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1D STABILITY ANALYSIS OF SINGLE-PHASE NATURAL CIRCULATION LOOPS

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Abstract. *Passive Cooling Systems (PCS') are engineering solutions to perform the function of heat transfer using the temperature difference between hot and cold sources to generate the driving force. These systems have increasing importance to nuclear power industry and are object of many studies since 1960 decade. One of the main concerns in PCS' is thermal-hydraulic stability. This work presents the results of a linear and a non-linear stability analysis of a single-phase Natural Circulation Loop. A 1D model has been developed to solve the steady-state and transient momentum and energy balances. Validation results against experimental data from literature are shown, as well as the effect of many process and geometrical parameters on the system's thermal-hydraulics stability.*

Keywords: *Single-Phase NCL, Thermal-hydraulic Instability, Passive Cooling System*

1. INTRODUCTION

After the events of March 11, 2011 in Fukushima, the role of passive safety systems in nuclear installations has increased considerably. Indeed, all new designs of nuclear reactors is provided with a passive safety cooling system (PCS), and some of them has such a system even for normal operation. The objective is to minimize the reliance on energy supply. In the current context of control over carbon emissions, nuclear energy appears as an alternative for a clean energy matrix, which places passive safety systems in the path of sustainable development. In Brazil, the last Energy National Plan (EPE, 2007), published by the Brazilian company responsible for planning the energy supply, indicated plans to construct at least four additional NPP's in Brazil before 2021, since almost all its hydro resources for electricity production is already being explored and other renewable energy technologies are not sufficient to fulfil current increasing demand.

Another application of natural convection occurs in nuclear spent fuel storage facilities for removal of the residual heat released from spent fuel assemblies. Currently there is a project under development in Brazil to construct a wet storage facility for spent fuel assemblies provided with a single-phase fully passive heat removal system.

PCS' can be classified as single-phase and two-phase systems. There is a third class which operate at very high temperatures and pressures: the supercritical systems, which are single-phase with characteristics of two-phase systems. Despite the recent growing interest in PCS', this is object of many studies since 1960 decade, mainly concerned about thermal-hydraulic stability. Boure *et al.* (1973) is one of the first works to provide a classification of instabilities in two-phase systems (including active systems) and is the basis for the classification used today by many authors for natural circulation loops (NCL's), both single and two-phase. Their work presents 10 types of instabilities, divided into two main classes: static and dynamic instabilities. Prasad *et al.* (2007) identified the types studied by Boure *et al.* (1973) as thermo-hydraulic instabilities, and added two other groups aside this: the instabilities associated to control systems and the ones associated to neutron kinetics.

Although two-phase systems are much more susceptible to instabilities, there are conditions in which single-phase systems can become unstable. This was experimentally confirmed by Creveling *et al.* (1975). They estate that, in previous works, instabilities in single-phase NCL's were only reported for systems operating at conditions close to the pseudo-critical point. They mention, however, previous analytical results which concluded that there are conditions under which subcritical single-phase systems present instabilities. Indeed, Keller (1966) had found theoretical evidence of instability in a rectangular single-phase loop. Welander (1967) extended this result and offered a very comprehensive explanation of the oscillatory mechanism. After these works, stability of single-phase NCL's started to received more attention from many researchers (Wacholder *et al.*, 1982; Zvirin, 1985; Lavine *et al.*, 1987).

In the beginning of 1990 decade, P. K. Vijayan and other researchers from the Bhabha Atomic Research Centre started to publish several works on NCL's. In Vijayan and Austregesilo (1994), a simplified scaling law for single-phase NCL's was proposed in which the system is characterized by a version of the Stanton and Grashof numbers. The authors showed experimental results and a 1D numerical model, which was much more conservative than the experiments. In Nayak *et al.*

