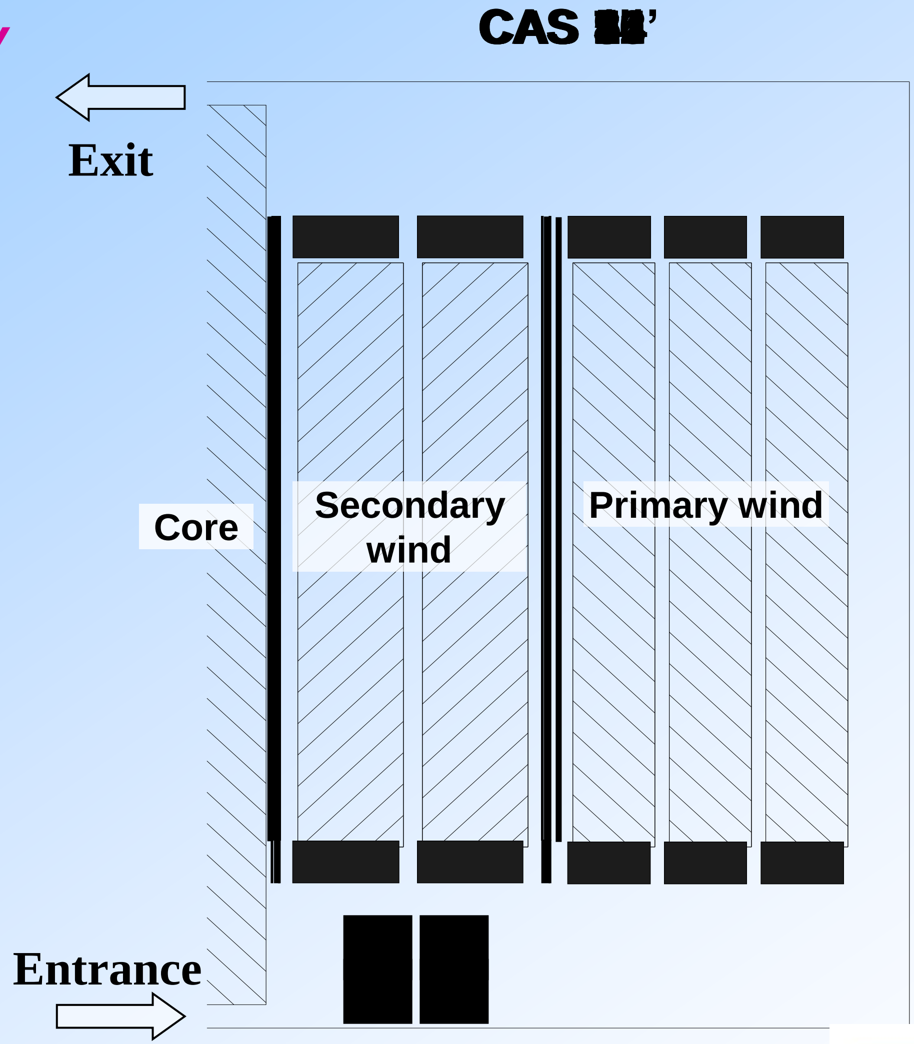
Steady state

uniform flux density ***imposed on the surfaces of the active parts***



Uniform source **imposed within the active parts**

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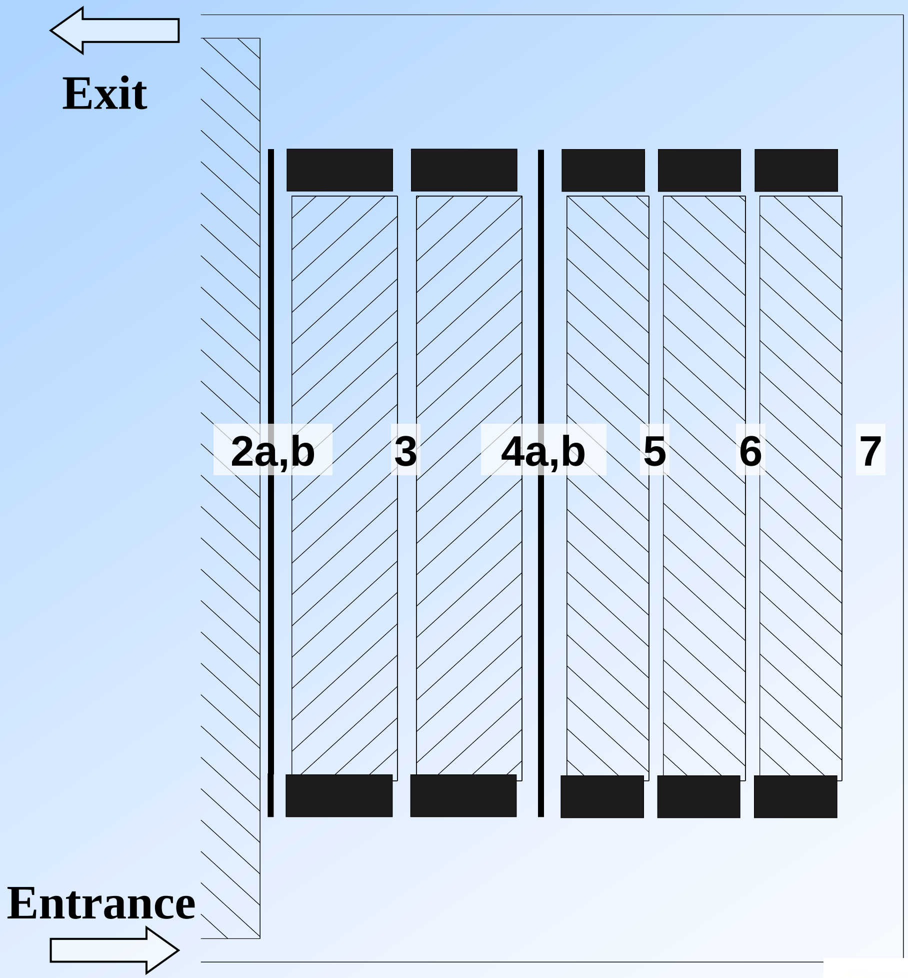
CASE 5



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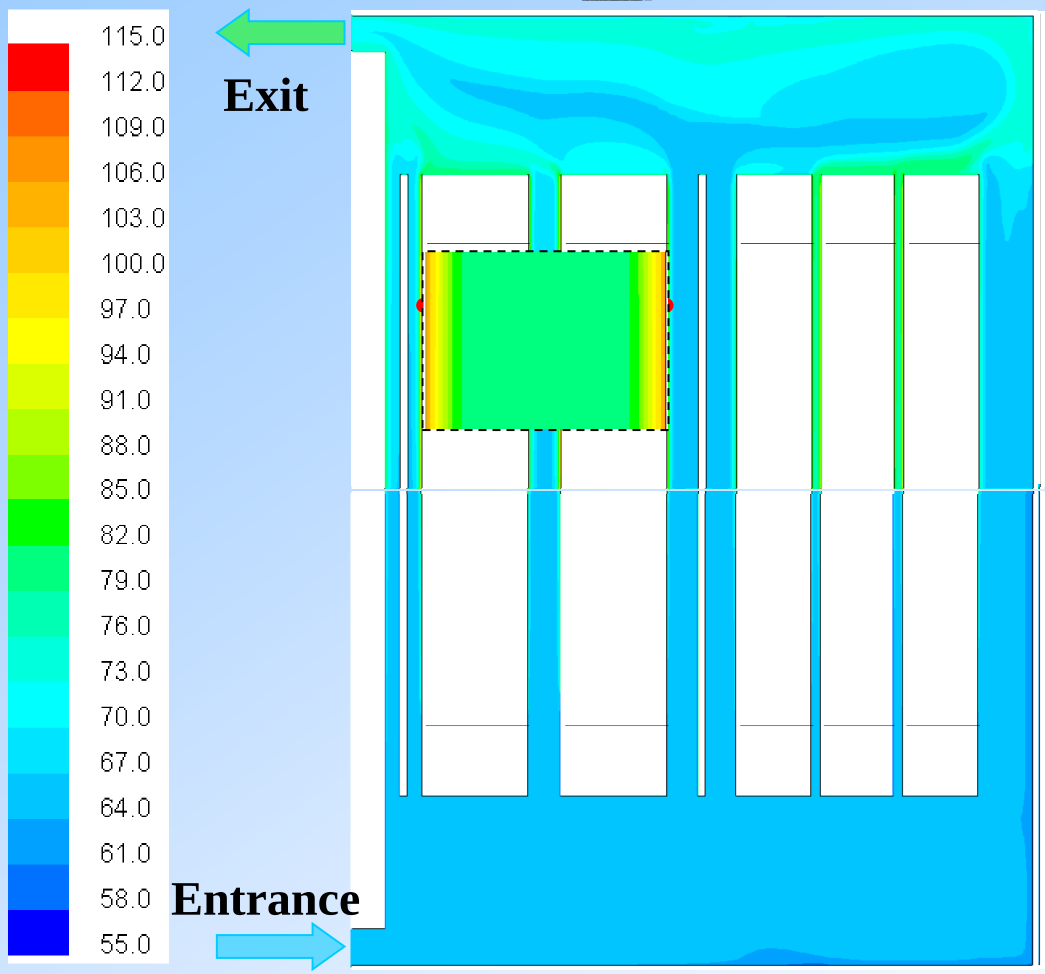
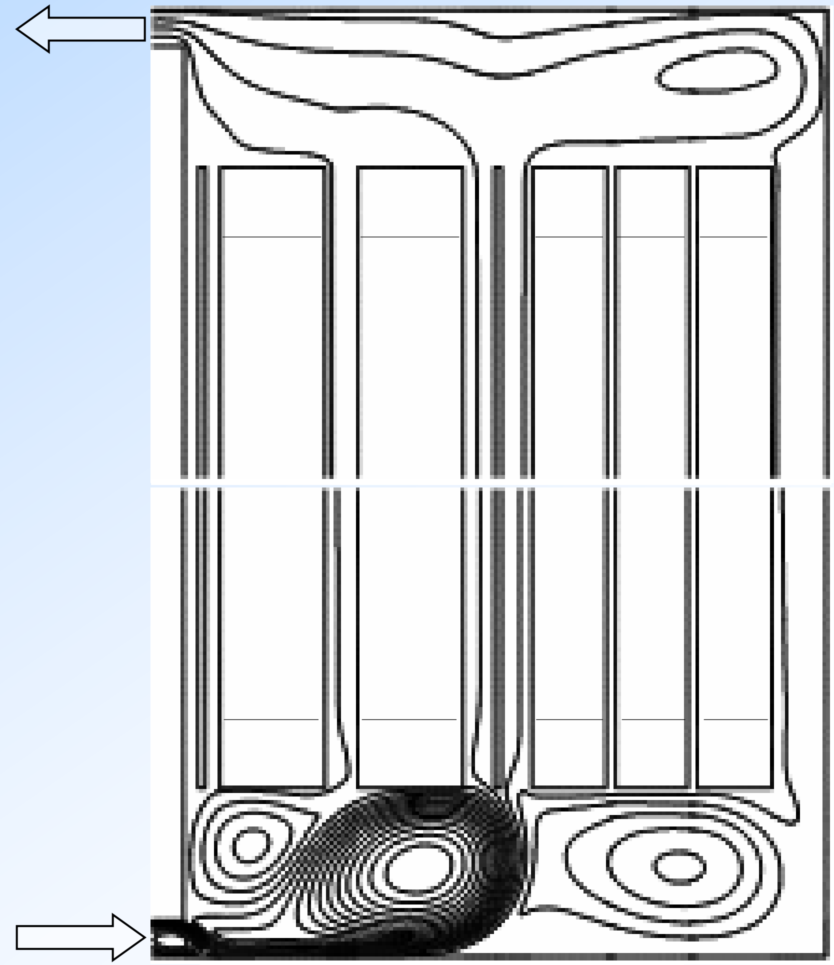
Temperature field and *streamlines*

T_{max} = [illegible] °C



U = [illegible] $m.s^{-1}$



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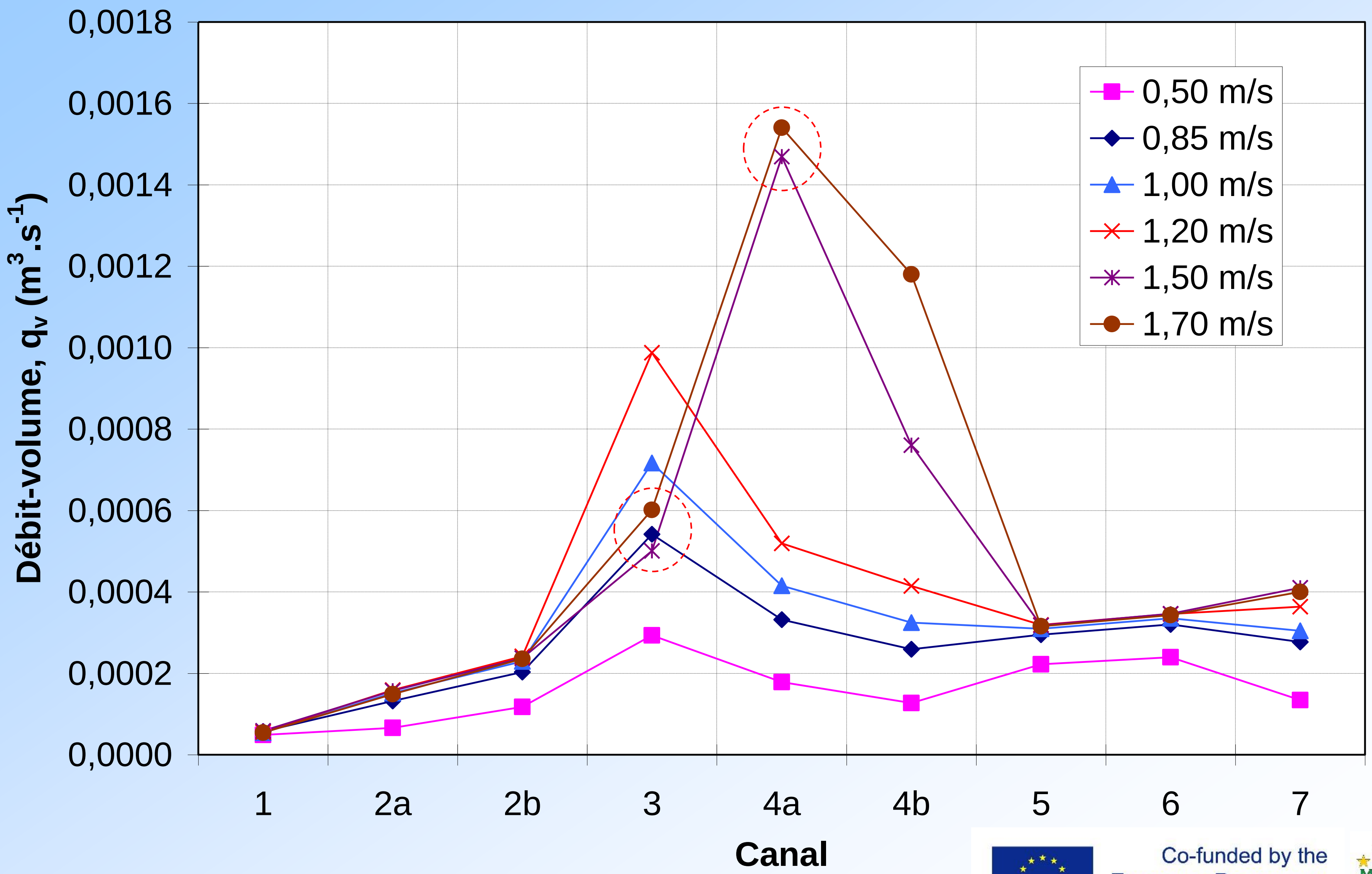
Flow volume in each channel



