



UNIVERSITÀ DEGLI STUDI DI TRIESTE

XXX CICLO DEL DOTTORATO DI RICERCA IN

Ingegneria e architettura
curriculum: Ingegneria meccanica, navale, dell'energia e della produzione

Finanziatore borsa di studio: Electrolux Professional Spa

ENERGY EFFICIENCY IMPROVEMENT IN PROFESSIONAL OVENS

Settore scientifico-disciplinare: ING-IND/08

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ANNO ACCADEMICO 2016/2017

To Viviana and Mariazzurra : the most important people in my life

Acknowledgments

Primary, I would like to express my deepest and sincere gratitude to Professor Diego Micheli, not only because he has believed in me, teach me and motivated, but also for his continuous presence during these three years. It has been a real pleasure and honor to work with him. These results and this thesis would not have been possible without him, his guidance and advices. Thank you a lot!

It gives me immense pleasure to express my sincere thanks and gratitude to Dott. Eng. Michele Simonato for his support, continuous guidance and for his inexhaustible patience during these three years. I would also express my gratitude for the second possibility that he gave me believing that I could also contribute to The Research Hub. All the discussion we had, made me grow and becoming the professional that I am. Thank you a lot!

I would like to thank gratefully Dott. Eng. Riccardo Furlanetto for giving me the opportunity to work at The Research Hub, as well as for his interest in these studies. Thank you very much for the continuous support and for all the passion you have transmitted to me. You are the person I look with a lot of admiration because you can manage, motivate and to smile even if you have so many thoughts considering your role. Thank you a lot!

I owe my gratitude to all the partners and employers involved in "The Research Hub", which has been funded by Electrolux Professional Spa and has constituted the framework of most of the activities conducted during my PhD program. In particular I would express my gratitude to Valentino Cucit, Nicola Battistella and Alessandro Morassut. I wish to express my gratitude to all my friends, former colleagues and mates not cited here, for their support during all my doctoral period.

I owe my gratitude to Prof. Romano Lapasin for helping me in conduction the rheological tests on Polyox WSR 301.

I owe my most warm and sincere gratitude to all my family, and in particular to my parents, Alessandra, Adriano and Eleonora for their help in each moment.

Finally, I reserve a special thank to the most important people of my life: Viviana and Mariazzurra. Thank you for helping me to overcome all the difficulties arisen in these years and for your support during these years. Thank you to both of you for the patience and understanding, in particular during the working weekends! Thank you for all the help! You are simply the most important people in my life. You are the best. To both of you I dedicate this thesis.

Abstract

The key priority for Electrolux is to be a leader in product energy efficiency. Electrolux Professional Spa, in order to satisfy this requirement has financed my PhD study in 'The Research Hub'. With the aim of reducing energy consumption and improving heat efficiency of professional appliances, this thesis is the results of the investment Electrolux Professional Spa has made. The research presents an analysis of the state of the art of the energy standards applied to professional ovens. It continues, identifying a methodology based on the first principle of thermodynamics for evaluating the energy efficiency of a combined oven structured on the experimental analysis of the balance of fluxes in coming and out coming from the oven in different cooking modalities. It emerges that the thermal insulation and the washing system are among the technical solutions with higher potential energy savings. For studying the insulation, a lumped capacitance model of a professional oven has been developed, characterized by the interactions between two thermal zones, i.e. the power zone and the cooking zone, and by the adoption of a not linearized radiation heat transfer approach. The results have been compared with a set of experimental data showing a good agreement in both the transitory and the stationary operating phases.

For studying the energy and water consumption of the oven washing plant, different benchmarks have been compared. It appears that the closed washing circuit configuration is the best solution in terms of water and energy consumption. With the aim of designing a new washing circuit, a numerical model has been developed. This analysis has been followed by a CFD analysis in order to evaluate the air mass flow rate losses due to two possible washing pipes entering the suction duct. During the experimental test on a prototype of the oven washing circuit, a possible cavitation problem has appeared. Pump processes a solution of water and detergent at 70 °C. High concentrated chemistry could impact on the working conditions of the pump and on cavitation. The wide range of variables affecting the phenomenon has led to the development of a laboratory rig.

This test rig permits measuring pump performances at various operating conditions, in order to obtain its characteristic curves, and also forcing cavitation to measure its Net Positive Suction Head required (NPSHr) at different flow rates. The pump test rig allows also testing various configurations of the pump at differ-

ent cavitation conditions, obtained by changing not only the suction pressure and temperature of the fluid but also its properties, adding detergents and additives. A representative chemical component present in detergents (Polyox WSR 301) has been selected and tested at different concentrations in the rheometer in order to identify the concentration values at which the solution switch from the diluted to the concentrated regime. For each solution, the resulting performance curves of the pump are then compared with those obtained with pure water. Furthermore, a method for detecting cavitation inception has been developed, based on a deep signal analysis of vibrations. The influence of different concentrations of Polyox WSR 301 on pump vibrations induced by cavitation has been studied. The results are quite impressive, because with the increasing of the concentration of Polyox WSR 301 there is a decreasing pump vibrations in the cavitation frequency range. In literature there are no studies analyzing the impact of detergent components on pump vibration induced by cavitation.

Sommario

La priorità chiave di Electrolux è essere leader nell'efficienza energetica dei prodotti. Electrolux Professional Spa, al fine di soddisfare questo requisito ha finanziato la mia borsa di dottorato all'interno del 'The Research Hub'. Questa tesi è il risultato dell'investimento effettuato da Electrolux Professional Spa, avente l'obiettivo di ridurre il consumo di energia e migliorare l'efficienza termica delle apparecchiature professionali. La ricerca presenta un'analisi dello stato dell'arte degli standard energetici applicati ai forni professionali. Continua, individuando una metodologia basata sul primo principio della termodinamica per la valutazione dell'efficienza energetica di un forno combinato strutturata sull'analisi sperimentale dell'equilibrio di flussi in entrata ed uscita provenienti dal forno in diverse modalità di cottura. Emerge che l'isolamento termico e il sistema di lavaggio sono tra le soluzioni tecniche ad alto potenziale di risparmio energetico. Per studiare l'isolamento, è stato sviluppato un modello a parametri concentrati di un forno professionale, caratterizzato dalle interazioni tra due zone termiche, vale a dire la zona di potenza e la zona di cottura, e dall'adozione di un approccio di trasferimento di calore radiante non linearizzato. I risultati sono stati confrontati con una serie di dati sperimentali che mostrano un buon accordo sia nelle fasi operative transitorie che stazionarie.

Per studiare l'energia assorbita ed il consumo d'acqua dell'impianto di lavaggio del forno, sono stati confrontati diversi 'benchmarks'. La configurazione del circuito di lavaggio a circuito chiuso è la soluzione migliore in termini di consumo di acqua ed energia. Con l'obiettivo di progettare un nuovo circuito di lavaggio, è stato sviluppato un modello numerico. E' stata effettuata anche un'analisi CFD al fine di valutare le perdite di portata massica d'aria dovute a due possibili tubi di lavaggio che entrano nel condotto di aspirazione. Durante il test sperimentale su un prototipo del circuito di lavaggio del forno, è apparso un possibile problema di cavitazione. La pompa elabora una soluzione di acqua e detergente a 70 ° C. La chimica altamente concentrata potrebbe avere un impatto sulle condizioni di lavoro della pompa e sulla cavitazione. L'ampia gamma di variabili che influenzano il fenomeno ha portato allo sviluppo di un impianto per testare le pompe.

Questo banco di prova consente di misurare le prestazioni della pompa in varie condizioni operative, al fine di ottenere le sue curve caratteristiche, e anche di forzare

la cavitazione per misurare la prevalenza netta di aspirazione positiva della pompa (NPSHr) a diverse portate. Il banco prova della pompa consente anche di testare varie configurazioni della pompa in diverse condizioni di cavitazione, ottenute cambiando non solo la pressione di aspirazione e la temperatura del fluido ma anche le sue proprietà, aggiungendo detergenti e additivi. Un componente chimico rappresentativo presente nei detergenti (Polyox WSR 301) è stato selezionato e testato a diverse concentrazioni nel reometro per identificare i valori di concentrazione a cui la soluzione passa dal regime diluito a quello concentrato. Per ciascuna soluzione, le curve di prestazione risultanti della pompa vengono quindi confrontate con quelle ottenute con acqua pura. Inoltre, è stato sviluppato un metodo per rilevare l'inizio della cavitazione, basato su un'analisi approfondita del segnale di vibrazione. È stata studiata l'influenza delle diverse concentrazioni di Polyox WSR 301 sulle vibrazioni della pompa indotte dalla cavitazione. I risultati sono piuttosto impressionanti, perché con l'aumento della concentrazione di Polyox WSR 301 si hanno vibrazioni della pompa decrescenti nell'intervallo di frequenza di cavitazione. In letteratura non ci sono studi che analizzano l'impatto dei componenti dei detergenti sulle vibrazioni della pompa indotte dalla cavitazione.

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Introduction

The challenge of energy sustainability represents one of the most important goals to be achieved by humanity in 21st century. The actual energy situation is crucial because there are certain aspects that must be considered, as the increasing global energy demand, the green house gas emission, the fossil fuel reserves and the water demand.

The global energy demand is expected to grow up dramatically up to 2040, due to both the new demands from emerging countries and the life-style changes in developed countries. In particular, the global energy demand is forecasted to increase of the a 30% to 2040 [1]. The energy related green house gas emission has increased constantly over the past decades and the fossil fuel reserves are sufficient only for a relatively limited period, then the extraction costs will be higher. Also the demand for water is forecasted to increase of the 40% by 2030 [2]. In order to mitigate energy expenditures, air pollution, water consumption and global warming, Electrolux has defined a roadmap till 2020. This roadmap tackles the urgency of climate change, chemical pollutants, access to water, energy and raw materials because they have a big impact also on the public perception of the company [2]. In particular, considering the market share all over the world, Electrolux, which sells annually over 60 million of home appliances [2], has a significant impact on society and environment, for which it can be considered as an industry involved in the climate change issue. With the aim of helping consumers and customers to live better lives, Electrolux is focusing on creating more efficient, high-performance appliances [2].

The Electrolux roadmap till 2020 states [2] that they want to be leader in product efficiency in their most important markets by 2020 and continuing to develop products with good environmental performances, focusing on energy efficiency. They also want continuing to drive the market for efficient products through awareness-building customer and consumer campaigns.

A key priority for researches in the R&D processes, in particular for me as a PhD student financed by Electrolux Professional Spa, is to tackle climate change by developing appliances that reduce energy consumption and greenhouse gas emissions and by saving water consumption, for example creating a washing system that is

more water efficient than previous solutions and in respect to washing dishes by hand.

This thesis deals with energy efficiency in professional appliances, in particular it considers Electrolux professional ovens as representative device for performing the study. During the analysis, a particular emphasis has been made also on other professional appliances components as the rack type dishwasher pump. The main aim of the work is to analyze the actual energy situation in professional appliances, to achieve a better knowledge on energy efficiency improvements techniques by means of literature analyses, numerical simulations, experimental activities and developing advanced components for energy efficient appliances. During the development of an efficient washing system for a professional oven, the necessity has arisen to study cavitation in centrifugal pumps processing a solution of water and detergent. The study has led *i)* to the selection of a component, the Polyox WSR 301, as representative of the chemical composition of detergents, *ii)* to the analysis of the rheological behavior of its solutions with water and *iii)* to the experimental evaluation of the impact of these solutions on pump performances and vibrations. This study will be the foundation for developing a sensor and a control system on a professional appliance in order to safer the pump from cavitation risks.

The thesis is organized according to the following outline:

Chapter 1 contains a detailed introduction on professional ovens, with an analysis and comparison of the energy efficiency standards EFCM, ENAC and ASTM, showing discrepancies among experimental results. The chapter continues with a systemic application of the mentioned standards to a specially instrumented prototype of professional oven. A new methodology for the energy efficiency evaluation of a combined oven is also presented. It is based on the first thermodynamic principle. The chapter concludes with an analysis and application of some technical solutions to a prototype.

Chapter 2 presents a numerical model of the transient thermal behavior of an oven. The developed code is based on a lumped capacitance method and had the aim of studying the behavior of the oven's insulation. The chapter shows that there are only few literature studies presenting an oven transient thermodynamic simulation by means of a lumped capacitance method. The code is also validated by comparison with a set of experimental data.

Chapter 3 describes the methodology used for developing an efficient oven washing system. At first, a comparison among different washing system representative devices has been carried out. The comparison shows that closed washing system is the best solution in terms of energy, water and detergent consumption. The analysis continues with a numerical model in order to calculate the pump operating conditions and to calibrate the washing plant circuit. It follows a CFD analysis of a scheme for

the washing plant pipes needed to allow the recirculation of the water, detergent and soil solution and the entrance of the fresh water. The chapter concludes with some considerations regarding the selection of a pump for an hydraulic washing circuit. In particular, due to the actual trend of increasing the chemistry in the washing processes, the need of a full study on cavitation in centrifugal pumps operating with different fluid solutions has emerged.

Chapter 4 contains a literature review of the possible effects of detergent components on pump performances and cavitation. The chapter continues with a detailed description of the newly realized test bench for characterizing pump performances and cavitation behaviour. Then it follows an analysis of the pump performances and vibrations with water solutions having different concentrations of the polymer Polyox WSR 301. This analysis shows that with the presence of this chemical component, chosen as representative of detergents, pump vibrations due to cavitation are lower than that obtained with pure water and that they are increasing with decreasing polymer concentration. The chapter concludes with the identification of the possible parameters of interest for developing a pump control system against cavitation.

Conclusions give a comprehensive overview of the research outcomes generated by the aims declared in the introduction of the thesis, with comments on the accomplished results and possible future developments.

Chapter 1

Energy efficiency improvement in professional ovens

1.1 Introduction

The challenging worldwide energy demand requires the economical and technical development of alternative energy sources and a restriction of the energy consumption by means of systems and machines that are more efficient. This permits an optimal management of energy fluxes. Food-service facilities, having an average energy use almost three times higher than other commercial activities, can be considered as an energy intensive field and, consequently, a sector with significant potentials for energy efficiency improvements. Moreover, eighty percent of the annual energy bill expenditures for commercial food services are wasted due to the use of inefficient equipments [3]. Energy performances of consumer and professional appliances are receiving more attention in the product development. For example, the European Community SAVE programme has promoted the efficient use of energy, in particular in domestic appliances [4]. This chapter focuses on professional ovens, which consume a large amount of energy. They need to satisfy high quality standards, high adaptability and reliability but they still do not have worldwide recognized standards for energy classification. The first purpose of this study is to analyse three test procedures for cooking appliances: EFCEM, ENAK and ASTM. They differ in the test methodology, load types, load conditions and on the definition of energy efficiency parameters. The analysis has highlighted an impossible comparison among the results and a consequent difficult evaluation of the energy efficiency of the oven. The subsequent step of the work presented in this chapter is the identification and the development of a detailed methodology for analysing the energy efficiency of the oven. The result of the methodology is then a guide in the identification of improved design technical solutions. Some of these are finally applied, showing remarkable results in the overall energy efficiency of the oven.

1.2 Energy efficiency tests comparison

Test procedures define a structured methodology for the evaluation of energy performances of professional combined ovens. They consider measurements, in different cooking modalities, of water consumption, energy consumption and variation of weight of the loads that are inside the cavity. An analysis of the parameters of interest is presented in [5]. For a better understanding of the thermodynamic behavior of the oven, a comparison among the following energy test procedures has been carried on:

- EFCEM [6]: it evaluates energy consumption in two cooking modalities: convection and steam. The thermal loads are fifteen water-saturated bricks in convective test and water-filled trays without lid in steam test;

- EFCEM [7]: it evaluates the energy efficiency for combined cooking mode. The thermal loads are water-filled trays with lids having a hole in the center;
- ENAK/SVGG [8]: it evaluates the energy consumption. The thermal loads are either water-saturated bricks or water-filled trays;
- ASTM F2861-10 [9]: it evaluates the energy efficiency. The thermal loads are potatoes.

A combined electric oven Electrolux AoS Touchline, type 10 GN 1/1 LW level, was used for the tests. It has a declared power of 17 kW in convection and steam mode, the internal cavity has a volume of 0.35 m^3 . The comparison was made on the energy efficiency yield, which is defined for an oven as the ratio between the heat given to the thermal load and the energy introduced in the system. The heat given to the thermal load can consider the presence of trays and lids or not. From Eq. 1.1 to Eq. 1.4 are presented different definitions of energy efficiency yields:

$$\begin{aligned}\eta_1 &= \frac{(m_{load} \cdot c_{load} \cdot \Delta T_{load})}{E_{el} + E_{GN} + E_{Lid}} = \\ &= \eta_{EFCEM,combi}\end{aligned}\tag{1.1}$$

$$\eta_2 = \frac{(m_{load} \cdot c_{load} \cdot \Delta T_{load})}{E_{el}}\tag{1.2}$$

$$\begin{aligned}\eta_3 &= \frac{(m \cdot c_{load} \cdot \Delta T_{load}) + (m_{tray} \cdot c_{tray} \cdot \Delta T_{tray})}{E_{el}} = \\ &= \eta_{ASTM,steam}\end{aligned}\tag{1.3}$$

$$\begin{aligned}\eta_4 &= \frac{(m \cdot c_{load} \cdot \Delta T) + (m_{tray} \cdot c_{tray} \cdot \Delta T_{tray}) + (\lambda \cdot \Delta m_{load})}{E_{el}} = \\ &= \eta_{ASTM,conv}\end{aligned}\tag{1.4}$$

Enak standard doesn't define any efficiency yields as the other procedures. This procedure defines how to measure the energy consumption in the cooking phase. Energy yield η_1 considers the energy supplied to the equipment minus the energy absorbed by the mass of the pans and the lids. Energy yield η_2 is introduced only to evaluate the contribution on the yield η_1 of the energy absorption by pans and lids.

The results of the analysis on the test procedures are reported in Tab. 1.1 [5], where the values of energy yields in respect to the different test methodologies are reported.

Cooking Mode	Test Method	η_1	η_2	η_3	η_4
Conv.	EFCM	41.0	41.0	44.3	83.4
	ENAK	24.7	24.7	25.6	68.2
	ASTM	32.4	32.5	33.9	88.5
Steam	EFCM	68.3	68.2	72.9	\
	ENAK bricks	21.4	21.4	22.5	\
	ENAK water	64.5	64.7	68.8	\
	ASTM	38.3	38.2	41.0	\
Combi	EFCM	59.5	59.8	65.8	69.9
	ENAK	44.2	44.2	47.5	89.6
	ASTM	\	\	\	\

Table 1.1: Energy yield [5]

The energy efficiency yields calculation in the combined cooking mode is not considered in ASTM conditions, that take into account only the temperature variability from one pot to another (cooking uniformity), using ice as load. Parameter η_4 was not calculated for the steam cooking mode, because weight losses due to the evaporation of the loads are hidden by the increased weight from the condensation of the steam, used as heating vector. Comparing the columns of Tab 1.1, it is possible to analyse the different definitions of energy efficiency yields. Instead, making a comparison between the rows of Tab 1.1, it is possible to analyse the different sets for each standard procedure. The small differences between η_1 and η_2 , calculated for the same test, show that the influence of weights and trapped energy of trays and lids is negligible. The values calculated for η_1 and η_3 show small differences and highlight that the different methods for considering the energy adsorbed by the trays do not significantly influence the results. Comparing η_3 and η_4 , it is possible to examine the impact of the latent heat of evaporation.

1.3 Oven energy characterization

The energy fraction absorbed by the food during cooking is low because a large portion of energy goes into the structure of the oven (e.g. walls, door and insulation), and is lost in the surrounding environment [10]. High-emissivity linings absorb the thermal radiation energy from the cavity and then it is lost through conductive bridges and convective leaks [11]. Furthermore, a lot of energy is lost through the venting of the evaporated moisture from the cavity. The analysis of the data from the test procedures show the need of a more comprehensive testing methodology for defining and calculating the balances of energy, which are required for a better characterization of the thermodynamic behaviour of the oven and to have an easier procedure for guiding the design phase. For calculating energy balances, the versions “6” of the draft EFCM was integrated with a series of measurements needed for characterizing the energy fluxes not considered in the test procedures. The aim of the measurements is the determination of the enthalpy and energy content of the fluxes which enter and leave the oven considered as a control volume [[12]- [13]]. Considering the schema of an oven as in Fig.1.1, it is possible to identify the power supplied (electricity or gas) and the fluxes of air inlet, water inlet, water discharge and exhaust fumes.

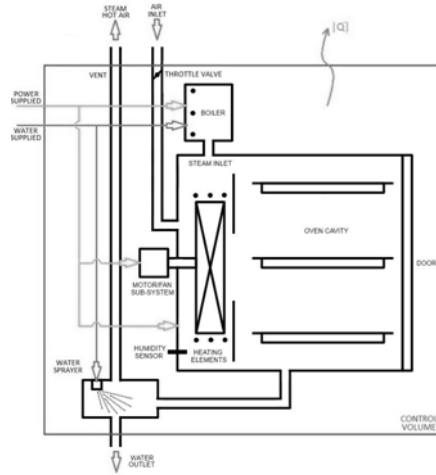


Figure 1.1: Oven schema

For calculating the balances of mass and energy two cooking modalities are considered [14]:

- Convection: in the center of the cavity a temperature of 160 °C is set and 15 water saturated bricks are used for the tests.
- Steam: in the center of the cavity a temperature of 100 °C and a relative humidity of 100% are set. Ten water trays with a welded top are used for the load. The welded top has a hole at the centre. Basically the oven during the cooking process

changes its thermodynamic state. It is not possible to consider steady state conditions, because parameters as work, energy and mass fluxes continuatively change during the test [15]. Generally speaking, the mass balance applied to a control volume during a transient period is given by Eq. 1.5, where the subscript i indicates the inflows, e the outflows, and the subscript cv the control volume.

$$\begin{aligned}
 \Delta m_{CV} &= \int_0^t \left(\frac{dm_{cv}}{dt} \right) dt = m_{CV}(t) - m_{CV}(0) = \\
 &= \int_0^t \left(\sum_i \dot{m}_i \right) dt - \int_0^t \left(\sum_e \dot{m}_e \right) dt = \\
 &= \sum_i \dot{m}_i - \sum_e \dot{m}_e
 \end{aligned} \tag{1.5}$$

The mass balance can be written in a more compact form as in Eq. 1.6 with reference to the convective modality test, and as in Eq.1.7 when steam modality is taken into account.

$$\Delta m_{CV} = |m_{WaterInlet} + m_{LoadWeightLoss} - m_{CondensedVent} - m_{LiquidDischarge}| \tag{1.6}$$

$$\Delta m_{CV} = |m_{WaterInlet} + m_{LoadWeightLoss} - m_{CondensedVent} - m_{LoadWeightGained}| \tag{1.7}$$

If the variations of potential and kinetic energy between inputs and outputs are negligible, the balance of energy fluxes is represented in integral form as in Eq.1.8 :

$$\begin{aligned}
 \Delta U_{CV} = U_{CV}(t) - U_{CV}(0) &= Q_{CV} - W_{CV} + \sum_i \left(\int_0^t \dot{m}_i h_i dt \right) - \\
 &\sum_e \left(\int_0^t \dot{m}_e h_e dt \right)
 \end{aligned} \tag{1.8}$$

If the initial and final test conditions are the same, the internal energy of the system does not change, and Eq. 1.8 can be further simplified as:

$$Q_{CV} - W_{CV} = \sum_e \left(\int_0^t \dot{m}_e h_e dt \right) - \sum_i \left(\int_0^t \dot{m}_i h_i dt \right) \tag{1.9}$$

With reference to the oven schema reported in Fig.1.1, Eq. 1.9 can be written as follows:

$$E_{vent} = E_{el} - E_{aux} - E_{load} - E_{liq} - E_{wall} - E_{door} + E_w \quad (1.10)$$

In Eq. 1.10 the energy adsorbed by the load in the cavity at the end of the cooking process, E_{load} , is given by Eq.1.11, where the specific heat of the load is determined with a weighted average as $c_{load} = \sum c_i x_i$.

$$E_{load} = (m_{load} \cdot c_{load} \cdot \Delta T_{load}) + (m_{tray} \cdot c_{tray} \cdot \Delta T_{tray}) \quad (1.11)$$

As example are reported the pie charts representing the percentage distribution of the energy fluxes in convection (left figure Tab. 1.2), and steam mode (right figure Tab.1.2).

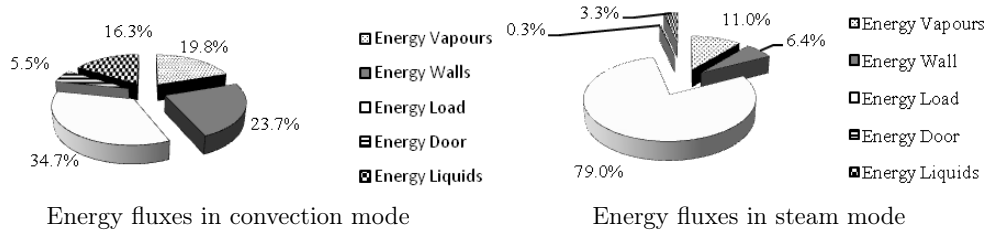


Table 1.2: Energy fluxes

Energy absorbed by the load is measured with Eq.1.11 weighing the mass after the cooking process and measuring the temperature in the centre of the load before and after the cooking process. In this measurement is considered also the energy given to the trays. Energy dissipation due to the opening of the door is also measured. During one hour test, three door openings, which last in three minutes, are made every twenty minutes. The resulting energy loss through the door is calculated by the difference between the power consumption in one hour when the door is opened and closed and the power consumption of maintenance. The last one is the energy rate dissipated through the walls, and is calculated with a measurement of the power consumption at a certain temperature on an un-loaded oven, in convective cooking mode. Energy introduced and dissipated by the liquids is calculated on the bases of temperatures and flow rates data and integrated by means of a proprietary program [16]. Energy loss through vapors is calculated by means of Eq.1.10.

1.4 Technical solutions for energy efficiency improvements

The proposed methodology has highlighted some technical solutions, capable to guarantee significant energy savings, an overall acceptance by the customers and also

a quite cheap industrialization. They are listed in Tab. 1.3, which is organized as the ones reported in [4], with reference to domestic ovens, to allow a comparison with the professional appliances here considered.

	Design Option	Energy saving	Consumer Response	Test in prototype
1	Improve thermal insulation (a,b)	0-11	Acceptable	Yes (Deeply investigated in chapter 2)
2	Improve cavity thermal insulation	7-8	Acceptable	Yes
3	Reduce mass of oven structure	10-18	Acceptable	Yes
4	Unglazed door	7-25	Unacceptable	No
5	Optimized glazed door design	4-12	Acceptable	No
6	Passive cooling for glazed door	0-8	Acceptable	No
7	Optimised vent flow	8 or 12	Acceptable	No
8	Aluminium foil on cavity walls	7-10	Acceptable	No
9	Reduce cavity volume	0-4	Acceptable	Yes
10	Reduce cavity opening access	0-4	Acceptable	No
11	Control with smaller oscillations	15	Acceptable	Yes
12	Reduce auxiliary energy	1-4	Acceptable	No
13	Revision of the washing system*	42-53	Only in professional appliances	Yes (results present in chapter 3)

Table 1.3: Technical solutions

For each design option, the potential energy savings are represented in column 3. These values derive from [4], except the last row that derives from an internal benchmark analysis (section 3.4.2 in the thesis). The fourth column represents the customer response to a new implemented solution, because an oven can be efficient but, if it is perceived as dangerous or the cooking performances are not good as before, the implemented new technical solution would be unacceptable for the customer. In this analysis, the interactions between design options have not been taken into account, even if it is known that energy saving is not a linear process. This means that the result of combined technical solutions could not simply be the sum of the effects of every single improvement. Solution 1 from Tab.1.3 can be achieved through the use of an additional layer of a low-cost standard material (solution a) or it can be constituted from a single layer of high performance material with higher cost (solution b) [17]. This solution will be explored in detail in chapter 2, simulating the transient thermal behaviour of the oven with the aim of evaluating oven energy efficiency improvements due to different insulation solutions.

Solution 2 from Tab.1.3 aims to reduce thermal bridges between the cavity and the rest of the oven structure with Teflon support elements (PTFE) [17].

Some of such technical solutions have been until now applied to an oven prototype, as indicated in the last column of Tab. 1.3. In Tab.1.4 the results of the analysis applied to the oven prototype during the cooking are represented and compared with the corresponding data of the actual oven One generation 10-1/1. The comparison is made with reference to the power consumption for the maintenance, the energy given to the structure and the cavity volume.

*This solution is not contemplated in the bibliographic reference but is a result of the analysis performed inside Electrolux Professional. This technical solution will be explored in detail in chapter 3, in particular considering the energy consumption, water consumption of the oven washing plant

The values are calculated from plot of measurements as the one represented in Fig.1.2. In abscissa is represented the time length and in ordinate are reported the measured temperatures. This measurements have been performed setting a convection cooking program with a fixed temperature inside the cavity of 180 °C.

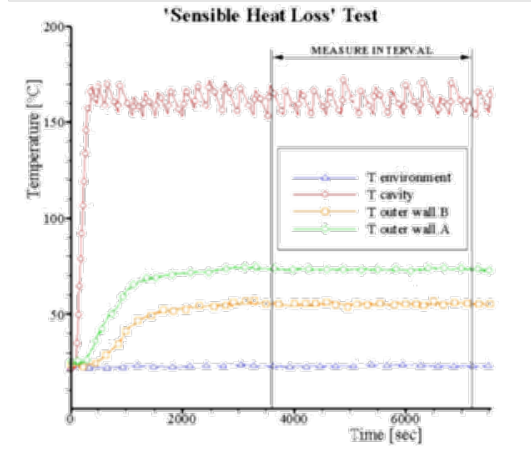


Figure 1.2: Plot of measurements

The values are detected in the center of the cavity and on its outer walls, in the locations where temperatures resulted to be higher. The temperature behaviour permits to study the energetic performances of the oven, characterized by different phases [18]. The energy consumed in the first hour represents the oven performances in the transition phase. After a transition phase, the oven reaches the operating conditions. The second hour of Fig.1.2 identifies the energy needed for maintain the operating conditions. The difference between the energy consumed in the first hour and the energy consumed in the second hour represents the energy absorbed by the structure. The cavity volume is an important factor affecting the energy performances. Small cavity volumes have less dispersion of energy and permit higher energy efficiency. A comparison between the results of the analyses on the oven One Generation 10 1/1 and the prototype is presented in the fourth column of Tab.1.4 .

	One 10-1/1	Prototype 10 1/1	%Difference
Energy for maintenance [KJ]	5595	3960	-29.2
Energy given to the structure [KJ]	3924	3285	-16.3
Cavity Volume [m^3]	0.35	0.316	-9.7

Table 1.4: Results of the Analysis

The results are remarkable because there is a 29.2% lower required energy for maintenance of the operating conditions in the prototype and a 16.3% lower energy given to the structure. The cavity volume in the prototype is 9.7% lower.