(1995) they present the model in more details. These attempts to simulate NCL's started to reveal the challenge of a proper selection of the friction factor correlation. In fact, that was the topic addressed by Ambrosini *et al.* (2004) for transitional flow regimes. The authors point the need for further investigation on the applicability of forced flow friction correlations to natural convection and argue that 3D simulations can make relevant contributions. The effect of several geometrical and process parameters on the stability of single-phase NCL's, such as heater and cooler orientations, loop diameter and input power have been studied in the past ten years (Vijayan *et al.*, 2008; Basu *et al.*, 2013a). Basu *et al.* (2013b) analysed the influence of geometrical and operating loop parameters and proposed an expression to calculate the threshold of stability based on the Richardson number applicable to rectangular single-phase NCL's within some constraints. Most of these works employed a 1D numerical model for the single-phase NCL, based on cross-section average properties, integrating the incompressible momentum equation along the loop's volume, using Boussinesq's hypothesis for the force term.

For a detailed review on single-phase NCL's, refer to Basu *et al.* (2014). Besides describing the different results obtained up to now, they show the upcoming research fronts on this topic.

In this paper, steady state predictions and stability of single-phase NCL's are studied from the point of view of impact of friction factor correlation and numerical model. For this purpose, a classical and simple finite differences model for non-linear stability analysis was implemented, with possibilities of changing friction factor and numerical discretizations. It will be shown that the reproduction of the oscillating behaviour of the loop is highly impacted by time advance scheme and space discretization, and that friction factor determines the amplitude and frequency of the oscillations, besides altering the steady state. The model is based on cross-section averaged properties and volume integrated momentum equation.

2. MATHEMATICAL MODEL

The basic structure of the NCL's considered in this work consists of a heater and a cooler connected by pipes, forming a rectangular circuit. Heater and cooler consist of tube sections with conducting walls and can be vertical or horizontal. In the scope of this work, only horizontal orientation is considered. In the mathematical model, all properties are assumed to be constant in the cross section area. The circuit walls are set as adiabatic along the entire loop except for heater and cooler. Axial heat conduction is neglected, as well as viscous heating. Introducing the space coordinate s in the axial direction and the axial velocity component u, the continuity equation reduces to

$$\frac{\partial u}{\partial s} = 0 \tag{1}$$

After this result, the axial component of momentum equation can be written in cylindrical coordinates as

$$\rho \frac{\partial u}{\partial t} = -\frac{\partial p}{\partial s} + \mu \left[\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial u}{\partial r} \right) \right] + \rho g_s \tag{2}$$

where t is time, r is the radial space coordinate, ρ is the fluid density, p is the pressure field, μ is the viscosity, g_s is the s component of the gravity acceleration vector $\mathbf{g}$. Boussinesq hypothesis is adopted to write density as a linear function of temperature in the gravity term of momentum equation to solve the inconsistency between incompressibility assumption and the presence of buoyancy. Thus, density is written as $\rho = \rho_0[1 - \beta(T - T_0)]$, where the subscript 0 represents a value based on mean temperature value and β is the thermal expansion coefficient (also evaluated at T_0). Now, let L, D and A be the length, diameter and cross section area (constant) of the loop, respectively, and let $w = \frac{1}{\rho A} \int_0^L u dA$ be the system's mass flow rate. Integration along the whole loop's volume then produces

$$\frac{L}{A} \frac{dw}{dt} = -\left(f \frac{L}{D} + K \right) \frac{w^2}{2\rho_0 A^2} + \rho_0 g \beta \oint T dz \tag{3}$$

Here, g is the module of gravity field. The friction factor f and the local losses coefficient K were introduced. Note that the pressure gradient vanishes and that the viscous term gives rise to the term associated to friction losses.