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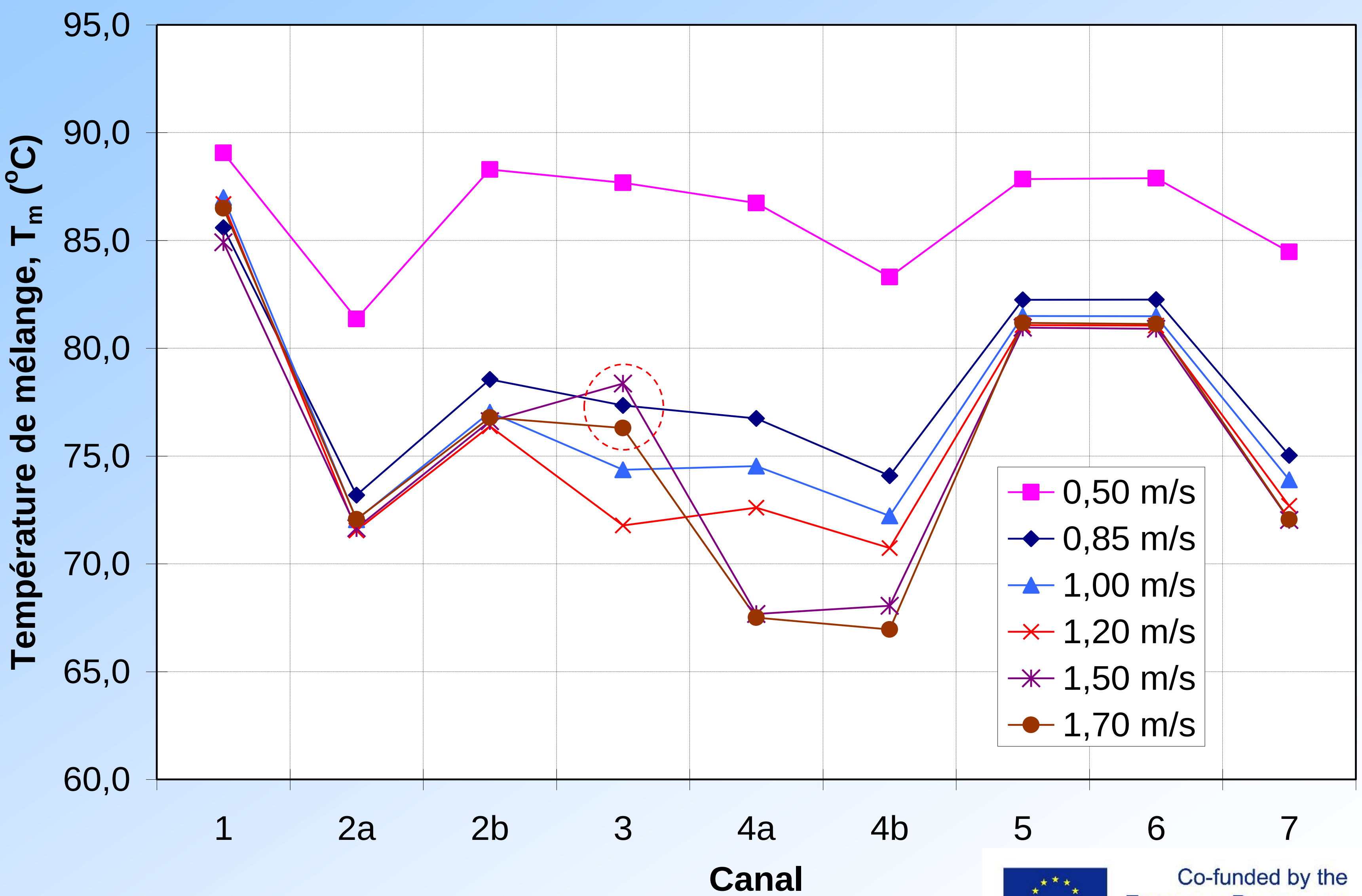
Mixing temperature at the outlet of each channel



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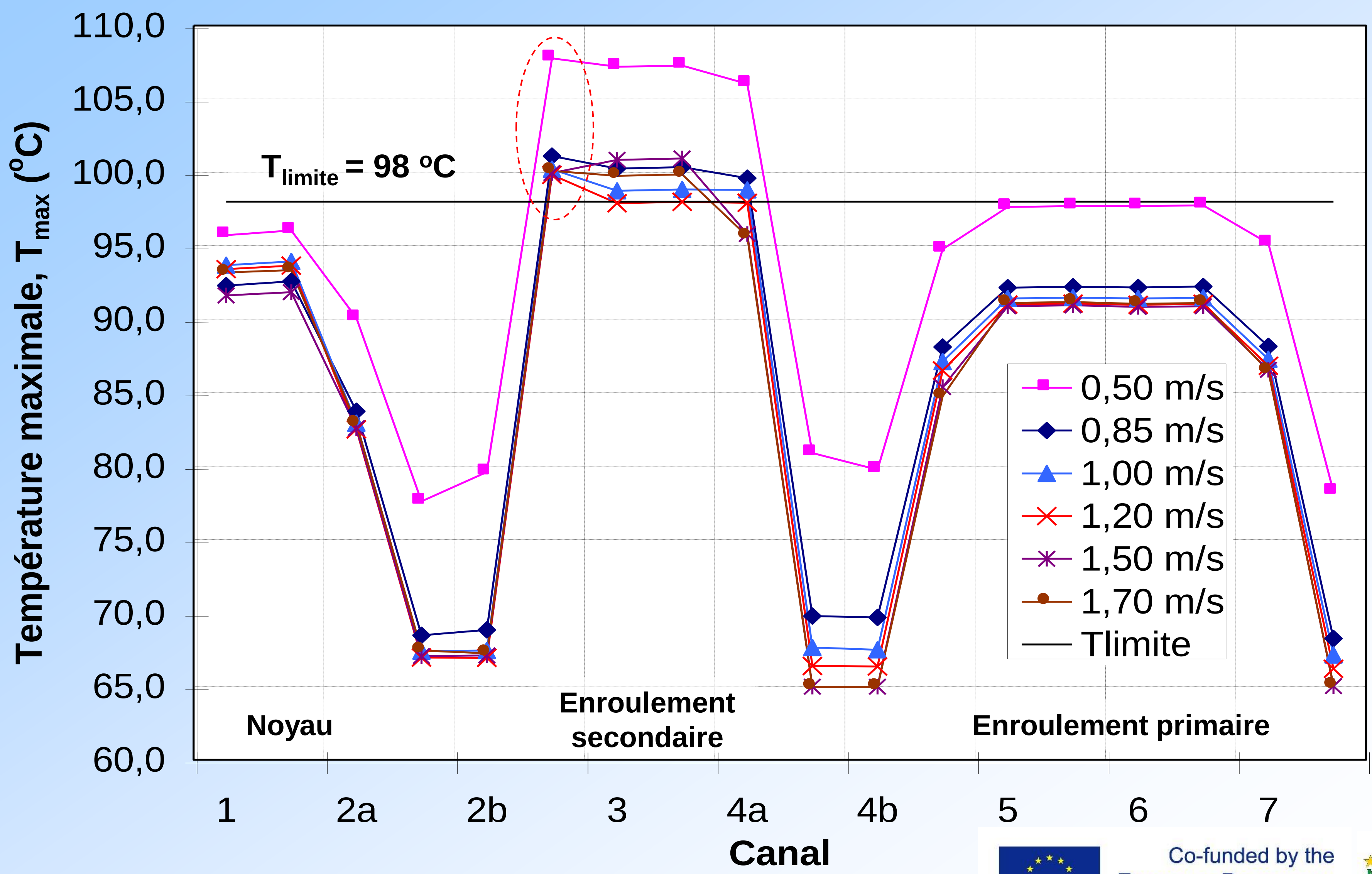
Maximum temperature of the walls of each channel



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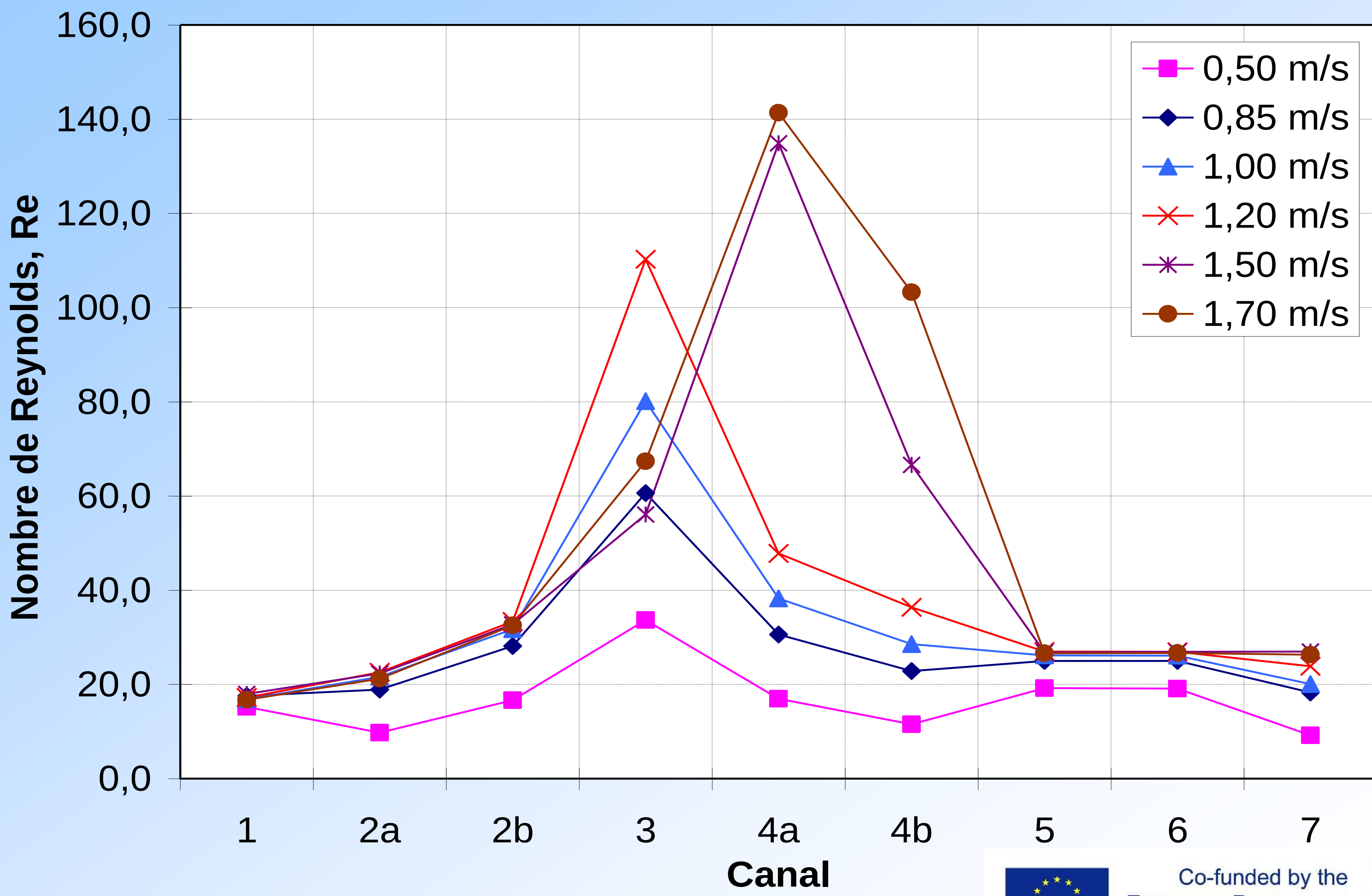
Reynolds number in the middle section of each channel



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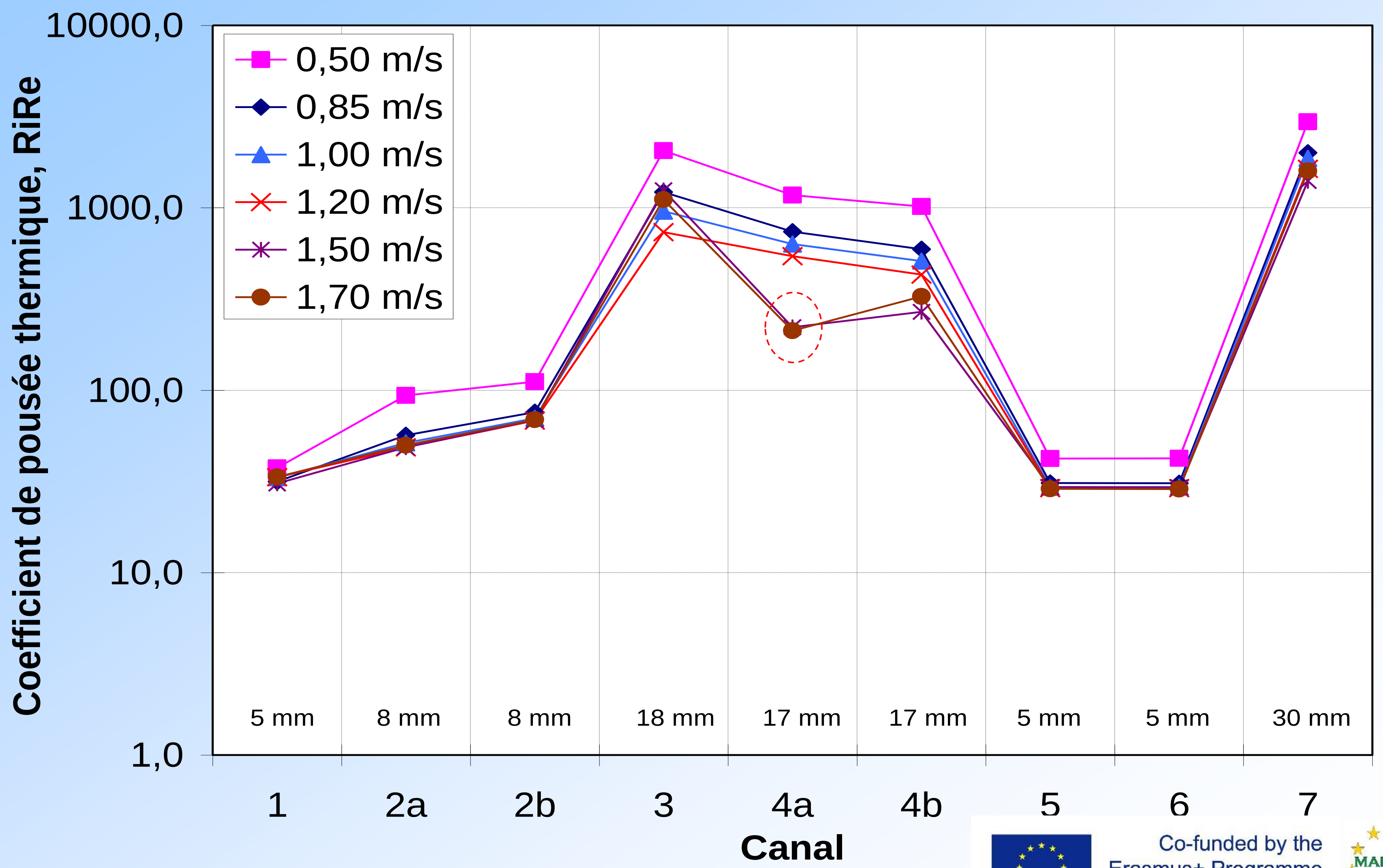
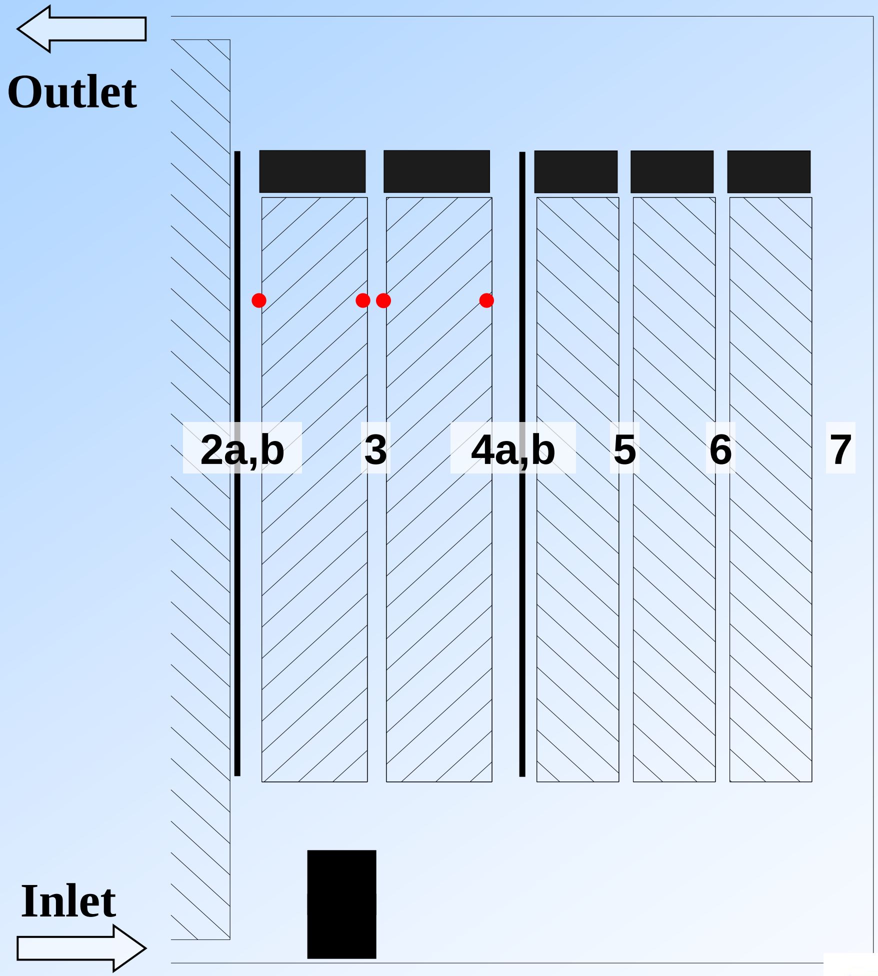
Thermal pressure coefficient in the middle of each channel