1.5 Conclusions

The methodological approach presented in this chapter is divided in three phases. In the first phase three different test procedures EFCEM, ENAK and ASTM were compared. These test procedures show differences in the settings and in the definition of the energy efficiency yields. In the second phase a methodology based on balance of fluxes entering and going out from the oven control volume is developed considering different cooking modes as the convection mode and steam mode. This methodology permits a fundamental understanding of the thermodynamic behaviour of the oven in respect to the results coming out from the application of the general procedures and standards. With the developed methodology is possible to establish an energy efficient design and to identify possible technical solutions for reach an efficient energy implementation. In the third phase some technical solutions were applied to an oven prototype and then analysed experimentally showing 29.2 % of energy savings. The first results coming out from the methodology applied to the oven prototype has present a lot of potentials in terms of energy efficiency improvements. The results presented have pushed the necessity of deeply investigating two of the most effective technical solutions : the oven thermal insulation and the oven washing system. In particular in the following chapter 2 is presented a simulation model of the behaviour of the oven during the transitory and steady state phases. With this code, it is possible to support the design phase of both the structure and the control strategy of the oven. It permits, for example, to get a general understanding of the best possible configurations and combinations of insulation materials for the cavity walls or, with reference to the control strategy, to simulate different cooking procedures, with the aim of optimizing the operating sequence of the oven, reaching the maximum energy saving without reducing the cooking quality. The code, validated by comparison with a set of experimental data obtained with a current production model, will be applied in the design phase of a new line of high efficiency professional ovens. In chapter 3 instead is presented the analysis of the possible efficiency improvements in the oven washing system in terms of energy, water and detergent consumption, in particular considering a closed washing solution.

Chapter 2

Thermal insulation analysis

2.1 Introduction

The current global environmental situation requires technical efforts for designing new systems and machines capable to minimize the overall energy consumption. Professional kitchens, having an average energy demand almost twice the amount used for domestic cooking, has a substantial energy saving potential. To meet the increasing demand for more sustainable kitchen products, it is necessary to develop suitable design strategies, based on theoretical models experimentally validated according to rigorous and significant test protocols on a prototype. The models should take into account not only steady state operating conditions, but also the transient behaviours of the device, which must be described with specially developed theoretical dynamic models. In fact, transitory situations are very common and time consuming during the operating life of a professional cooking appliance: the operating profile of an oven, for example, consists of a sequence of unsteady phases (cavity heating-up, food introduction and extraction, switching from one cooking mode to another) interspersed with steady cooking steps. To characterize the energy performances of the system it is necessary to identify and quantify the incoming and outgoing energy flows from the device, including the evaluation of internal and external heat transfers during both the transitory and the steady state phases. The scope of such an analysis is not only to do an energetic performance evaluation of a particular appliance but, more generally, to give support to the design phase, the material selection, the cooking phases evaluation and the control logic definition from the beginning. The aim of this chapter is the definition and validation of a model of a professional oven. Several studies dealing with the modelling of an oven can be found in the open literature: they can be organized according to five possible approaches, i.e. three or two dimensional CFD models, system modelling for control purposes, algebraic modelling and applications of the lumped capacitance method, as briefly discussed in the following review. The most accurate and detailed 3D-CFD approach is adopted in [19], [20], [21]. In [19] authors present a model of an electric bread-baking oven, comparing the internal temperature profiles obtained with three radiation models.

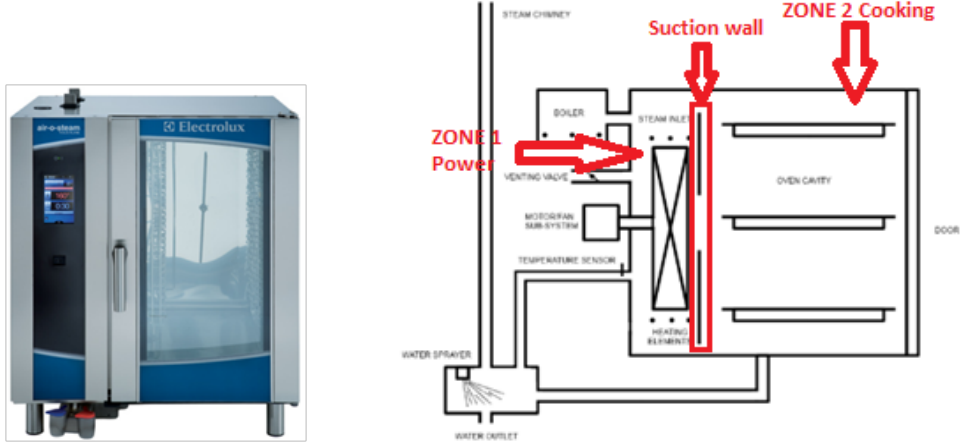
In [20] the authors evaluate heat transfer and fluid flows inside a domestic gas oven. Reference [21] presents the model of a pilot scale convection oven, used to study the reduction of the energy consuming pre-heating time. All these models are validated by comparison with experimental data with fairly limited discrepancy, also at different thermal levels and geometries of the internal baffle plate [21]. Other authors have analysed the thermal behaviour of an oven by means of 2D-CFD numerical models. In [22], an electric static oven, used for bread baking, is analysed to calculate the heat exchanged with the product. Authors present a comprehensive methodology for evaluating the heat fluxes around the test material, considering natural convec-

tion, infrared radiation and conduction through a cement slab. Heat equations are solved on a cross section of the cavity, for all components of the oven, with the finite element method, using a parametric solver. Numerical results are in good agreement with heat flux measurements on the upper surface of a polymeric cylindrical sample. The scope of the model was to test the influence on energy consumption of different operating conditions, obtained lowering the cooking temperature, increasing the incident radiative heat flux and lowering the thermal capacity of the oven. Another approach, used in the designing of control systems, considers simple thermal models, able to describe the temperature dynamics of an oven cavity. In [23], the authors present a set of mathematical models, which relate the input power and the air temperature inside a forced convection oven, based on experimentally determined transfer functions. A modelling approach based on simple algebraic models is presented in [24]. The authors predict the heat transfer to a load positioned in an electric oven, considering the contributions of natural convection and radiation. To allow an analytical solution of the model equation, the radiative transfer term was linearized considering the temperature differences between the oven walls and the surface of the thermal load, instead of being driven by the fourth-power. The analysis takes into account changes of size, shape, materials, radiation surface properties and oven set point temperatures, showing discrepancies of about 1% between predicted and experimental data. Another approach, based on lumped capacitance method, is presented in [25]. The authors state that such an approach, used for building energy simulation [26] with apparently good results, have never been used to model ovens before. It simplifies the heat transfer equations by considering the system as a discrete set of thermal capacitances and resistances, permitting to have good results in transient heating and cooling problems [25], [27]. It has been developed a lumped capacitance model of a professional oven, characterized by the interactions between two thermal zones, i.e. the power zone and the cooking zone (Fig. 2.1 (b)), and by the adoption of a not linearized radiation heat transfer approach. As a whole, the main features of this model are: the analysis of the energy exchange between the two zones of the oven; the modeling of external energy exchange, taking into account the temperature outside the glass door; it considers all the heat exchange mechanisms (radiation, forced convection due to the fan and conduction); the number of nodes inside the material is customizable, to evaluate the impact of layers of different insulation materials on the oven efficiency; it can be easily modified for evaluating new designs with low computational resources.

2.2 Thermal model of a professional oven

2.2.1 The oven

The oven used for the analysis is an Electrolux AoS Touchline 10 GN 1/1 length-wise. The technical characteristics of the oven are presented in chapter 1.2 and in [28]. The oven cavity is composed of two adjoining sections: the rear part, or power zone, occupied by the fan and the heat exchanger elements, and the cooking zone (Fig. 2.1 (b)). The suction wall, that separates the two zones, has two functions: guiding the air flow to the fan and directing the heated air into the cooking zone [28]. The power zone has the following dimensions: height = 0.76 m, width = 0.65 m, length = 0.2 m, while the cooking zone has height = 0.76 m, width = 0.65 m and length = 0.48 m.



(a) Electrolux AoS Touchline 10 GN 1/1; (b) Scheme of Electrolux AoS Touchline 10 GN 1/1.

Figure 2.1: Oven AoS Touchline 10 GN 1/1

2.2.2 The thermal model

Aim of the model is the evaluation of the energy performance of the oven during transient and steady state operations. The numerical model considers two zones, which contain only dry air and oven accessories (fan, trays, etc.); these have a high impact on the thermal inertia of the system. The presence of the food is not considered. All the components are modeled with the lumped capacitance method, so they interact as lumped elements of an electric circuit: the potential nodes refer to the temperature of each element, a set of capacitors reproduce the thermal capacity while electric resistances are used to indicate the convective, radiative or conductive thermal interaction between the elements. The model considers the two zones of the professional oven separately, as previously described: the power zone (hereinafter indicated with the subscript P) and the cooking zone (hereinafter indicated with the

subscript C). Spatial discretization is considered for the walls (hereinafter indicated with the subscript w_i for the i -th wall), divided in one dimensional isothermal layers. The layers are numbered starting from the external one (indicated with the subscript w_{i1}) to the internal superficial one (indicated with the subscript w_{is}), see Fig.2.3. The total number of walls in each zone is indicated with N .

2.2.3 The power zone

The logical scheme of the power zone is shown in Fig.2.2. This zone is bounded by five continuous walls and interfaced with the cooking zone through the suction wall, Fig.2.1 (b).

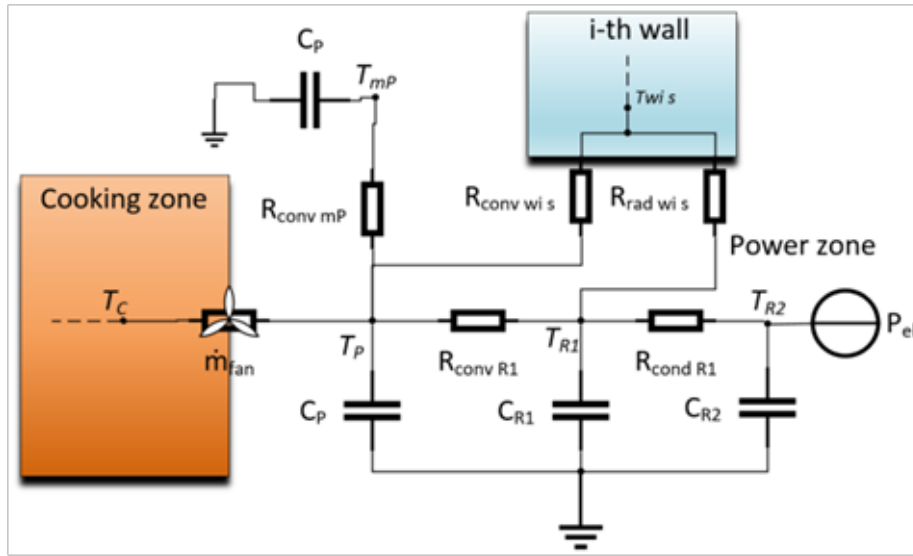


Figure 2.2: Functional scheme of the power zone of the oven.

The power zone is characterized by the presence of the heaters (electric resistor); they are modeled with two nodes (one internal and one external) due to the resistor high thermal inertia. The thermal power, P_{el} , is generated only inside the volume of the inner node. The external node interacts both with the air through convection and with the walls through radiative heat transfer. The presence of the fan and other accessories is considered in the thermal mass node (indicated with the subscript mP). The fan is responsible of the enthalpy flux exchange between the power and the cooking zones, $\dot{m}_{fan} c_P (T_P - T_C)$. The corresponding energy balance equations for every node are given below (Eqs.2.1-2.5).

$$\begin{aligned}
 V_P \rho_P c_P \frac{T_{P,n} - T_{P,n-1}}{t_n - t_{n-1}} = & \sum_i^N h_{conv w_i s} A_{w_i s} (T_{w_i s, n} - T_{P, n}) + \\
 & + h_{conv m P} A_{m P} (T_{m P, n} - T_{P, n}) + \\
 & + h_{conv R1} A_{R1} (T_{R1, n} - T_{P, n}) + \\
 & + \dot{m}_{fan} c_P (T_{C, n} - T_{P, n})
 \end{aligned} \tag{2.1}$$

$$\begin{aligned}
 V_{R1} \rho_{R1} c_{R1} \frac{T_{R1, n} - T_{R1, n-1}}{t_n - t_{n-1}} = & \frac{A_{R1}}{R_{cond R1}} (T_{R2, n} - T_{R1, n}) + \\
 & + h_{conv R1} A_{R1} (T_{P, n} - T_{R1, n}) + \\
 & + \sum_i^N P_{rad w_i s, n}
 \end{aligned} \tag{2.2}$$

$$V_{R2} \rho_{R2} c_{R2} \frac{T_{R2, n} - T_{R2, n-1}}{t_n - t_{n-1}} = \frac{A_{R1}}{R_{cond R1}} (T_{R1, n} - T_{R2, n}) \tag{2.3}$$

$$V_{mP} \rho_{mP} c_{mP} \frac{T_{mP, n} - T_{mP, n-1}}{t_n - t_{n-1}} = h_{conv m P} A_{m P} (T_{P, n} - T_{m P, n}) \tag{2.4}$$

$$\begin{aligned}
 V_{w_i s} \rho_{w_i s} c_{w_i s} \frac{T_{w_i s, n} - T_{w_i s, n-1}}{t_n - t_{n-1}} = & h_{conv w_i s} A_{w_i s} (T_{P, n} - T_{w_i s, n}) + \\
 & + \frac{\lambda}{\Delta x} A_{w_i s} (T_{w_i s-1, n} - T_{w_i s, n}) + \\
 & + \sum_j^N (1 - \delta_{ij}) P_{rad w_j s, n}
 \end{aligned} \tag{2.5}$$

In Eq.2.5, the subscript j indicates a wall different from the i^{th} one.

2.2.4 The cooking zone

The logical scheme of the cooking zone is shown in (Fig.2.3). This zone has the same number of bounding walls of the power zone, but it is characterized by the presence of the door, which is composed by two glasses with an air gap between them. The door is modeled with two nodes: the internal node (subscript g1) and the outer node (subscript g2). The internal node interacts with the cooking zone by convection (the radiation with the walls is neglected since the door is made with a low-emission glass) and with the outer node by means of an overall thermal resistance, indicated

in the figure as R_g . Furthermore, the area of inner glass is equal to the area of the outer glass, $A_{g1} = A_{g2}$.

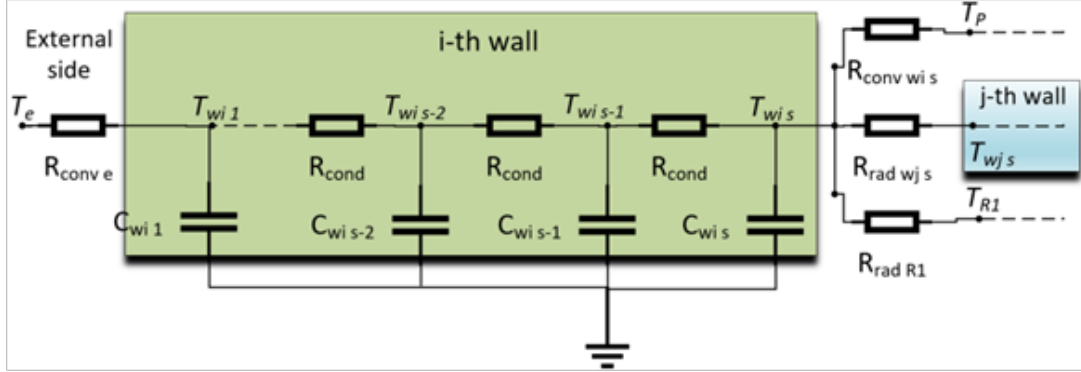


Figure 2.3: Functional scheme of a composite wall.

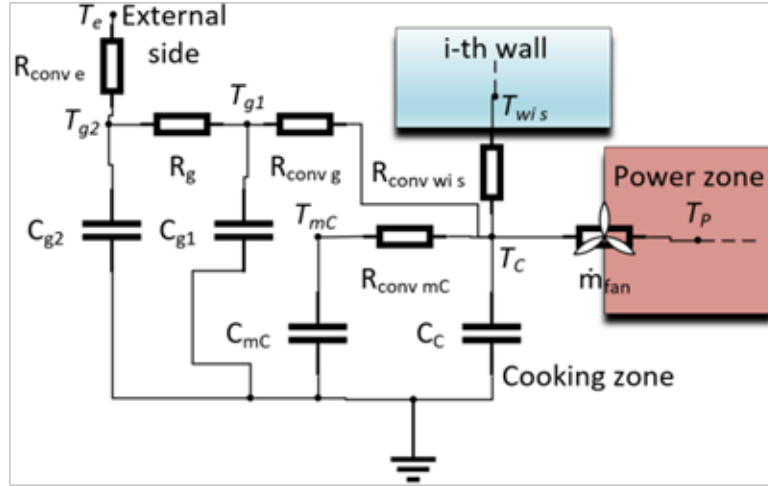


Figure 2.4: Functional scheme of the cooking zone of the oven.

The corresponding energy balance equations for every node are given below (Eqs.2.6-2.12).

$$V_C \rho_C c_C \frac{T_{C,n} - T_{C,n-1}}{t_n - t_{n-1}} = \sum_i^N h_{conv w_i s} A_{w_i s} (T_{w_i s,n} - T_{C,n}) + h_{conv mC} A_{mC} (T_{mC,n} - T_{C,n}) + \dot{m}_{fan} c_C (T_{P,n} - T_{C,n}) \quad (2.6)$$

$$V_{mC} \rho_{mC} c_{mC} \frac{T_{mC,n} - T_{mC,n-1}}{t_n - t_{n-1}} = h_{conv mC} A_{mC} (T_{C,n} - T_{mC,n}) \quad (2.7)$$

$$\begin{aligned}
 V_{w_i s} \rho_{w_i s} c_{w_i s} \frac{T_{w_i s, n} - T_{w_i s, n-1}}{t_n - t_{n-1}} = & h_{conv w_i s} A_{w_i s} (T_{C, n} - T_{w_i s, n}) + \\
 & + \frac{\lambda}{\Delta x} A_{w_i s} (T_{w_i s-1, n} - T_{w_i s, n}) + \\
 & + \sum_i^N (1 - \delta_{ij}) P_{rad w_j s, n}
 \end{aligned} \tag{2.8}$$

$$\begin{aligned}
 V_{w_i s-1} \rho_{w_i s-1} c_{w_i s-1} \frac{T_{w_i s-1, n} - T_{w_i s-1, n-1}}{t_n - t_{n-1}} = & \frac{\lambda}{\Delta x} A_{w_i s-1} (T_{w_i s-2, n} - T_{w_i s-1, n}) + \\
 & + \frac{\lambda}{\Delta x} A_{w_i s-1} (T_{w_i s, n} - T_{w_i s-1, n})
 \end{aligned} \tag{2.9}$$

$$\begin{aligned}
 V_{w_i 1} \rho_{w_i 1} c_{w_i 1} \frac{T_{w_i 1, n} - T_{w_i 1, n-1}}{t_n - t_{n-1}} = & h_{conv e} A_{w_i 1} (T_{e, n} - T_{w_i 1, n}) + \\
 & \frac{\lambda}{\Delta x} A_{w_i 1} (T_{w_i 2, n} - T_{w_i 1, n})
 \end{aligned} \tag{2.10}$$

$$\begin{aligned}
 V_{g1} \rho_{g1} c_{g1} \frac{T_{g1, n} - T_{g1, n-1}}{t_n - t_{n-1}} = & h_{conv g1} A_{g1} (T_{C, n} - T_{g1, n}) + \\
 & \frac{A_{g1}}{Rg} (T_{g2, n} - T_{g1, n})
 \end{aligned} \tag{2.11}$$

$$\begin{aligned}
 V_{g2} \rho_{g2} c_{g2} \frac{T_{g2, n} - T_{g2, n-1}}{t_n - t_{n-1}} = & h_{conv e} A_{g2} (T_{e, n} - T_{g2, n}) + \\
 & \frac{A_{g2}}{Rg} (T_{g1, n} - T_{g2, n})
 \end{aligned} \tag{2.12}$$

2.2.5 Radiation heat transfer

Radiation has a big impact in the heat transfer process in a professional oven, in particular in the power zone where the resistor radiates with high temperature difference towards the walls and the other elements. In building energy simulation, but also in the case of ovens analysis [24], radiation is usually modeled with the linearization theory. With such an approach, a coefficient of radiative heat transfer multiplies the temperature difference between two surfaces, so that the radiation term becomes linear, but some assumptions are required to estimate the coefficient itself, which depends on several factors (geometry, temperatures, etc.). In building simulation, the geometry is simple and temperature differences are quite low thus, the linear model is sufficient to have a good evaluation of radiative heat transfer. In the present case, due to both the complex geometry of the oven and the high

temperatures, the linear model is not taken into account. The radiation heat transfer is here modeled directly with an explicit scheme, without linearization and with the evaluation of the view factors between the resistor and the walls. For the last purpose, it has been used a MATLAB[®] script developed by [29]: it is a function that uses CDIF (Contour Double Integral Formula) to calculate view factors between planar surfaces (polygons). Consequently, the main assumption of the model is that the resistor is seen by each surface as a plane surface. The heat transmitted by radiation is then calculated with the approach called the net radiation method for enclosures. A formal explanation of this theory is presented in [30] and correspond to Eq. 2.13:

$$\begin{aligned} \sum_{j=1}^N \left(\frac{\delta_{ij}}{\epsilon_j} - F_{i-j} \frac{1 - \epsilon_j}{\epsilon_j} \right) \frac{P_{radwjs,n}}{A_j} &= \sum_{j=1}^N (\delta_{ij} - F_{i-j}) \sigma T_{j,n-1}^4 \\ &= \sum_{j=1}^N F_{i-j} \sigma (T_{i,n-1}^4 - T_{j,n-1}^4) \end{aligned} \quad (2.13)$$

where i has the value $1, 2, \dots, N$ for each surface. Temperatures at the time step $n - 1$ are used to calculate the radiative heat exchanged at the time step n (explicit scheme). Air is considered transparent to radiation.

2.2.6 Tuning procedure

The following four parameters are selected as tuning parameters for setting up the model:

- the averaged convective coefficient between air and walls: h_{convws} ;
- the averaged convective coefficient between air and thermal resistance, multiplied by resistor surface: $h_{convR1}A_{R1}$;
- the averaged convective coefficient between air and thermal masses, multiplied by the interface area: $h_{convmC}A_{mC}$, $h_{convmP}A_{mP}$.

A tuning procedure based on the trial and error method has been used for setting up the four parameters. A comparison between numerical and experimental data was made analyzing the slope of the transitory state (initial heating up phase), see Fig. 2.5 and of the ramp up and ramp down in steady state phases, see Fig. 2.6 . In order to check the physical meaning of the parameters, for two of them (h_{convws} and $h_{convR1}A_{R1}$) it was possible to compare the results with those obtained with theoretical correlations. The average convective coefficient between air and walls is calculated through the averaged Nusselt number obtained with the heat transfer correlation, valid for isothermal flat plates, defined by Eq. 2.14 [31]:

$$\bar{Nu} = \frac{h_{convws}L}{\lambda} = 0.037Re_L^{\frac{4}{5}}Pr^{\frac{1}{3}} \quad 0.6 < Pr < 60$$

$$5 \times 10^5 < Re_L < 10^7$$
(2.14)

Re_L is calculated with a velocity of 10 m/s, obtained from known CFD values, and a characteristic length of the walls $L = 0.7$ m. The averaged convective coefficient between air and heaters was compared with a heat transfer correlation used for the crossflow across tube banks [31]. The averaged Nusselt number is given by Eq. 2.15:

$$\bar{Nu} = \frac{h_{convR1}D}{\lambda} = C_2CRe_D^mPr^{0.36}\left(\frac{Pr}{\bar{Pr}}\right)^{\frac{1}{4}}$$
(2.15)

$$0.7 < Pr < 5000 \quad 1 < Re_D < 2 \times 10^4$$

where the coefficients, which depend on the geometry of the tubes, have been assumed equal to $C_2 = 0.76$, $C = 0.4$, $m = 0.6$ [31]. Also in this case the mean velocity, 15 m/s, is a result of known CFD analysis for this type of professional oven, while the heaters diameter $D = 0.006$ m is considered as characteristic length. For the other two parameters $h_{convmC}A_{mC}$ and $h_{convmP}A_{mP}$ no theoretical considerations can be made. Tab. 4.39 presents the values of the parameters obtained with the tuning procedure.

Parameter	Tuning Value	Correlation Value
$h_{convws} [W/m^2 K]$	20	29 (Tuning parameter from Eq.2.14)
$h_{convR1}A_{R1} [W/K]$	25	35 (Tuning parameter from Eq.2.15)
$h_{convmC}A_{mC} [W/K]$	70	-
$h_{convmP}A_{mP} [W/K]$	75	-

Table 2.1: Tuning parameters

The correlation values are different from the tuning ones: in fact the heat transfer correlations are valid for heaters tubes in perfect crossflow and fluid flow parallel to the surface, while in the oven these conditions are only partly satisfied. Moreover the differences are due also to the use of the averaged value of velocity.

2.3 Results and discussion

The control logics used in the model is on/off for the resistor power, with a fixed set point, and a deadband for the activations of the heating power, based on the power zone temperature. With the chosen values of the tuning parameters the model shows a good agreement with the experimental data as shown in Figs. 2.5 and 2.6.

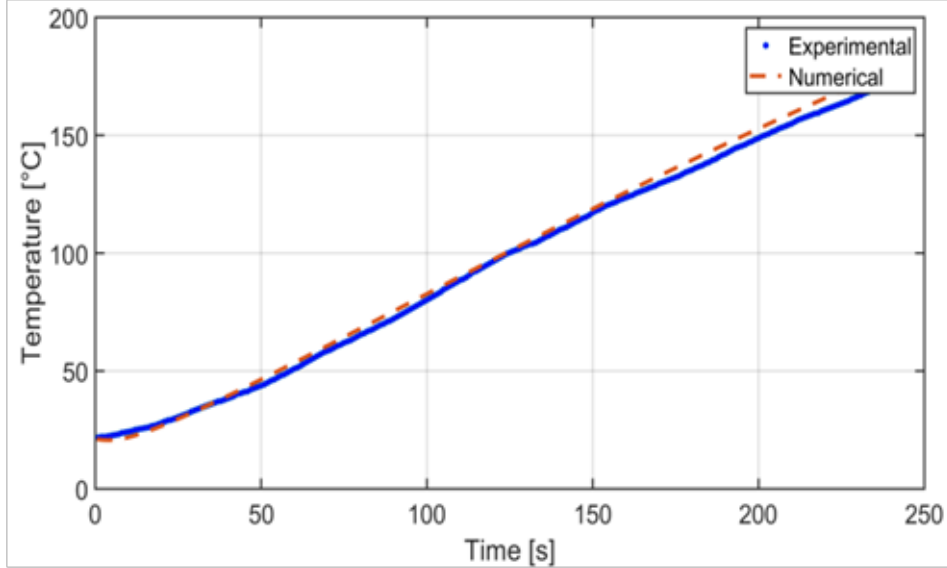


Figure 2.5: Comparison between numerical and experimental temperature profiles in the cooking cavity (transient state).

With the present model, it is therefore possible to face the challenging task of studying the thermodynamic of a professional oven, which has to ensure high productivity with high quality. Advanced control strategies can also be developed, to increase the performance of the device according to several parameters, i.e. the monitored temperature of a sample inside the cavity, or the energy exchanged with it. Figs. 2.7 and 2.8 present some preliminary numerical results. In the first, heaters temperature and activation times are reported together with the air temperatures in the two oven zones. In the second, the radiation heat transfer exchanged by the heaters is shown: it is clear how radiation influences the transient state, while during steady state it has a lower impact, since the temperature difference between the heaters and the surrounding components is lower.

2.4 Conclusions

In this chapter, a numerical model of a professional oven has been presented. This model is based on the lumped capacitance method, that has demonstrated the capability of predicting the thermodynamic performances of the oven with low

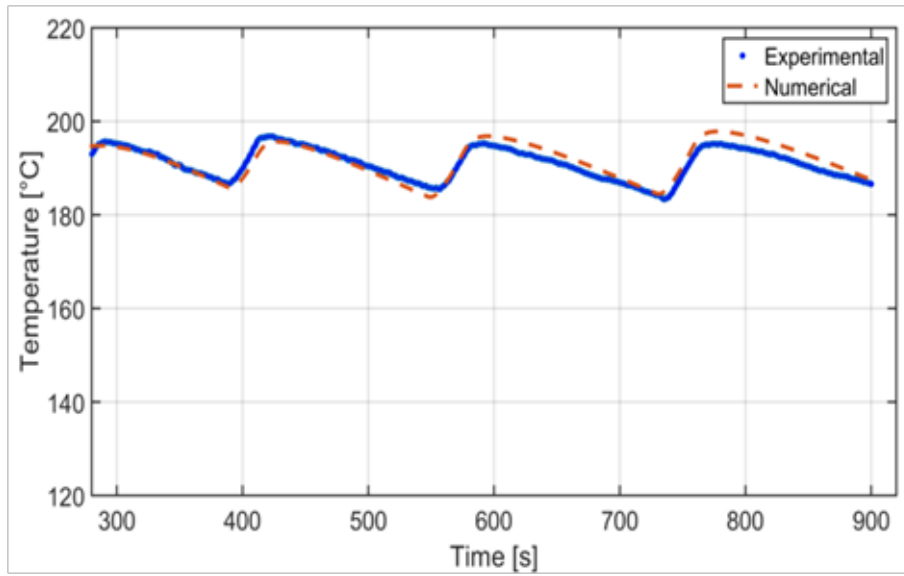


Figure 2.6: Comparison between numerical and experimental temperature profiles in the cooking cavity (steady state).

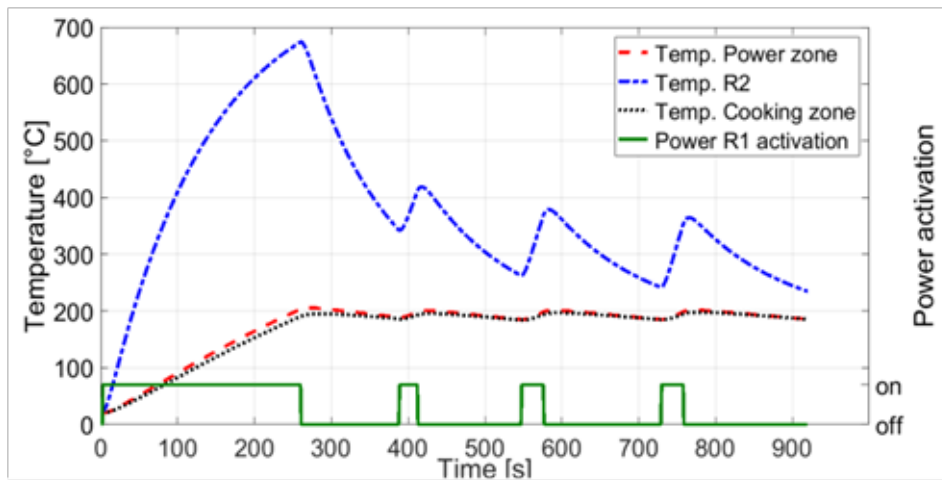


Figure 2.7: Numerical results: temperature and power diagrams.