Applying the assumptions described, energy conservation is expressed by

$$\frac{\partial h}{\partial t} + u \frac{\partial h}{\partial s} = \bar{q} \tag{4}$$

here, h is the specific enthalpy and $\bar{q}$ is a heat source per unit mass. Introducing the specific heat at constant pressure c_p given by $c_p = \frac{\partial h}{\partial T}$ the energy equation can be written in terms of T as

$$\rho_0 c_p \left(\frac{\partial T}{\partial t} + u \frac{\partial T}{\partial s} \right) = \dot{q} \tag{5}$$

Note that the heat source is now a thermal load per unit volume $\dot{q}$. Let's define L_h as the heater's length and Q as the total thermal power injected in the circuit (W in the SI). After integration on the cross section, equation 5 can be expressed as

$$\frac{\partial T}{\partial t} + \frac{w}{\rho_0 A} \frac{\partial T}{\partial s} = \frac{\dot{q}}{\rho_0 c_p} \tag{6}$$

It is now convenient to split the energy equation into heater, cooler and adiabatic tube sections, which implies in different forms of the heat source $\dot{q}$. We therefore arrive at

$$\text{heater: } \frac{\partial T}{\partial t} + \frac{w}{\rho_0 A} \frac{\partial T}{\partial s} = \frac{Q}{L_h A \rho_0 c_p} \tag{7a}$$

$$\text{cooler: } \frac{\partial T}{\partial t} + \frac{w}{\rho_0 A} \frac{\partial T}{\partial s} = -\frac{4U(T - T_s)}{D \rho_0 c_p} \tag{7b}$$

$$\text{pipes: } \frac{\partial T}{\partial t} + \frac{w}{\rho_0 A} \frac{\partial T}{\partial s} = 0 \tag{7c}$$

Note that the heat transferred by the cooler Q_c is modelled by $Q_c = A_c U(T - T_s)$, where U is the cooler's heat transfer coefficient (assumed constant), A_c is the cooler's surface area and T_s is the temperature of the cold source. This is the same model as that presented by Vijayan and Austregesilo (1994).

3. NUMERICAL MODEL

Momentum and energy equations are solved by finite differences method. The equilibrium values of mass flow and temperature distribution is obtained from the steady state form of equations 3 and 7, whose solution is obtained by means of an iterative procedure. From the mathematical point of view, the instability of a dynamical system can be associated to the competition between multiple solutions to the set of equations (Vijayan and Nayak, 2005), and the calculation of the steady state is subject to such multiplicity of solutions. Consequently, the procedure to solve the steady state of a NCL may not converge. To solve this, a controlled time step was introduced to the system, determined by a relaxation parameter based on the residuals of the previous iteration. So the "artificial" transient character prevents the solution from diverging, holding it towards the stationary solution. Additionally, the energy conservation equation follows an implicit formulation.

In this work, non-linear stability analysis is performed by solving the transient behaviour of the steady state solution after a perturbation (imposed on the mass flow). Some aspects have to be dealt with in the numerical simulation of such a system.

With respect to the energy equation, Vijayan and Austregesilo (1994) proposed explicit Euler for time advance with first order upwind discretization for the temperature convection term. If CFL $= 1$ (CFL $= u\Delta t/\Delta s$ is the Courant number), then this formulation is capable to transport temperature without losses in each time step, following a Lagrangian behaviour, which suits well for the hyperbolic character of equation 7. However, to set CFL $= 1$ is not a straightforward task because the flow velocity is constantly changing. Moreover, when the system is close to reversal (if that is the case), time increment Δt can be too large, and the time step can grow larger than the oscillating period. A solution is to set an upper limit for Δt, but this implies the introduction of numerical diffusion (due to the first order approximation) and requires a criterion for definition of maximum Δt. So the chosen upper bound value was the time increment used by Ambrosini *et al.* (2004), i.e., $\Delta t_{\max} = 0.1$. The explicit forward Euler form for the discretized energy equation for time step n and mesh node i is given by

$$T_i^{n+1} = \Delta t \left(-\frac{w_n}{\rho_0 A} \frac{T_i^n - T_{i-1}^n}{\Delta s} + \frac{Q}{L_h A \rho_0 c_p} \right) + T_i^n \tag{8a}$$

$$T_i^{n+1} = \Delta t \left(-\frac{w_n}{\rho_0 A} \frac{T_i^n - T_{i-1}^n}{\Delta s} - \frac{4U(T_i^n - T_s)}{D \rho_0 c_p} \right) + T_i^n \tag{8b}$$

$$T_i^{n+1} = \Delta t \left(-\frac{w_n}{\rho_0 A} \frac{T_i^n - T_{i-1}^n}{\Delta s} \right) + T_i^n \tag{8c}$$