CASE 14



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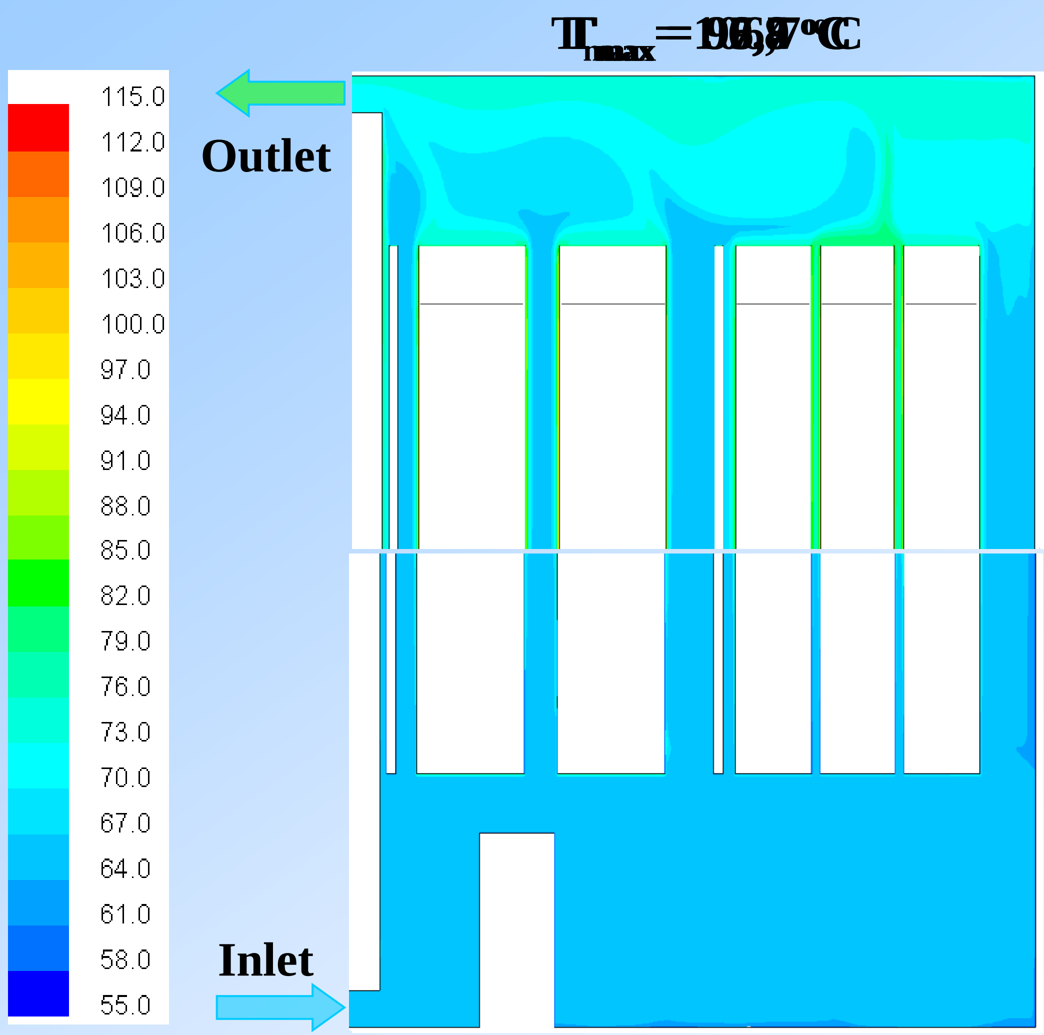
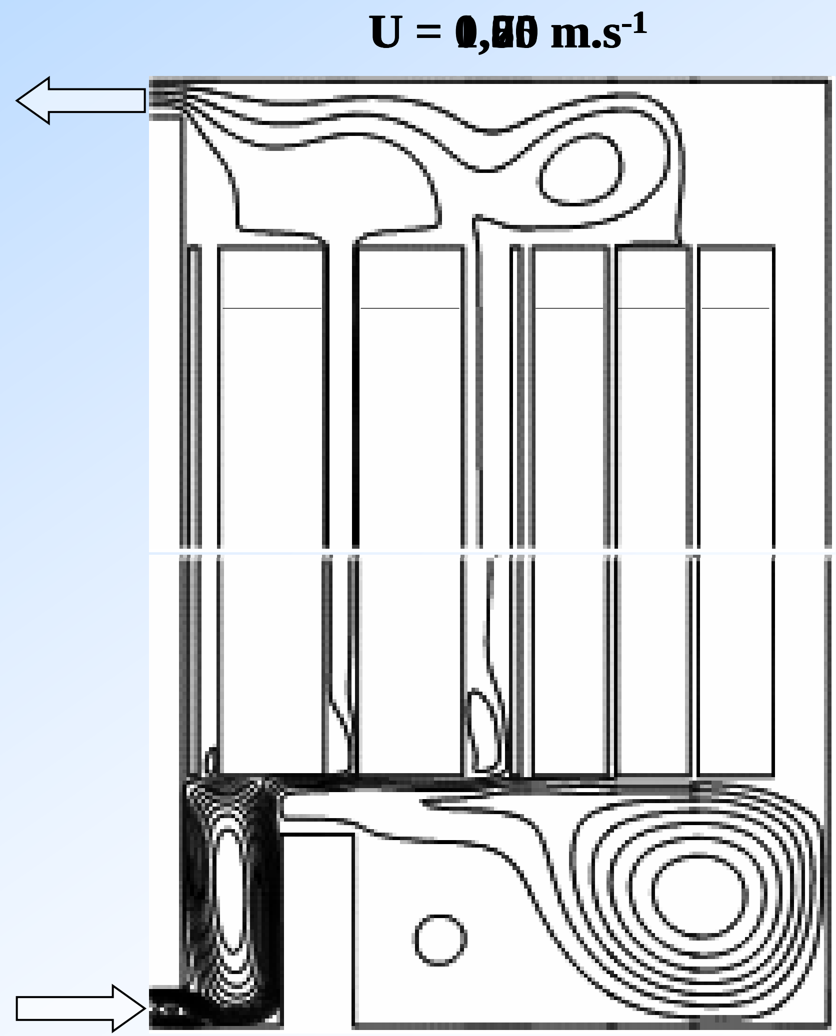
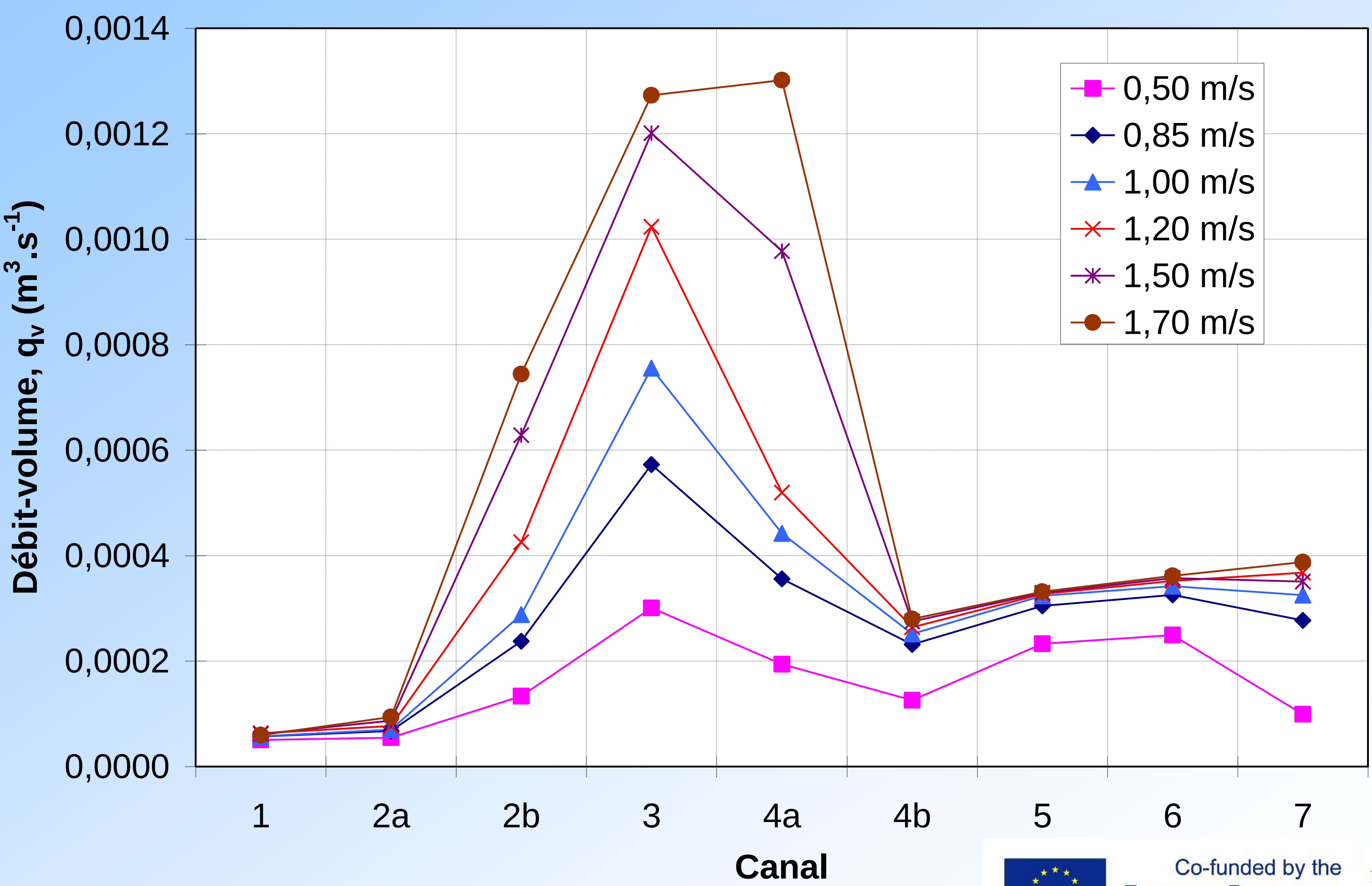
Temperature field and streamlines





Volume flow rate inside each channel



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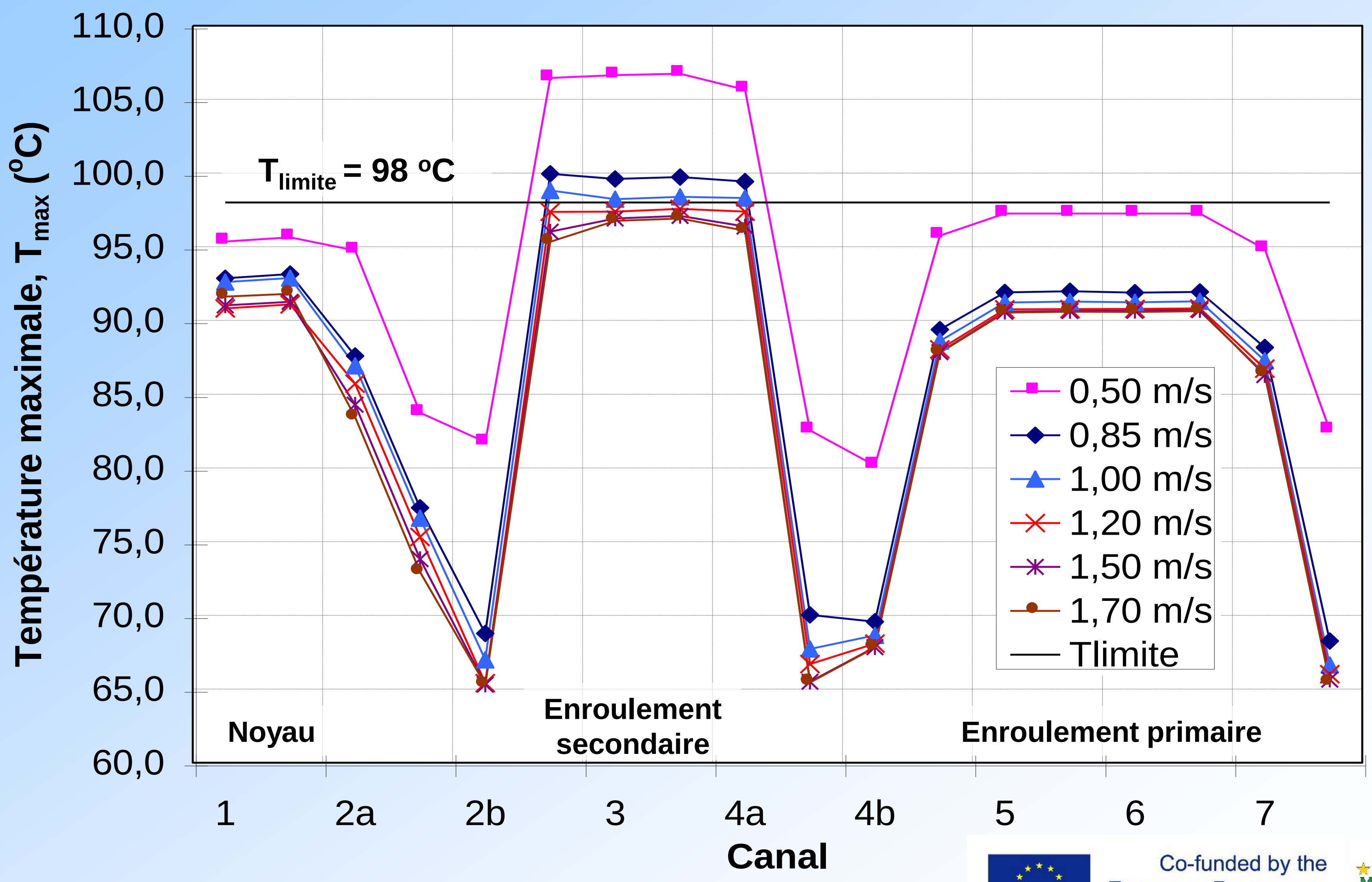
Maximum temperature on the walls of each channel