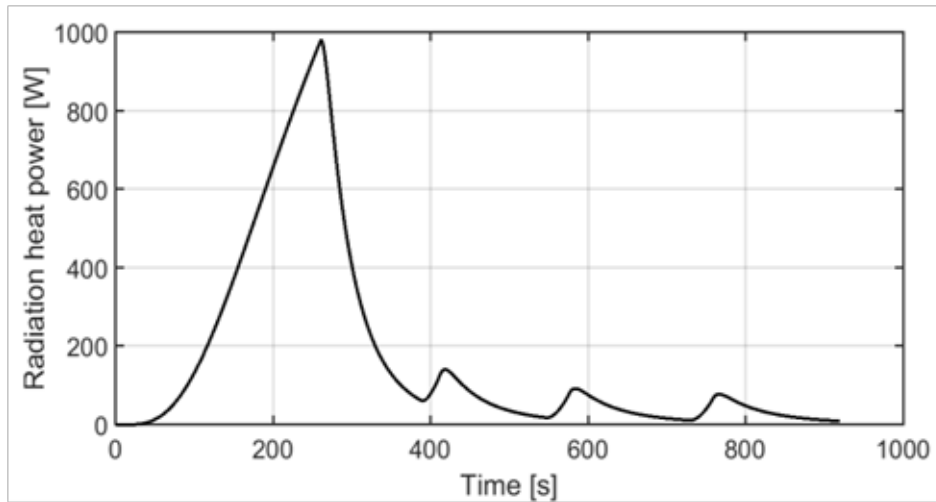


Figure 2.8: Numerical results: radiative heat from the heaters.

computational efforts compared to other numerical techniques (i.e. CFD). The oven presents two distinct zones: the power zone and the cooking cavity. The model is capable of predicting the thermodynamic behavior of both the zones and it permits to evaluate the overall energy performances of the device. The oven works mainly in forced convection so it was required an estimation of the convection parameters with a tuning procedure to find their averaged value. The contribution of radiation heat transfer has also been modeled with a quite accurate approach based on the so-called net radiation method for encloses. The results have been compared with a set of experimental data showing a good agreement in both the transitory and the stationary operating phases. The last part of the chapter presents the capabilities of the model to predict in detail the thermodynamic performances of the oven in given operating conditions. Future work will focus on the following activities:

- comparison with more experimental data at different cooking modes;
- an optimization study to find correlations of the tuning coefficients according to different operating conditions;
- introduction of the hygrometric balance in a combined oven (presence of steam produced by a dedicated boiler);
- simulation of the food thermodynamic behavior inside the oven.

In the next chapter 3, as introduced in chapter 1, is presented an analysis of the energy efficiency of the oven washing system in terms of energy and water consumption.

Chapter 3

Analyses of oven washing systems

3.1 Introduction

In professional ovens the cleaning process is performed by a dedicated automatic washing circuit. The washing phase involves an energy absorption that has been analyzed among different technical solutions as open and closed washing circuits. Furthermore, in the washing operations, different factors have to be considered for establishing the effectiveness of the washing process, as illustrated in the next paragraph.

3.2 The Sinner Circle

Detergency is considered the action performed by a detergent solution for removing the food soils and residuals. The working solution plays a key role in the detergency performance. In the washing process of a professional oven, a solution of water and cleaning product enters in contact with different types of soils. The cleaning process necessitate to be highly efficient in terms of energy consumption and time duration of the cleaning process. A chemical engineer from Henkel, Herbert Sinner, outlined the four basic factors of a cleaning process: chemical action, temperature, mechanical action and time [32]. In a cleaning scenario, the four cleaning factors can be manipulated such that increasing one factor allows the decreasing of other three factors. A good combination of the cleaning factors permit to have good washing performances supporting energy efficiency improvements. These factors are listed as hereinafter.

- Chemical action: is defined as the chemical reactions and interactions of detergent molecules with the soil. Washing performances are dependent on the detergent concentration, such that increasing detergent concentration leads to higher interactions between the solution and the soil due to the higher quantity of detergent molecules. Such trend is limited by the possible deterioration phenomena of the oven materials. There are also limits related to possible environmental pollution.
- Temperature action: temperature enables the action of the chemical species in particular it affects the kinetics of the chemical action (surfactants) and the chemical hydrolysis. The temperature of the washing process is related to the thermal energy transferred to the washing cycle. In the designing of the washing cycle it is important to select an appropriate temperature that considers the maximum yield of the detergent and avoids the deterioration of certain chemical components.
- Mechanical action: mechanics is represented by the forces and frictions generated in the washing cycle due to water flux and pressure that controls the

quantity of soil removed from the substrate.

- Time: the length of time at which a certain temperature is maintained in the washing cycle influences the disinfection effects. Too long washing cycles could imply into soil re deposition.

The actual washing processes trend is to reduce the use of three parameters increasing in particular the chemical action. In this chapter this aspect will be explored in detail.

In professional appliances high performance detergent are used. They contain high concentrations of sodium hydroxide and phosphate builders [33]. In Tab.3.1 is presented a general formulation guide and the range of concentration of the various detergent components.

Components	Functions	Concentrations
Builder	Sequestration, soil suspension, alkalinity, emulsification, soil peptization	10 – 20 %
Caustic alkali	Alkalinity, soil hydrolysis, soil removal	10 – 50 %
Surfactant	Wetting agent, soil removal, spot-film prevention, sheeting action, soil dispersion, drying aid	0 – 3%
Silicate (optional)	Anticorrosion, alkalinity, soil hydrolysis, soil removal sequestration	0 – 20 %
Bleach (optional)	Soil removal, stain removal, sanitation, disinfection	0 – 3%
Defoamer (optional)	Foam prevention, washing efficiency	0 – 1 %
Thickener (optional)	Product stability, aesthetic enhancement, binder (for liquid product)	0 – 2 %
Color – perfume	Aesthetic enhancement	< 0.5 %
Water (optional)	Solvent, carrier, flow property	Balance

Table 3.1: Detergent components [32]

Builders provide high and rapid sequestration of water hardness permitting an efficient cleaning process. They avoid that hard water metal ions form insoluble systems with soils, surfactants and detergent materials. Common builders, as Sodium Tripolyphosphate or Silicate, are multi functional cleaning agents that are also able of powering the surface activity of surfactants. Thus they operate as dispersing and buffering agents suspending the soil and preventing its redistribution. Silicate adds also corrosion protection. There are even polymers secondary builders, as the polyacrylates, which reduce also the crystal growth of calcium precipitate. Surfactants in warewashing detergents are used at low concentration, especially in mechanical dishwashers. In liquid detergents for warewashing machines it is preferred to use low foaming NI (non-ionic) surfactants, because the foam generated from protein soil and surfactant can cause a drop in water pressure, so decreasing the speed of the spray arm rotor, if any, with consequences on the cleaning effectiveness. Common not foaming substances are polyethylene glycols, fatty acid alkanolamide and EO/PO (Ethylene Oxide / Propylene Oxide) block polymeric surfactants.

Bleach and oxidizing agents are used for remove oxydable food stains coming from vegetables, wine and color food. Liquid gel or slurry detergents contain thickeners [33].

3.3 Open and closed washing circuits

A professional oven has a washing system in order to clean the cavity, the quenching system and the boiler. There are two possible washing solutions as the closed and the open washing systems.

- The *open washing system* is characterized to have a solution of water and detergent entering and going out from the system without recirculation and after the detergent action .

The Electrolux Professional One Oven generation have an open washing system. The water from the network or a solution of water and detergent is sucked by the action of the pump from an external dedicated tank (Fig. 3.1 green circle). The hydraulic circuit is connected to a water spray arm positioned inside the oven cavity (Fig. 3.1 red circle), that rotates due to the action of the water pressure and distributes the solution on all the surface of the oven cavity. When the solution of water and detergent is sprayed into the cavity, the oven cavity its temperature is maintained constant by means of a thermostat for permitting an effective chemical action of the detergent. In the bottom part of the oven cavity there is the drainage section where the solution directly exits from the oven cavity. After the cleaning phase performed with the detergent, there is the rinse phase performed with water. In Fig 3.1 is presented the One Oven-10/1.

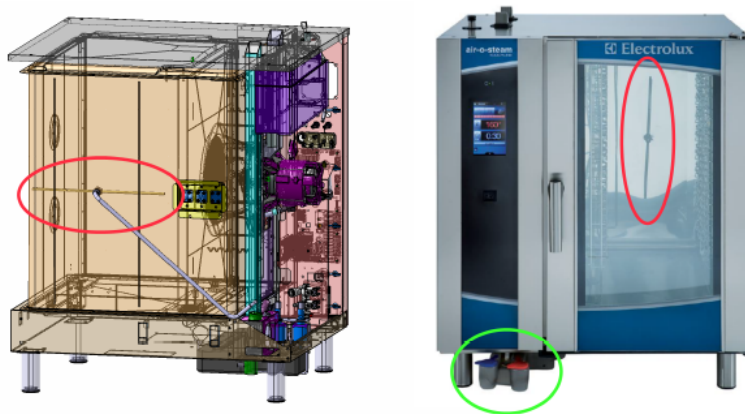


Figure 3.1: One Oven-10/1

- The *closed washing system* is characterized to be a system where the entering solution of water and detergent recirculate inside the system till the solution losses its effectiveness.

The actual trend is to substitute open washing systems with closed washing systems. An example of closed washing system is the Grand Cousin oven present in Electrolux Professional. Grand Cousin is a professional oven dimensioned for the domestic luxury sector but having all the characteristics of a professional oven. In the washing circuit of Grand Cousin the same solution recirculates inside the oven various times. The closed hydraulic circuit was designed for reaching low water consumption during the washing cycle. In this configuration all the detergent potential is used resulting in lower detergent consumption compared to the open circuit. In Fig 3.2 is presented the CAD design of the Grand Cousin oven. A recirculating tank filled with water or detergent solution (Fig. 3.2 red circle), is positioned under the cavity. A pump permits the re-circulation of the fluid between the tank and the top of the oven cavity (Fig. 3.2 green circle). The fluid is sprayed through the axis of the rotating oven fan, such that the fluid is axially sucked and radially sprayed through all the cavity. The rinse phase in Grand Cousin is characterized to have also a brief cavity saturation performed with steam.

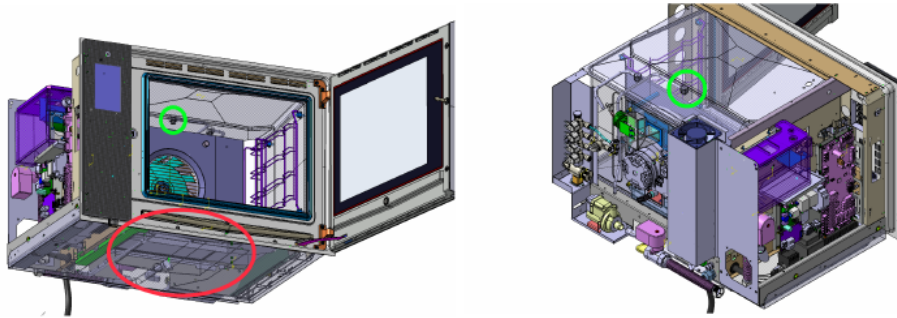


Figure 3.2: Grand Cousin Oven

For understanding what is the best configuration among closed and open washing systems, in the next section is presented an energy efficiency analysis of the two possible solutions. After this analysis, a technology demonstrator has been developed in these years with a self cleaning system characterized to have low energy consumption with a closed hydraulic circuit. [32]

3.4 Energy efficiency analyses of actual washing systems

In this paragraph is presented a comparison between open and closed washing systems. The results coming out from this analysis justify the development of a



Figure 3.3: Grand Cousin Oven

technology demonstrator with a closed washing system.

The washing cycle in a professional oven consume a lot of energy and water. Open washing systems are characterized to be less efficient compared to the closed washing systems in terms of energy and water absorption but also in terms of quantity of chemistry used during the washing phases. Energy efficiency improvements in a washing system could be also reached acting on the parameters of the Sinner's circle, by using shorter washing cycles conducted at lower temperatures.

3.4.1 Washing cycles composition

Considering the Electrolux One Oven Generation as representative device, four washing cycles are present as shown in Tab. 3.2: soft, medium, strong and extra-strong. Every washing cycle is composed by different combined macro phases presenting different time lengths. In Tab. 3.2, time length is normalized with the maximum time length of the washing cycle due to confidentiality agreements [34].

Every macro phase is then composed by a sequence of sub phases as presented in Figs. 3.4, 3.5, 3.6, 3.7, 3.8. The diagrams presented from Fig. 3.4 to 3.8 are indicative in the data of the temperature (normalized temperature with reference to the data of the maximum temperature in the macrophase) and time length due to confidentiality agreements [34].

Phases A, B are two different types of washing macro phases, C is a descaling phase, D is the drying phase and E is the rinse phase.

Before the initial stage of the washing cycle, an initial heating or cooling phase (with cold water) is scheduled for stabilizing the initial temperature conditions in the oven cell.

A trend of the water consumption and the energy consumption during the different washing cycles is presented in Fig. 3.9. All the data are normalized respectively to the values of the maximum time length, water consumption and energy consumption [34].

Cycle	Phases composition	Normalized duration of the washing cycle (t/t_{max})
Soft	B+C+E+D	0.3
Medium	A+C+E+D	0.4
Strong	2xA+C+E+D	0.6
Extra - strong	3xA+C+E+D	1

Table 3.2: Washing cycles characteristic

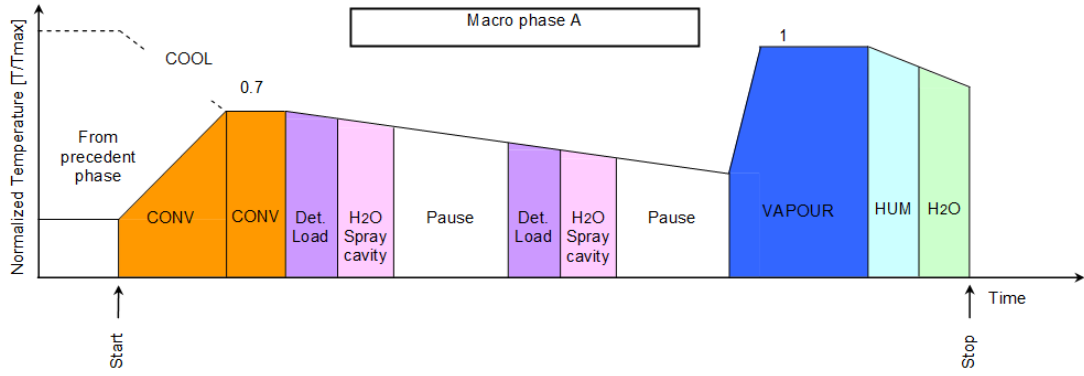


Figure 3.4: Macro phase A : Washing phase

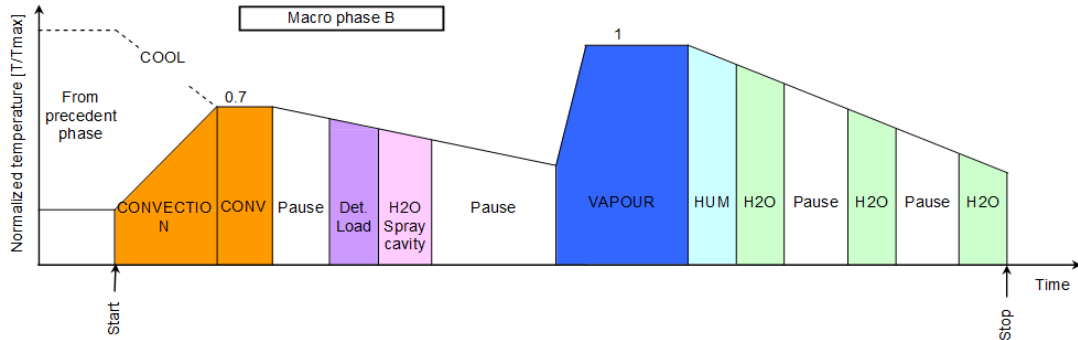


Figure 3.5: Macro phase B : Washing phase

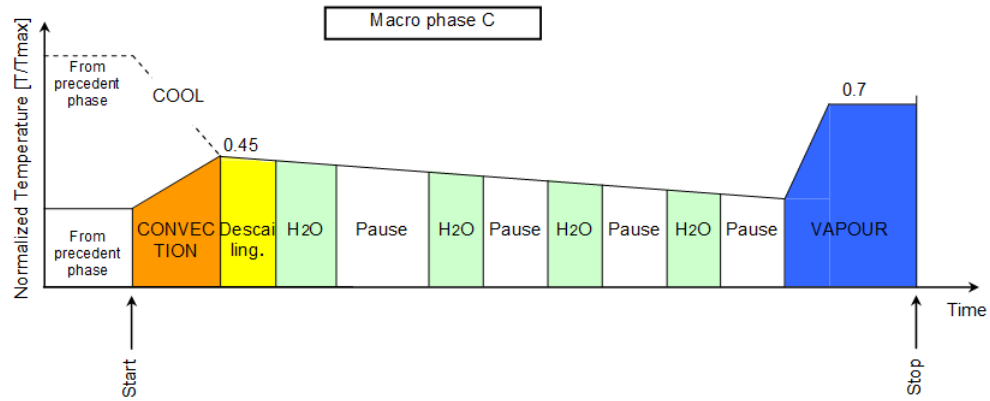


Figure 3.6: Macro phase C : Descaling phase

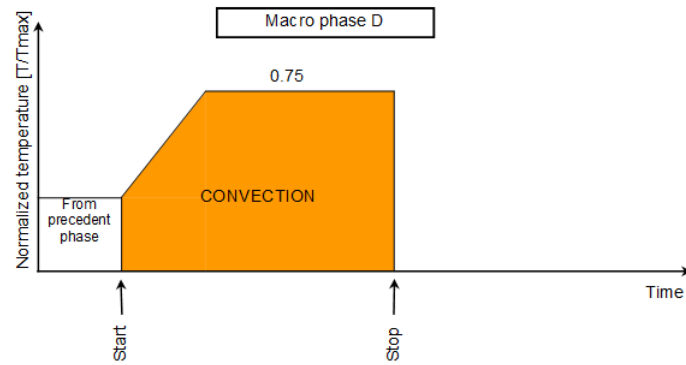


Figure 3.7: Macro phase D : Drying phase

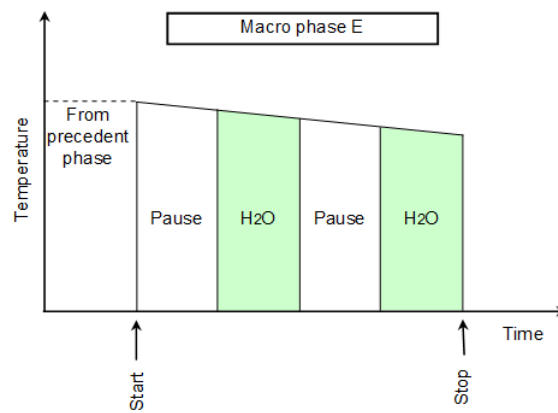


Figure 3.8: Macro phase E : Rinse phase

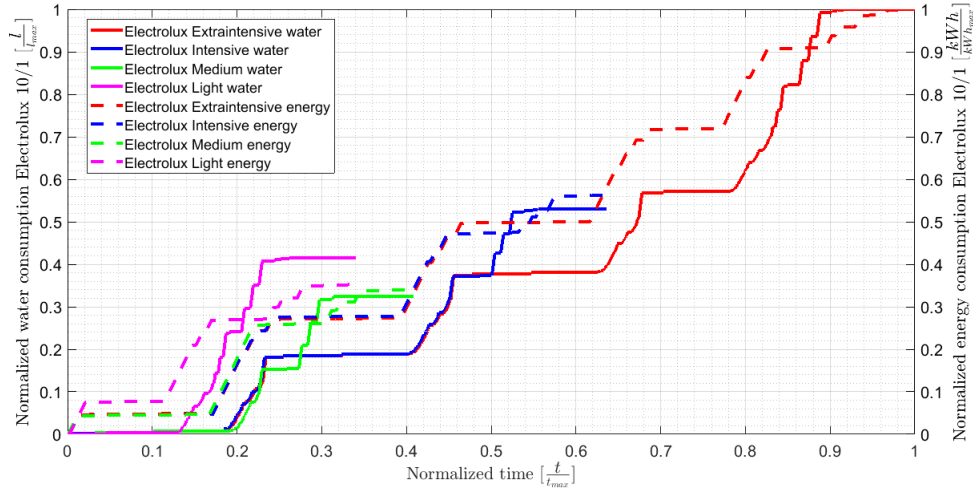


Figure 3.9: Trend of the water consumption and the energy consumption during the different washing cycles of Electrolux One Oven-10/1

The energy and water consumed in the different cycles increases from the soft cycle to the extra intensive cycle.

3.4.2 Comparison between representative devices

Some representative devices of professional ovens washing plants were selected. These representative devices have the same size. Two washing circuits are representing open washing systems (Electrolux and Benchmark 2) and the other two (Benchmark 1 and Benchmark 3) are representing closed washing systems. In order to perform a comparison on the water and energy consumed among the four professional representative devices, test were made fixing one common washing cycle, and comparing the water loaded and energy consumed between the different representative devices as represented in Fig. 3.10. The selected washing cycle is the extra strong.

Benchmark 1 presents the longest washing cycle, instead Benchmark 2 presents the shorter washing cycle. Percentage normalized differences (P.N.D.) are used to compare the different washing cycles, i.e. the difference between the value (liters - KWh - quantity of detergent and rinsing agent) obtained at the considered washing cycle and that obtained at the end of Benchmark 2 (considered as the solution that consumes more water) divided by the last one, expressed in %. In Fig.3.11 are represented the percentage normalized differences of the energy and water consumption.

Fig. 3.11 shows a clear difference between open systems and closed systems in terms of water consumption and energy consumption at the end of the washing cycle. The quantity of water used in the closed cycle configurations is lower compared to the one used in the open cycle configuration in particular Benchmark 3 and

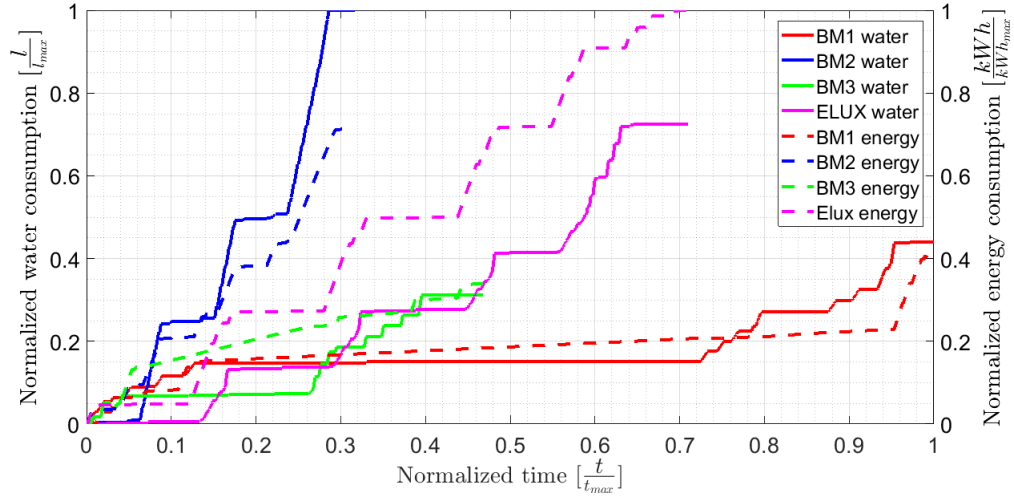


Figure 3.10: Trend of the water consumption and the energy consumption during the different washing cycles

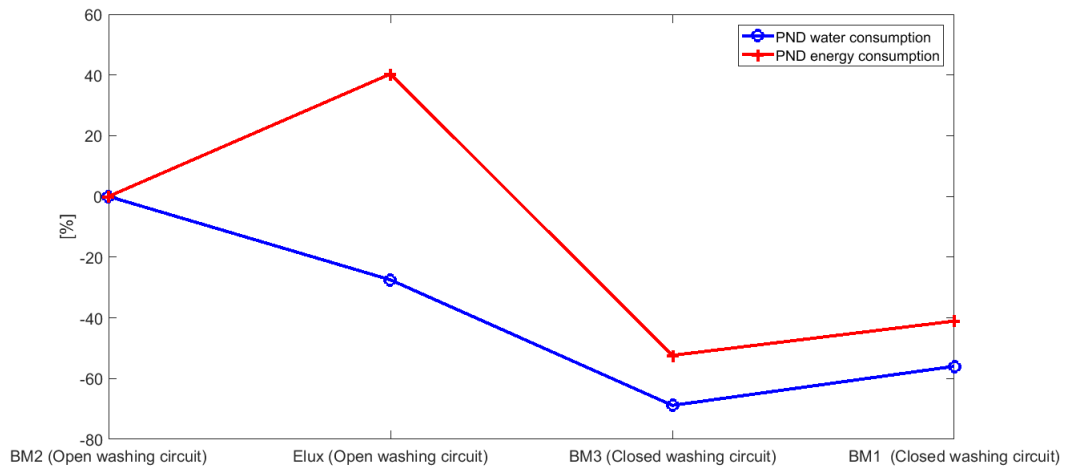


Figure 3.11: Percentage normalized differences between different configurations on the energy and water consumption

Benchmark 1 consume respectively 70 % and 55 % less water compared to Benchmark 2. Also energy consumption is lower in closed systems, in particular Benchmark 2 consume respectively 53% and 42 % more energy compared to Benchmark 3 and Benchmark 1.

Detergent and rinsing agent consumption have also been analyzed with the percentage normalized differences considering the reference the Benchmark 2 on the whole length of the washing cycle. Benchmark 2 consumes respectively 75 % and 95% more rinsing agent compared to Benchmark 3 and Benchmark 1. Considering the detergent agents, Benchmark 2 consumes respectively 36% and 67% more compared to Benchmark 3 and Benchmark 1.



Figure 3.12: Percentage normalized differences between different configurations on the quantity of detergent and rinsing agent used

Fig.3.11 , Fig. 3.12 show that closed configuration gives an overall less consumption in terms of energy, water, rinsing agent and detergent justifying the closed circuit as an efficient solution. The main parameters of the Sinner circle were analyzed with normalized histograms on the maximum value of the considered parameter in order to understand how they impact on the different washing processes of the representative washing plants. Quantity of detergent, water consumption, time needed for the washing cycle, energy used and maximum temperature reached inside the cavity have been compared among different representative professional ovens washing plants.

As represented in Fig. 3.15 Benchmark 1 has the longest washing cycle compared to the other solutions.

The maximum temperature among different cycles (Fig. 3.17) shows that all the washing solutions present the same maximum temperature inside the cavity except Benchmark 1 that presents a maximum temperature of 160 °C.

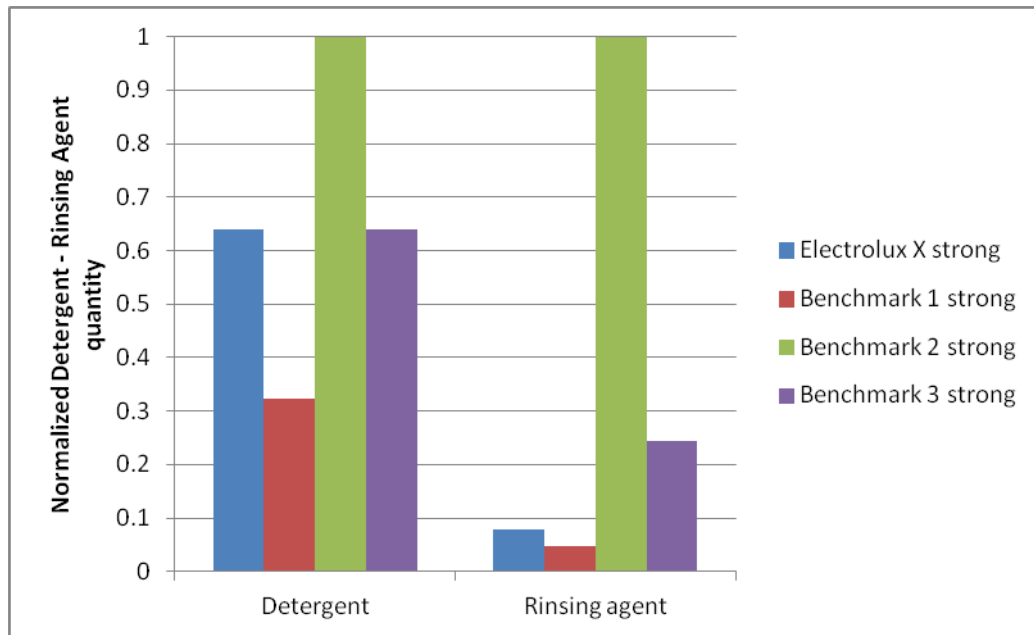


Figure 3.13: Bar diagrams comparison on the quantity of detergent and rinsing agent