Some details concerning the solution of the momentum equation (an ODE) are worth to mention as well. After time discretization, the momentum equation, in implicit form, is written as

$$\frac{w_{n+1} - w_n}{\Delta t} = -\left(f\frac{L}{D} + K\right)\frac{w_{n+1}^2}{2\rho_0 A^2} + \rho_0 g\beta \oint T^{n+1} dz \tag{9}$$

so that the solution w_{n+1} are the roots of a parabola, which means that two solutions will be available. Hence, a criterion to select one of the roots must be established. In most cases, a positive and a negative root will satisfy 9, and the positive one would be selected. But this is not a general case, because of the possibility of flow reversal. The criterion adopted in the present model was to select the root corresponding to the closest time derivative to that of the previous value (at time step n). Since the momentum equation has a quadratic form of the mass flow, friction losses are always acting in the same direction, which is not true if flow reverses. To overcome this limitation, the flow is always considered to be in the same direction, i.e., w_n is always positive during the solution procedure.

4. FRICTION FACTOR

Solutions for 3 and 7 are highly sensible to the friction factor correlation. Additionally, most single-phase NCL flows lye in the transitional regime (between laminar and turbulent ones), where friction correlations are more scarce. On this matter, Ambrosini *et al.* (2004) made a study comparing two friction factor correlations: Churchill's correlation, given by

$$\begin{aligned} f &= 8\left[\left(\frac{8}{\mathrm{Re}}\right)^{12} + (A + B^{-1,5})\right]^{1/12} \\ A &= \left\{-2,457\ln\left[\left(\frac{7}{\mathrm{Re}}\right)^{0,9} + \frac{0,27e}{D}\right]\right\}^{16} \\ B &= \left(\frac{37530}{\mathrm{Re}}\right)^{16} \end{aligned} \tag{10}$$

and the maximum between Poiseuille's and Colebrook's correlations, i.e.,

$$\begin{aligned} f &= \max(f_P, f_C) \\ f_P &= 64/\mathrm{Re} \end{aligned} \tag{11}$$

$$\frac{1}{\sqrt{f_C}} = -2\log\left(\frac{e}{3.7D} + \frac{2.51}{\mathrm{Re}\sqrt{f_C}}\right) \tag{12}$$

These two correlations are equal except for the transitional region, where Churchill's correlation presents a non-monotonic shape in the curve of f as function of Re (see fig.1).

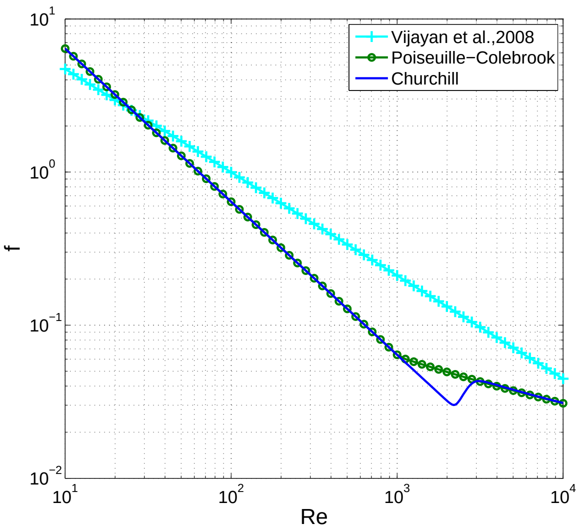

Figure 1. Comparison of friction correlations, taking into account $D = 26.9$ mm and $e = 10^{-7}$ (wall roughness).