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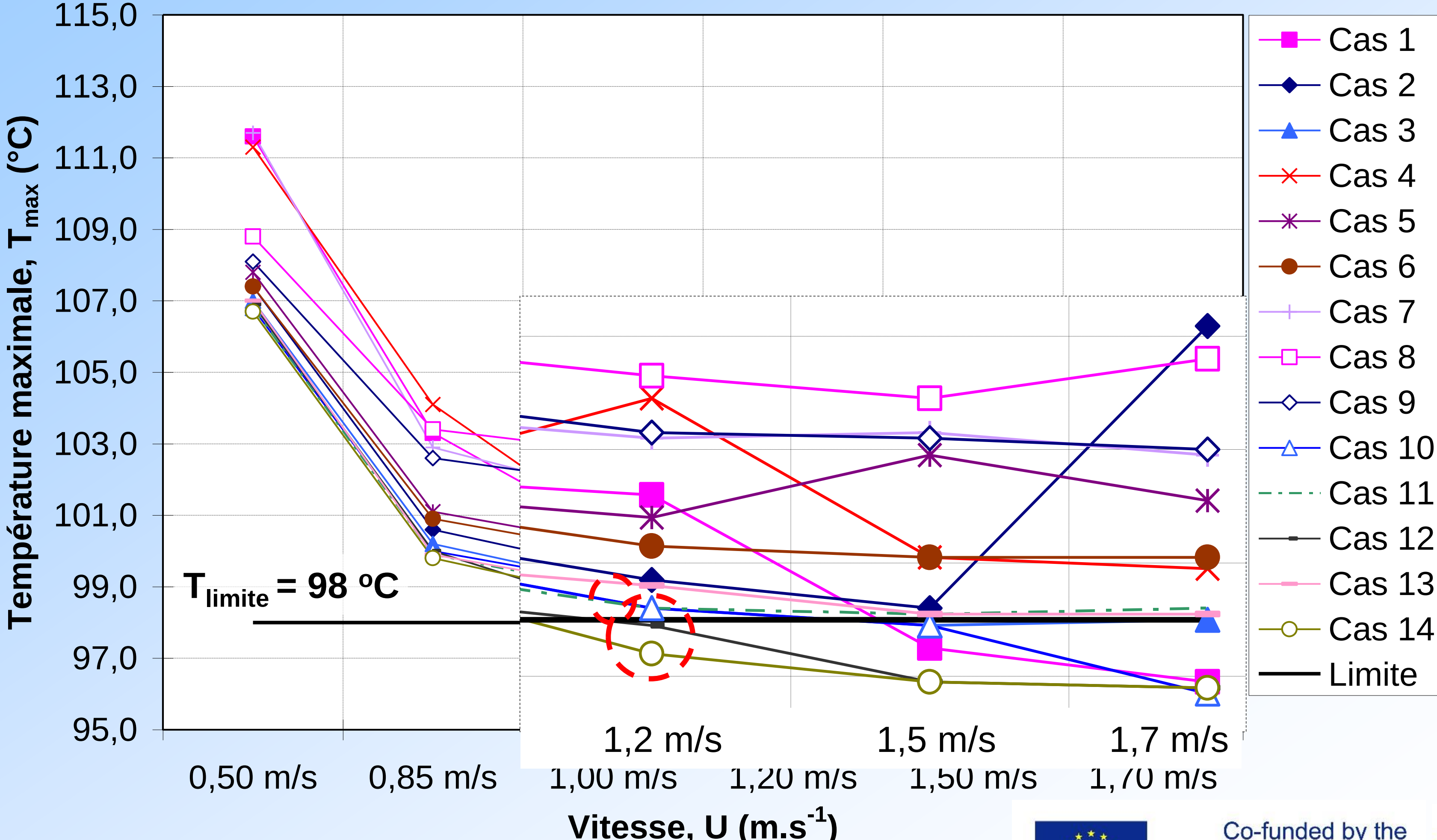
Comparisons between different cases

***Maximum temperature** calculating inside the power transformer*



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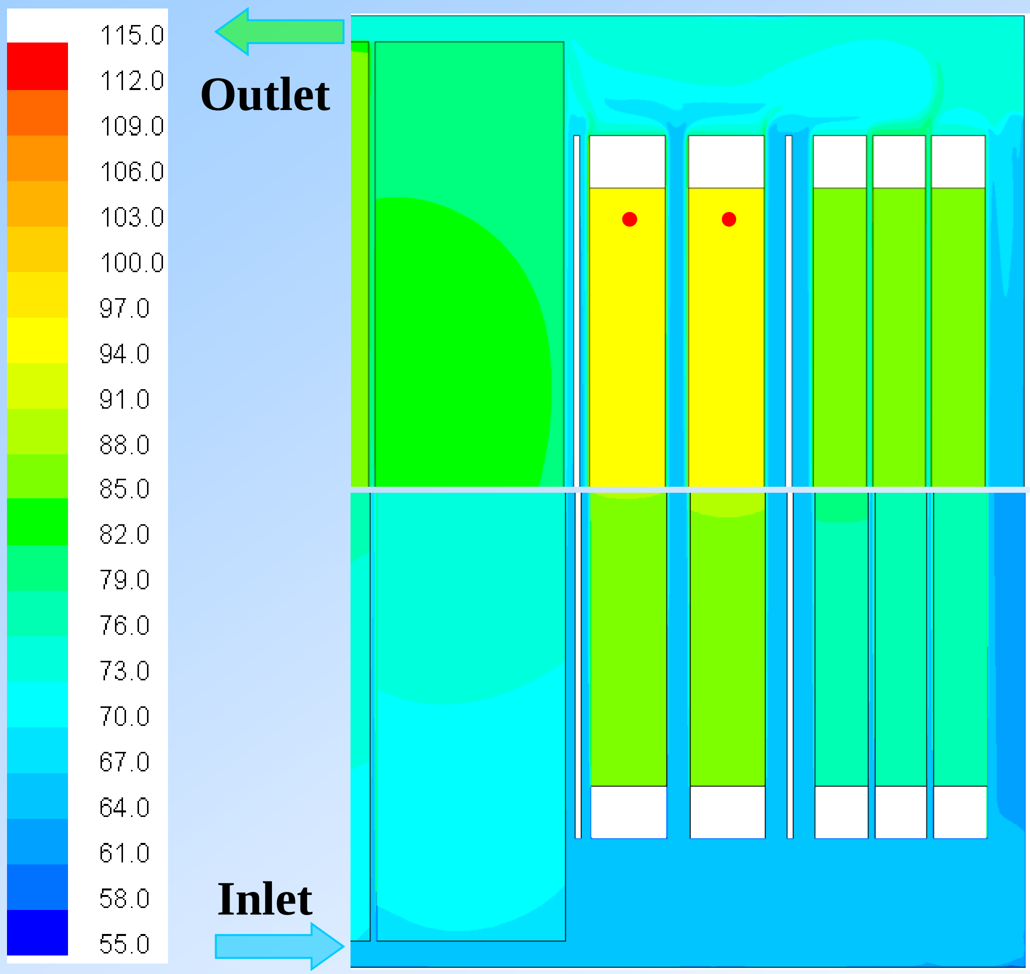
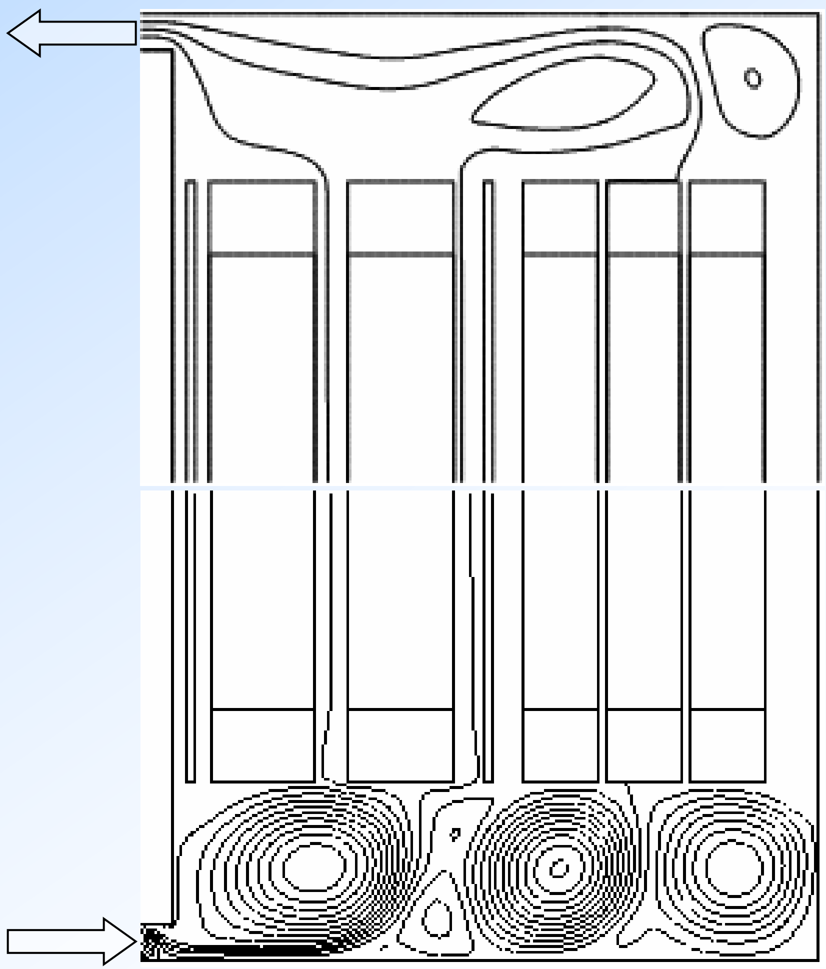
CASE 5'

Temperature field **and** *streamlines*

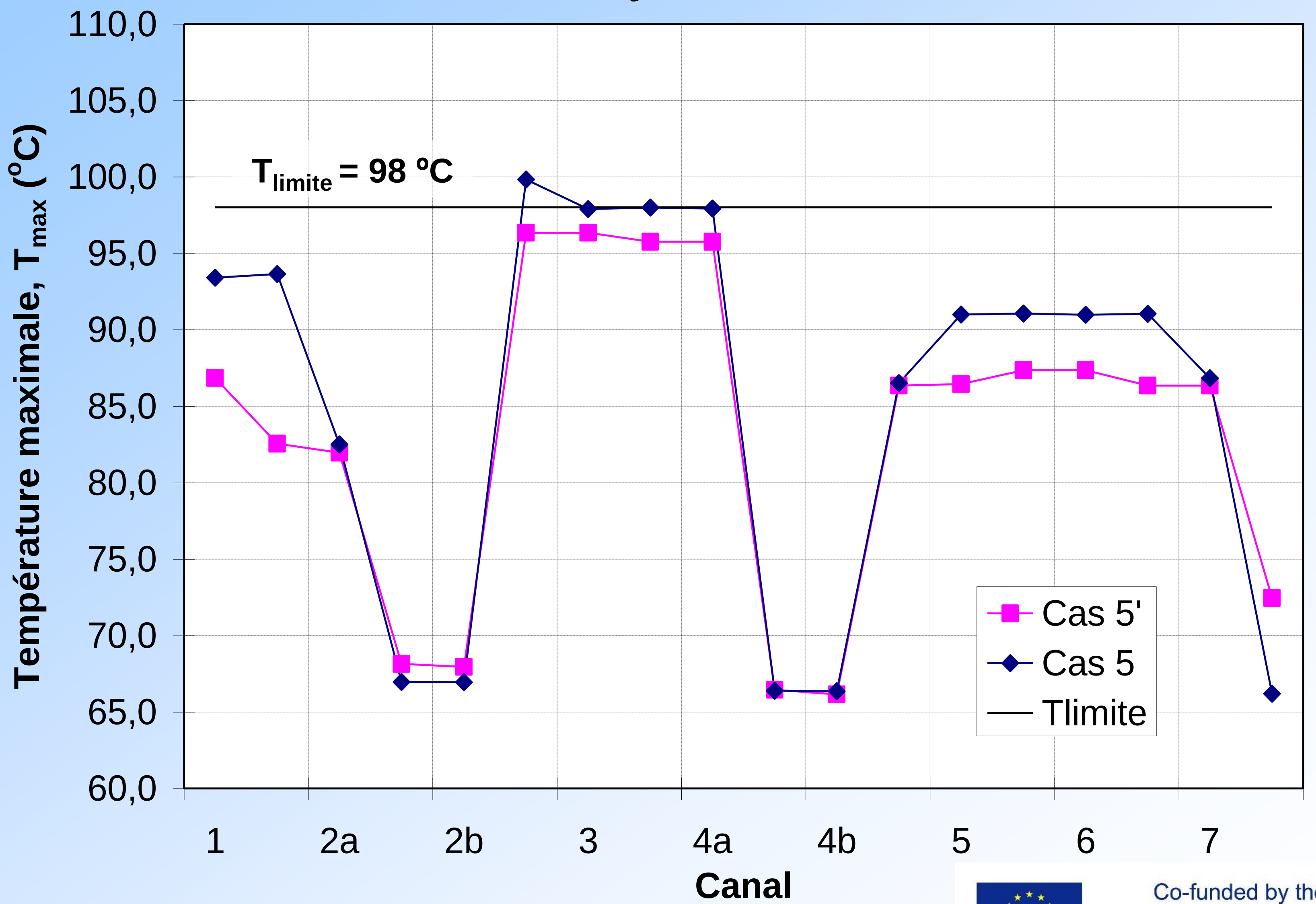
T_{max} = 96,4 °C



U = 1,20 $m.s^{-1}$



Maximum temperature on the walls of each channel for flow velocity of 1,2 $m.s^{-1}$