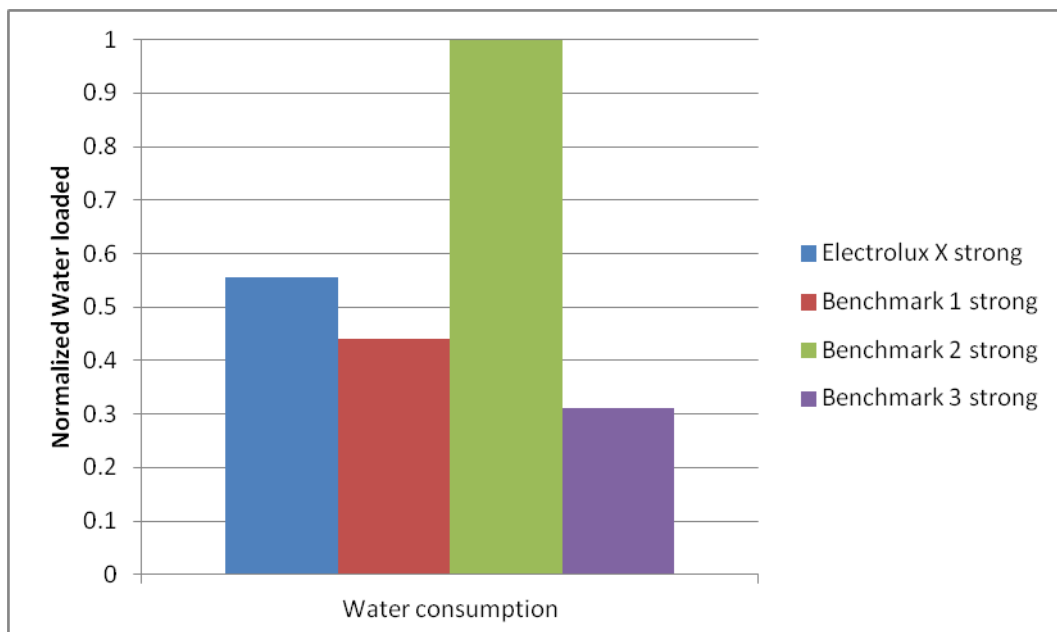


Figure 3.14: Bar diagrams comparison on the quantity of water

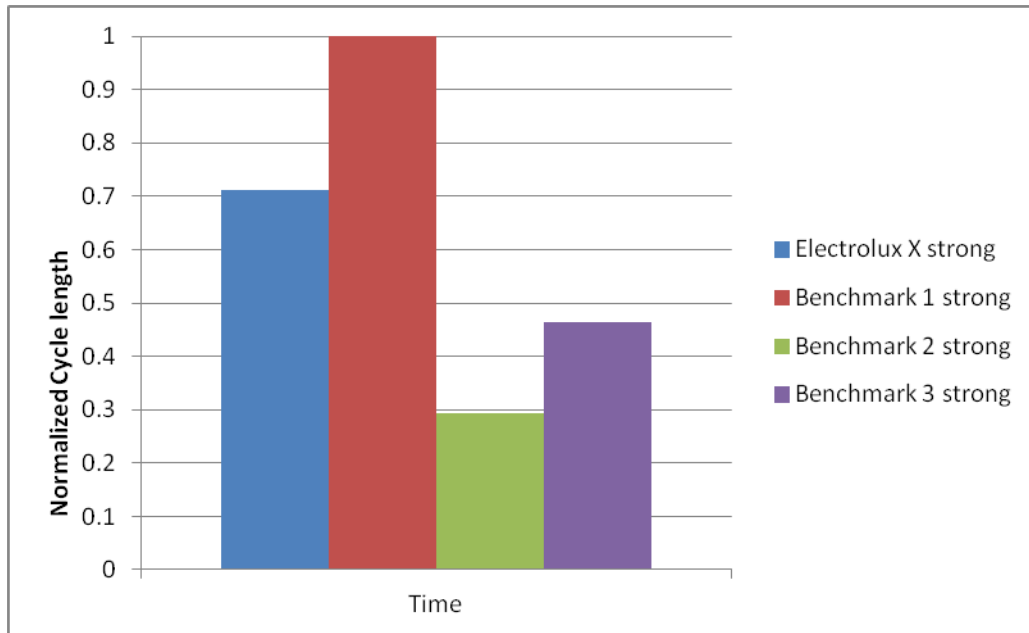


Figure 3.15: Bar diagrams comparison on the time length of the cycles

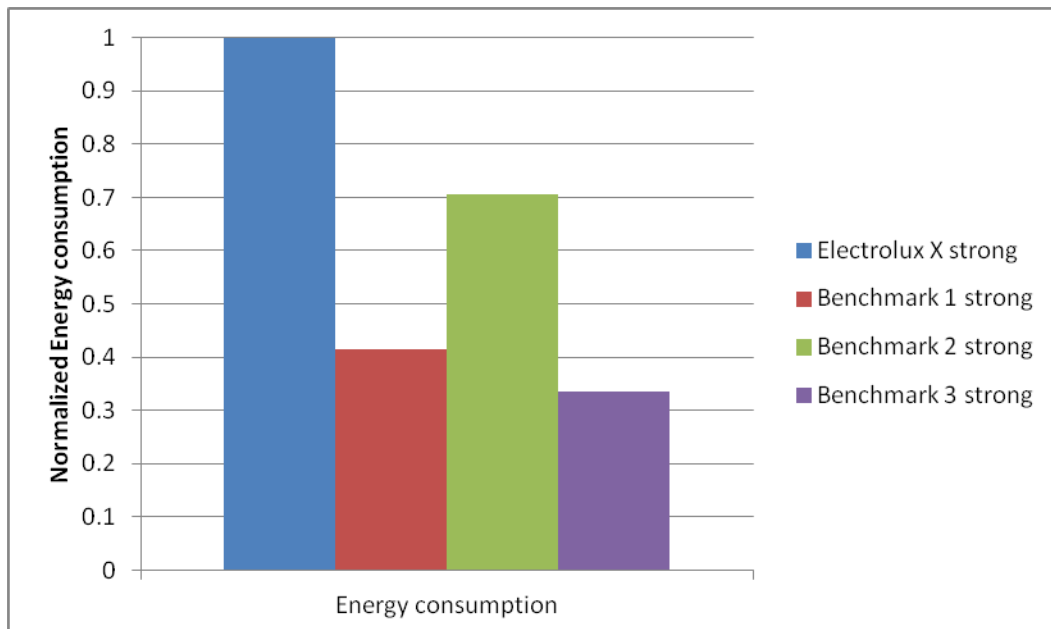


Figure 3.16: Bar diagrams comparison on the energy consumption

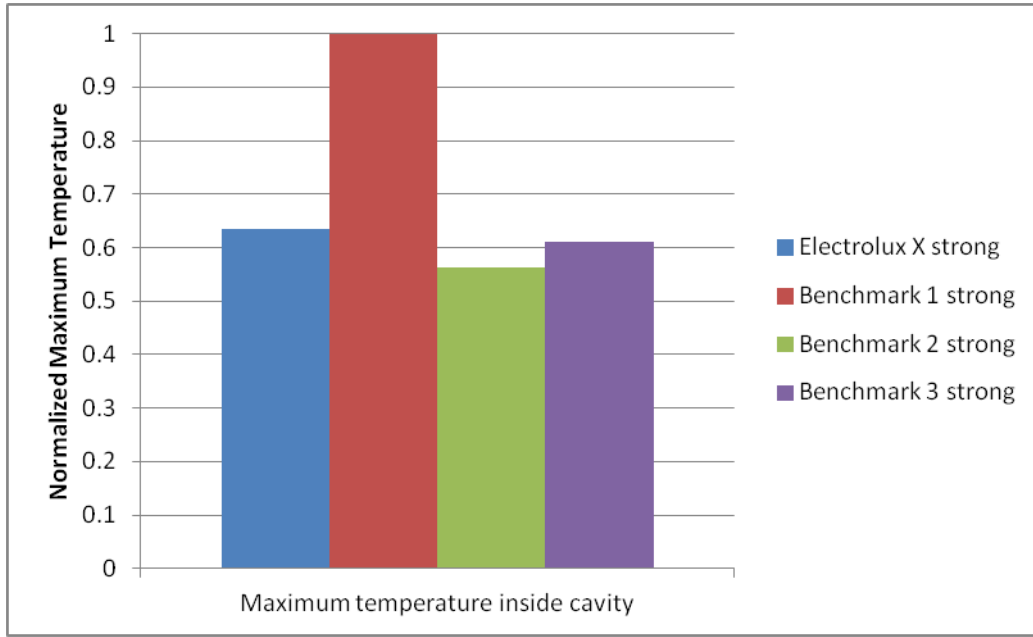


Figure 3.17: Bar diagrams comparison on the maximum temperature inside the cavity

3.5 Development of a new closed washing circuit

The comparison among closed and open washing solutions shows that closed washing systems present best performances in terms of energy, water, detergent and rinsing agent consumption. This trend justify the development of a new concept of closed washing system.

In this section is presented a methodology for developing a washing system based on a first preliminary numerical analysis with the aim of simulating and predicting the behavior of the system and its main component, the pump. A first comparison between experimental and numerical analysis on the hydraulic circuit has been conducted leading to a definition of the general hydraulic lay out of the professional oven. The activity is followed by a brief analysis on a possible solution for cleaning up the aspiration duct, filling up the hydraulic circuit and recirculate the solution of water and detergent in a professional oven. The proposed solution has two pipes of the washing circuit entering in the suction duct of the oven used for the air entrance inside the oven cavity. Before developing a physical prototype of the aspiration duct, the analysis was performed with a CFD tool in order to understand if the pipes entering in the suction duct could influence the air flow rate entering into the oven cavity. At the end of the chapter is presented a brief analysis on the possible selection criteria for the pump used in the hydraulic circuit of the professional oven.

3.6 The closed washing circuit model

The simulation software, Open Modelica is an open-source Modelica-based modeling and simulation environment used for industrial and academic purposes. Modelica is an object oriented, declarative, multi-domain modeling language for component-oriented modeling of complex systems, e.g. systems containing mechanical, electrical, electronic, hydraulic, thermal, control, electric power or process-oriented sub components. Open source Modelica gives a modeling, compilation and simulation environment based on free language developed by the non-profit Modelica Association.

Open Modelica permits to connect different components model. The connection (interaction) of a model component to other components is achieved by physical ports (connectors) that define how models can interact (connect) with other models in particular defining the variables of the model shared with other models. [35]

For modeling the oven washing plant the Open Modelica fluid library was used applying the following packages:

- Vessels : devices storing the fluid;
- Pipes;
- Centrifugal pump;
- Sensors;

The washing system lay out considered for the analysis is presented in Fig. 3.18, instead the lay out converted into Open Modelica is presented in Fig. 3.19

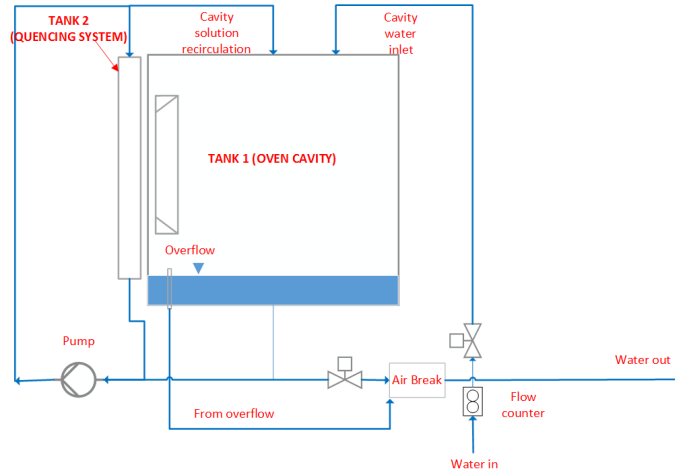


Figure 3.18: System lay-out of the oven washing system

In the Open Modelica schema of the oven two special tanks called TankWithTop-Ports are used. These tanks represent the oven cavity (Tank1) and the quenching

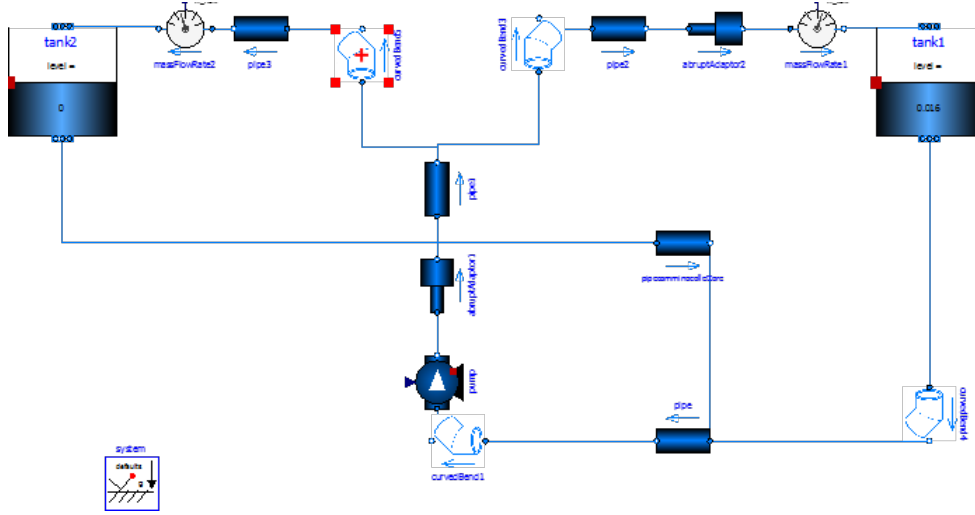


Figure 3.19: Open Modelica system lay-out of the oven washing system

system (Tank2). The Tank1 is filled with water (8 l , level 0.016 m) and is assumed to have uniform temperature. The pump permits the re circulation of the fluid between Tank1 and Tank2. The fluid coming out from the quenching system (Tank2) goes into the pipe before the pump. For understanding what is the fluid flow that goes through the cavity (Tank1) and the quenching system (Tank2), two mass flow sensors are used. For detecting the operating point of the pump, the characteristic curve is used as input for the model. In the model are considered the pressure losses due to the pipes, the curve bends and the abrupt adaptor. The model simulates the circuit in the transitory state.

3.6.1 Equations used in the model

The Open Modelica model solve the next equations.

- Mass balance (Eq. 3.1)

$$\frac{\partial(\rho A)}{\partial t} = -\frac{\partial(\rho A v)}{\partial x} \quad (3.1)$$

- Energy balance (Eq. 3.2)

$$\frac{\partial(\rho u A)}{\partial t} = -\frac{\partial(\rho v(u + \frac{p}{\rho})A)}{\partial x} + v A \frac{\partial p}{\partial x} + v F_F + \frac{\partial}{\partial x}(k A \frac{\partial T}{\partial x}) + \dot{Q}_e \quad (3.2)$$

- Momentum balance (Eq. 3.3)

$$E \frac{\partial(\rho v A)}{\partial t} = -\frac{\partial(\rho v^2 A)}{\partial x} - A \frac{\partial p}{\partial x} - (F_F + A \rho g \frac{\partial z}{\partial x}) \quad (3.3)$$

The pressure losses due to wall friction in pipes is computed as the product of

the dynamic pressure and the loss factor as presented in Eq. 3.4.

$$\Delta p = \zeta \frac{\rho v^2}{2} \quad (3.4)$$

Where the loss factor is defined by Eq.3.5.

$$\zeta = \lambda \frac{l}{d} \quad (3.5)$$

Also the pipes pressure difference due to static head is modeled in the program as Eq.3.6.

$$\Delta p = \rho g z \quad (3.6)$$

The pipeline in the Open Modelica model is slightly simplified with respect to the circuit present in the technology demonstrator of the professional oven. The fluid considered in the circuit is water.

3.6.2 Analysis of the results

In order to achieve knowledge of the simulated washing plant of the professional oven 10GN 1/1, an experimental activity has been scheduled and conducted. A comprehensive view of the oven prototype during a washing cycle is presented in the Fig. 3.20 . A particular of the two branches connecting the oven cavity (Tank 1 with letter A) and the quenching system (Tank 2 with letter C) is presented in Fig. 3.21 .

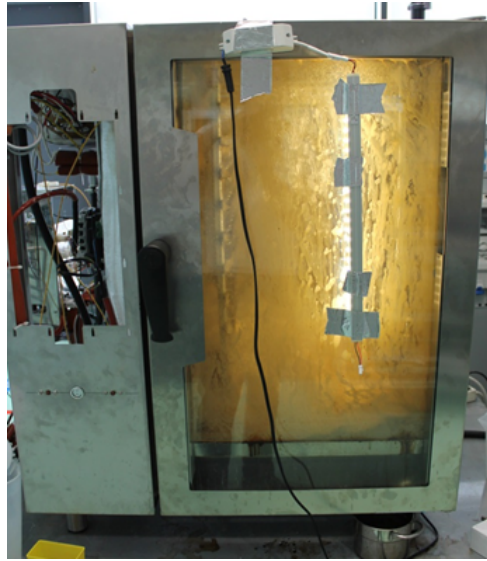


Figure 3.20: Experimental analysis on the prototype

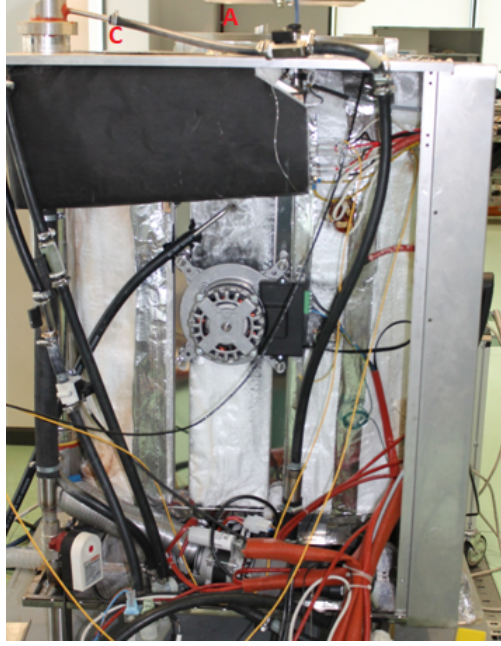


Figure 3.21: Experimental analysis on the prototype

The first oven prototype washing system is set up in a closed loop as presented in Fig. 3.18. The first tests conducted on the system had the aim of verifying the rough correspondence between measured and expected performances, in particular on the operating point of the pump in the circuit in order to select the best pump for the application. The data acquired from the oven were the fluid flow going through the quenching system and to the oven cavity.

The results are presented in the Fig. 3.22. The theoretical resistance curves are calculated starting from the known head provided by the pump and deduced from experimental analysis and numerical calculations. Pump head must exactly match the static head plus friction heads (calculated analytically) in both numerical and experimental analysis at a certain fluid flow.

The measured fluid flow going to cavity, chimney and the NPSHa compared to the numerical results are presented in the Tab. 3.3

Parameters	Numerical result	Experimental results
Fluid flow to oven cavity (Tank 1)	14.7 l/min	13 l/min
Fluid flow to quenching system (Tank 2)	10.8 l/min	8 l/min
NPSHa	10.11 l/min	

Table 3.3: Detergent components

The trend of the oven cavity level (Tank 1) and the quenching system (Tank 2) are represented respectively in Fig. 3.23 and Fig. 3.24.

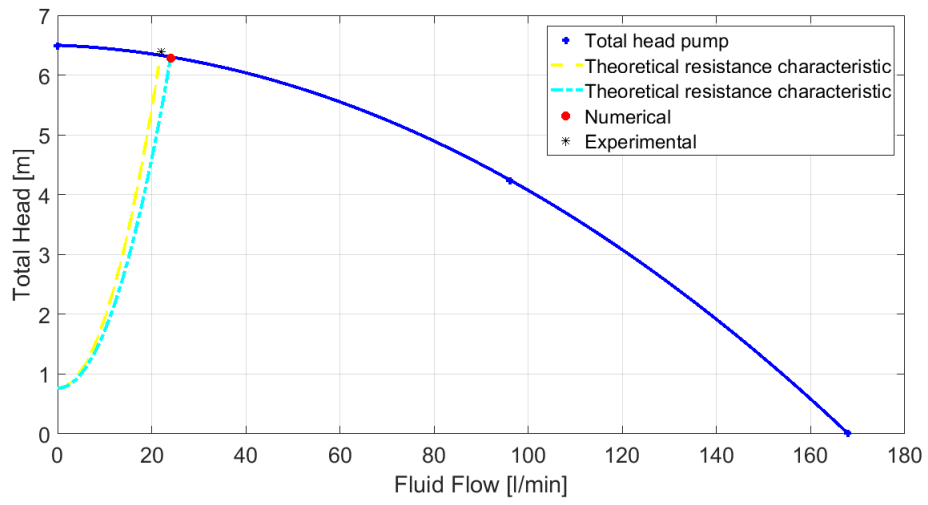


Figure 3.22: Experimental analysis on the prototype

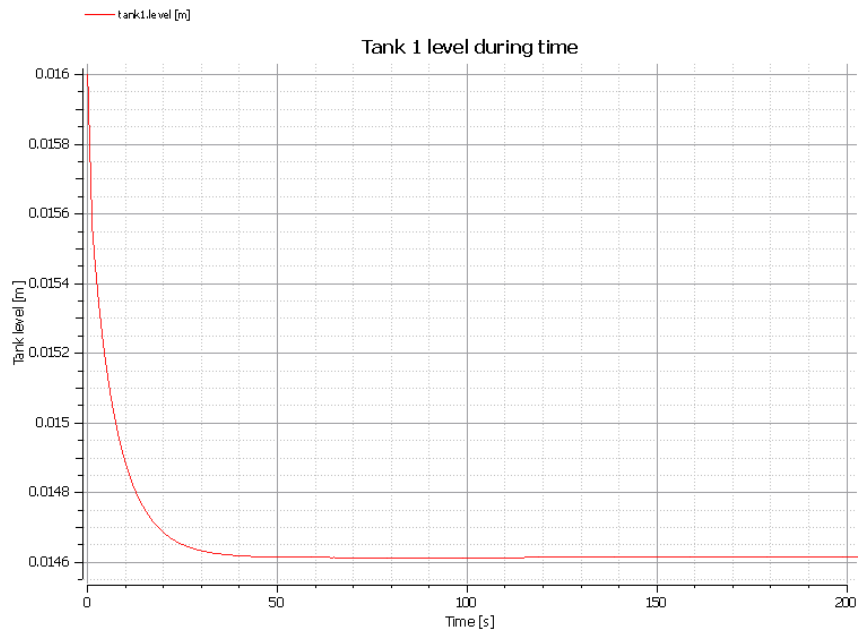


Figure 3.23: Numerical results on the tank level trend during transitory state in Tank1

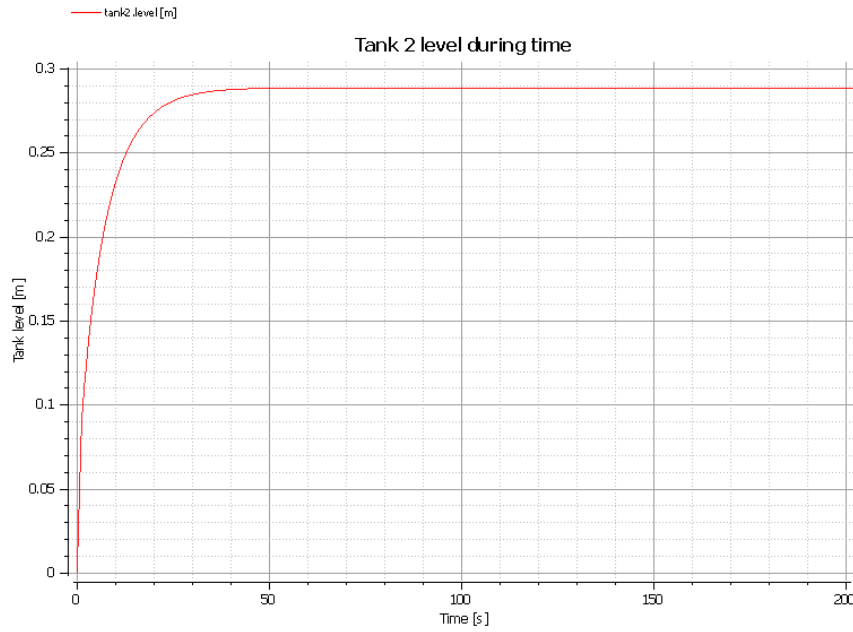


Figure 3.24: Numerical results on the tank level trend during transitory state in Tank2

3.7 Detailed analysis of the suction duct

In the hydraulic circuit of the professional oven it was necessary to evaluate also the interactions among aerodynamics (air entering the inlet of the suction duct) and hydraulics (introduction of two pipes of the washing circuit inside the suction duct).

The target of this analysis is to evaluate with a CFD tool the following objectives:

- Test the performances of a new suction duct positioned in front of the suction wall. This new solution is different from the solution present in One Oven 10GN 1/1 configuration where the suction duct is positioned behind the fan.
- Understanding of what can be the influence of the geometry of the new suction duct on the mass flow rate due to the presence of two pipes (pipe A represents the re circulation pipe in the circuit and pipe B the water entrance in the cavity Fig. 3.18 and Fig. 3.28) inside the suction duct;

The validation of the CFD model is not performed due to the lack of experimental data. The suction duct present in the One Oven 10GN 1/1 (positioned behind the fan) has a squared cross section as represented in Fig. 3.25 . The new suction duct positioned in front of the suction wall in the technology demonstrator is represented in Fig. 3.26.

3.7.1 Numerical fluid-flow analysis

The CFD analysis is composed by the subsequent phases:

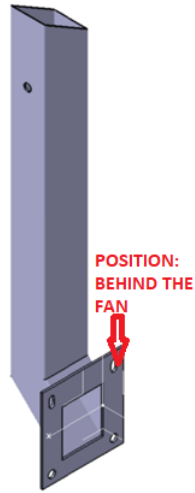


Figure 3.25: Suction duct One Oven Generation 10/1 (Position: behind the fan)

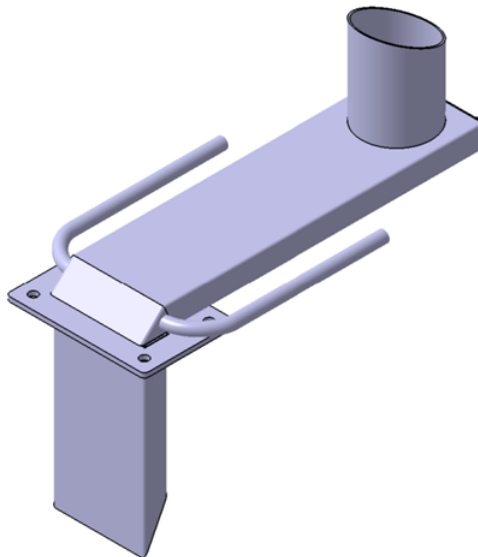


Figure 3.26: Suction duct of the technology demonstrator (Position: in front of the fan)

- CAD input;
- Simplified CAD;
- Mesh and Model set-up;
- Solving;
- Post-Processing;

A CAD geometry was selected for having a uniform flow distribution inside the suction duct minimizing the pressure drops and permitting an aspiration in front of the suction wall.

The CAD simplification aims to generate a set of clean surface (*.iges / *.stl) for extracting the fluid part from the CAD geometry in the CFD software. Fig. 3.27 and Fig. 3.28 represent the simulated fluid volume outcoming from the simplified CAD necessary for the simulation of the suction duct.

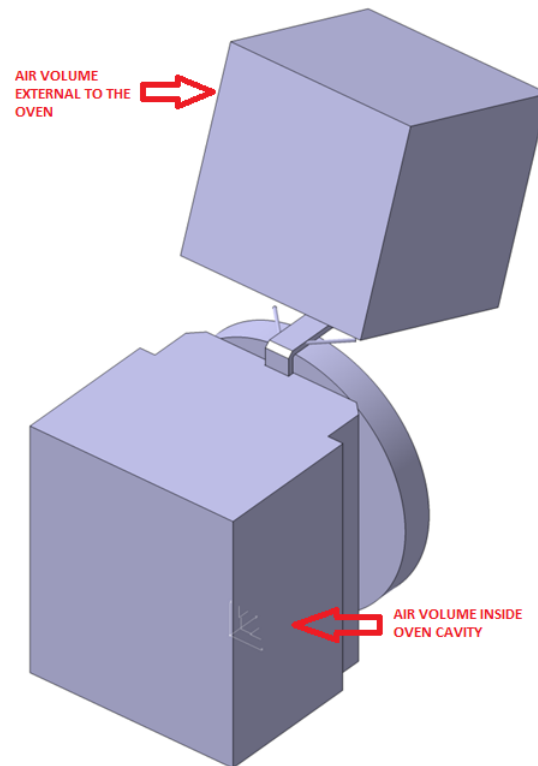


Figure 3.27: Fluid Flow Volume

3.7.2 Model description

In the computational domain the items hereinafter listed are physically represented.

- Inlet air suction duct;
- Suction duct;

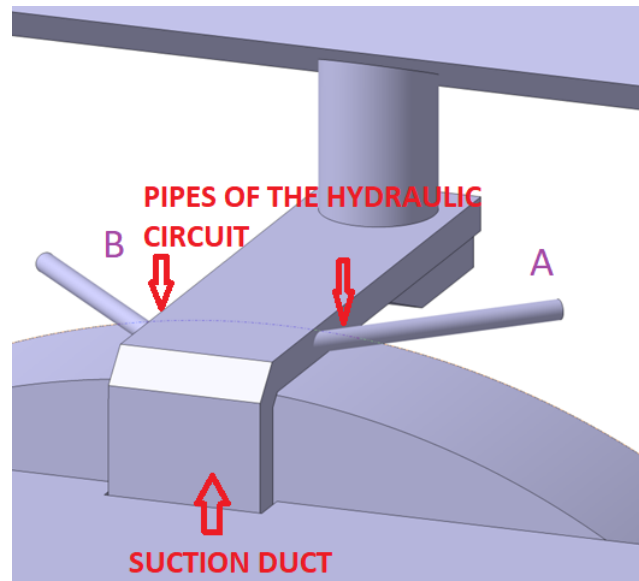


Figure 3.28: Particular of the inlet duct air volume

- Fictitious cavity;
- Suction wall;
- Fictitious fan;

The mesh is composed by 400.397 polyhedral cells and the boundary layer is represented with extruded prisms.

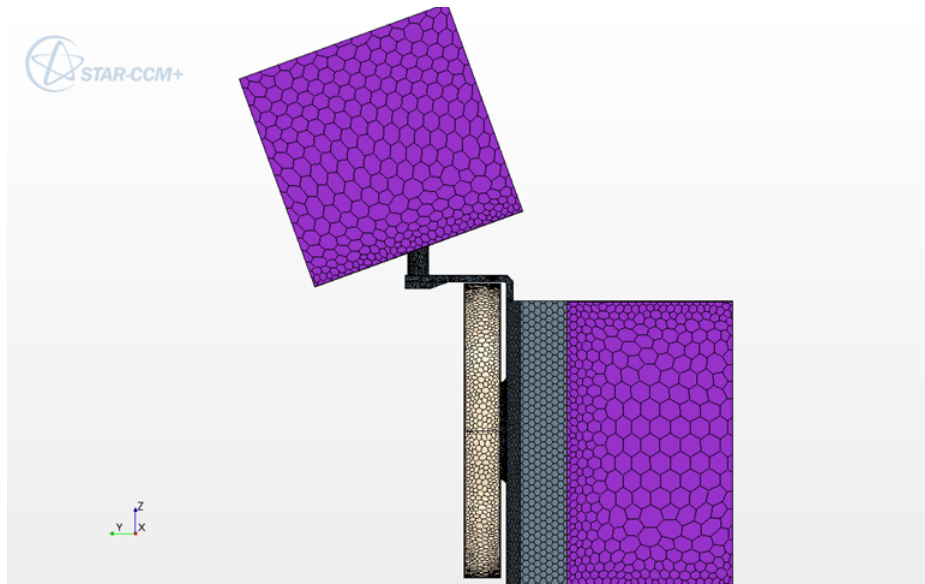


Figure 3.29: Mesh

The simulation considered is steady state and the physics implemented is hereinafter defined.

- Turbulence model: Reynolds Averaged Navier Stokes equation K- ϵ (standard);
- Equation of state: segregated flow (ideal gas law with constant density);

The Boundary conditions are:

- Inlet air duct : stagnation Inlet;
- Inlet air cavity: stagnation Inlet;
- Outlet air fan: mass flow inlet (in this case it was considered a physic value of 0.918801 [Kg\ s])
- All the other parts of the geometry are considered as walls.

The y^+ is verified to be in the suited range for the turbulence model chosen. The y^+ shows a good level of superficial mesh.

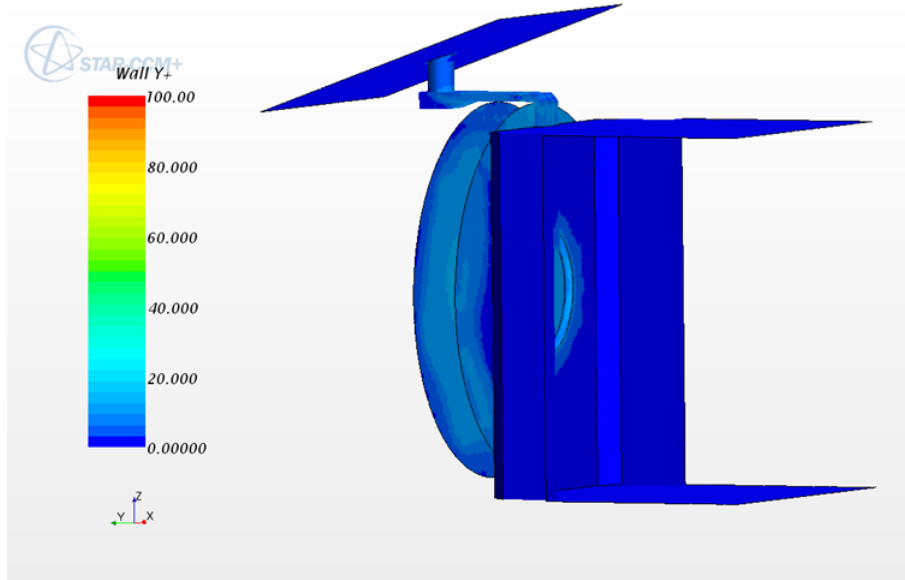


Figure 3.30: Wall y^+

3.7.3 Model solving and results

The solver is Star CCM+ v.9.06 and the solver time were 2 hours for 500 iterations (5 cores).

Pressure of the duct In Fig. 3.31 is represented the pressure distribution along the suction duct. Fig. 3.31 presents also the pressure distribution along a single streamline in the y direction and z direction.

The pressure distribution in a section of the suction duct is represented in Fig. 3.32.