Another aspect addressed by Ambrosini *et al.* (2004) was the applicability of forced flow correlations to natural convection systems. They mentioned that previous works (Creveling *et al.*, 1975; Vijayan and Austregesilo, 1994) have pointed that such correlations could be underestimated with respect to friction effects in natural circulation systems. Vijayan and Austregesilo (1994) have performed an experimental campaign and arrived at $f = 22.26\,\mathrm{Re}^{-0.6744}$. Indeed, there is a number of authors which employed friction factor correlations of the form $f = a\,\mathrm{Re}^{-b}$, a and b being positive constants (Naveen *et al.*, 2014). Attempting to cover all flow regimes, Vijayan *et al.* (2008) proposed adequate values of a and b, i.e.,

	a	b	flow regime
$\mathrm{Gr}\,D/L \leq 2 \times 10^5$	64	1	laminar
$2 \times 10^5 < \mathrm{Gr}\,D/L \leq 10^{10}$	22.26	0.6744	transition
$\mathrm{Gr}\,D/L > 10^{10}$	0.316	0.25	turbulent

Table 1. Selection criterion of constants for the friction factor correlation according to Vijayan *et al.* (2008).

where Gr is the Grashof number defined as $\mathrm{Gr} = D^3 \rho \beta g Q H (\mu^3 A c_p)^{-1}$, where H is the elevation difference between the centre line of heater and cooler. Figure 1 compares the correlations proposed by Vijayan *et al.* (2008) with those already mentioned here. Results for each of these three friction correlations – hereafter referred to as Churchill, Poiseuille-Colebrook and Vijayan *et al.* (2008) – will be shown in the following sections.

5. RESULTS AND DISCUSSION

5.1 Steady state

Experimental data presented by Naveen *et al.* (2014) will be taken as reference for the results of this work. The NCL under consideration has the following geometrical characteristics:

dimension	value	unit
internal diameter, D	26.9	mm
height, H	2.200	m
width, W	1.415	m
heater length, L_h	0.620	m
cooler length, L_c	0.800	m
heater orientation	horizontal	
cooler orientation	horizontal	
local pressure losses, K	1.800	

Table 2. Geometrical configuration of the NCL taken for simulations.

Figure 2 shows steady state results with the model using the three friction correlations described in section 4. Results are presented in the plane $\mathrm{Re} \times \mathrm{Gr}$.

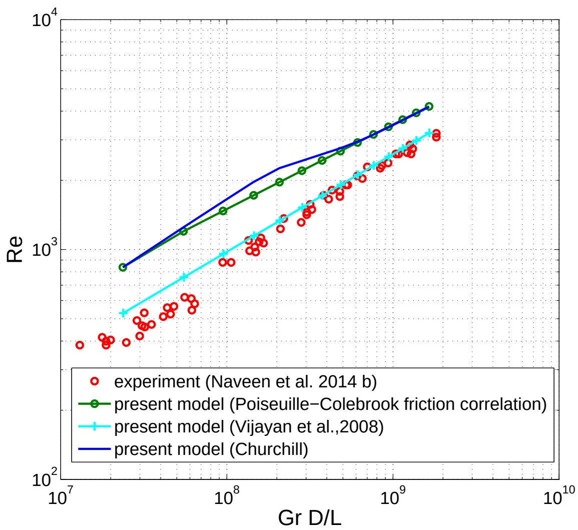

Figure 2. Steady state results considering the three friction correlations.

As expected, Vijayan's correlation fits best, though experimental points are a bit more distant for $\mathrm{Gr}\, D/L < 10^8$, $\mathrm{Re} < 10^3$, close to laminar regime, and despite table 1 suggests a "laminar" correlation only for $\mathrm{Gr}\, D/L < 10^5$. And as it can be expected as well, results for the other two correlations provide larger Re numbers, coherent with the values of f shown by fig.1, i.e., lower wall friction allows for larger mass flow. Additionally, the region corresponding to the transitional regime can be clearly observed in the curve for Churchill's correlation.

5.2 Transient state

Employing the implementation described in section 3, the transient behaviour of the NCL was simulated for a heat power of 220 W, same as presented by Naveen *et al.* (2014). The cooler was simulated by means of a constant heat transfer coefficient $U = 450$ W/m^2/K, for a cold source temperature of 30.4 ^{o}C. Figure 3 shows simulation results using two of the three friction correlations presented here: Vijayan's correlation produces stable results, whereas experimental data shows instability, so the corresponding results will be shown in a separate graph.