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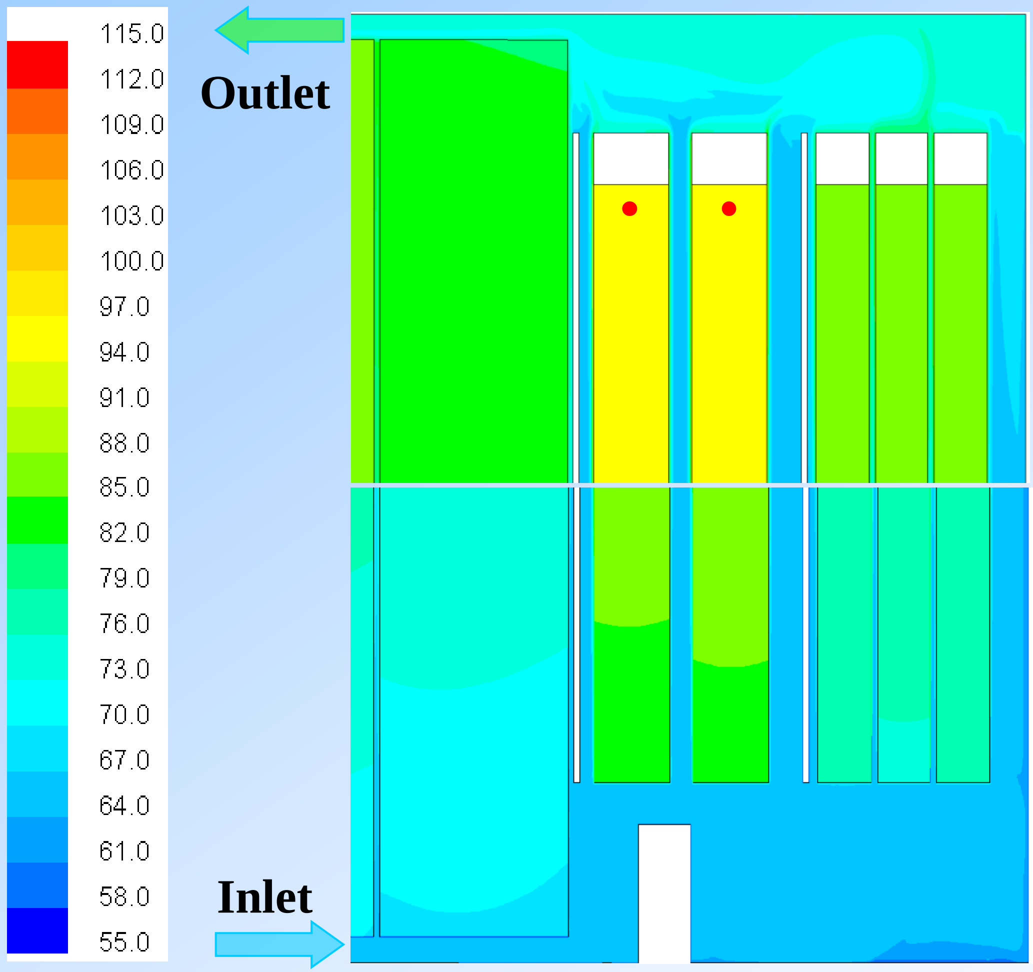
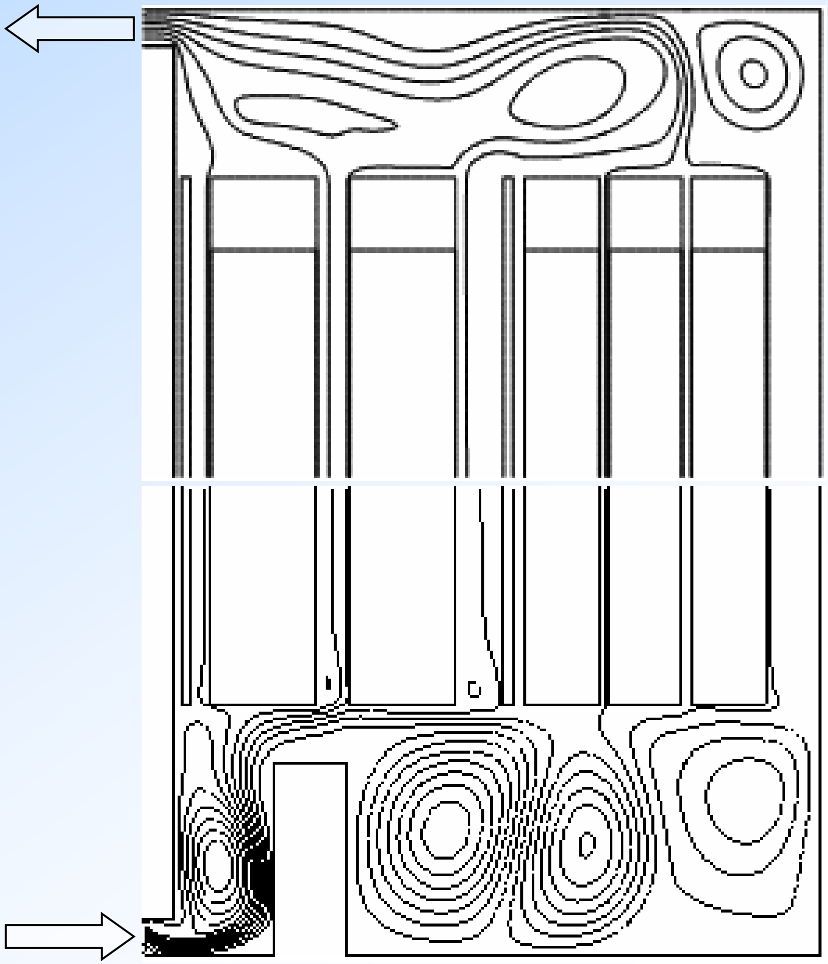
CASE 14'

Temperature field and streamlines

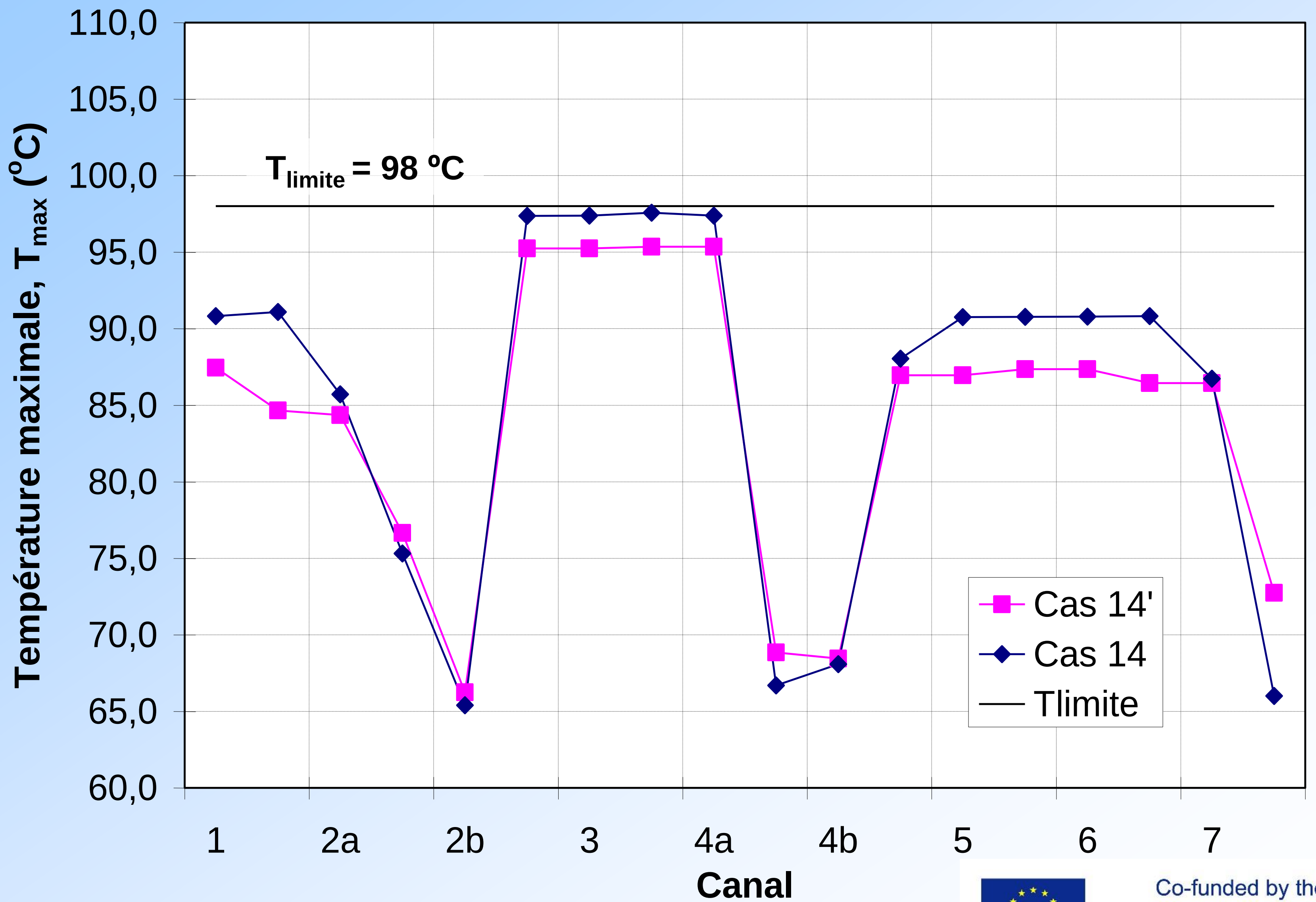
T_{max} = 95,4 °C



U = 1,20 m.s^{-1}



Maximum temperatures on the walls of each channel for flow velocity of 1,2 $m.s^{-1}$



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Conclusions

- The flow of oil inside of such a system is very complex and recirculations may occur

- The hotspots were located on:
  - the surfaces of the secondary winding in the case of a flux density
  - Inside the secondary windings in the case of a volume source.

- The limit of 98 ºC is respected in the *case 14* where:
  - a big *obstacle* is placed closer to the entrance
  - the longitudinal insulations are *moved inside the channel*
  - the oil velocity at the entrance of the power transformer is *1,2 m.s^{-1}*

- Temperatures are *overestimated* in the case of a flux density compared to the case of a volume source

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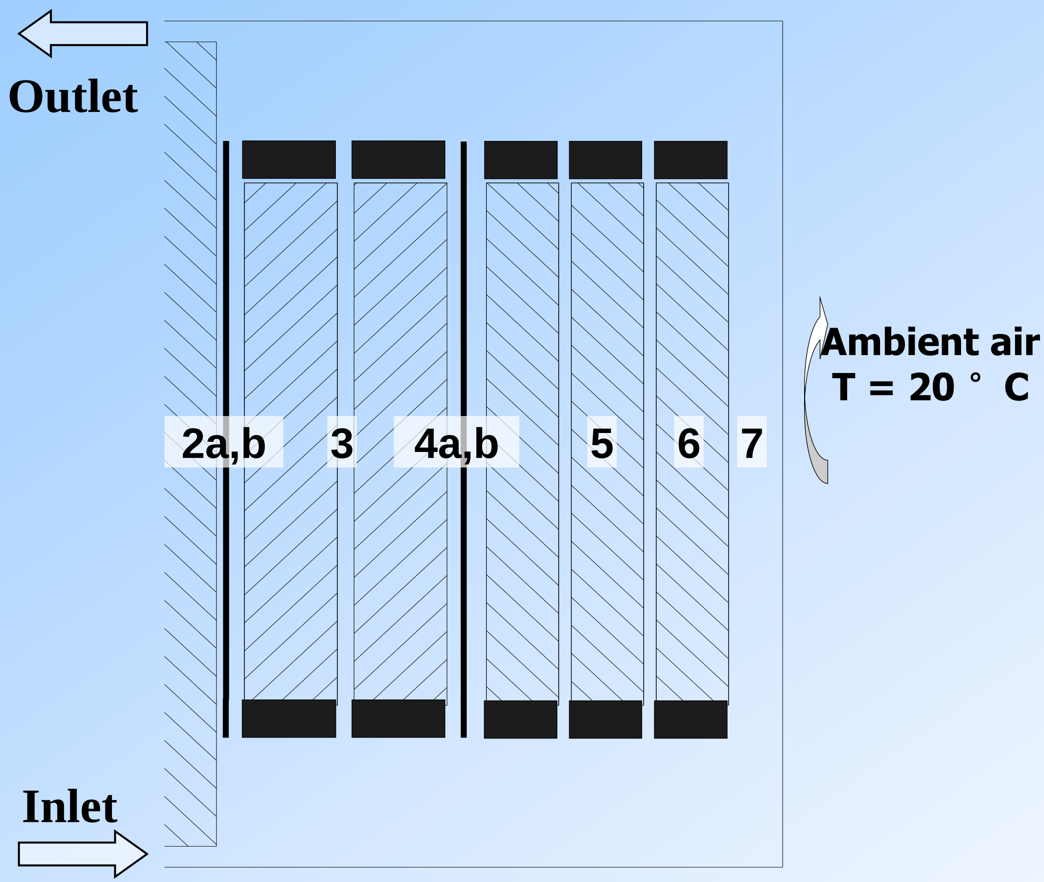
Experimental simulations

CASE 5



U = 1,50 m.s^{-1}
T = 65 ºC

ELECTROPUTERE CRAIOVA

- Determining the average temperature of the windings

Method of electrical resistance variation :

$$T_2 = \frac{R_2}{R_1}(235 + T_1) - 235$$

- Measuring the temperature of the mineral oil to the output of the transformer

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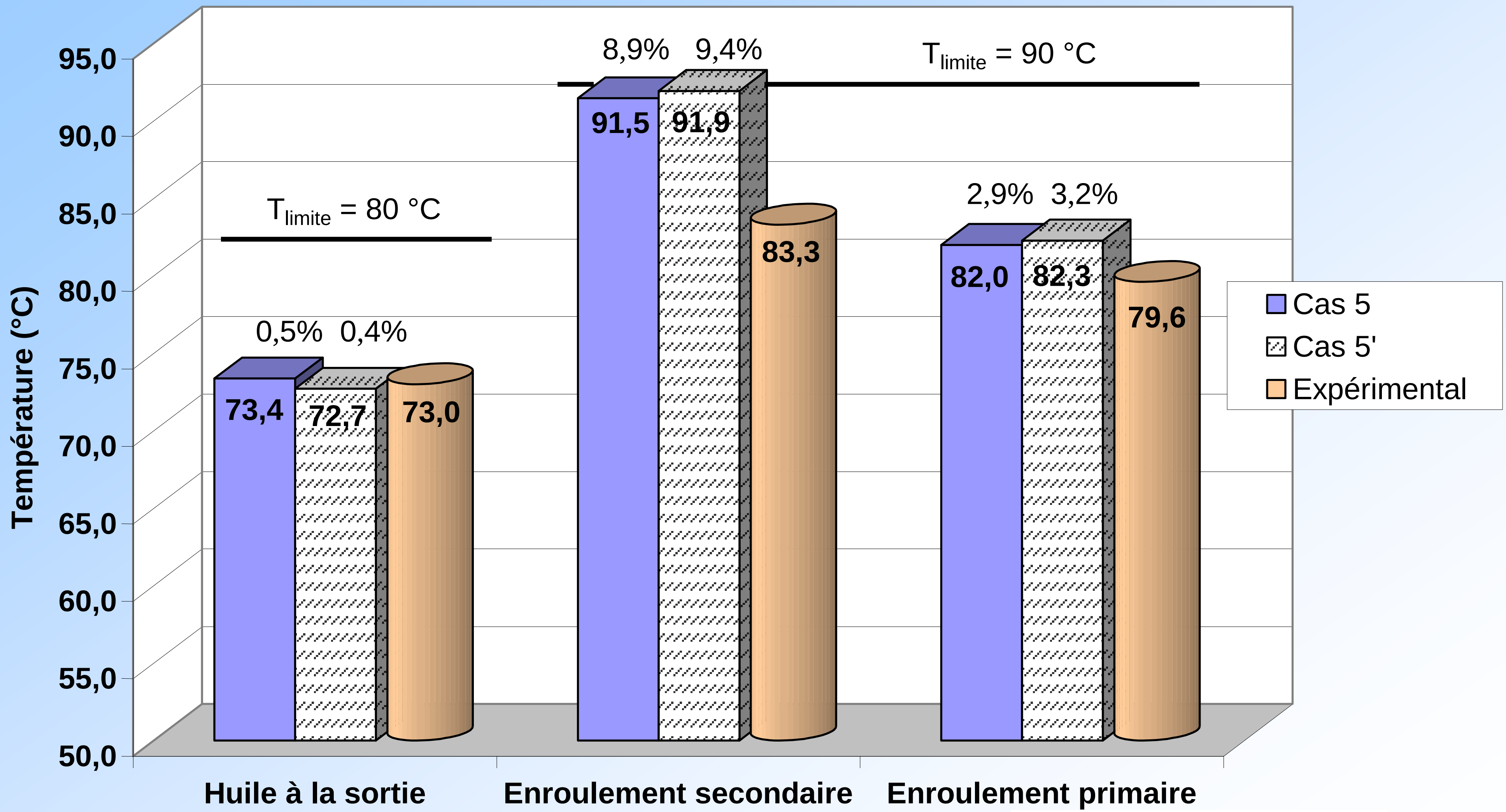
Comparison between *experimental* and *numerical* *results*