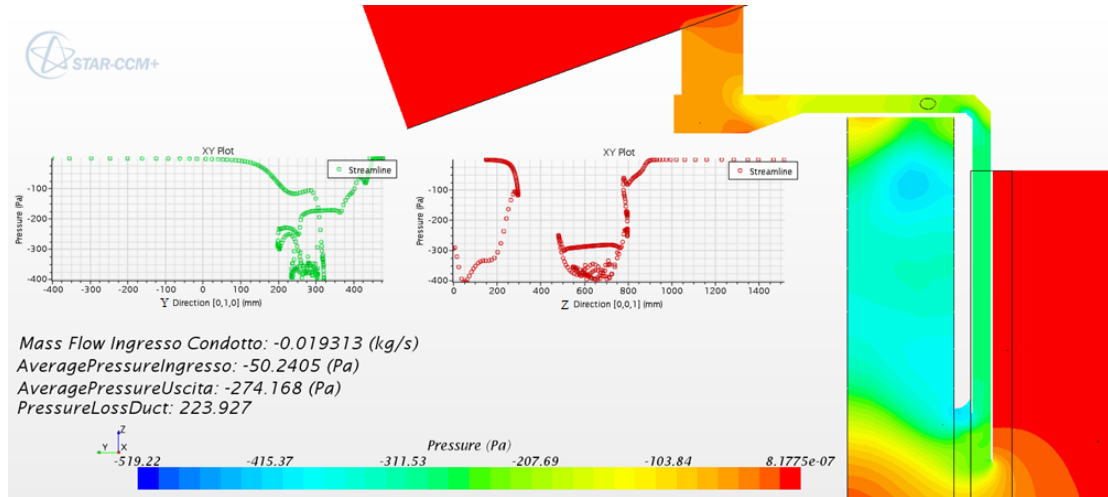


Figure 3.31: Pressure distribution along the duct

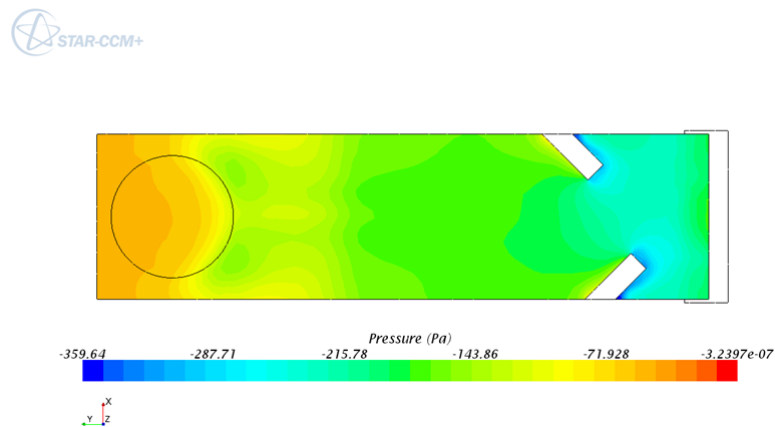


Figure 3.32: Pressure distribution in a section

Velocity plot of the duct In Fig. 3.33 is presented a velocity vector plot of the duct. The velocity vectors are following the radial direction of the fictitious fan.

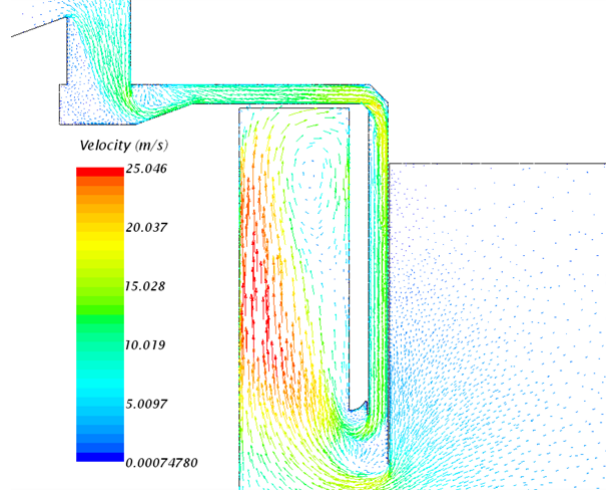


Figure 3.33: Velocity vector plot

The line integral convolution is represented in Fig. 3.34 .

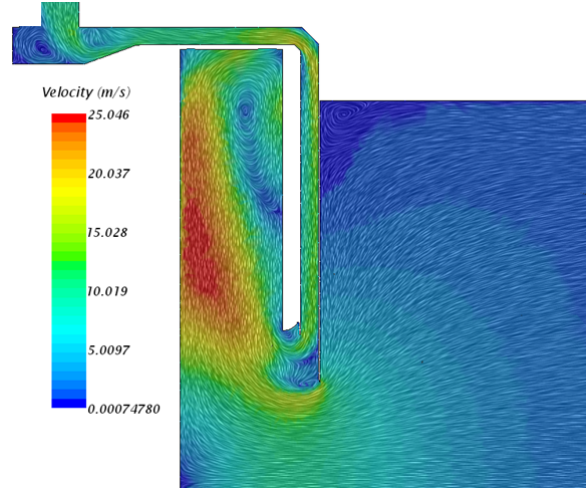


Figure 3.34: Line integral convolution

The velocity field in a section of the duct is represented in Fig. 3.35. The vectors represent the direction of the velocity field and the color map represents the magnitude of the velocity field.

3.7.4 Validation and data analysis

The model validation was not performed due to the lack of experimental data. The main aim of the model was to understand what are the possible influences of the two pipes entering the new suction duct on the air mass flow rate in the suction

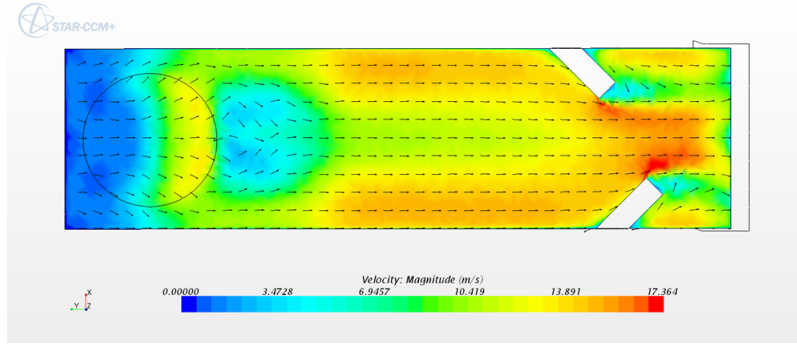


Figure 3.35: Velocity field in a section

duct. For this reason we did not perform an experimental analysis, but the results of the simulation were used only for comparative analysis with the same simulated geometry without pipes.

In Tab. 3.4 are presented some results. The first column of Tab. 3.4 represents the experimental values measured on the One Oven Generation 10GN 1/1 (suction duct positioned behind the fan). The second and third columns represent respectively the simulated values for the new duct (section : 80mm x 20mm) in the representative device without and with the entering pipes of the oven washing plant. The new duct has a flow rate 39,1% higher compared to the duct present in the One Oven Generation 10GN 1/1 with a section area 20% smaller. The new duct without pipes has 0.7 % higher mass flow compared to the duct with pipes.

	One 10GN 1/1 Elect.	T.D.80x20 Simulated without pipes	T.D. 80x20 Simulated with pipes
Air Flow inlet velocity [$\frac{m}{s}$]	4,1	/	/
Air flow rate [$\frac{m^3}{s}$]	0,011	/	/
Mass Flow Rate [$\frac{kg}{s}$]	0,014	0,0194	0,0193
Section area [m^2]	0,00200	0,00160	0,00160
% Difference Mass Flow Rate	/	39,1	38,4
% Difference Section area		-20,0	-20,0

Table 3.4: Comparison among numerical and experimental results

3.8 Required pump characteristics in the hydraulic circuit of a professional oven

A centrifugal pump in the hydraulic circuit of a professional oven ensures fluid flow circulation as well as the kinetic energy needed for the soil removal.

The recirculating pump in a professional oven is generally characterized to have backwards and radial blade geometries (the yield of the backwards geometries is higher compared to the other geometries but it has lower specific energy (kinetic energy) compared to the other geometries).

The choice of the recirculating pump is an important aspect in the design process of a washing plant. The most general selection criteria for a pump are the rpm of the pump, the delivered flow rate, the total head between suction and discharge and the cavitation margin. Based on the prior analysis of the hydraulic circuit of the professional oven some preliminary centrifugal pumps have been identified.

The selection process of the pump for particular applications like the hydraulic plant of professional cooking appliances (oven) or dishwashing appliances (rack type or hood dishwasher), cannot rely only on such general selection criteria, but it depends also on other factors as the operating temperatures of the solution of detergent and water, the compatibility of detergents with the pump materials and the effect of detergent on pump performances and cavitation. In particular, detergent solutions are critical fluids, since in professional appliances, the washing environment can reach harshly levels, degrading the materials and components of the pump. Also, high concentrated chemistry could impact on the working conditions of the pump because the fluid properties are different from those of pure water.

In fact, as introduced before, the actual trend in this kind of professional appliances is to reduce both energy consumption and time needed for cleaning process: this involves short washing cycles conducted at low temperatures with a solution of water and highly concentrated chemistry. Detergents contain different components and additives, as polymers and surfactants, which can affect the performance of the pump, including cavitation inception conditions. Cavitation leads to flow instabilities, affecting pump performances, inducing an increment in the level of vibrations and noise and in long term it can destruct pumps internal components.

While cavitation phenomena in Newtonian fluids is well known, particularly as far as pure water is concerned, in literature there are also various studies on cavitating flows in presence of diluted solutions of polymers additives in water [36] (studies performed on blunt bodies, constriction elements as orifice and nozzles), but only few studies are available regarding the effect of detergent components on pumps cavitation and, in general, on pumps performances [36]. The wide range of variables affecting the phenomenon has led to the development of a laboratory rig for testing centrifugal pumps with aqueous solutions representative of those used in the warewashing sector [36]. In the test rig it is possible to test different pump sizes .

This test rig permits measuring pump performances at various operating conditions, in order to obtain its characteristic curves, and also forcing cavitation to measure its Net Positive Suction Head required (NPSH_r) at different flow rates. The pump test rig allows also testing various cavitation conditions, obtained by changing not only the suction pressure and temperature of the fluid but also its properties, adding detergents and additives. Cavitation inception can be detected measuring both the corresponding prevalence decrease and the change of vibration and noise level.

For a general understanding of the problem it was decided to select and study a pump used in the dishwashing sector due to its bigger dimensions compared to the others pumps present inside Electrolux Professional. This pump permits to have a scaled view of the influences of detergent components on performances, cavitation and vibrations compared to the centrifugal pump used for example in a oven. Furthermore, the selected pump is the only pump inside Electrolux Professional with an impeller made of AISI 304. All the other pumps have an impeller made of plastic materials. Pumps subjected to various cavitation test with an impeller made of plastic materials have a lower lifetime compared to pumps made of metallic materials.

3.9 Conclusions

In the first part of this chapter is presented an analysis of the four basic factors of a cleaning process (Sinner circle) : chemical action, temperature, mechanical action and time. A good combination of the cleaning factors allows to have good washing performances supporting energy efficiency improvements. The actual trend in washing processes is to reduce the use of the four parameters increasing in particular the chemical action.

The first analysis is followed by an investigation on a typical phases composition in a washing cycle. Different washing cycles present in the Electrolux Professional One oven 10GN 1/1 have been analysed in terms of energy absorption, length and water consumption. Subsequently, the investigation continues with a comparison among open and closed washing systems in professional oven benchmarks, where the results show that closed washing solutions are better compared to open washing systems in terms of energy, water, detergent and rinsing agent consumption. For developing a new line of professional ovens it was decided to prosecute the development of the washing system with a closed solution.

The behavior of closed oven washing system has been studied using before a simple worksheet and then a numerical tool considering the transitory state in order to define a lay-out of the plant, select the pump for the circuit and understand the potential performances achievable in a specific configuration increasing the accuracy and decreasing the time needed for developing prototypes. The pressure losses, the working point of the pump and the NPSHa have been simulated with an Open Modelica script, in particular for understanding the tank level trend (oven cavity - quenching system) and the fluid flow available at the two branches (oven cavity - quenching system) necessary for calibrating the pressure losses at each branch. After a first screening with the model, in terms of feasibility, a more accurate analysis has been performed experimentally for understanding if the predictions are accurate and if the cleaning results in the oven are fulfilled.

Furthermore, a CFD analysis of the oven suction duct interacting with with the

pipes of the washing circuit has been presented. The model validation has not been performed due to the lack of experimental data. The new duct has a flow rate 39,1% higher compared to the duct present in the One Oven Generation 10GN 1/1 with a section area 20% smaller. The introduction of pipes decreases of the 0,7 % the mass flow rate, compared to the duct without pipes.

In the last section of the chapter, a general pump selection process criteria is presented. The wide range of variables affecting the cavitation phenomena and pump performances has led to the development of a laboratory rig for testing centrifugal pumps with aqueous solutions representative of those used in the warewashing sector. A pump used in the dishwashing sector has been selected because it permits to have a scaled view of the influences of detergent components on performances, cavitation and vibrations compared to the centrifugal pump used for example in an oven.

In the next thesis chapter is presented the developed test rig with all its capabilities and the analysis performed for understanding the influence of detergent components on pump performances, cavitation, vibrations and noise.

Chapter 4

Analyses of washing circuits pump performances

4.1 Introduction

Pumps used in professional appliances operate in demanding conditions because they process a solution of hot water and detergents. Certain components of detergents impact on pump performance also in terms of cavitation inception and behavior. For these reasons, in this chapter a bibliographic analysis is presented of the interactions between detergent components and rheological properties of the water detergent solutions. Then, it is described the test rig used for measuring the performance of pumps in operative conditions typical of professional appliances. The obtained results are analyzed in detail.

4.2 Effect of detergent components on cavitation and pump performance

The choice of the pump has a key role in the design process of the hydraulic circuit of a professional appliance, because both the fluid flow circulation and the kinetic energy needed for the soil removal are supplied by the machine, usually of the centrifugal type. As already described in chapter 3, a pump is suitable for these applications only if it satisfies a series of material compatibility, safety, cavitation margin and performance requirements due to the pumped water detergent solution and to the temperatures involved in a washing cycle, as presented in chapter 3.8.

In this chapter, a first literature research is presented on the possible influences of detergents, or of some detergent components, on pump operating conditions. Aims of the research is to understand what are the most influencing variables and to identify a general component of detergents that could be tested as representative of their class, from the point of view of the strongest effect on the pump performance and cavitation inception.

Generally speaking, the cleaning process is performed in three main steps. The first step consists of a physical process and of chemical reactions that aim to displace the soil from the substrate. The second phase has the function of dispersing the soil into the cleaning medium. The last phase aims to prevent the soil re-deposition. The cleaning process is then guided by four main factors that are chemistry, mechanics, temperature and time. These factors and their interactions have been explained in paragraph 3.2, where it was pointed out that the actual trend in professional appliances is to reduce the time needed for the cleaning process, with short washing cycles conducted at low temperatures [33]. This requires a concentrated chemistry for having good washing results, and high concentrated chemistry impacts also on pump performance, especially on cavitation.

A cleaning product is characterized by different components as surfactants, al-

kalis, acids, builders and other substances. Basically, the detergent effect in a professional appliance is made by the action of builders and surfactants. They and the other detergent components affect the water-detergent solution properties in such a way that often it can no longer be considered as a Newtonian fluid, like pure water [37], [38]. Despite the fact that builders and surfactants can have an influence particularly on cavitation inception [39], in literature there are only a few studies on this subject [40], [41], [42], [43], [39], [44], [45]. The wide range of variables affecting the phenomenon has then pushed the necessity to realize a test rig that permits to study cavitation in centrifugal pumps running in operating conditions and with the detergents used in the professional appliances sector. The test rig will be presented in the following paragraph 4.3. The thesis continues with an experimental analysis of vibrations and noise during the overall pump operating conditions and with the selection of the most suitable statistical parameters for evaluating the vibration components induced by cavitation. This analysis has been carried on in order to identify a sensor and to develop a control system for avoiding pump cavitation in professional appliances. The chapter concludes with the study of the effects of various solution of the representative detergent component on pump performances, using the previously developed experimental and signal analysis tools .

4.2.1 Brief analysis of cavitation inception phenomena

Cavitation inception happens when bubbles appears in the fluid flow. The list hereinafter classifies hydrodynamic cavitation cases according to [39].

- Travelling cavitation:
 - Characteristics : Individual transient bubbles which expand and shrink. When bubbles enter in a region of high pressure they collapse.
 - Zones : Bubbles appear in low pressure zones along solid boundaries, or in moving vortices.
- Fixed cavitation :
 - Characteristics : Develops after inception. Flow detaches from the rigid boundary of an immersed body or a flow passage.
 - Zones : Bubbles appear in highly turbulent surfaces, in particular in the low pressure side of the blades.
- Vortex cavitation :
 - Characteristics : Bubbles form in low pressure cores of vortex regions of high shear.
 - Zones : Bubbles appear in open and ducted propellers.

Considering the relative flow of an object immersed in a fluid, cavitation inception can be reached by lowering the values of local pressure on the surface of the object until a critical pressure p_c . If the surface tension of the bubble is neglected, p_c is also the value of the pressure inside the bubble [39], equal to vapor pressure p_v . An index of dynamic similarity in such condition is the incipient cavitation number, Eq.4.1, where p_0 e v_0 are pressure and relative velocity of the undisturbed liquid at some distance of the object.

$$\sigma_i = \frac{(p_0 - p_c)}{(0.5\rho v_0^2)} \quad (4.1)$$

While the incipient cavitation number is obtained decreasing the pressure until cavitation inception is achieved, a similar index, known as the desinent cavitation number, is obtained increasing the pressure from a cavitation situation until no cavitation is reached. These numbers are not always equal [39]. Cavitation inception and development are strictly related to the bubble dynamics that is associated with the number, distributions and size of gaseous weak spots in the liquid, called nuclei, which dimensions can range from few micrometers to some hundreds of micrometer. Free, or undissolved, gases can act as cavitation nuclei while the dissolved gas content affects their number, size and growth [39]. A free nucleus can be considered to be a spherical bubble, containing vapor and some permanent gas, which stability condition is given by Eq.4.2 [46], where S is the surface tension and R_b the bubble radius.

$$p_v - p_c = \frac{4S}{3R_b} \quad (4.2)$$

Eq. 4.2 shows that the critical pressure for cavitation inception, if the surface tension is taken into account, is less than the vapor pressure but approach it for sufficiently large values of R_b [46]. Considering a pressure field that is constituted by a local mean pressure component p_{l0} and a time varying component, it is possible to calculate with Eq. 4.3 [46] the threshold amplitude of the pressure oscillation, p_{tv} , needed for a nucleus of an initial radius R_n to grow, according to a vaporous growth mechanism, up to the critical radius at which nuclei become unstable.

$$p_{tv} = p_{l0} - p_v + \frac{4S}{(3\sqrt{3}R_n)} [1 + (p_{l0} - p_v) \frac{R_n}{2S}] \quad (4.3)$$

Another mechanism responsible for bubble growth in time varying pressure fields is gas diffusion, also called rectified diffusion [47]. If there is an oscillation in the ambient pressure, when the bubble is larger than its mean size gas in the solution tends to enter, due to the reduction of its partial pressure in the bubble. On the contrary, gas tends to dissolve when the bubble size is smaller than its mean size. The influx of gas is usually higher than the efflux, so that the bubble tends to increase. This is due to the fact that when the bubble is larger there is a thinner and stretched

diffusion boundary layer, with respect to the smaller bubble phase, but there is also a higher surface area [47].

The result is that a small bubble in the fluid under a steady pressure field can disappear but in case of an unsteady pressure field it can grow due to the rectified diffusion. Eq.4.4 [46], where C_∞ is the actual concentration of the gas, permits to calculate the threshold pressure amplitude, p_{td} , needed for a nucleus to growth by rectified diffusion. This threshold pressure can be an order of magnitude smaller than that for vaporous cavitation for nuclei greater than 1 m.

$$p_{td} = \frac{\sqrt{2}}{3} p_{l0} \left[1 + \frac{2S}{R_n P_{l0}} - \frac{C_\infty}{C_0} \right]^{0.5} \quad (4.4)$$

Another parameter that influences cavitation inception is fluid viscosity. It affects the flow turbulence level that determines, in particular, the development of boundary layer and all the related viscous effects. As a matter of fact, cavitation inception occurs mostly in this region, where laminar flow separations and local turbulent transitions are locations of high pressure fluctuations. So nuclei size modifications can occur through mass transfer, as previously discussed, depending also on residence time, which is also affected by viscosity. In [46] the effect of possible different trajectories of a bubble in a boundary layer are described in order to discuss the effects of different pressure fields and residence times on bubble evolution. A bubble involved in boundary layer transition has more residence time in a low and unsteady pressure field. If it goes through a short laminar separation region, the bubble is subjected to a more complex pressure field, characterized by a steady value followed by pressure oscillations in the reattachment zone. A bubble entering in a recirculation region and then into the reattachment zone is subjected to an intermittent steady pressure near the separation region, and an unsteady pressure near reattachment. In dishwashing machine, detergents change the rheology, viscosity and vapor pressure of the solution due to their contents of polymers, surfactants and other substances described in the next paragraph. As a consequence, detergents play a significant and hardly predictable role in the cavitation inception behavior of the pump. Experimental analysis is so necessary to underline the influences of ware washing detergents on cavitation inception in the real operating conditions of a professional dishwashing pump.

4.2.2 How detergents/polymers components impact on viscosity based effects of cavitation

Detergents cover the entire spectrum of viscosity from low viscosity Newtonian fluids to semisolid pastes [38]. Moreover, the surface layer of a detergent solution can have mechanical properties different from those of the solution itself [37]. This impacts on the stability of foams, because highly stable foams present higher surface

viscosity than not stable foams [37]. The desired viscosity and aesthetic formulation in detergents is guaranteed with some appropriate rheology modifiers [38]. A significant impact on the rheology is due to surfactants, which are present in particular in all liquid detergents. At low surfactants concentration, liquid detergents have low viscosity and are approximately Newtonian, but if the concentration of surfactants is high, surfactants tend to diverge from the Newtonian behaviour [38]. In automatic dishwashing machines the common rheology modifier used are carbomer and cross-polymers, having a high degree of pseudo plasticity [38]. An analysis of the viscosity on some detergents is presented in [48], [49] and [50]. The viscosity behaviour of cationic detergents tetradecyltrimethylammonium chloride in water and various sodium chloride solutions is presented in [48] and represented in Fig.4.1. The viscosity behaviour of three different aqueous solutions of ionic detergents (potassium caprate, sodium oleate and dodecylamine hydrochloride) is presented in [49] and represented in Fig.4.2. From Fig.4.1, 4.2 is possible to highlight the increasing trend of the viscosity with the solute's concentration [48], [49].

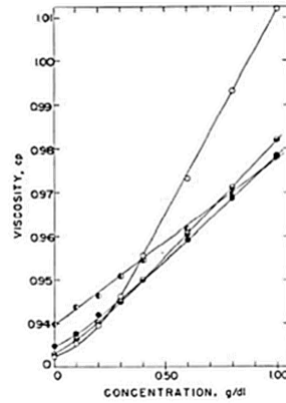


Figure 4.1: Viscosity behaviour of tetradecyltrimethylammonium chloride in water and various chloride solutions [48]

The wide rheological spectrum of a liquid detergent requires to examine the water-detergent solution behaviour (Newtonian / Non Newtonian). In literature there are studies highlighting that many polymer solutions are characterised to be Non Newtonian and they have particular effects on the flow as drag reduction, pressure drop reductions and vortex inhibition [39]. In detergents, particularly on ware washing ones, polymers are usually present. Tab.4.1 [33] shows the characteristic components used in a high alkaline ware washing block detergent [33]. For each component, it is highlight if it is a polymer. In cavitation literature different studies are presented on the effects of addition of polymers in water aiming to study the influence of polymer on cavitation inception [39, 40] [42] [43] [44] [51]. The effects of a polymer solution concentration in different cavitation conditions as jet cavitation, vortex cavitation and cavitation around blunt bodies are presented in [39]. Considering for example a

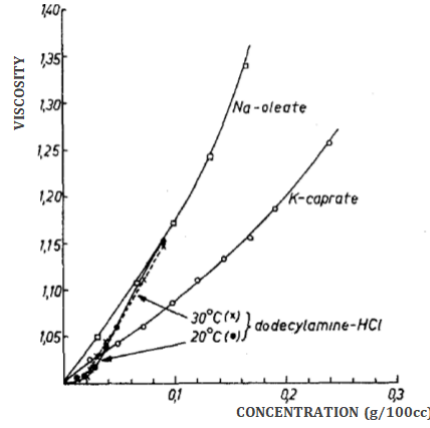


Figure 4.2: viscosity behaviour of three different aqueous solutions of ionic detergents [49]

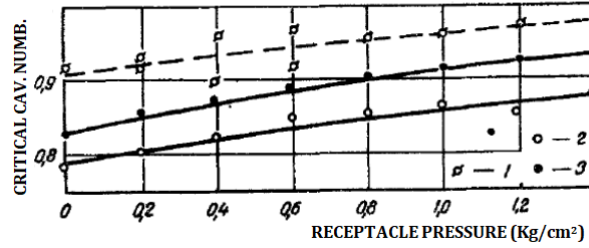


Figure 4.3: critical cavitation number versus pressure in the tank for 1) water ; 2) 1000 ppm Polyox WSR-301; 3) Butyl alcohol [40]

jet cavitation, that occurs in constriction elements as orifices, nozzles. When there is a reduction of static pressure to a critical value there is an increase in the nuclei formation, but a very small amount of polymers additives (diluted solution of flexible polymers, possessing very high values of extensional viscosity) is able to suppress the inception of cavitation. Polymers additives can also affect the cavitation noise, in particular decreasing the time averaged shock pressure [39]. A similar research is presented in [40] where experiments were conducted with a hydrophone placed in pulsed cavitation equipment. The author studies the effect of polymer additives (Polyox WSR-301 used in soaps). Cavitation number diminishes of 15% for a 0.1% Polyox solution, and 8% for a 0.47% butyl alcohol solution. This behaviour of the cavitation number is represented in Fig.4.3 [40]. Other studies presented in [40] show that surface active agents impact on cavity formation and can reduce the cavitation noise, lengthen the time before inception and decline the erosion. In [43] it is shown that there is an increasing cavitation threshold and a reduced incidence of damage with an increasing amount of polymer additives. The mechanism of cavitation suppression by polymer additives is discussed in [39]. It is not completely understood, but in the case of cavitation around blunt bodies, it is due to an earlier transition

to turbulence, suppressing the laminar separation region, where cavitation usually happens. Instead in tip vortex cavitation, the ejection of polymer from the wing tip reduces the tangential velocity increasing the pressure in the core of the vortex, retarding the cavitation inception. In [42] authors investigate the role of polymers on the delay of tip vortex cavitation. Polymers increase tip vortex core radius delaying and lowering cavitation inception. In vortex chamber with polymer solution, cavitation inception is delayed appearing to be due to an increased turbulent kinetic energy rather than a growth of the viscous core or a decreased rotational speed [44]. A homogeneous polymer solution suppresses cavitation in foils acting on tangential velocities modifying circulation and lift [39]. Some polymers in viscoelastic fluids suppress cavitation inception affecting extensional viscosity, lowering the speed of the solution and increasing resistance and static pressure [39].

Ingredients	Type of component	Polymer	Phosphate	Nonphosphate
Water	-	-	3 – 10%	8 – 15 %
Sodium hydroxide (50%)	Inorganic compound	No	7 – 20 %	7 – 20 %
Sodium polyacrylate (50%)	Anionic polyelectrolyte with negatively charged carboxylic groups in the main chain	Yes	8 – 20 %	2 – 6 %
Phosphino polycarboxylate	Anionic polymers	Yes	-	-
Desquet 2010	-	Yes	-	-
Nonionic surfactant	E.g. Polyethylene glycol. Forms:oligomer or polymer of ethylene oxide.	No	1.5 – 3	1.5 – 3
Sodium hydroxide bead	Inorganic compound	No	40 – 45	40 – 45
Sodium sulphate	Inorganic compound	No	5 – 10	5 – 10
Sodium chloride	Inorganic compound	No	3 – 6	3 – 6
Sodium tripolyphosphate	Inorganic compound	No	30 – 45	-
Phosphate ester defoamer	Deareating wetting agents	No	0.2	0.2
Solid chloride bleach source	Salt of hypochlorous acid	No	7.5	7.5

Table 4.1: High alkaline ware washing block detergent compositions - US Patent 4846993 [32]

4.2.3 Surfactants impact

When a bubble is placed in an oscillating pressure field, rectified mass diffusion can occur [52] [53], as seen in paragraph 4.2. Surfactants influence rectified diffusion [40] [45]. Surfactants are present in detergents and tend to form aggregates such as micelles [54], constituted by a hydrophilic head directed through the water interface and a hydrophobic tail directed through inside and attracted by the soil. The main properties and actions of the surface active agents are wetting, emulsifying, peeling and foaming actions. The bubble growth rate is enhanced by the action of the surfactants due to their impact on surface tension, interfacial resistance to mass transfer and surface rheological properties. An effective plot of these effects is reported in Fig.4.4 [41] ,4.5 [41] ,4.6 [41] where it is possible to see the variation in bubble radius and growth rate. In Fig.4.5 the initial bubble radius is 25 μm and is smaller than the threshold bubble radius for rectified diffusion that is 32 μm at 0.22 bar and 22.1 kHz [41]. In Fig.4.4 it is shown that the size of a bubble of initial radius 20 μm remains constant over 300 s and then it dissolves.

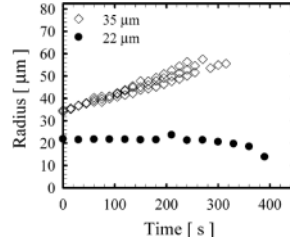


Figure 4.4: variation of bubble radius as a function of time in air saturated water [41]

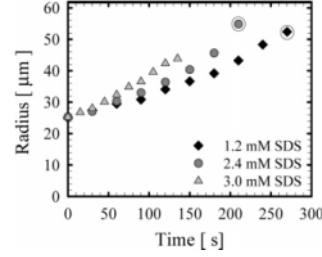


Figure 4.5: variation of bubble radius as a function of time for different bulk concentrations of SDS [41]

4.2.4 How detergents components impact on vapor pressure

Detergents have the property of lowering the surface and interfacial tensions other than solubilising and emulsifying abilities. These properties are related with the thermodynamics of the solution and, in particular, the trend of the partial vapour tension of the solvent in respect to that of the pure substance is given by the Raoult law. The properties of nineteen solutions of detergents at various concentrations are presented in [55]: considering for example a potassium soap (potassium caprate) diluted in water, the Raoult law is represented in Eq.4.5.

$$p_A^p = x_A p_A^{ps} = \frac{m_{H_2O}}{m_{H_2O} + m_{PC}} \times p_A^{ps} \quad (4.5)$$

Given the molality of the solution, it is possible to obtain the moles of solute, m_{pc} , and the corresponding grams, w_{pc} , with Eq.4.6 where, for potassium caprate, MM_{PC} is 210.35 g/mol.

$$w_{PC} = MM_{PC} \times m_{PC} \quad (4.6)$$

The grams of solvent are calculated subtracting the grams of solute from one kg of solution, and the subsequent moles of water, m_{H_2O} , are obtained dividing by its molar mass. Applying Eq.4.5, the resulting variation of the partial vapour tension of water as a function of the molality of potassium caprate is obtained at a given temperature, as shown in Fig.4.7.