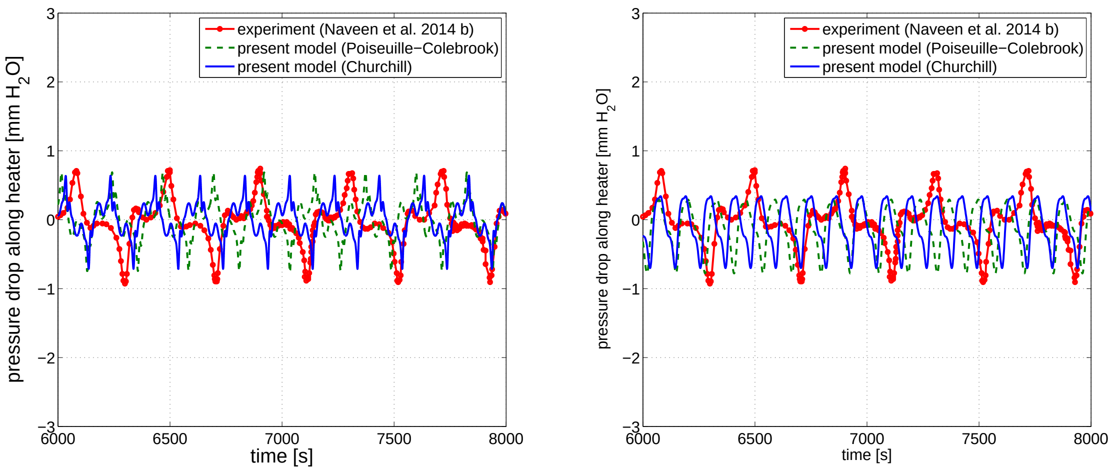

Figure 3. Comparison of predictions from the model with experimental data for 220 W heat power circulation, using Churchill and Poiseuille-Colebrook friction correlations with explicit formulation (left) and implicit formulation (right).

The left side graph of fig.3 shows that the model produces the same shape of oscillations but with different period compared to the experiment.0 The amplitude is qualitatively close, but predicted amplitudes are slightly underestimated for both correlations. It can also be observed that Poiseuille-Colebrook's correlation performed better than Churchill's. But once the steady state results already showed that these two friction factors, developed for forced flow, are smaller than that for natural convection, this result was expected.

An implicit formulation was implemented for the energy equation, with centred finite differences. The results are on the right side graph of fig.3, and predicted quite different oscillating behaviour compared to the experiment. Differently than in the explicit discretization, $\mathrm{CFL} = 10$ and $\Delta t_{\max} = 1$ were employed to take profit from the numerical stability of the formulation. This result endorses the strategy of employing upwind scheme with $\mathrm{CFL} = 1$ to avoid numerical diffusion.

Since the period of simulated results is smaller than that of the experiment, perhaps diffusion could bring predictions closer to the reference. That's what fig is showing. With $\mathrm{CFL} = 0.1$ the period increased a little, indicating that the smaller CFL, consequently smaller time step, compensates friction losses not accounted for with the used correlation.

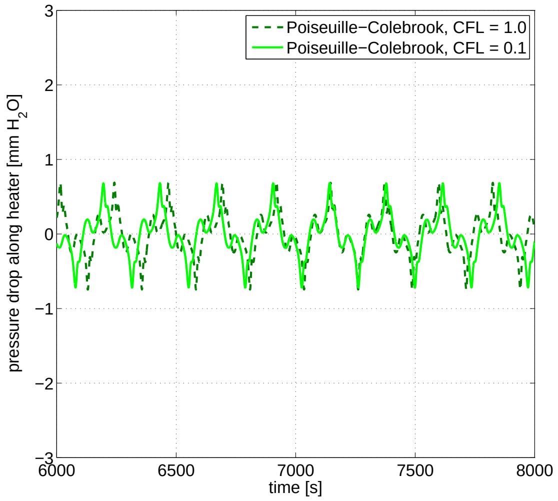

Figure 4. Comparison of Poiseuille-Colebrook's friction factor with CFL $= 1$ and CFL $= 0.1$.