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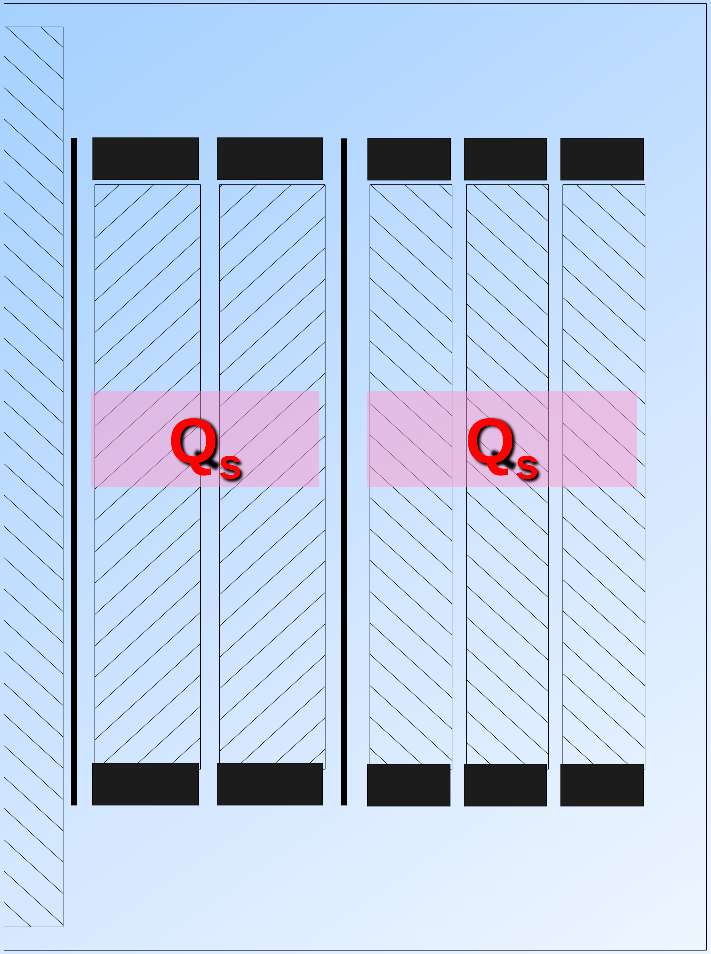
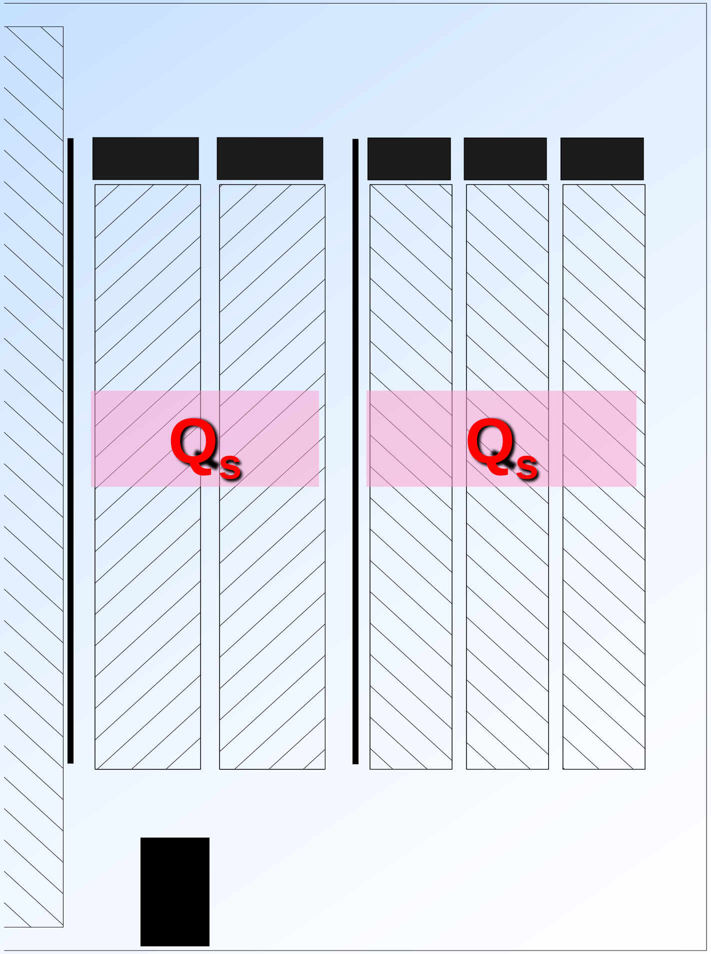
Unsteady state

Overload of 60% $(I_s = 1,6\ I_n) \rightarrow Q_s = 2,56\ Q_n$

CASE 5'

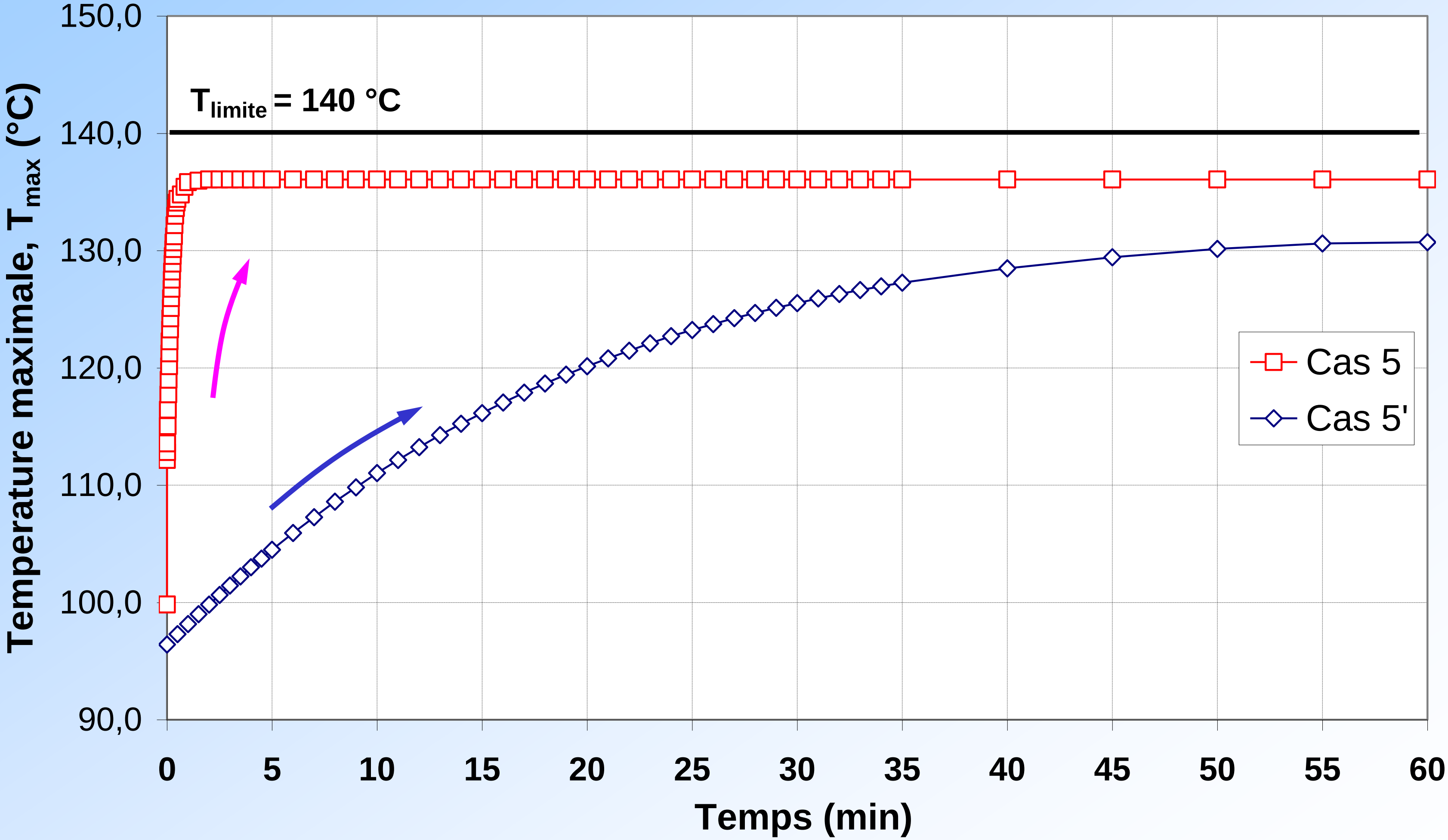


CASE 14'

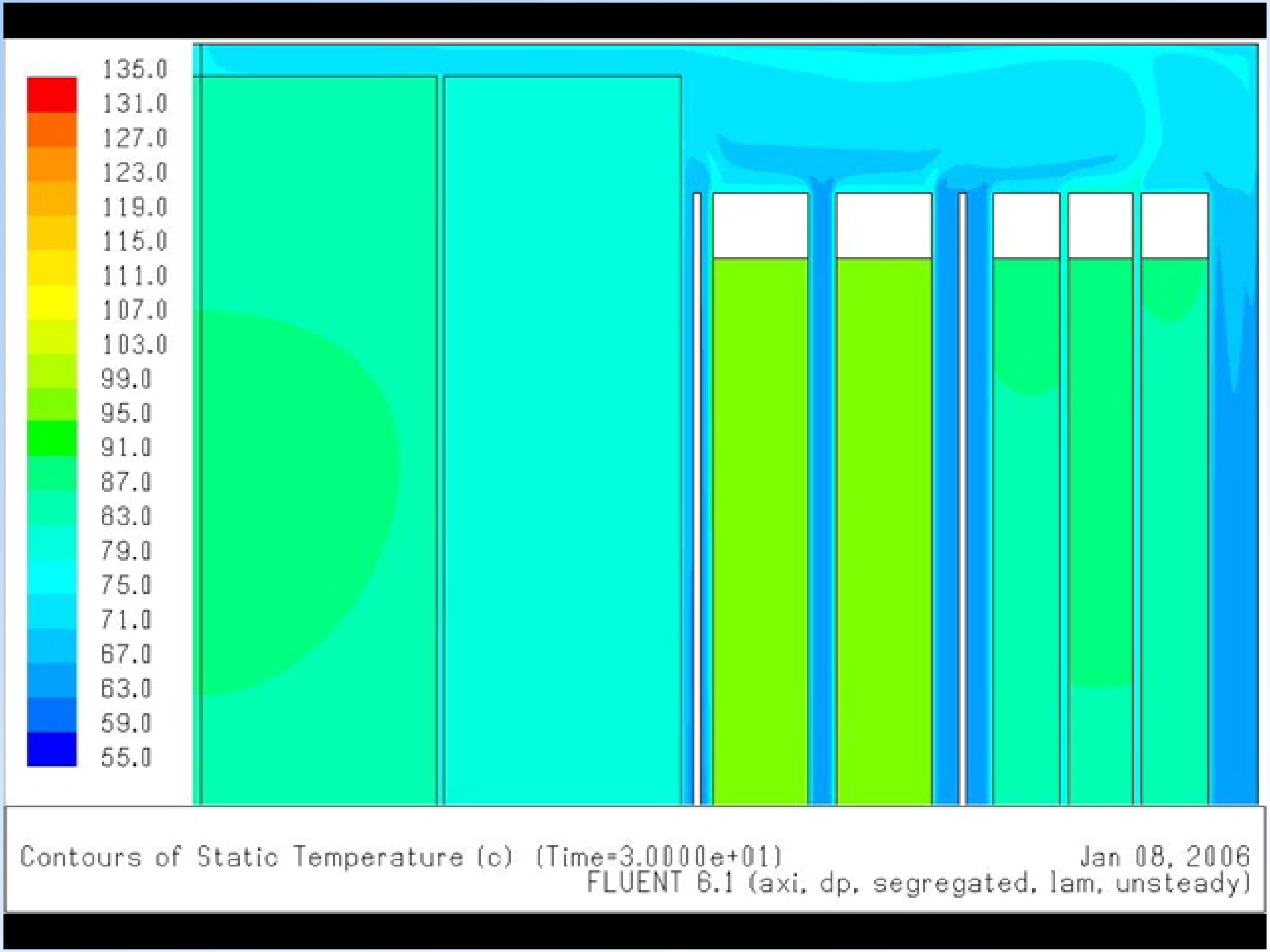


CASE 5 and 5'

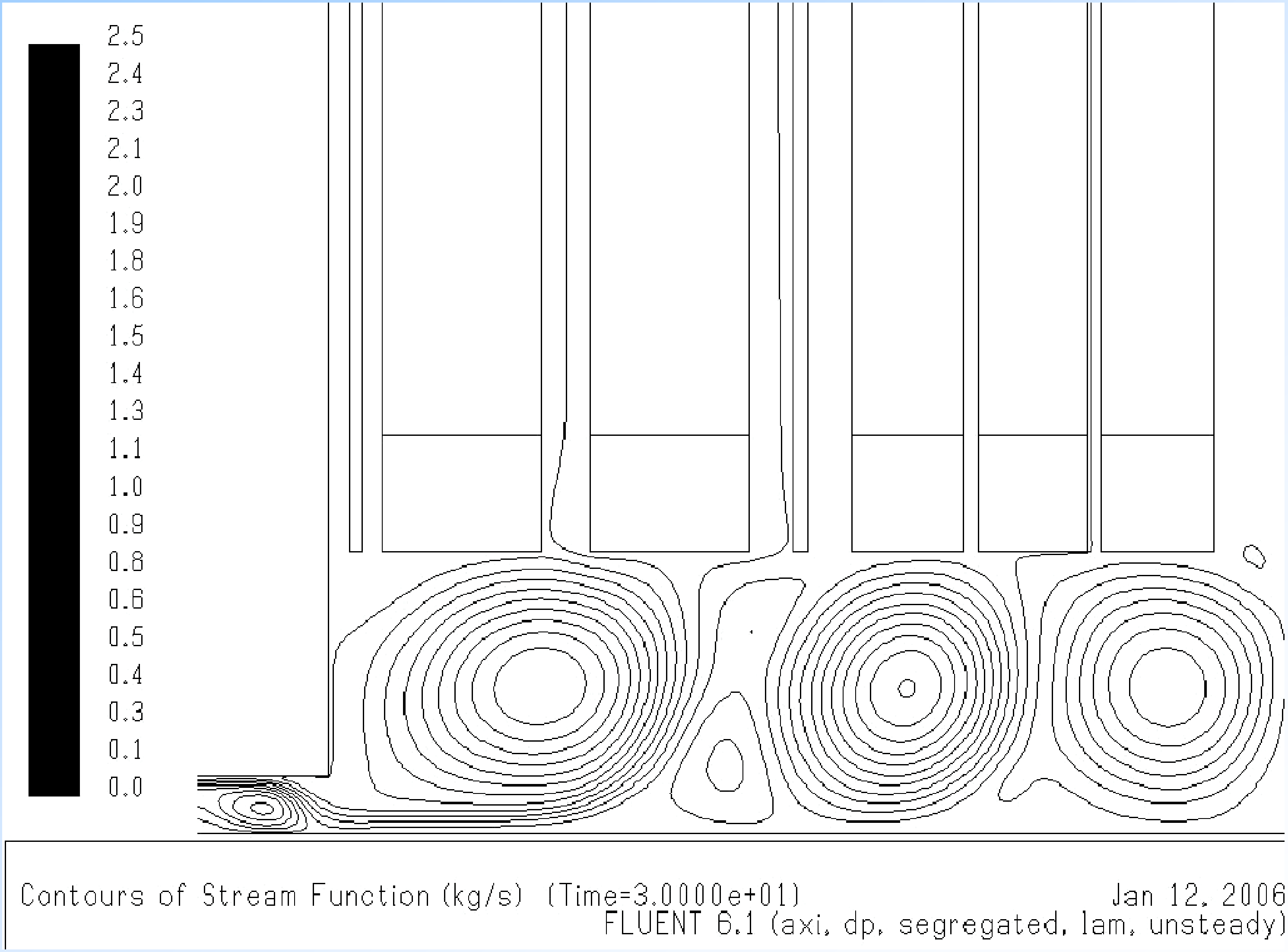
Variation of *maximum temperature* at inside the electrical power transformer during one hour



Temperature field at superior part of the transformer during *one hour* for *U = 1,20 m.s^{-1}*