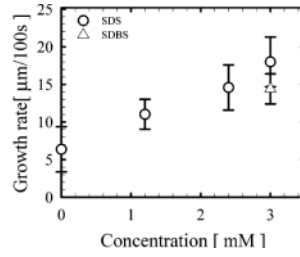


Figure 4.6: bubble growth rate as a function of SDS bulk concentrations [41]

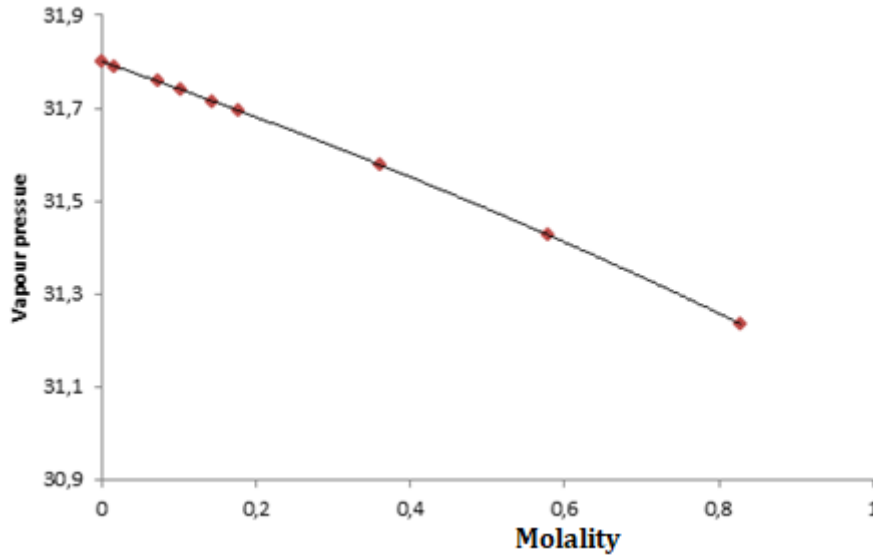


Figure 4.7: Calculated partial vapour pressure (in mmHg) vs. molality of water potassium caprate solutions at 30 C, based on the data presented in [55]

4.2.5 Final considerations

From the literature research appears that there are only few studies relating cavitation inception with the presence of detergents in water solutions. These detergents could affect cavitation due to their content of surfactants - polymers, which alter the rheology, surface tension, viscosity and vapor pressure of the solution with respect to the case of pure water. It emerges that surfactants enhance the bubble growth rate, impacting on rectified diffusion and polymers decrease the incipient cavitation number. With the aim of studying and monitoring the influence of these parameters on cavitation inception, an experimental pump test rig has been constructed (paragraph 4.3). Cavitation inception can be detected measuring both the corresponding total head decrease and the change of vibration and noise levels.

4.3 Test rig description

A test rig has been designed and constructed according to the International Standard 9906:2012 [56] and other literature studies [57], [58]. It allows measuring the characteristic curves and NPSHr of different pumps, with various fluids (e.g. water - polymers solution and detergent at different concentrations). The pump is installed in a closed circuit, as presented in Fig.4.8, whose main pipes have external diameter DN 65. In the NPSHr test the fluid flow is fixed and the NPSHa is progressively reduced. Such reduction is achieved acting on the pressure of the free surface level in the tank, until a specific value of pressure is reached (see Fig.4.10). In this plant it is also possible to induce cavitation at a fixed fluid flow by changing the water temperature by means of two electric resistances (one of 17 kW and the other of 9 kW). The tests have been performed at a fixed rotation speed of the pump (2900 rpm). A 3% head decrease at a certain fluid flow has been considered as cavitation signal [56,57]. The tank is designed to avoid inclusions of gases in the suction of the pump. Inside the tank are installed two calm screens, a spray nozzle device and the connection to a vacuum pump. The fluid-free surface pressure is measured with a vacuum-pressure gauge. Two relative pressure transducers by Trafag are positioned at a distance of two diameters from suction and outlet sections of the pump (symmetric uncertainty on the confidence level of 95 % is 0.002 bar, precision 0.15% on full scale). The aspiration transducer has a measurement range of ± 1 bar (power supply: 24 V-DC, signal output: 4-20 mA) instead the pressure transducer at the delivery has a measurement range between 0 bar and 2.5 bar (power supply: 24 V-DC, signal output: 10 V-DC). The pump flow rate is measured with an electromagnetic flow meter provided by Endress Hauser with a measurement interval 0 – 950 l/min (power supply 230 V-AC, output signal 4-20 mA, error 0.5% on the read value). Flow rate measurements accuracy is reached positioning the flow meter after ten diameters from the pipe curve and at five diameters from the branch to the deaeration nozzle. The flow rate regulation is obtained by throttling the valve 2 in Fig. 4.8, installed in the discharge pipe. The pump is driven by a three-phase AC electric motor of 2.2 kW. A twelve diameters length straight transparent Plexiglas pipe is installed at the suction side of the pump, for reaching a uniform distribution of velocity and pressure. Another transparent Plexiglas pipe is also positioned at the delivery section of the pump. These transparent pipes are used for visualising the flow in cavitation conditions. The temperature of the solution in the tank is measured with a Pt100 (symmetric uncertainty on the confidence level of 95 % is 0.2 °C, measurement interval is – 50°C to 600 °C) and a K thermocouple is used for monitoring the resistance temperature. All the measurements are acquired by means of a LabVIEW® controlled N.I. CompactRIO® system, and the data are then processed with MATLAB® codes. A Yokogawa digital power meter is used for measuring the

electric parameters (accuracy $\pm 0.1\%$ in reading, $+ 0.05$ of range).

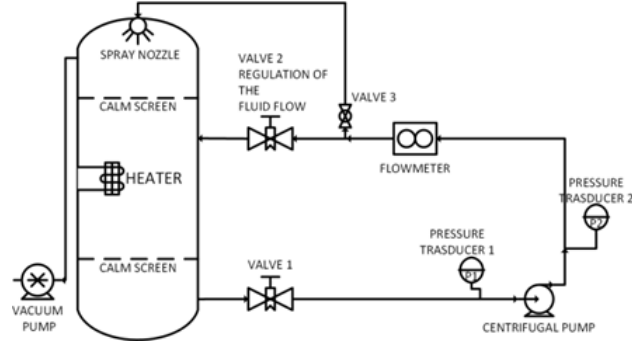


Figure 4.8: Functional scheme of the test rig



Figure 4.9: Picture of the test rig

4.3.1 Measurement of pump characteristic curves and NPSHr

The tests have been performed at nine fluid flow values, from 0 l/min up to 800 l/min with a step of 100 l/min. Fluid flow, absorbed power and temperature measurements have been acquired at low sample frequency (10 Hz), while the others at high sample frequency. The pressure sensors signals have been acquired in the initial phase of the experimental campaign at 12800 Hz, then the sampling frequency was shifted to 25600 Hz, as explained in paragraph 4.3.2.

Eq.4.7 gives the total head of the pump:

$$H_T = \frac{\Delta p}{\rho g} + \frac{(u_d^2 - u_s^2)}{2g} + \Delta z \quad (4.7)$$

where Δp is the pressure difference between the pump delivery and suction sections, u_d is the flow velocity at the delivery while u_s is the flow velocity at the suction. These velocities are calculated according to the measured flow rate and the diameters of the corresponding sections. The density ρ is obtained on the basis of the temperature measured during the test and Δz is the height difference between the

two pressure transducers. The total efficiency of the pump is calculated according to Eq.4.8 :

$$\eta = \frac{(\rho g Q H_T)}{\sqrt{3VI}} \quad (4.8)$$

where V and I are the voltage and intensity of the current measured at the pump electric motor. The required net positive suction head, $NPSH_r$, is defined in Eq.4.9:

$$NPSH_r = \frac{(p_s)}{\rho g} + \frac{(u_s^2)}{2g} - \frac{(p_v)}{\rho g} \quad (4.9)$$

where p_v is the vapor pressure at the test temperature and p_s the pressure at the suction section of the pump.

In a typical test, pump rotating speed and flow rate are maintained constant during signal sampling process. The test is performed lowering slowly the tank pressure by means of a vacuum pump, until the total head decreases more than 3% at every analyzed fluid flow (Fig. 4.10). The pump characteristic curves are then obtained changing the mass flow rate and interpolating the obtained data as shown in Fig. 4.11.

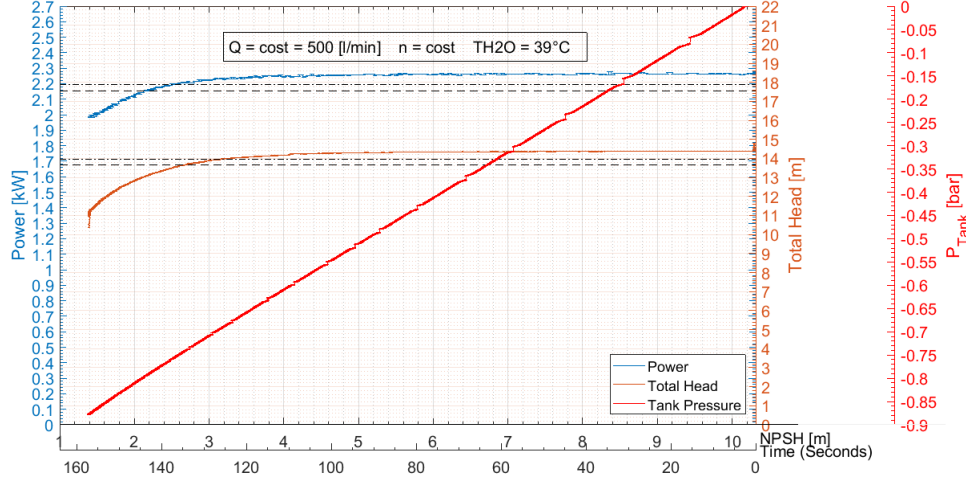


Figure 4.10: Total head and power decrease due to the decreasing of the tank pressure

In the initial stage of the experimental analysis, some test have been performed acquiring the sensors of the test rig during the lowering and the rising of the tank pressure in order to detect cavitation hysteresis. A plot of the pump Δp in dependency of the $NPSH_a$ of the test rig is represented in Fig. 4.12 where is possible to see cavitation hysteresis.

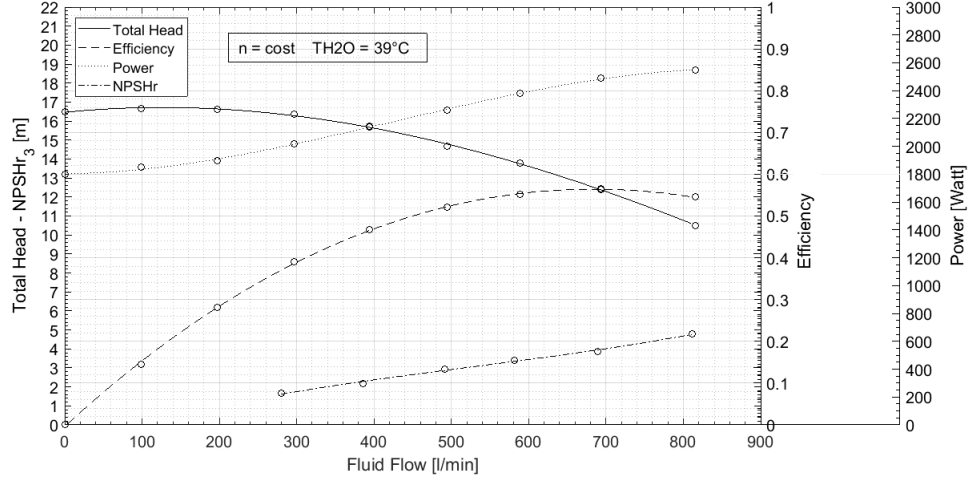
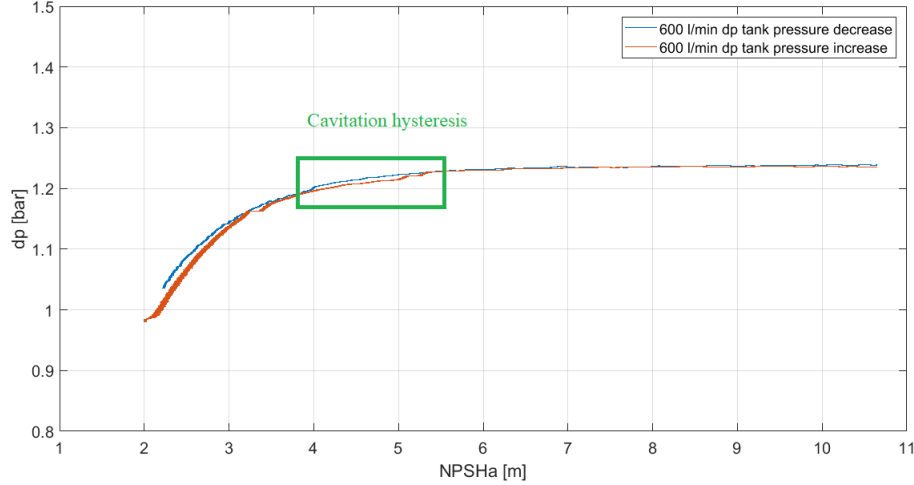
Figure 4.11: Total head, absorbed power, efficiency and NPSH_r of the pump

Figure 4.12: Cavitation hysteresis

4.3.2 Measurements of vibrations and noise

Vibration measurements have been acquired with three 352C03 ICP [®] accelerometer sensors. The accelerometer sensitivity is ($\pm 10\%$) 10 mV/g, measurement range ± 500 g, frequency range ($\pm 10\%$) 0.3 to 15000 Hz, resonant frequency > 50 kHz.

Acoustic measurements have been acquired with a PCB Piezotronics free-field microphone model 130E20, placed 50 cm from the pump, perpendicular to the pump axis. The microphone frequency response is in a range of 20 to 20000 Hz (± 5 dB). Vibration and noise measurements have been acquired, as the pressure measurements, at a sampling frequency of 12800 Hz or of 25600 Hz for collecting high frequency signals. The sample rate is a compromise between a not overloaded measure and the capability to detect cavitation process associated with random signals / high

frequencies spectrum.

Accelerometer sensors have been positioned as in Fig.4.13, covering three normal axis along significant directions in the pump volute. In particular, the two accelerometers positioned along the Z and Y directions acquire radial accelerations, instead the sensor positioned along the X direction acquires the axial acceleration.

Pump inlet and outlet have been connected with the plant pipes with flexible joints, while the pump base was fixed on dumpers, in order to reduce the influence on the vibration signals of the pump of other vibrations coming out from the experimental system.

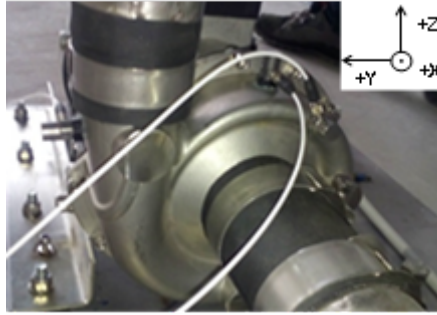


Figure 4.13: Position of the accelerometers

LabView software have been used for recording at the chosen sampling frequency all the acquired sensors data: three pressures (at pump suction and delivery and at water tank surface level), the temperature in the tank, the fluid flow, the sound level, three accelerations and the power absorbed by the pump. All data have been then processed with MATLAB[®].

Measurements have been carried out continuously from non cavitation regime to cavitation regime. In particular, the data analysis on vibrations and noise signals have been performed in the first phase of the analysis continuously, considering the whole data and in a second phase considering small data intervals at defined tank pressure steps. Decreasing the NPSH of the plant, different cavitation conditions can be observed and measured with the accelerometers and the microphone, from non-cavitation to severe cavitation.

4.4 Cavitation induced vibration analysis

4.4.1 Noise and vibration in pumps

Noise and vibrations in pumps are generated by various mechanisms. Noise and vibrations are characterized by being generally broadband in the spectrum with pronounced discrete frequencies. Generally speaking, vibration measurements in pumps are used to monitor and detect unbalances, misalignments, defective bearings and resonances [59]. Pumps vibrations result from:

- hydraulic forces;
- turbulence, cavitation and recirculation.

Vibrations due to hydraulic forces are related to pressure pulsations and are generated when the impeller vane passes a stationary diffuser or the volute tongue. Hydraulic forces are caused by all the pumping events in each revolution of the impeller and they are calculated considering the number of blades multiplied by the shaft rotational speed and its associated harmonics, Eq.4.10 [60–65]

$$f_p = BPF = \frac{(n \times rpm \times Z)}{60} \quad (4.10)$$

where BPF is the Blade Passing Frequency, n is the harmonic number, rpm is the shaft rotational speed in revolution per minute and Z is the number of pump events per revolution that is linked to the number of impeller vanes. In [63] is reported that BPF and its tonal noise are independent from the load.

In centrifugal pumps, hydraulic pulsations are reduced to small values when the impeller is centrally aligned with pump diffuser and there is also enough clearance between the impeller blades and the diffuser or volute tongue [59].

In pumps, operating conditions at fluid flows different from the design value cause the appearance of radial and rotodynamic forces. These forces are generated by an uneven static pressure distribution in the circumference of the impeller and in long term can cause radial hydraulic unbalance. When the hydraulic forcing mechanism matches the shaft natural frequency, the rotor of the pump can be excited significantly causing radial hydraulic unbalance. Generally speaking, when a centrifugal pump operates at a flow rate lower than the design one, there is an increasing of vibrations and noise in a narrow band so that the blade passage pressure pulsations and the radial hydraulic imbalance may increase significantly. This phenomenon is generated because the fluid flow does not smoothly matched the solid boundaries in the pump [59].

Authors in [59–66] report that a pump spectrum, in addition of the discrete frequencies tones due to hydraulic forces, has also a broadband component due to turbulence, cavitation and re circulation. Turbulent noise has different sources depending on the operating points of the pump. At the design flow rate the turbulent noise reaches its minimum value. Turbulent noise is generated by the blade interactions with vortices in the radial and axial clearance due to the different pressure distribution in the semi open rotor blades and to the fluid viscous friction.

In Fig. 4.14 [67] is presented the characteristic curve of a generic pump where the different events that can take place at different operating conditions are described [67].

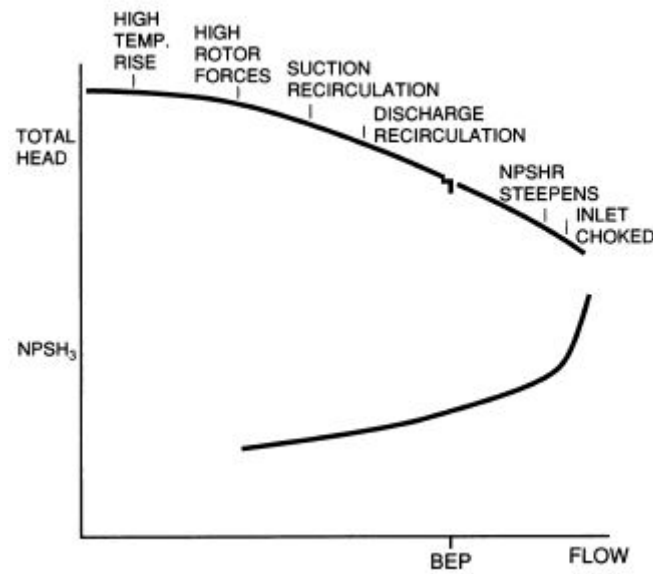


Figure 4.14: Pump events at different operating conditions [67]

Pumps operating at low flow rates present unstable operating conditions, as the stall and the surge. In particular, rotating stall is generated by internal re circulation in the suction and discharge area of the impeller, resulting in increased pressure pulsations [67].

Internal re-circulations at the suction are a possible source of very intense vortices, causing a significant lowering of the static pressure at that location due to the high velocities in the core. Cavitation and severe pressure pulsations can be generated these vortices, affecting significantly the vibration and noise levels of the pump with random frequencies. Pumps operating at off design conditions can present also re circulation at the impeller discharge. Also this phenomena can produce vortices characterized by low pressure regions potentially dangerous for cavitation formation and random pressure pulsations. Re-circulation at the impeller discharge can cause severe problems but is less frequent compared to internal re-circulation at the suction [67]. Generally, high broad band vibrations are generated by flow separation, re-circulation and increased fluid structure interactions.

When a pump operates at flow rate higher than the design flow value, it is subjected to laminar and turbulent boundary layer vortex shedding on the pressure side of the blade, that increases toward the trailing edge. This phenomenon causes an increased turbulence and generated noise [60,61].

Autoscillating cavitation, or surge cavitation, is a mechanism that causes very large pressure and mass flow rate oscillations through the pump [52] [68]. The generated noise and vibrations are random and broadband because bubbles appear randomly and chaotically. Cavitation noise can be described as the characteristic crack-

ling or hissing sounds produced by the pulsation of the pump rotor, the generated bubbles and the implosion of the vapor bubbles in the high pressure regions.

Autoscillating cavitation phenomena can be further classified into two cavitation phases: the incipient cavitation and the developed cavitation. Incipient cavitation is the beginning phase of the cavitation process where the bubbles are small and few. Cavitation inception bubbles when collapse can cause micro-shocks. At fully developed cavitation, bubbles collapse on the impeller walls and they cause shock waves on the surfaces. The damage caused by the impact of the shock waves depend on the number of bubbles and their size but also on the material of the impeller [69]. An high number of bubbles, that denote strong cavitation, produces a broadband frequency range of noise and vibrations because bubbles implode on a larger surface. Large bubbles are characterized by high volumes, causing stronger cavitation that impacts the low frequency range, due to their damping effect [69]. Open or semi open impellers are characterized to propagate stronger noise in the outside part of the pump (pump casing). In a closed impeller the pitting shock are less conveyed through the casing of the pump but more through the bearings, leading to an high damping and scattering effect of noise and vibrations [69].

4.4.2 Signal analysis methods classification

As introduced in the previous section, there are different hydraulic excitation that induce vibrations in hydraulic machinery. In short, pump vibrations can be classified as generated by flows instabilities in the pump sump or intake, cavitation, hydro elastic vibrations (phenomena caused by incorrectly shaped hydraulic profiles of discharge edge on the blades), self excited vibrations (phenomenon caused by the movement of mechanical parts, seals and clearances, that interact with the flow around or within) [70]. This section presents a methodology for measuring and detecting cavitation in centrifugal pumps measuring the vibrations induced by the phenomena. Generally speaking, a pump working at a fixed fluid flow, in both normal or cavitation conditions, can be considered as a dynamic system. A dynamic system is defined as physical system changing over the time, characterized by one or more inputs and one or more output responses.

Vibration signals can be classified in terms of frequency content in three classes, known as [59]:

1. periodic signals: they are characterized by the repetition after a certain time, called period;
2. random signals (stochastic processes): these signals at each time instant are independent of the values of the signal at other time instants. These signals are not predictable for the future time instants;

3. transient signals: signals characterized by having a limited length.

Cavitation in pumps generate random high frequency broadband energy [71]. Considering a general vibration signal acquired for machine diagnostics purposes, the signal can be furthermore classified as deterministic or random, accordingly to the following definitions:

1. deterministic signals: they are mathematically represented with an addition of sine waves for all the entire signal and for all the times (past and present). Periodic signals are part of this class. Deterministic signals show discrete line frequency spectra, as for example the vibrations from a rotating shaft. Deterministic signals are described as being almost periodic or quasi-periodic, in particular when there are not harmonic relationships between the various frequency components;
2. non deterministic signals: they cannot be evaluated by a short observation in the past or present time, because their values change randomly.

In [59] it is presented another general signal analysis classification used in the so called "condition monitoring", reported in Fig. 4.15. The author divides the process by considering four techniques: magnitude domain analysis, time domain analysis, frequency domain analysis and dual signal analysis. This classification can be adopted for applying condition monitoring to a centrifugal pump, as presented in [72]. In particular, the signal analysis procedure applied in this dissertation is presented in Fig. 4.16, where in red color are highlighted the parameters that have been not used.

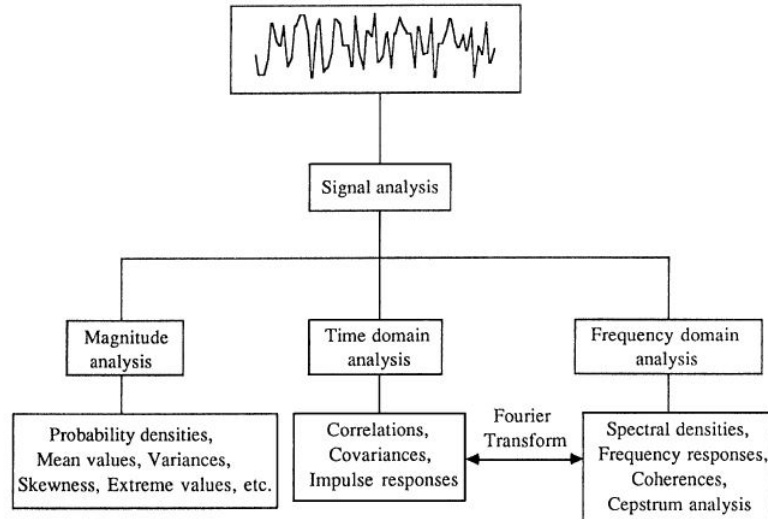


Figure 4.15: General signal classification [62]

Magnitude and time domain analysis give basic information about the signal, instead frequency domain and dual signal analysis (not considered in Fig. 4.15) provide more detailed information, but they are more sophisticated and computationally expensive.

Magnitude analysis and time domain analysis are used in particular by maintenance engineers because they permit to evaluate the signal continuously. Instead, frequency domain analysis is used mostly for periodic signals. Generally speaking, signals are random in nature [73] and they change in their dynamic conditions. Signal changes are detected analyzing changes in the statistical properties of the signal [74] [75] [72].

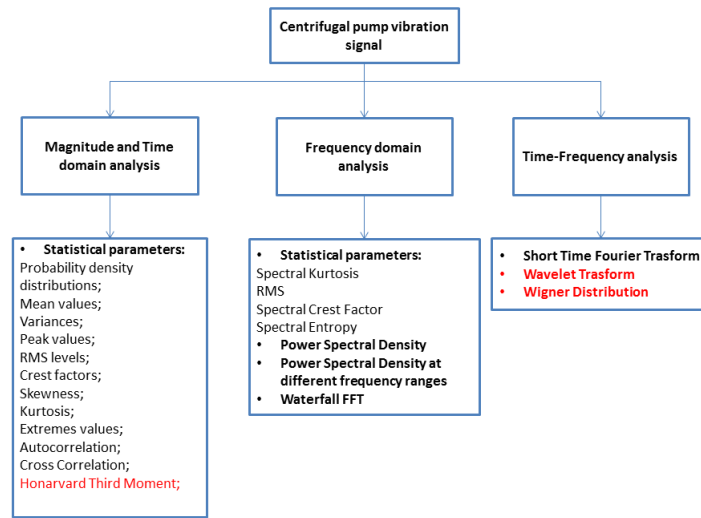


Figure 4.16: General signal classification [72]

4.5 Statistics of random processes applied to pump cavitation

In the initial phase of the experimental analysis on pump cavitation, the methods used for analyzing the acquired signals have been based on all the three approaches presented in Fig.4.16 (magnitude and time domain analysis, frequency domain analysis and time frequency analysis) in order to identify the best approach for detecting and controlling cavitation inception and development.

4.5.1 Time domain analysis

Time domain analysis is based on the time histories of the signals, considering their levels and particularities. Time domain analysis in most cases is not useful for prognostics purposes, leading to unsatisfactory results. The prediction of a certain failure or event can even turn out to be impossible to perform with this kind of analysis, because they do not present pronounced temporal changes before that failure or event. For example, mechanical degradation can only be detected at a very late stage with time features, and the residual time until failure is insufficient to make a prognostic. Failure appearance can only be detected in a late stage with these techniques. An example of acquisition of data issued from accelerometer and acoustic sensors during a typical test on the pump undergoing to cavitation conditions is reported in Fig. 4.17, where a_x , a_y , a_z are the acceleration measurements in the axial direction x and the radial directions y - z, while the last picture shows the measured noise. In Fig 4.17, the acquisition process of the sensor signals is performed going from normal to cavitation conditions and it shows that vibrations amplitude and noise levels are increasing during the test. These signals are analyzed in the following paragraphs by means of some time domain techniques.

Probability density function Vibrations are considered to be random when the motion is unpredictable: if a variable $\alpha(t)$ is random, it means that its values cannot be predicted in advance [76]. The first order probability density function describes the distribution of values of a random variable and is defined by Eq. 4.11.

$$Prob(\alpha \leq \alpha(t_0) \leq \alpha + d\alpha) = p(\alpha)d\alpha \quad (4.11)$$

Considering a random variable and assuming that the statistical characteristics of $\alpha(t)$ are not changing with time, it is possible to use the time history up to now of the variable $\alpha(t)$ to calculate its probability density function.

Assuming a time interval T as in Fig. 4.18, the value $\alpha(t)$ lies in the band of values α and $\alpha + d\alpha$ for a total time of $(dt_1 + dt_2 + dt_3 + dt_4)$ and the probability density function is the fraction of the total elapsed time for which $\alpha(t)$ lies in α and

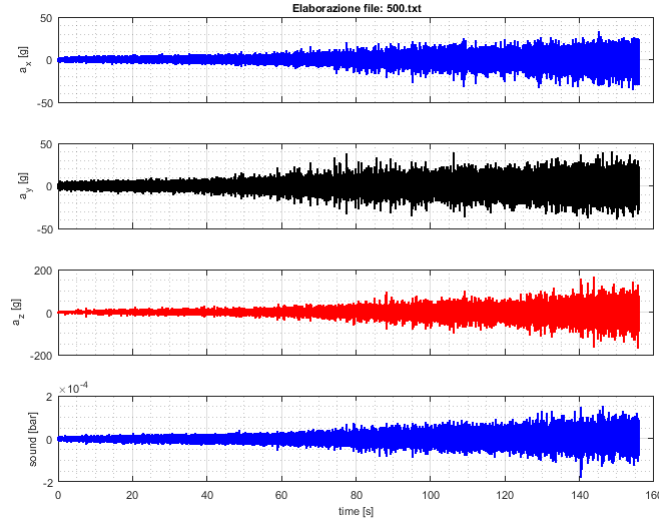


Figure 4.17: Acceleration signals

$\alpha + d\alpha$ band (Fig. 4.18).

$$p(\alpha)d\alpha = \frac{(dt_1 + dt_2 + dt_3 + dt_4)}{T} = \frac{\sum dt}{T} \quad (4.12)$$

In practice, the probability density function represents a continuous histogram of signal amplitudes measuring the time spent in each band values of α .

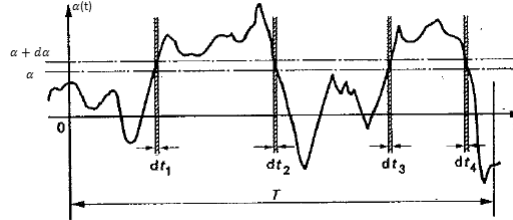


Figure 4.18: Time interval of a signal

The probability distribution function is defined by Eq. 4.13.

$$P(\alpha) = \int_{-\infty}^{\alpha} p(\alpha)d\alpha \quad (4.13)$$

The probability distribution function is the integral of the probability density function $p(\alpha)$ and represents the probability $p(\alpha)d\alpha$ to find the signal $\alpha(t)$ in the range $\alpha(t)$ and $\alpha(t + \Delta t)$ and its maximum value is 1 as shown in Eq. 4.14.

$$Prob(-\infty \leq \alpha \leq +\infty) = \int_{-\infty}^{+\infty} p(\alpha)d\alpha = P(\alpha = \infty) = 1 \quad (4.14)$$

The probability density function $p(\alpha)$ represents the slope of the probability

distribution function [76, 77] as expressed in Eq.4.15.

$$\frac{dP(\alpha)}{d\alpha} = p(\alpha) \quad (4.15)$$

A true random signal is considered to have a bell shaped probability density distribution because stationary random signals are generally Gaussian in nature and have a well known Gaussian probability distribution [59] (central limit theorem). The central limit theorem, states that [77]:

<<Considering $X_1, X_2 \dots X_N$ independent random variables with their own probability distribution, the sum of random variables, $S_N = \sum_{K=1}^n X_k$, tends to have a Gaussian distribution as n becomes larger, independently from the individual distribution of the variable X_k .>>

A normally distributed random variable has a probability distribution that can be described by the mean and the variance, such that the probability density function is given by Eq. 4.16.

$$p(\alpha) = \frac{1}{\sigma_\alpha \sqrt{2\pi}} e^{-(\alpha - \mu_\alpha)^2 / 2\sigma_\alpha^2} \quad (4.16)$$

If $\mu_\alpha = 0$ and $\sigma_\alpha^2 = 1$ then it is called 'standard normal distribution'.