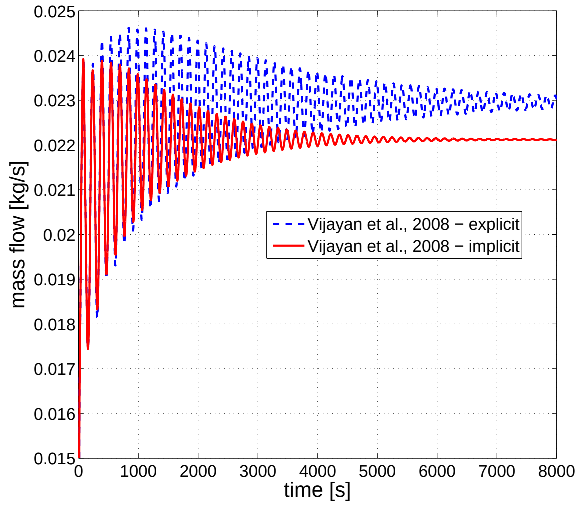

Figure 5. Transient mass flow simulated with Vijayan's correlation, using explicit and implicit formulations.

However, using Vijayan's correlation, the predictions resulted in stability for this heat power, both for explicit and implicit forms, as shown by fig.5.

5.3 Linear stability analysis

A linear stability analysis can provide advanced information about the system's stability, avoiding the computation time of the transients, and has been used in this work. The linear method consists on the decomposition of the unknowns in equations 3 and 7 into a mean component, which is the steady state solution, and a fluctuating component. After inserting the decomposed fields in the conservation equations, the quadratic terms of fluctuations are neglected, and the system of equation results in an eigenvalue problem, where the eigenvalues are complex numbers whose complex part is associated to the frequency of the oscillations and whose real part is the growth rate of the oscillation amplitudes. If at least one of the eigenvalues has a positive real part, then the system is considered unstable. To be stable, all oscillating modes must be damped along time, which means all eigenvalues must have negative real parts. The result of the linear analysis is shown by fig.6. Note that the curves are in accordance the other results, and note that, indeed, the analysis for Vijayan's correlation shows stability for heat powers up to 400 W.

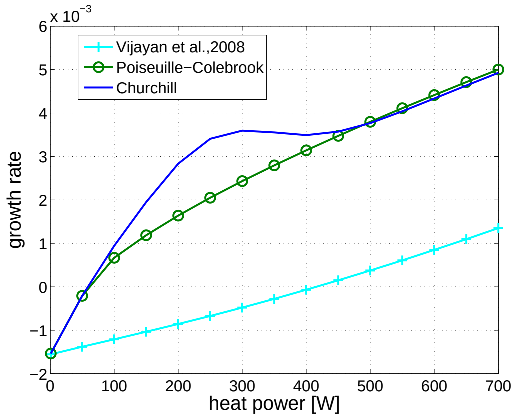

Figure 6. Results from linear analysis method of stability; the ordenates represent the largest real part of the eigenvalues.

6. CONCLUSIONS

This study raised the following conclusions:

- The high dependence of friction factor correlation was confirmed in the present model;
- two forced flow correlations performed better than the one developed from NCL experiment;
- the correlation which performed well for steady state may produce unrealistic result in the transient calculation;

- from the point of view of systems design, the forced flow correlations employed in this work – Churchill's and Poiseuille-Colebrook's – have indicated to be reliable and provided a level of conservatism;
- Vijayan's correlation (described by table 1) is applicable to horizontal heater and cooler. Other orientations should be modelled by different correlations;
- The use of $\mathrm{CFL} = 1$ is suggested since it imitates a Lagrangian scheme and therefore minimizes numerical diffusion due to time advance;
- A special algorithm had to be implemented in order to prevent the time increment from growing too large when the flow approached the reversal.

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