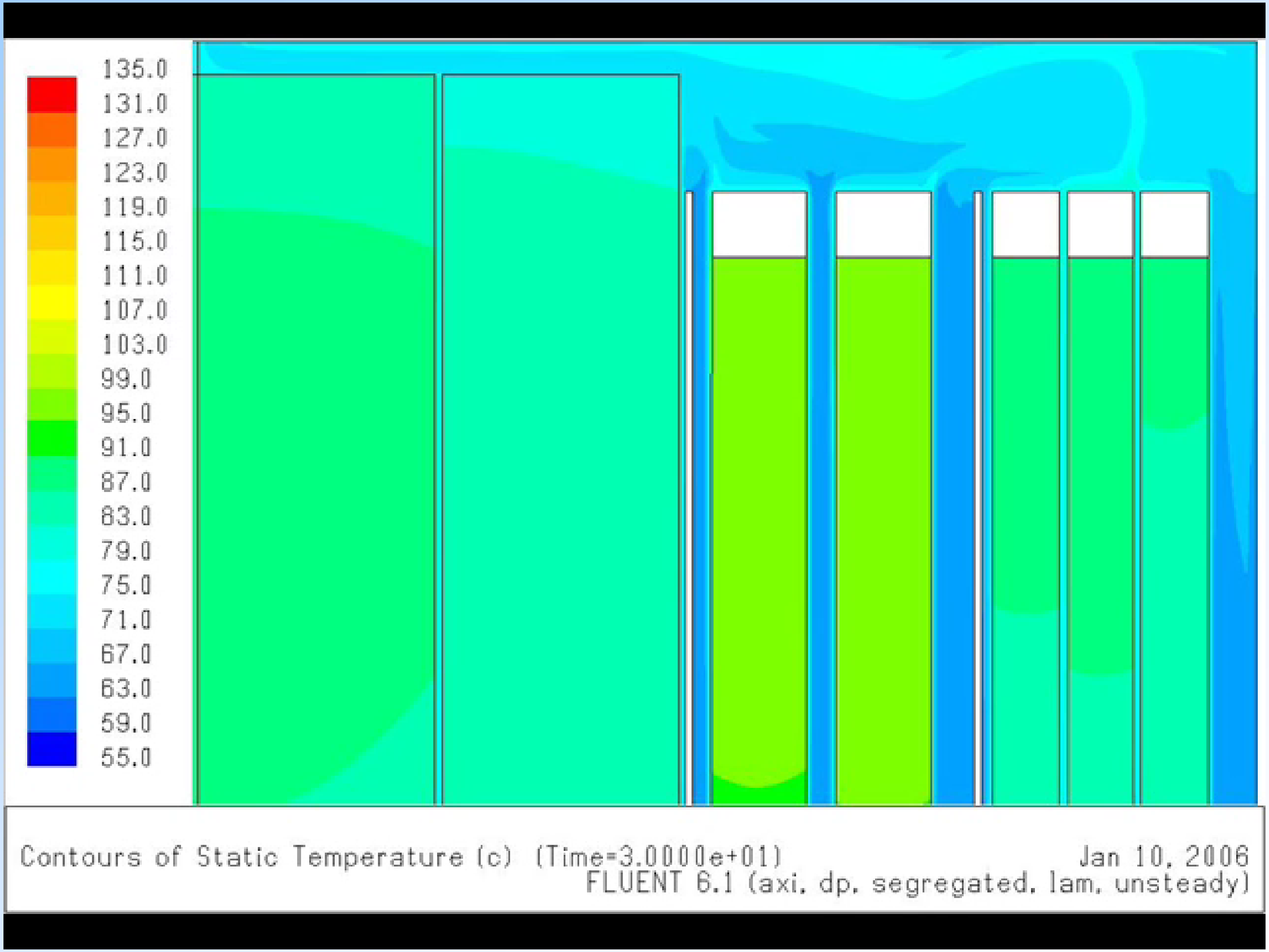


Streamlines at superior part of the transformer during ***one hour*** for ***$U = 1,20\ m.s^{-1}$***

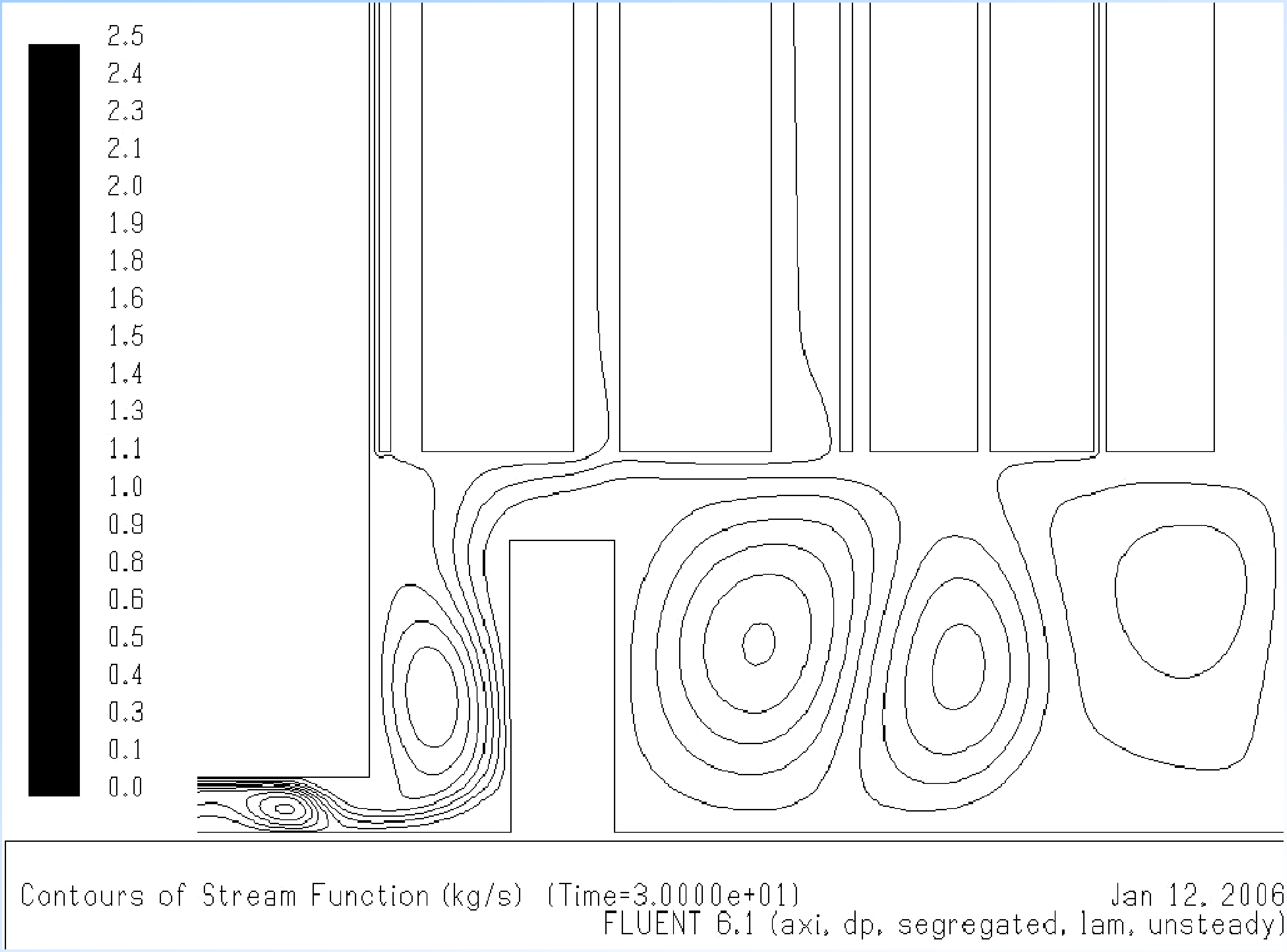


CASE 14'

Temperature field at superior part of the transformer during *one hour* for *U = 1,20 m.s⁻¹*



Streamlines at superior part of the transformer during *one hour* for $U = 1,20\ m.s^{-1}$



Conclusions

- The temporal evolution of the hot spot temperature is:
  - gradually in the case of a bulk source
  - abruptly in the case of a flux density

- Hot spots are always located within the secondary winding

- The hot spot temperature does not exceed the limit of 140 ºC imposed in transient state

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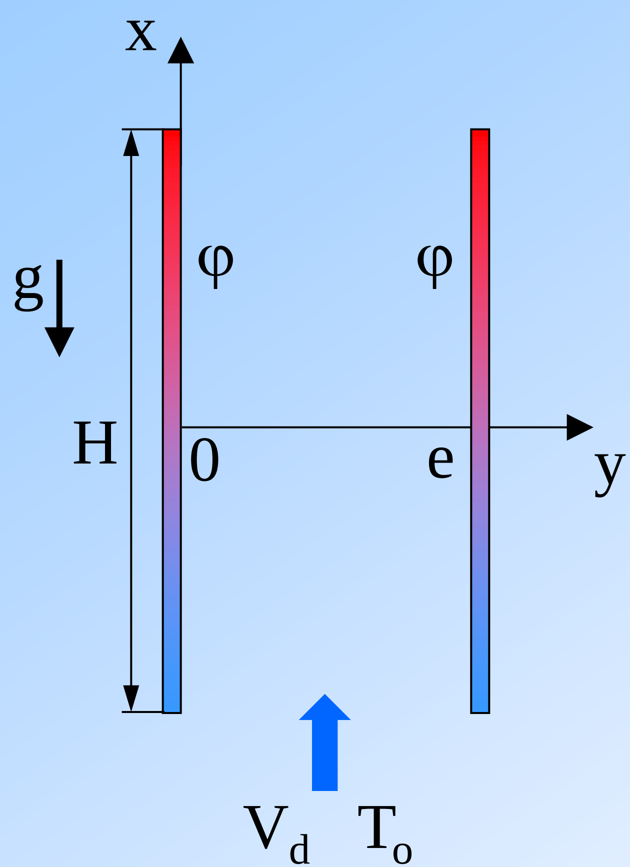
Assumptions



- ***Assisted mixed convection*** **in** ***steady state***
- **Ascending and laminar flow**
- ***Velocity and temperature of fluid*** **uniformly imposed at the entrance**
- **Convective heat transfer on smooth surfaces**
- ***Uniform flux density imposed on the walls***
- **thermo-physical properties of the fluid are independent of temperature**

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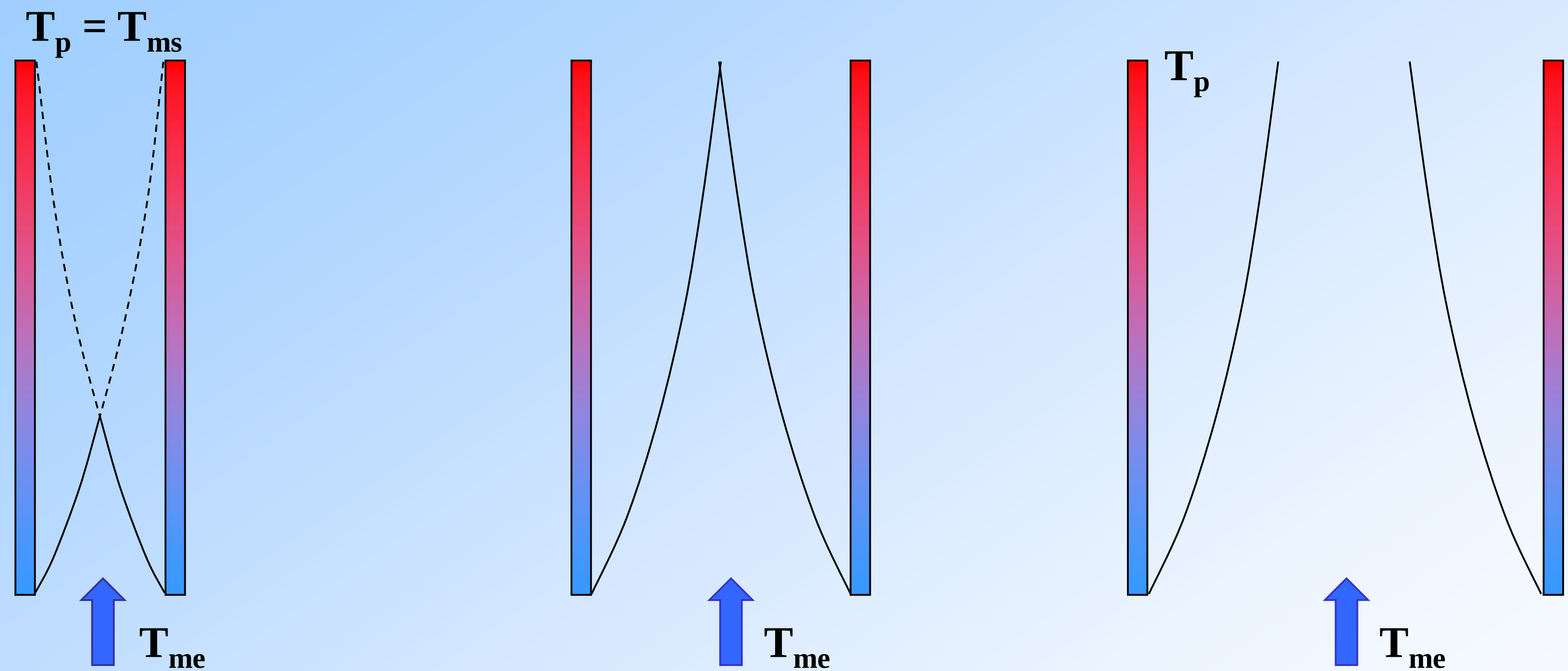
Narrow spacing | Optimal spacing | Large spacing



$$Q_{m1} = V_d\, e\, \rho\, C_p (T_p - T_m)$$

$$Q_{m2} = 2\, H\, h (T_p - T_m)$$

$$Q_{m1} = B e_H\, \Delta T\, \lambda\, e^3 H^{-3} \left[0{,}04 + 0{,}04 \left(1 + 0{,}057 \frac{g\, H\, \beta\, \mu\, \varphi\, e}{\Delta p^*\, \nu\, \lambda} \right)^{1/2} \right] = Q_{m2} = 14{,}76\, Re_H^{1/2} \left[1 + 0{,}05 \left(Ri_H^* / Re_H^{1/2} \right)^{3/5} \right]^{1/3} \lambda (T_p - T_m)$$

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Optimal distance

- **Mixed convection :**

$$e_{m\,opt} = 2{,}45H\left[\frac{\overbrace{\left(\frac{V_d H}{\nu}\right)}^{Re_H}{}^{1/2}\left[1+0{,}05\left(\overbrace{\frac{g\,\beta\,\varphi H^2}{\lambda V_d^2}}^{Ri_H^*}\overbrace{\left(\frac{V_d H}{\nu}\right)}^{Re_H}{}^{-1/2}\right)^{3/5}\right]^{1/3}}{0{,}04+0{,}04\left(1+0{,}057\frac{\rho\, g\,\beta\,\varphi\, e_{m\,opt}\, H}{\Delta p^*\,\lambda}\right)^{1/2}}\right]^{1/3}\overbrace{\left(\frac{\Delta p^* H^2}{a\,\mu}\right)}^{Be_H}{}^{-1/3}$$

- **Natural convection[1] :**

$$e_{n\,opt} = 2{,}3\,H\overbrace{\left(\frac{g\beta\,\Delta T\,H^3}{\nu\,a}\right)}^{Ra_H}{}^{-1/4}$$

- **Forced convection[1] :**

$$e_{f\,opt} = 2{,}7\,H\overbrace{\left(\frac{\Delta p^* H^2}{\mu\,a}\right)}^{Be_H}{}^{-1/4}$$

[1] *A. Bejan, "Shape and Structure, from Engineering to Nature", Cambridge University Press*