Considering for example the condition monitoring applied for failure detection, a defect increasing in time changes the probability density distribution curves both in shape and amplitude. These changes are detectable by means of the time trend of the probability density distribution curves with a waterfall plot of the PDF. As previously said, the probability density function can be implemented by the use of the amplitude histograms, where every value of the signal is positioned in a certain bin and counted (the total number of bins is the total number of intervals considered to divide a certain signal length).

In the experimental activity, pump behavior have been studied measuring continuously the total head and the increasing vibrations and noise of the pump due to the progressively decreasing NPSH of the test rig .

Fig. 4.19 represents the probability density function, calculated at different NPSH values, of the three accelerometer signals (indicated with a_x, a_y, a_z in Fig. 4.19) and of the noise signal (indicated with m in Fig. 4.19). The test has been performed with pure water and the pump fluid flow was 700 l/min; accelerometers and noise signals data are collected in 50 bins every one second of acquisition, in order to compute the probability density function.

The abscissa axis "Value" gives the value of the measured variable, the number of counted values in every bin (frequency) is represented in the ordinates axis "F" and the NPSH values in the abscissa axis "NPSH". The probability density function tends to increase in its frequencies and to enlarge in the values of the measured variables with the decreasing trend of NPSH.

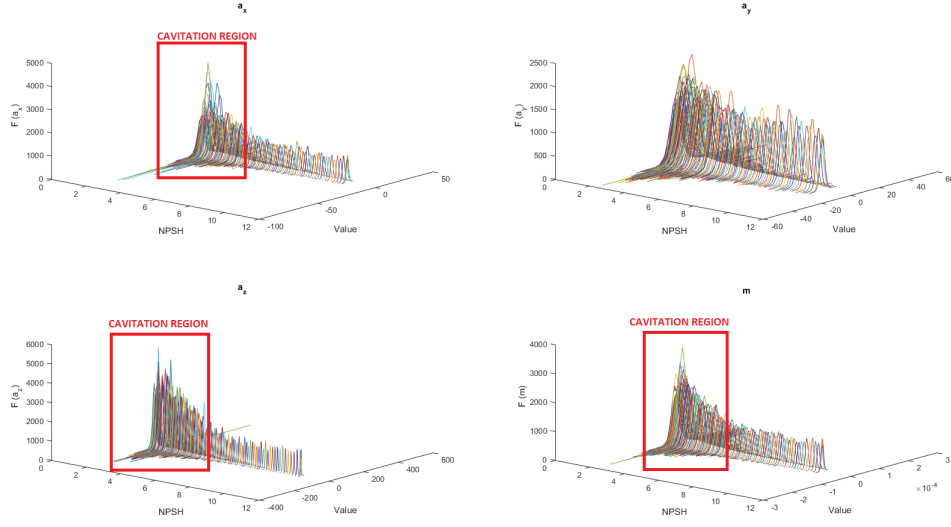


Figure 4.19: Waterfall Probability Density Function of the three accelerometers and the microphone

Considering only the peak values of the frequencies as a function of the NPSH values, Fig.4.20, it is possible to identify a trend such that at a certain NPSH value, the filtered signal of the frequency peaks shows an increment in slope.

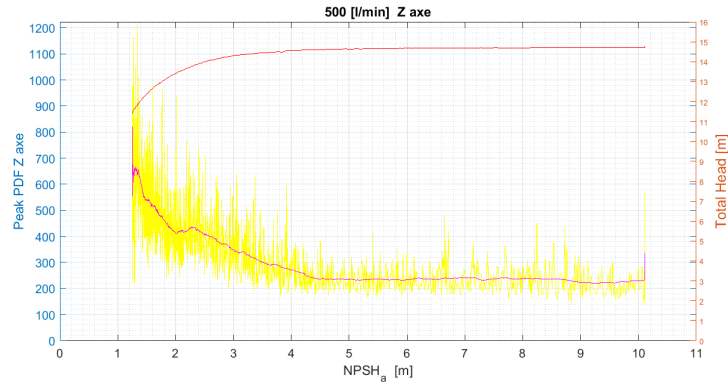


Figure 4.20: Peak PDF calculated in 50 bins every 2560 acceleration samples for the Z axis filtered with a moving average filter

The probability density function, as anticipated, provides information related to the probability characteristics of the random variables, but to get information about the expectations of a process it is necessary to introduce the statistical moments.

Statistical moments In a discrete process, for example, considering a random variable α that can assume different values with their own probability, if an α_i occurs n_i times in N total amount of experiments, the following probability of occurrence of the α_i variable is defined as the occurred times of the variable divided by the total number of experiments $f_i = \frac{n_i}{N}$. In an infinite number of experiments, empirical

probability approaches theoretical probability and the average value of α is defined in Eq. 4.17.

$$\bar{\alpha} = \frac{1}{N} \sum_i n_i \alpha_i = \sum_i \alpha_i f_i \quad (4.17)$$

In a continuous process, the probability is replaced with the probability density defined as $p(\alpha_i)$. The expected value of α , represented by $E(\alpha)$ is the first statistical moment of the probability density distribution (mean value), described in Eq. 4.18.

$$E[\alpha] = \int_{-\infty}^{\infty} \alpha p(\alpha) d\alpha \quad (4.18)$$

The second statistical moment of the probability density distribution is the mean square value. It is expressed in Eq. 4.19.

$$E[\alpha^2] = \int_{-\infty}^{\infty} \alpha^2 p(\alpha) d\alpha \quad (4.19)$$

The central moments are known instead, as the moments about the mean. The second moment about the mean (variance) is represented in Eq.4.20.

$$Var(\alpha) = \sigma_\alpha^2 = E[(\alpha - \mu_\alpha)^2] = \int_{-\infty}^{+\infty} (\alpha - \mu_\alpha)^2 p(\alpha) d\alpha \quad (4.20)$$

The standard deviation is given by Eq. 4.21.

$$\sigma_\alpha = \sqrt{Var(\alpha)} \quad (4.21)$$

For a transient process that changes distribution from a Gaussian to a non-Gaussian one, some interesting information related to probability density functions are given by higher moments. Generally speaking, odd statistical moments provide information about the disposition of the peaks relative to the median value, while even statistical moments give information about the shape of the probability distribution curve [77].

A generalized form of the moment is reported in Eq.4.22.

$$M'_k = E[\alpha^k] = \int_{-\infty}^{\infty} \alpha^k p(\alpha) d\alpha \quad (4.22)$$

Considering a k^{th} moment about the mean (central moment) the general formulation is represented in Eq.4.23.

$$M_k = E[(\alpha - \mu_\alpha)^k] = \int_{-\infty}^{+\infty} (\alpha - \mu_\alpha)^k p(\alpha) d\alpha \quad (4.23)$$

Skweness The third statistical moment is the skewness. The skewness of the distribution allows to understand if the data distribution is not Gaussian and to study the distribution of the peaks or extreme values of discrete events. In a non dimensional form the skewness is calculated with Eq. 4.24.

$$skw = \frac{E[(\alpha - \mu_\alpha)^3]}{\sigma_\alpha^3} \quad (4.24)$$

Skewed probability density functions are presented in Fig. 4.21. A symmetric probability density function within the mean assumes a skewness value of zero, while with positive skewness the probability density function is shifted to the left and with negative skewness is shifted to the right.

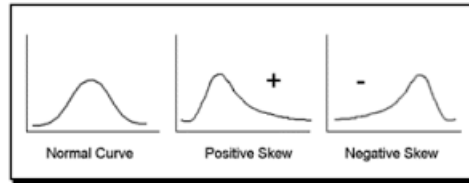


Figure 4.21: skewness effect

When a probability distribution is significantly skewed compared to a Gaussian distribution, different techniques are used for estimating the probability of exceed from a non Gaussian defined level as the Weibull distribution of peaks and Gumbel logarithmic relationships [59] .

In Fig. 4.22 it is represented the skewness of the accelerometer sensor signal positioned in the Z direction and calculated at various NPSH. The skewness is calculated every one second signal length, showing that at different NPSH steps it assumes values around zero. On the whole pump cavitation test the probability distribution of the acceleration sensor follows a normal distribution.

Kurtosis The fourth moment, called kurtosis, is a measure of the degree of flat-tering of the probability density function near its mean [77]. The formulation of the Kurtosis is presented in Eq.4.25:

$$Kurt = \frac{E[(\alpha - \mu_\alpha)^4]}{\sigma_\alpha^4} \quad (4.25)$$

Kurtosis is weighted along the values of the tails of the probability density distribution and is related to the spread of the distribution [59]. It provides information associated to the disposition of the peaks relative to the median value. The common Kurtosis value for a Gaussian distribution is 3. High Kurtosis shows a spread of higher signal values in respect to a Gaussian distribution. This parameter is useful for detecting the presence of an impulse with the signal, in particular to distinguish

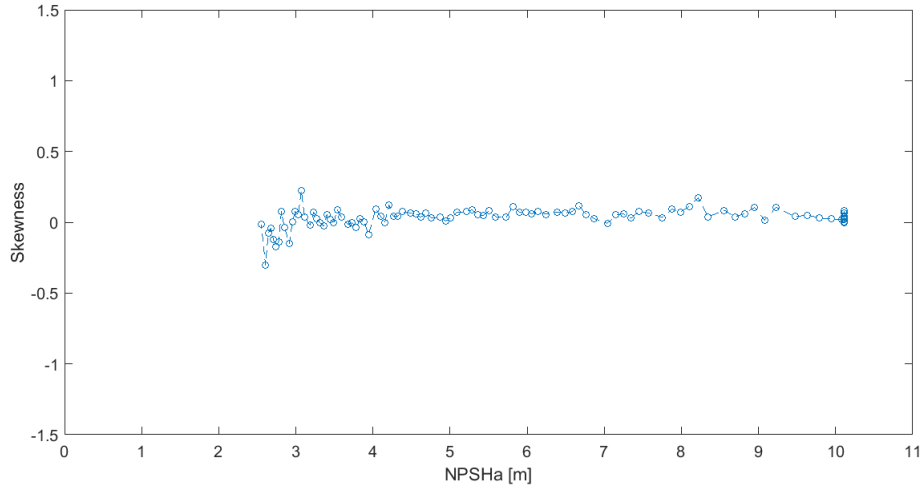


Figure 4.22: Skewness at 500 [l/min] performed every 25600 samples - sampling frequency 25600

if there are non linearity because nonlinear systems present non-Gaussianity [77]. Considering for example a machine, constituted by different components generating different types of random vibrations, the measured vibrations have a Gaussian behavior. When there is for example a fault in a component, there is a changing in the signal behavior of the machine. The signal change could produce an oscillatory behavior changing the probability distribution from a Gaussian distribution to another type. Generally a distribution that is more peaky than a Gaussian distribution is called leptokurtic and its Kurtosis is positive, instead a flattened distribution is called platykurtic [77]. The probability density functions with different values of Kurtosis are represented in Fig. 4.23.

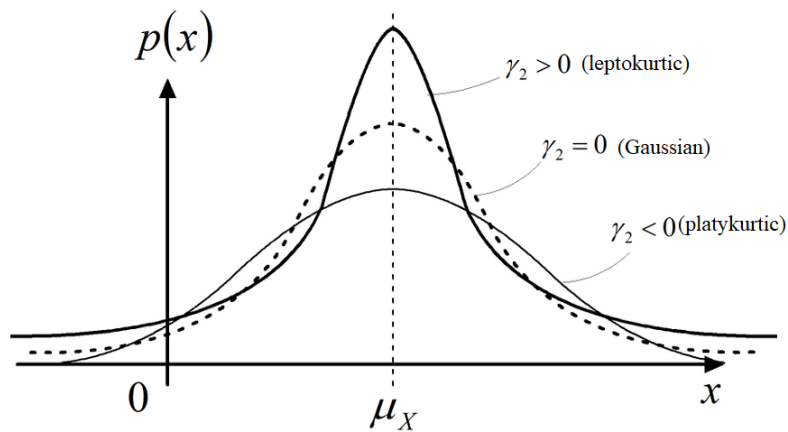


Figure 4.23: Probability Density Function at different Kurtosis values

Kurtosis is a sensible parameter but its application could lead to erroneous results.

Considering for example a bearing damage, when it becomes more distributed, the impulsive content of the signal decreases with a consequent decrease in the value of the kurtosis.

Fig. 4.24 represents the calculated Kurtosis at a flow rate of 500 l /min of the acceleration and noise signals at different NPSH. Kurtosis is calculated every 1280 values of the signals (sampling frequency of the signal divided by 10) and is represented with a yellow line. The signal is filtered with the MATLAB® function "rlowess" over 100 samples and plotted in a purple color. Kurtosis signal show an increasing trend when the total head of the pump decreases in all three accelerometer axes. The Kurtosis of the sound signal does not show trends during the whole test.

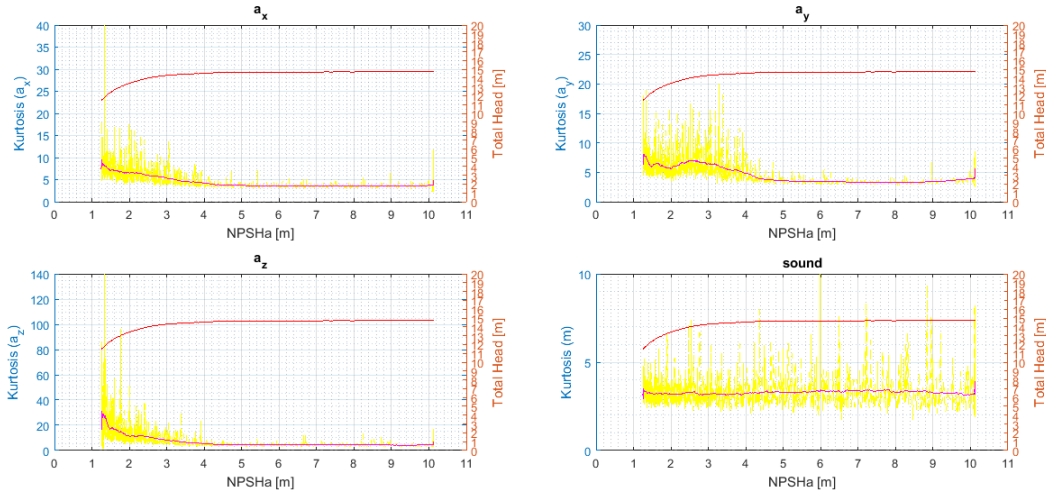


Figure 4.24: Kurtosis of the vibration and sound signals (flow rate 500 [l/min]) - Calculation performed every 1280 samples - sampling frequency = 12800 Hz

Various analysis have been performed for measuring the value of the Kurtosis at developed cavitation considered at the 3% total head decrease of pump. An example of this analysis, calculating the filtered Kurtosis at the 3% total head decrease is presented in Fig. 4.25.

Peak values In magnitude domain analysis, peak levels are used for detecting different behaviors in the time domain history of the signal. This parameter is used in particular for detecting defects (e.g. bearings) in the time history of the signal corresponding to an impulse.

The maximum acceleration values, evaluated from the signals at every 1280 values of the signals (sampling frequency of the signal divided by 10), are plotted in Fig.4.26 for the test performed at 500 l/min. As the Kurtosis signals, also peak values show a clear increasing trend at the decreasing total head of the pump.

Various analysis have been performed for measuring the value of the acceleration peaks at developed cavitation considered at the 3% total head decrease of pump. An

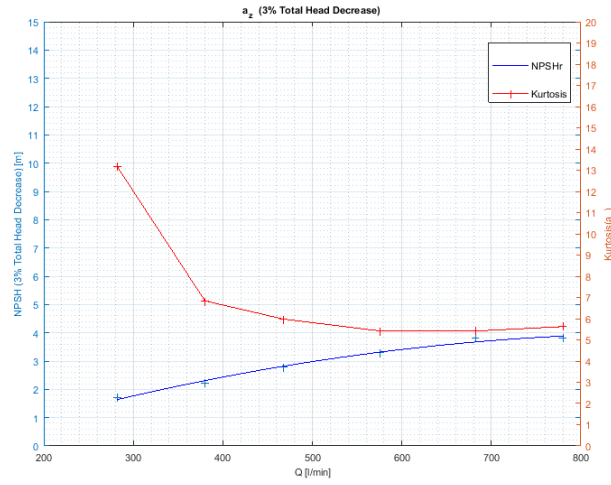


Figure 4.25: Kurtosis of the vibration in the Z axis at the three percent total head decrease for every fluid flow

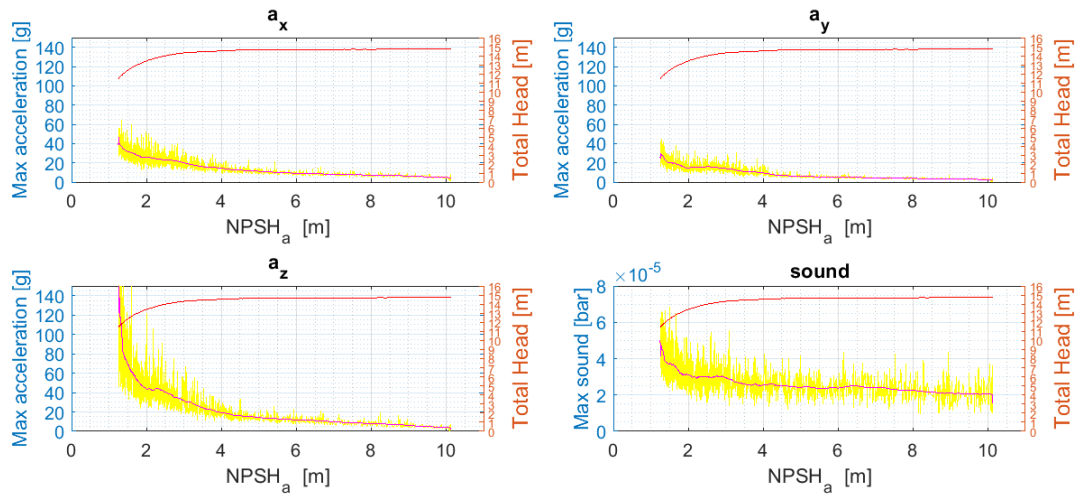


Figure 4.26: Maximum values of the vibration and sound signals (flow rate 500 [l/min]) - Calculation performed every 1280 samples - sampling frequency = 12800 Hz

example of this analysis is presented in Fig. 4.27.

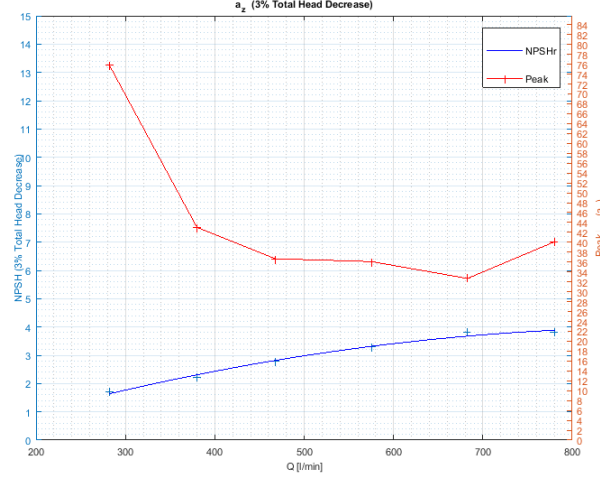


Figure 4.27: Peaks of the vibration in the Z axis at the three percent total head decrease for every fluid flow

RMS RMS (Root Mean Square) is an important parameter outcoming from the definition of standard deviation with zero mean value [59, 77]. It represents a continuous measure of the signal energy and is performed with Eq.4.26.

$$alpha_{RMS} = \sqrt{\frac{1}{N} \sum_{n=0}^{N-1} \alpha_n^2} \quad (4.26)$$

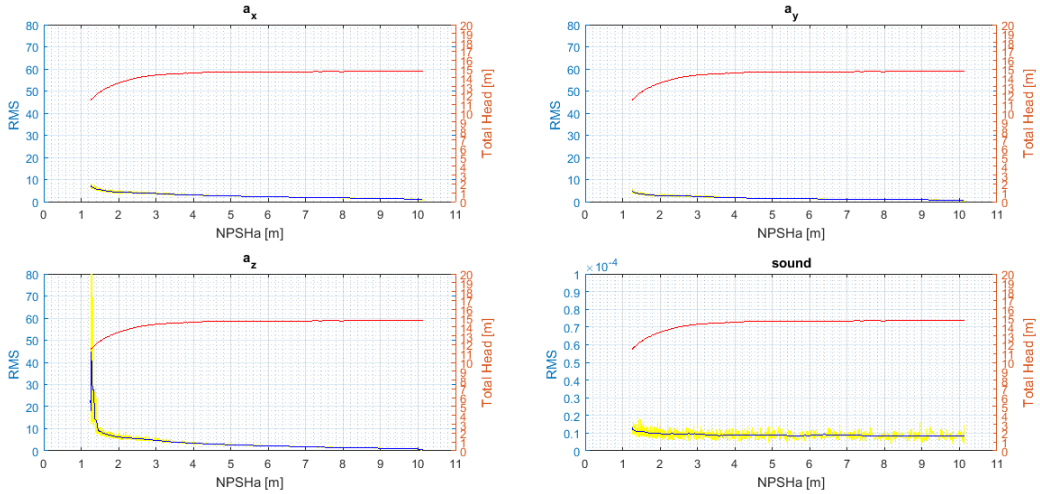


Figure 4.28: RMS of the vibration and sound signals (flow rate 500 [l/min]) - Calculation performed every 1280 samples - sampling frequency = 12800 Hz

In order to develop a control system for detecting and avoiding cavitation, various

analysis have been performed for measuring the RMS value of the accelerometer signals at the 3% total head decrease conditions and for relating these values to the pump NPSHr, as presented in Fig. 4.29.

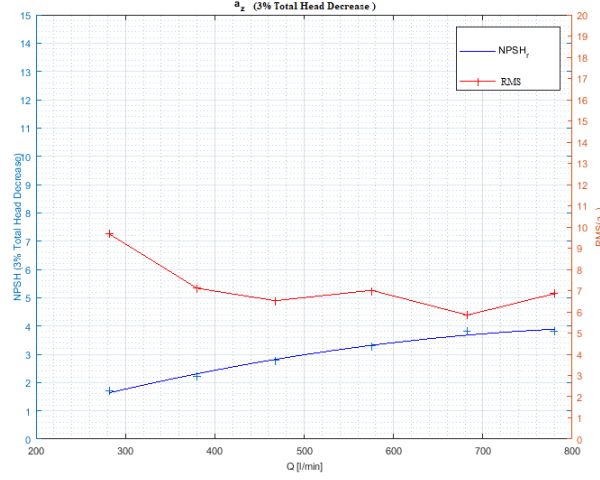


Figure 4.29: RMS of the vibration in the Z axis at the three percent total head decrease for every fluid flow

Crest Factor Crest factor is defined as the ratio of the peak levels divided by the RMS levels. Crest factor is a measure of the signal impulsiveness (noise and vibration signal). Generally, a truly random noise has a crest factor value less than 3 and usually it is applied for detecting shocks, impulsive events and noise. Damaged bearings have a crest factor higher than 3.5.

The formulation of the crest factor is given in Eq. 4.27.

$$CF = \frac{Peak}{\alpha_{RMS}} \quad (4.27)$$

In Fig. 4.30 it is represented the crest factor elaborated every 1280 samples and then filtered with a moving average on 100 data. Crest factor does not show an evident increasing trend with the decreasing of the pump total head except for the data out coming from the accelerometer sensor positioned in the Z axis.

Autocorrelation Autocorrelation measures the degree of association among variables in the signal history. It allows to detect if the signal is periodic or random. Periodic signals have periodic autocorrelation, random signals have an autocorrelation that tends to 0 for large time values [77].

The formulation of the autocorrelation is reported in Eq. 4.28.

$$R_{\alpha\alpha}(\tau) = E[(\alpha(t)\alpha(t + \tau))] \quad (4.28)$$

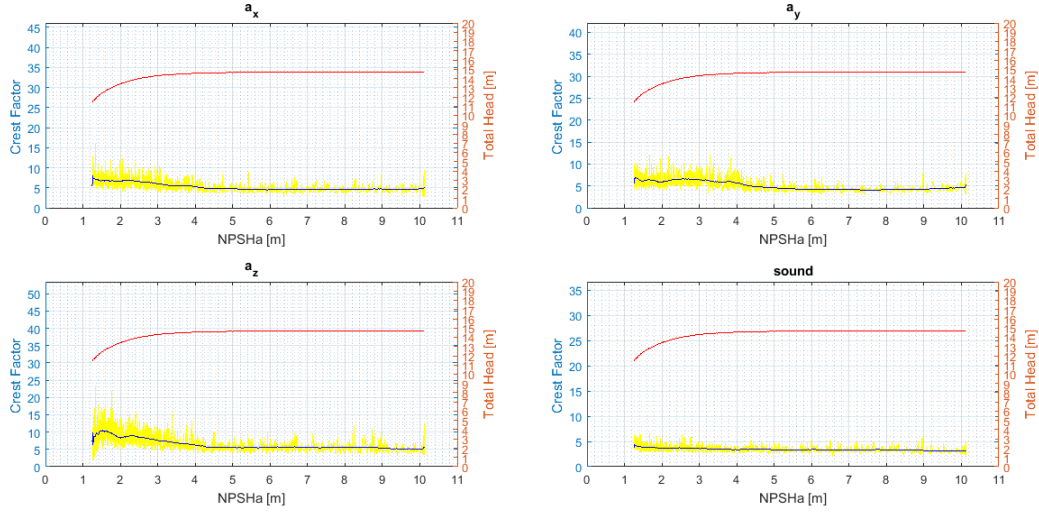


Figure 4.30: Crest Factor of the vibration and sound signals (flow rate 500 [l/min])
 - Calculation performed every 1280 samples - sampling frequency = 12800 Hz

An example of the autocorrelation calculated at a fluid flow of 400 l/min for the three accelerometer and the noise signals is presented in Fig. 4.31: it is clearly shown that all the signals are randomic.

Cross correlation The cross-correlation is similar to the autocorrelation, but it is applied to more signals. Considering two stochastic processes, as for example two accelerometer signals $\alpha(t)$ and $\beta(t)$: the joint-moment between these two signals is the cross co-variance function, defined in Eq.4.29.

$$C_{\alpha\beta}(t_1, t_2) = E[\alpha(t_1)\beta(t_2)] \quad (4.29)$$

Cross correlation allows to measure the degree of association between the signal $\alpha(t_1)$ at t_1 and the signal $\beta(t_2)$ at t_2 . The above formulation can be rewritten as Eq. 4.30.

$$C_{\alpha\beta}(t_1, t_2) = E[\alpha(t_1)\beta(t_1 + \tau)] \quad (4.30)$$

4.5.2 Frequency domain analysis

Waterfall Fast Fourier Transform (small time sample) The waterfall Fast Fourier Transform (waterfall FFT) has been performed as a first analysis on the sensors data sampled at 12800 Hz. This analysis permits to understand how the frequencies are influenced by the decreasing of the plant NPSH. The discrete FFT, Eq.4.31, has been applied on small intervals of the signal (every one second of acquisition time in a slowly changing process) because generally the Fourier methods can be applied only on deterministic phenomena.

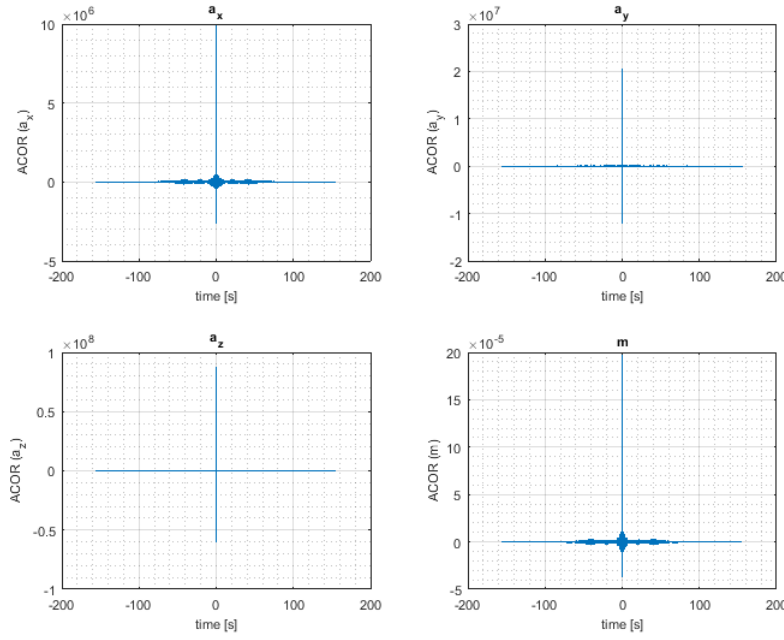


Figure 4.31: Autocorrelation of the vibration and sound signals (flow rate 500 [l/min])

$$X(k) = \sum_{n=0}^{N-1} x(n)e^{-j(\frac{2\pi}{N})nk} \quad (4.31)$$

The waterfall plot of the vibration signal at different NPSH steps of the test rig is presented in Fig.4.32.

The frequency range between 2500 Hz and 6400 Hz shows a gradual increase in the spectrum level, in particular with reference to Y and Z axis and at low NPSH values. For this reason, to have a better understanding on the cavitation phenomena, the sample rate has been subsequently increased to 25600 Hz, in order to see if such a trend, that characterize cavitation, was continuing at higher frequencies.

Power Spectrum using the Periodogram and the Welch methods As opposed to periodic signals, random signals have continuous spectra, which contain all frequencies, not only discrete frequencies. The signal must be described with a density type of spectrum. The power spectral density function of a process is represented in Eq.4.32.

$$S_{\alpha\alpha} = F[R_{\alpha\alpha}(\tau)] = \int_{-\infty}^{+\infty} R_{\alpha\alpha}(\tau)e^{-j2\pi f\tau} d\tau \quad (4.32)$$

where $R_{\alpha\alpha}(\tau)$ is the autocorrelation function and $S_{\alpha\alpha}$ is the forward Fourier transform of the autocorrelation function. This formulation states that the average power of the process (variance) is decomposed into the frequency domain through the power

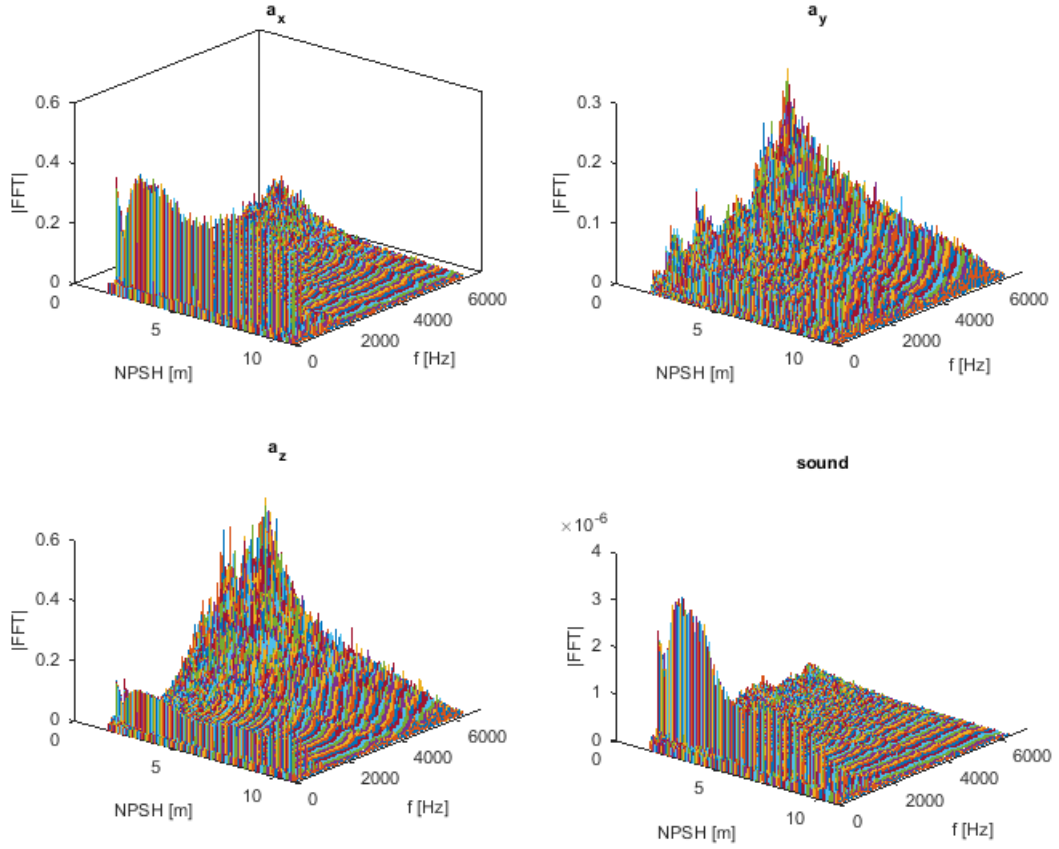


Figure 4.32: Waterfall FFT of the vibration and sound signals (flow rate 500 [l/min]) where the FFT is calculated every second and the sampling frequency is 12800 Hz

spectral density $S_{\alpha\alpha}(f)$ (Wiener – Khinchin relations) [77] [78].