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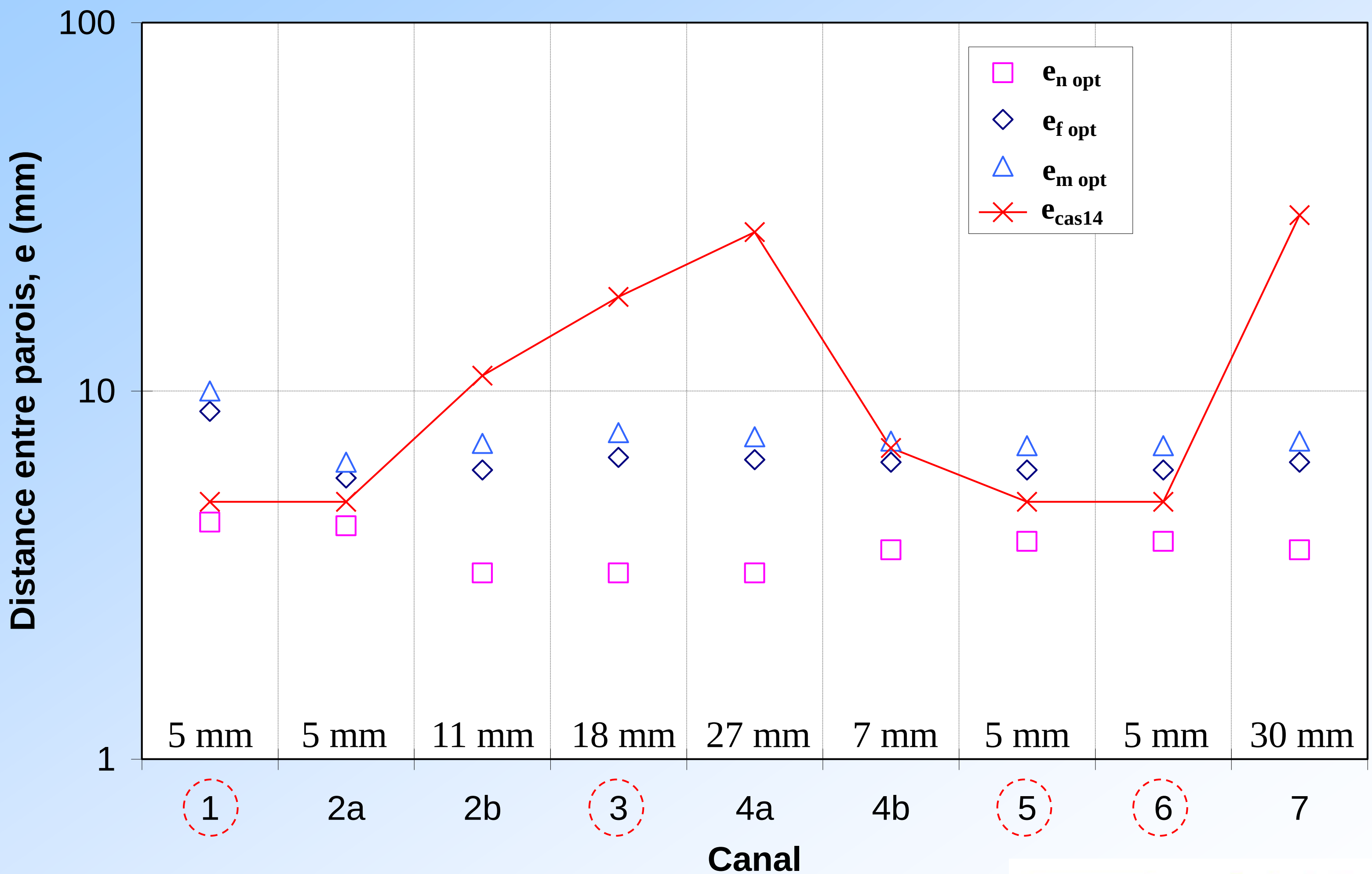
Optimal spacing and spacings used in *cas 14* for each channel and different convective regimes



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- ➢Conclusions

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Selection criteria

➢***Uniform temperatures imposed on the walls***

- **Ist : comparison of the friction stresses on the walls**
- **IInd : comparison of gravitational and viscous forces**
- **IIIrd : comparison of gravitational forces and pressure forces**
- **IVth : comparison of the total gravitational and kinetic energies**

➢***Uniform flux densities imposed on the walls:***

- **IInd : comparison of gravitational and viscous forces**
- **IIIrd : comparison of gravitational forces and pressure forces**

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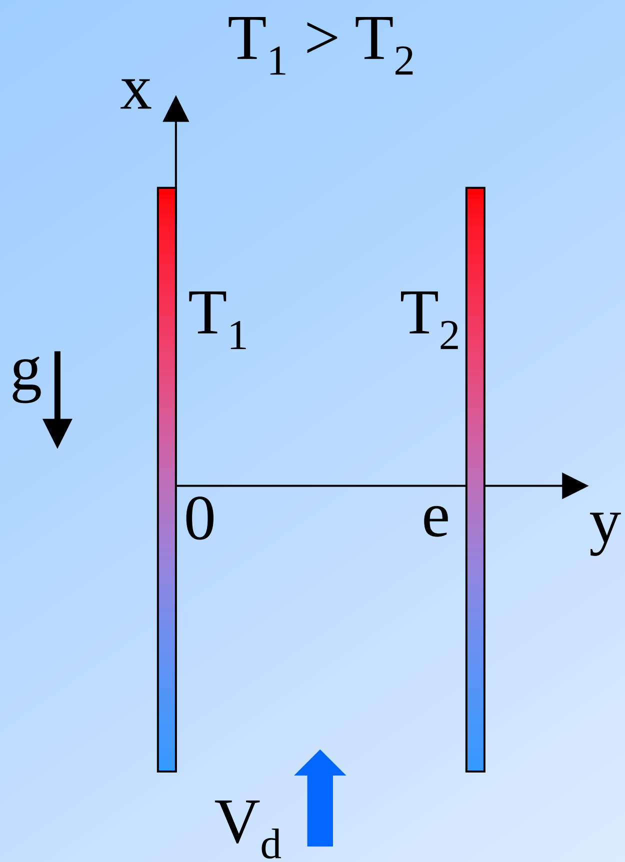
Uniform temperatures



Assumptions

- ***Uniform temperatures*** **iimposed on the walls**
- ***Assisted mixed convection*** **in** ***steady state***
- **Laminar and ascending flow**
- ***Velocity and temperature of fluid imposed at the entrance***
- **thermo-physical properties of the fluid are independent of temperature.**

Boundary conditions:

$$y = 0 : T = T_1 ; U = 0$$

$$y = e : T = T_2 ; U = 0$$

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✓ Continuity equations :

$$\frac{\partial U}{\partial x}+\frac{\partial V}{\partial y}=0 \qquad \rightarrow V=cte=0$$

✓ Momentum equations:

a) Gravitational forces — b) Pressure forces — c) Viscous forces

$$U\frac{\partial U}{\partial x}+V\frac{\partial U}{\partial y}=g\beta\left(T-\frac{T_1+T_2}{2}\right)-\frac{1}{\rho}\frac{\partial p^*}{\partial x}+\nu\left(\frac{\partial^2 U}{\partial x^2}+\frac{\partial^2 U}{\partial y^2}\right) \qquad \rightarrow 0=\underbrace{g\,\beta\left(T-\frac{T_1+T_2}{2}\right)}_{a)}\underbrace{-\frac{1}{\rho}\frac{dp^*}{dx}}_{b)}\underbrace{+\nu\frac{d^2U}{d\,y^2}}_{c)}$$

$$U\frac{\partial V}{\partial x}+V\frac{\partial V}{\partial y}=-\frac{1}{\rho}\frac{\partial p^*}{\partial y}+\nu\left(\frac{\partial^2 V}{\partial x^2}+\frac{\partial^2 V}{\partial y^2}\right) \qquad \rightarrow 0=\frac{\partial p^*}{\partial y}$$

✓ Energy equation :

$$U\frac{\partial T}{\partial x}+V\frac{\partial T}{\partial y}=a\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \qquad \rightarrow 0=\frac{d^2T}{d\,y^2}$$

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➢ I^st criterion: comparison of the friction stresses on the walls

Natural convection

$$\frac{\tau_{p1} - \tau_{p2}}{\tau_{p1}} < 10\% \rightarrow (RiRe)_e \geq 5472$$

Forced convection

$$\frac{\tau_{p1} + \tau_{p2}}{\tau_{p1}} < 10\% \rightarrow (RiRe)_e \leq 15{,}2$$

➢ II^nd criterion: comparison of gravitational and viscous forces

a) Gravitational forces

c) Viscous forces

$$0 = g\beta\left(T - \frac{T_1 + T_2}{2}\right) - \frac{1}{\rho}\frac{dp^*}{dx} + \nu\frac{d^2U}{dy^2}$$

$$P^2 = \frac{\overline{a^2}}{\overline{c^2}} = \frac{(RiRe)_e^2}{(RiRe)_e^2 + 27648}$$

$$(RiRe)_e = \frac{166{,}28P}{\sqrt{1-P^2}}$$

Natural convection

$$P > 0{,}95 \rightarrow (RiRe)_e \geq 505{,}9$$

Forced convection

$$P < 0{,}05 \rightarrow (RiRe)_e \leq 8{,}3$$

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➢ IIIrd criterion : comparison of gravitational forces and pressure forces

a) Gravitational forces — b) Pressure forces

$$0 = \underbrace{g\,\beta\left(T - \frac{T_1 + T_2}{2}\right)}_{\text{a) Gravitational forces}} \underbrace{- \frac{1}{\rho}\frac{dp^*}{dx}}_{\text{b) Pressure forces}} + \nu\frac{d^2U}{dy^2}$$

$$\Gamma^2 = \frac{\overline{a^2}}{b^2} = \sqrt{\frac{(RiRe)_e^2}{27648}}$$

$$(RiRe)_e = 166{,}28\,\Gamma$$

NATURAL CONVECTION

$\Gamma > 0{,}95 \rightarrow \mathbf{(RiRe)_e \geq 332{,}56}$

FORCED CONVECTION

$\Gamma < 0{,}05 \rightarrow \mathbf{(RiRe)_e \leq 8{,}3}$

➢ IVth criterion : comparison of the total gravitational and kinetic energies

$$K_e = \frac{(RiRe)_e}{\sqrt{580608 + (RiRe)_e^2}} \rightarrow (RiRe)_e = \frac{762\,K_e}{\sqrt{1 - K_e^2}}$$

NATURAL CONVECTION

$K_e > 0{,}95 \rightarrow \mathbf{(RiRe)_e \geq 2318}$

FORCED CONVECTION

$K_e < 0{,}05 \rightarrow \mathbf{(RiRe)_e \leq 38{,}2}$

(RiRe)$_e$ number corresponding to each criterion for uniform imposed temperature

Transition	First criterion	Second criterion	Third criterion	Fourth criterion
Mixed / natural convection	***5 472***	***505, 9***	***3 325,6***	***2 318***
Mixed / forced convection	***15,2***	***8,3***	***8,3***	***38,2***

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Flux density uniformly imposed

- **II^nd criterion:** comparison of gravitational and viscous forces

$$P^2 = \frac{\overline{a^2}}{\overline{c^2}} = \frac{1{,}06.10^{-7}\left(892786 + (Ri\,Re)_e^{*}\right)^2}{\left(299935 + (Ri\,Re)_e^{*}\right)^2}(Ri\,Re)_e^{*2} \qquad (Ri\,Re)_e^{*} = 34058 + 85126\sqrt{16 - P}$$

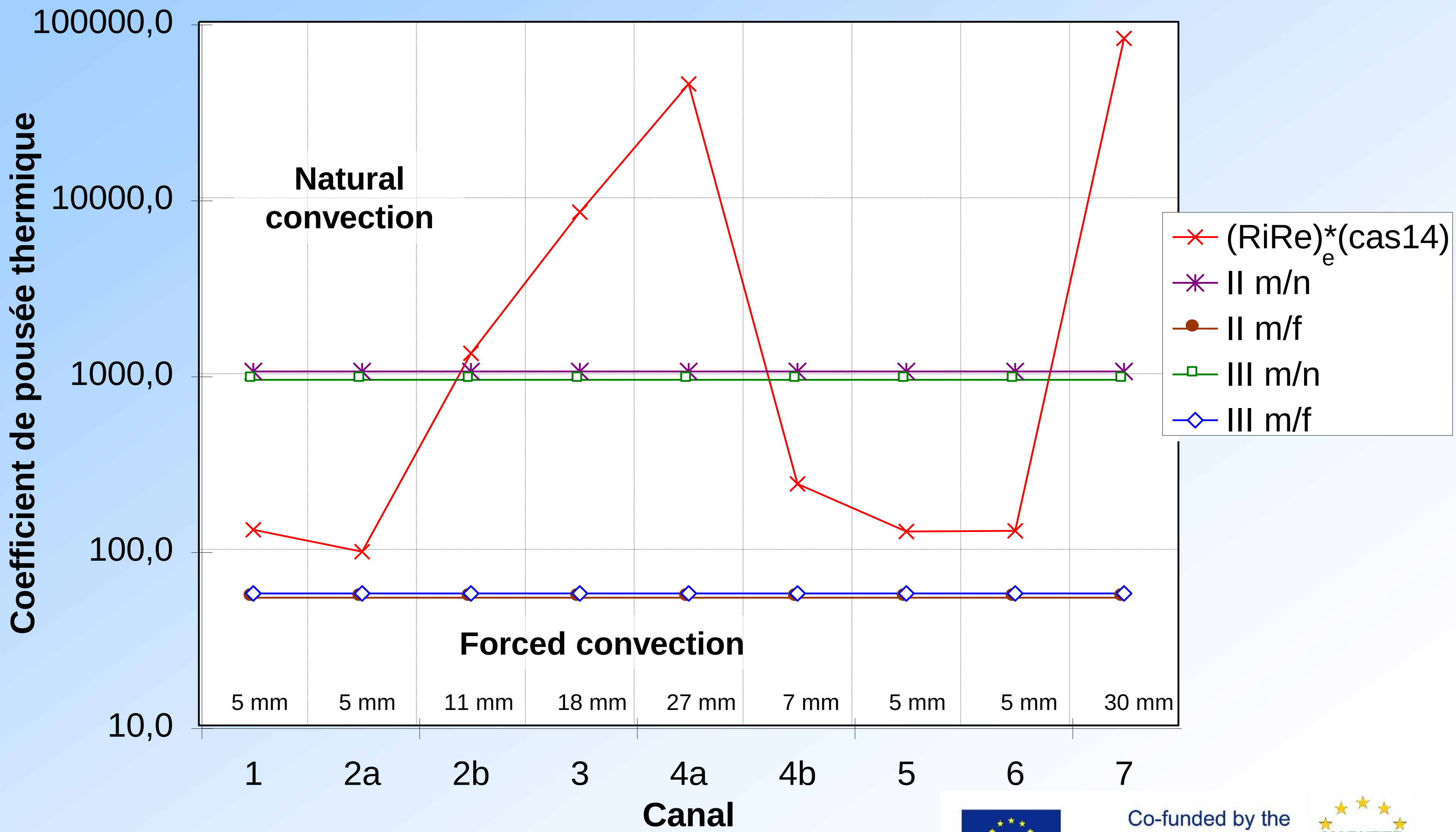
NATURAL CONVECTION	FORCED CONVECTION
$P > 0{,}95 \rightarrow \mathbf{(RiRe)_e^{*} \geq 10261}$	$P < 0{,}05 \rightarrow \mathbf{(RiRe)_e^{*} \leq 532}$

- **III^rd criterion :** comparison of gravitational forces and pressure forces

$$\Gamma^2 = \frac{\overline{a^2}}{\overline{c^2}} = 1{,}70.10^{-14}(Ri\,Re)_e^{*2}\left(1{,}2.10^{-5}(Ri\,Re)_e^{*} + 0{,}082\right)^2\left(892786 + (Ri\,Re)_e^{*}\right)^2$$

NATURAL CONVECTION	FORCED CONVECTION
$\Gamma > 0{,}95 \rightarrow \mathbf{(RiRe)_e^{*} \geq 8736}$	$\Gamma < 0{,}05 \rightarrow \mathbf{(RiRe)_e^{*} \leq 519}$

Thermal pressure coefficient corresponding to different selection criteria in *case 14* for oil velocity at the entrance of *1,2 m.s⁻¹*



Summary

- Objectives
- Description of thermoconvective model
- Steady state
- Experimental simulation
- Unsteady state
- Optimal spacing between vertical flat plates
- Selection criteria for distinguishing different convection regimes
- Conclusions

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Conclusions

- Using an obstacle to direct the oil from entering the transformer helps to cool its active parts at lower cost

- The comparison of numerical and experimental results shows good agreement with lower spreads 10%

- The hypothesis of a flux density imposed on surfaces that were justified in the steady state is no longer valid in the transient state

- The semi-analytical analysis of the optimal spacing showed that the heat transfer is more effective within narrow channels

- The heat transfer inside narrow channels is realized by mixed convection while in the wider channels by natural convection

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