The autocorrelation for a stochastic time signal is presented in Eq. 4.33: [77].

$$R_{\alpha\alpha}(\tau) = E[\alpha(t_1)\beta(t_1 - \tau)] \quad (4.33)$$

where E, as introduced in section 4.5.1 is the expected value of a random variable. Vibration measurements have been undertaken for all test conditions and their spectrum has been also visualized. A comparison among power spectral densities allows to understand the pump behavior in different regimes, i.e. non-cavitation regime and cavitation regime. The power spectral densities of vibrations signals, respectively in the Z direction (Fig. 4.33) and Y direction (Fig. 4.34) are reported as an example at two values of NPSHa, respectively 9.42 m in non cavitation regime and 5.75 m at initial cavitation stage.

The figures shows clearly that cavitation regime influences the power spectral density at all frequencies but, in particular, in a high frequency band.

A zoom of the power spectral density in the Z axis between 0 and 1000 Hz (Fig. 4.35) shows the characteristic Blade Passing Frequencies (BPF) and Rotational

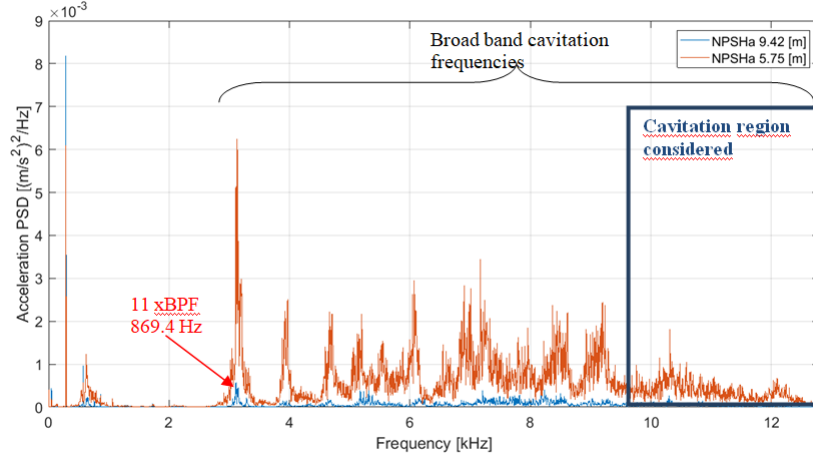


Figure 4.33: Power spectral density of the signal Z axis

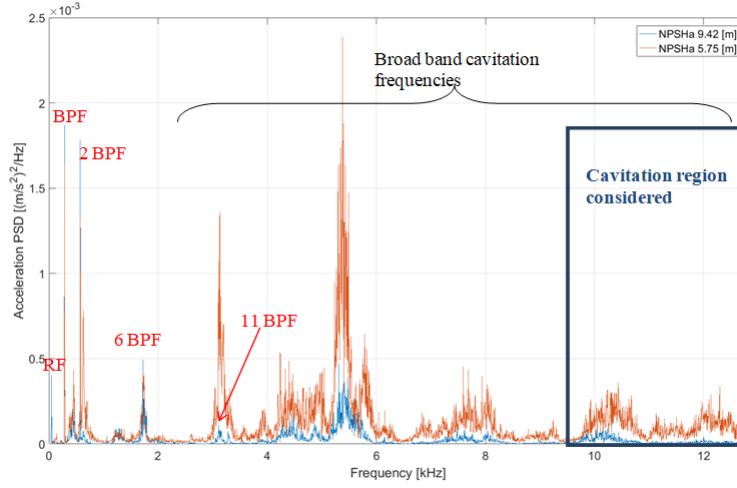


Figure 4.34: Power spectral density of the signal Y axis

Frequency (RF). The formulation for the Blade Passing Frequency (287 Hz) is given in Eq.4.10 while the Rotational Frequency is calculated with Eq. 4.34.

$$RF = \frac{rpm}{60} \quad (4.34)$$

These characteristic frequencies are also shown in more details in the FFT diagrams of acceleration along the Z and Y axes in 0 - 2000 Hz frequency range of Fig. 4.36 and Fig. 4.37.

According to theoretical analysis [79], [80], [81], [82], [83], [84] high frequency noise is excited during the collapse of cavitation bubbles, which have a particular effect on the vibration energy in high frequency bands. In [79], authors present the vibration spectra of a gerotor pump at 2000 to 5000 rpm, in cavitation and non cavitation conditions. They sampled at a frequency of 102.4 kHz. Authors affirm

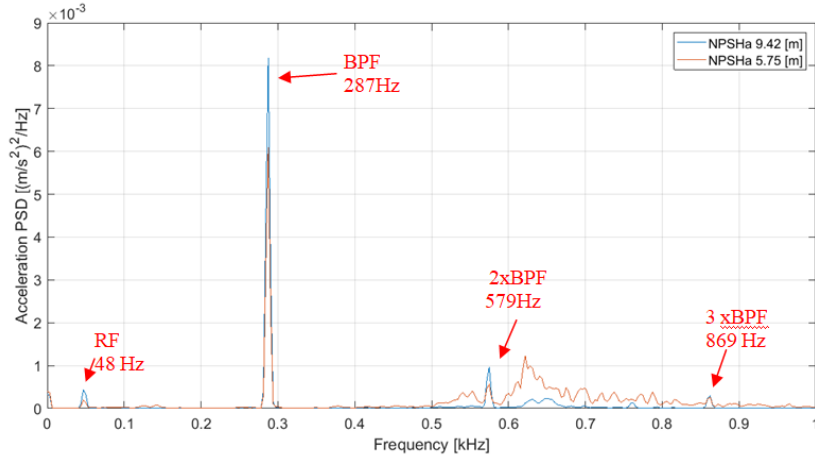


Figure 4.35: Power spectral density of the signal at Z axis between 0 - 1 kHz

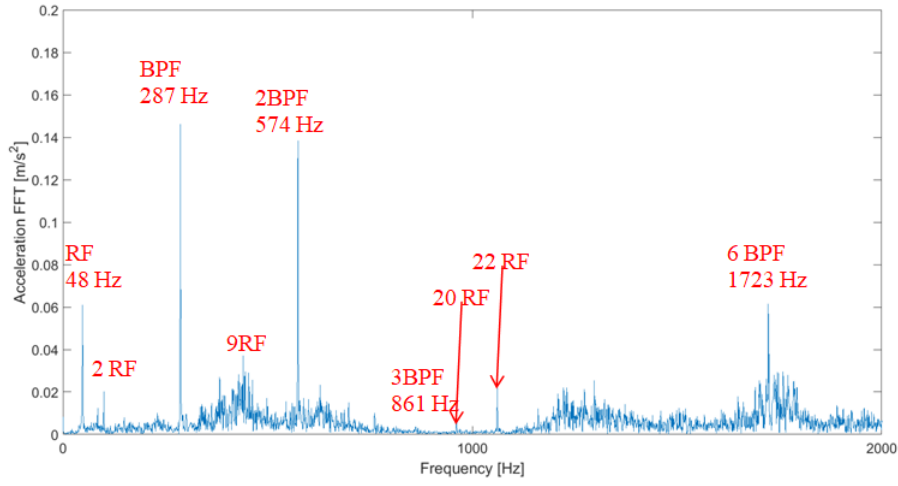


Figure 4.36: FFT of the signal at Z axis (NPSHa 9.42 m)

that, in vibration spectra, cavitation exhibits in high energy signals at frequencies from 1000 Hz to 10000 Hz. In [80] authors analyze vibrations spectra induced by cavitation in a centrifugal pump. Test are made at a sample rate of 20000 Hz, positioning the accelerometers in three normal directions. Authors have used also an accelerometer fixed on the base of the pump. In the axial axis analysis, vibration frequencies in cavitation are distributed between 2 kHz and 10 kHz. During the NPSHa decrease, authors have found that intensity and amplitude fluctuations are distributed between 6 kHz and 9 kHz but, when cavitation inception point is reached, frequencies become more distributed and, in particular, in critical cavitation conditions they are distributed between 2 kHz and 10 kHz. In one of the radial direction, cavitation vibration frequencies are present in the signals between 4 kHz and 10 kHz while in the other radial direction, broadband fluctuations are present

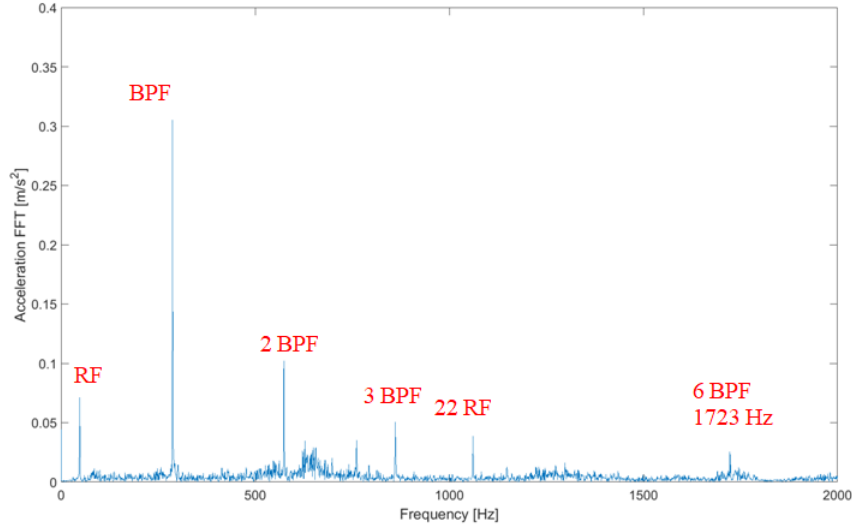


Figure 4.37: FFT of the signal at Y axis (NPSHa 9.42 m)

in a range between 6 kHz and 10 kHz. Authors conclude that, generally, frequency bands from 6 kHz to 10 kHz can be considered typical of cavitation in centrifugal pumps. In [81] authors analyzed the vibration characteristics of a pump at 1450 rpm during non-cavitation and cavitation regimes at different frequency ranges, with a sampling frequency of 80 kHz. Vibration signals in non-cavitation regime have been high pass filtered at 10 Hz and analyzed considering the vibration energy vs. flow rate in the low frequency band, between 10 – 500 Hz. In cavitation regime, frequency signals have been analyzed in four different frequency bands, 10-500Hz, 0.5 – 5 kHz, 5 – 10 kHz and 10 – 25 kHz. Authors affirm that vibration energy in 5 – 10 kHz and 10 – 25 kHz bands increased by the effect of cavitation. Instead, frequencies between 10 – 500 Hz and 0.5 – 5 kHz bands are less influenced from the cavitation noises. Moreover, frequency bands between 10 – 25 kHz are more sensitive to the onset of cavitation and, in particular, vibration energy is affected much earlier before the 3% head drop point. Authors in [82] divided vibration signals into four frequency bands, 10-500Hz, 0.5 – 10 kHz, 10 - 25 kHz and 25 – 51 kHz. They state that cavitation has a significant influence on vibration spectrum such that the whole frequency band of the signals is affected. Furthermore, high frequency signals in 10000 Hz – 51200 Hz are firstly affected by cavitation. Authors analyzed in particular also the low frequency signals in the band between 10 and 500 Hz, that is related to the cavitating flow structures in the blade channel: in no cavitation conditions, the blade rotation disturbs the liquid and generate a vibration pulse at the blade passing frequency, while in cavitation regime the pump is filled with liquid and vapour bubbles and, due to their compressibility, the vibration level is partially damped. A different approach is presented in [85] where authors investigate cavitation in low signal

frequency band. In particular, they present various vibration spectrums up to 15 kHz for five different operating cases in both non cavitation and cavitation regime. Authors show that cavitation produces high energy signals at high frequency ranges (between 1000 Hz and 15 kHz) but also that, as pump operation moves towards cavitation, the vibration level at the rotational speed increases while the vibration level at the blade passing frequency decreases. In [86], authors present a list of sources of vibration in centrifugal pumps, i.e. mechanical, hydraulic and peripheral ones. Authors state that cavitation generates random, high frequency broadband energy which is sometimes superimposed with the blade pass frequency harmonics. In [87], authors present the simultaneous survey of the vibration spectra at the rotational and blade passing frequencies as a detection method of cavitation. In their study the sampling rate was 52.2 kHz. In [88] authors have studied cavitation on a Francis prototype suffering from erosion on the runner blades by means of a on-board sensor, fixed to the rotating shaft, with a limited bandwidth of 6 kHz combined with accelerometers to analyse frequency contents up to 20 kHz. Authors state that it is reasonable to use a frequency band from 3 to 6 kHz for a detailed investigation of cavitation induced vibrations. In [70] three categories of flow induced vibrations are presented: global, local and by radial rotodynamic forces. Considering global flow vibrations, cavitation noise has dominant frequencies located between 1- 20 kHz, while rotating cavitation presents in a frequency range between 1.1 -1.2 times the rotating frequency. Partial cavitation or supercavitation has a frequency range lower than that of the rotating speed. This is a particular phenomenon where the length of the vapor cavity approaches that of the blade, in such a way that the cavity implodes in the region of the trailing edge. A study on unsteady cavitation is presented in [89] where authors have sampled accelerations at 20 kHz founding unsteady cavitation evidence in a frequency range between 0 and 300 Hz.

These literature indications have been used for setting up the experimental research and the data analysis because they show that, for detecting incipient cavitation in centrifugal pumps, high frequency vibration signals are more effective. Furthermore in [90] the authors filtered frequency signals below 1 kHz due to the high level of vibrations coming from the shaft and the blade passing. In [91] author state that for determining cavitation inception by means of noise measurements, an high-pass filter must be set. Its value has to be higher than the frequency range of the background noise (e.g. 10 kHz). With this method the measure will be sensitive enough to highlight the noise increase generated by the first bubble implosions from the background noise. A detailed observation of the power spectral density presented in Fig. 4.35 between 0 and 1000 Hz and of the FFT presented in Fig. 4.36 and 4.37 demonstrates the presence of the shaft frequency and its harmonics in the signal. An estimation of the power calculated from the power spectral density at different NPSHa, for the accelerometers positioned respectively in the Y and Z axes, is pre-

sented in Fig.4.38-4.39. The broadband intensity and amplitude power fluctuations have been found to be weaker in the Y direction in respect to the Z direction, due to the different propagation distances from the shock waves generated by the cavitation phenomena (Fig.4.38 - 4.39). The comparison of the vibration signals spectra shows also that in non cavitation regime, the spectrum has predominantly lower frequency components in comparison to the cavitation regime. Another aspect coming out from Figs. 4.33- 4.34 and Fig.4.38 - 4.39 is that the signal is more prosperous in frequency components in cavitation regime in comparison with non cavitation regime. In [79] it is explained this phenomena: in cavitation regime the pump is processing a multiphase flow and the blade rotation disturbs both the liquid and vapor phases. Due to the compressibility of the vapor bubbles, the effect of the disturbance is amplified and the results are conspicuous in higher level of vibrations spectrum, such that the structure resonances of the pump are excited. In Fig.4.38 - 4.39 the dominant frequency of the pump at every NPSHa step are clearly visible. Another important aspect coming out from the estimated power is that the vibration spectrum amplitude gradually increases with the decrease of NPSHa and at the same time also the fluctuations of the signal gradually increase.

Cavitation flow induced vibration characteristics: RMS in frequency bands

In the first step of the analysis on the vibration frequencies induced by cavitation, the vibration data in the Y and Z axes have been analyzed by means of the Fast Fourier Trasform computed for one second of sampling interval. Considering the test at 500 l/min (with reference in this example to the use of a solution of water and 1600 ppm of polymer) the frequency spectrum was calculated every 0.01 bar of pressure variation in the tank (corresponding to an equivalent number of NPSHa steps), resulting in 84 frequency spectra. For finding the frequencies influenced by pump cavitation, the variance of each frequency across all the 84 spectra was calculated by means of Eq. 4.35 :

$$Var = \frac{\sum_{n=1}^N |x_n - \mu_x|^2}{M - 1} \quad (4.35)$$

where M is the total number of spectra, x_n is the frequency amplitude and μ_x is the mean value of the amplitude over all the spectra. The resulting variance amplitude of the frequencies from 1 to 12800 Hz over the entire range of NPSHa values up to cavitation is reported in Fig.4.40 and Fig. 4.41 with reference to the accelerometers in the Y and Z directions.

The figures show that cavitation has a significant influence on a lot of frequency ranges of the vibration spectrum. In the Y axis these frequency ranges are mainly three:

1. from 2900 Hz to 6500 Hz

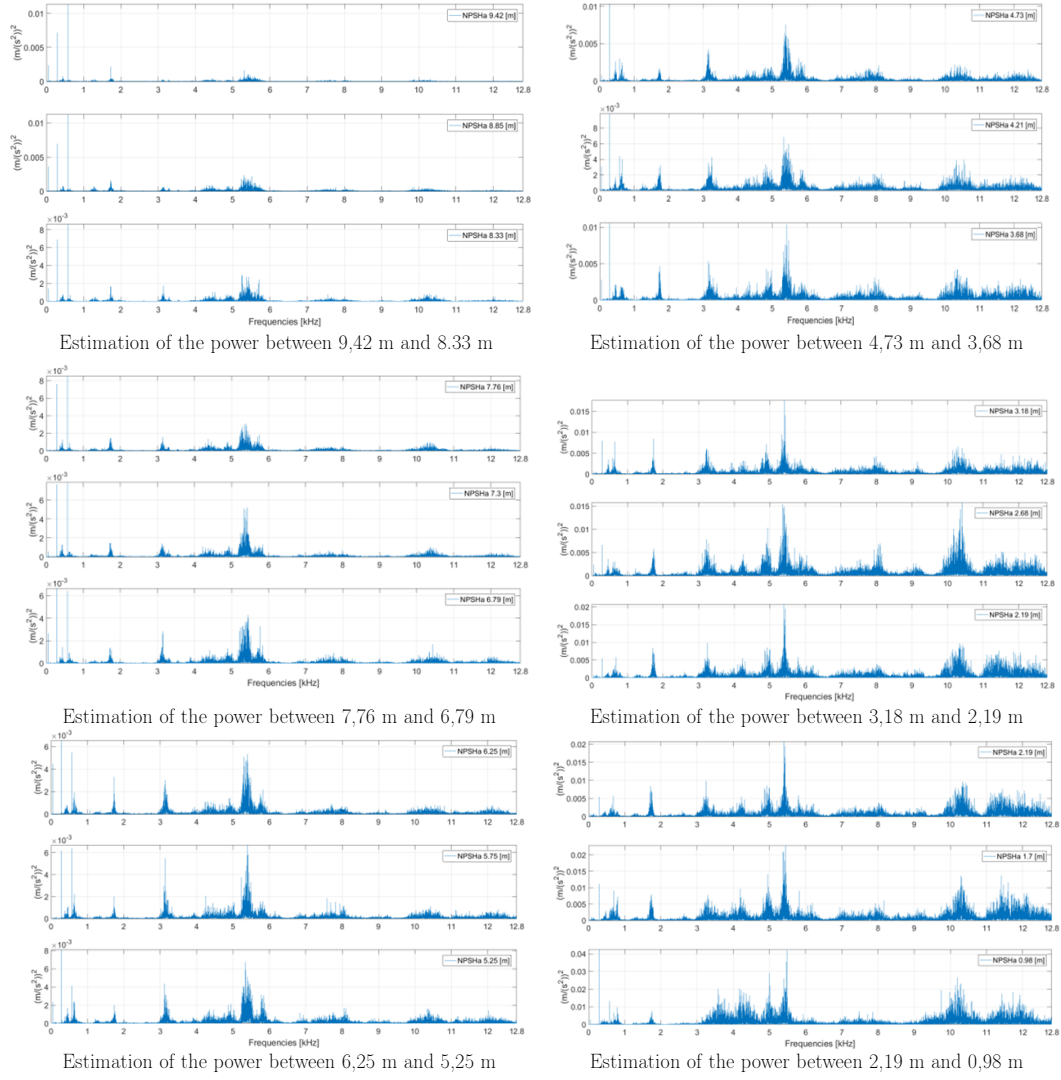


Figure 4.38: Power estimation at different NPSH steps Y axis

2. from 9550 Hz to 10080 Hz
3. from 10080 Hz to 12800 Hz

On the contrary, in the Z axis the frequency range mainly sensitive to cavitation is in the range between 2700 Hz and 12800 Hz. In order to evaluate vibration energy versus cavitation number in different frequency bands as in [82], the RMS method is applied. For a random signal, the area under the PSD in a specific frequency range is the square of the RMS, Fig. 4.42. So the energy of the signal is calculated with Eq.4.36.

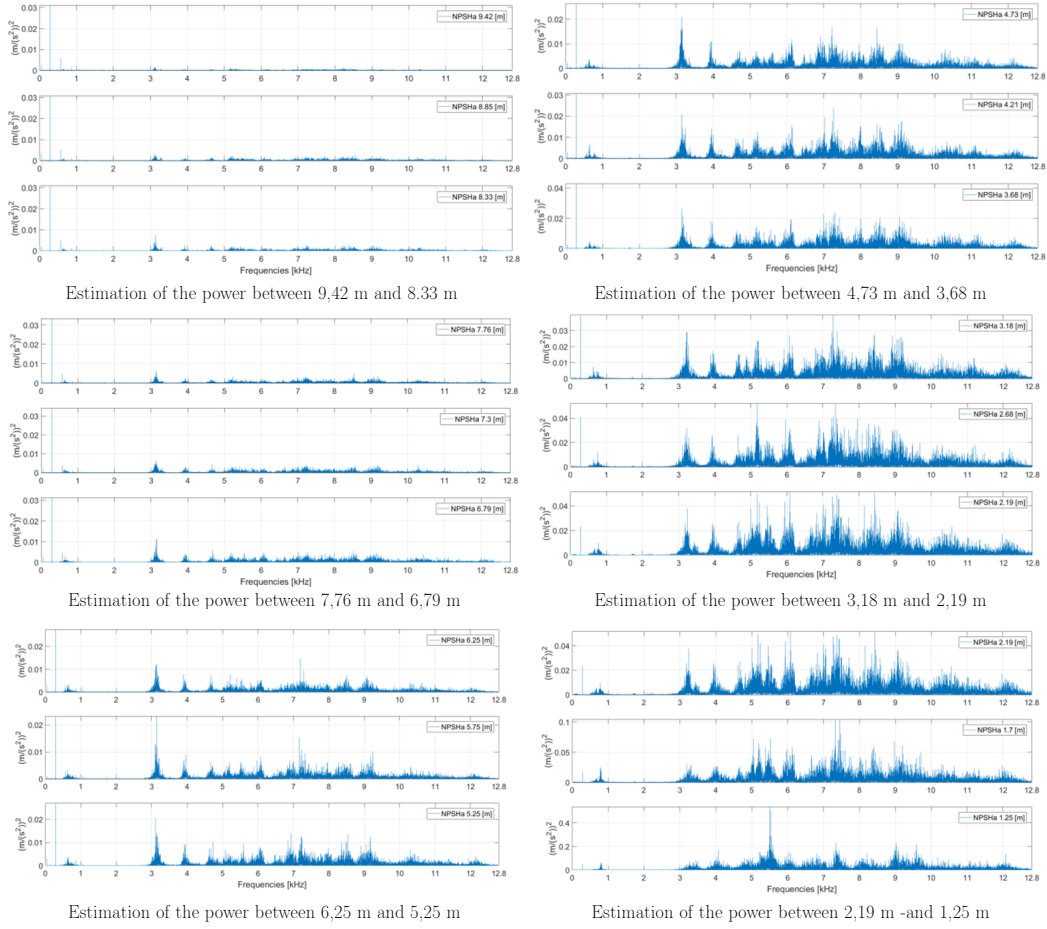


Figure 4.39: Power estimation at different NPSH steps Z axis

$$Energy = \alpha_{RMS} = \sqrt{\int_{f1}^{f2} 2R_{\alpha\alpha}(f)df} = \sqrt{Areaundercurve} = \int_{f1}^{f2} PSD(f)df \quad (4.36)$$

The vibration data are filtered at different frequency bands. The power estimation is performed using the non-parametric periodogram method. This procedure was performed because cavitation intensity is considered proportional to the energy of the signal in the frequency band of interest. From the variance analysis in the Z axis (Fig.4.41), the RMS in the frequency range between 2700 Hz and 12800 Hz is calculated and the resulting trend is presented in Fig. 4.43.

Fig. 4.43 presents a continuous trend of the RMS in the frequency range between 2.7 and 12.8 kHz up to NPSHa of 3 m. Cavitation inception is not clearly visible before the total head decreases, so that a further investigation on different frequency bands is needed. In literature, as presented before, it is widely accepted that as the

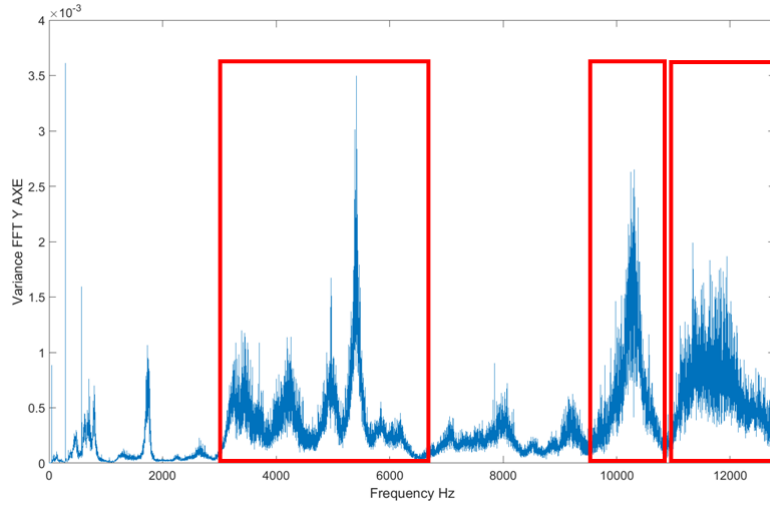


Figure 4.40: Vibration ranges sensitive to cavitation analyzed with the variance in the Y axis

NPSHa decreases, high frequency signals are the first to be affected by cavitation and in this frequency range it is then possible to detect incipient cavitation. In [81], as introduced before, authors state that high frequency signals, in their particular case between 10 and 25 kHz, are more sensitive to the onset and collapse of the small bubbles generated by cavitation. In [92], author state that the trend of vibration for detecting cavitation inception is particularly visible in the frequency bands of 10 – 15 kHz and 15 – 20 kHz. In [81], [82] authors compare different vibration energy trends during the cavitation development in a centrifugal pump. In particular at the beginning, with the NPSHa decreasing, small bubbles develop near the leading edge. These small bubbles travel towards blade exit and collapse at high pressure regions which affect the vibration energy in high frequency bands. Author in [81], [92] state that a closer correlation between vibration energy and the development process of cavitation can be found and, in particular, that vibration energy is affected much more early than the 3% decrease of the total head.

In our experiments, it was decided to analyze in particular the frequency range between 10 and 12.8 kHz due to the literature findings and also because from Fig.4.38 - 4.39 it is clear that, while the frequency ranges before 10000 Hz show an increasing trend of the fluctuations from the start of the NPSHa ramp down, the frequency range between 10 and 12.8 kHz shows an increasing trend from a certain value of NPSHa. This confirms that cavitation inception could be determined, before the total head decreases, analyzing the energy signal on this range.

In Fig.4.44 it is represented the RMS in the frequency range 10 – 12.8 kHz, where it is possible to detect the cavitation inception point at an NPSHa of 5 m, where the vibration level increases and there is a change in the slope of the RMS. The slope

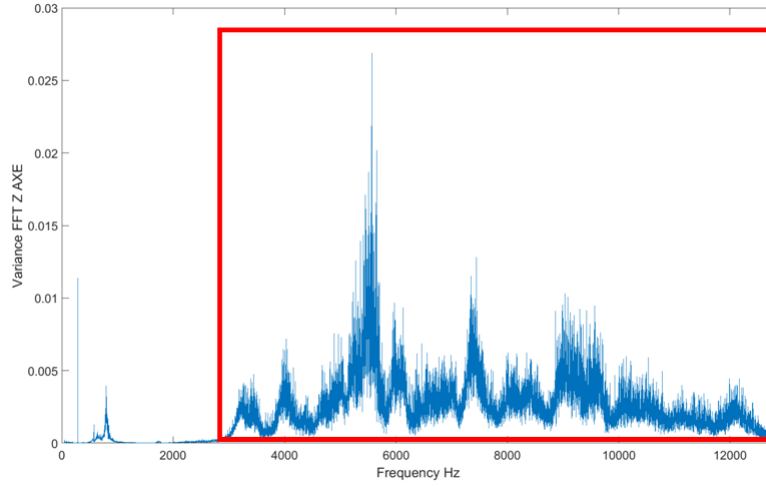


Figure 4.41: Vibration ranges sensitive to cavitation analyzed with the variance in the Z axis

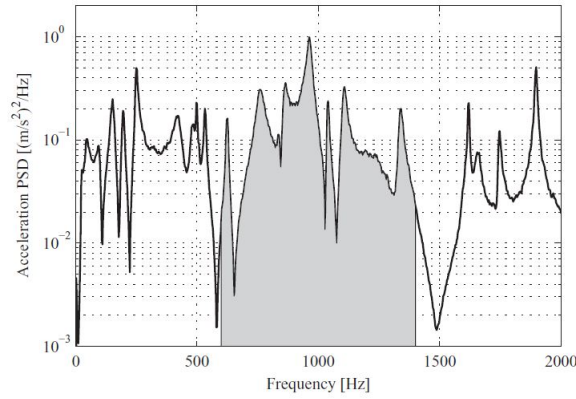


Figure 4.42: Area under the power spectral density [78]

change is more visible in the Y axis in comparison to the Z axis. It is also possible to define the critical cavitation point according to [81] where the vibration level achieve a maximum, at $NPSHa = 1.24$ m. In this point, severe cavitation erosion danger may exist. In the region between $NPSHa = 5$ m and $NPSHa = 1.24$ m cavitation near the suction side of blade inlet expands. Under $NPSHa = 1.24$ m, vibration energy decreases drastically. An explanation can be found in [81]: after the maximum, the emitted noise during the collapse of the bubbles is attenuated because the interior flow field in the impeller is in a state of two phase flow. This state is characterized by being highly compressible.

Fig.4.46 and Fig. 4.47 show the RMS in the frequency ranges 10 – 500 Hz and 500 Hz – 10 kHz. Vibration energy trend in the range between 10 and 500 Hz needs a further investigation (at a smaller RMS interval and highlighting the vibration amplitude at the first BPF) because it shows a trend that could be related to the

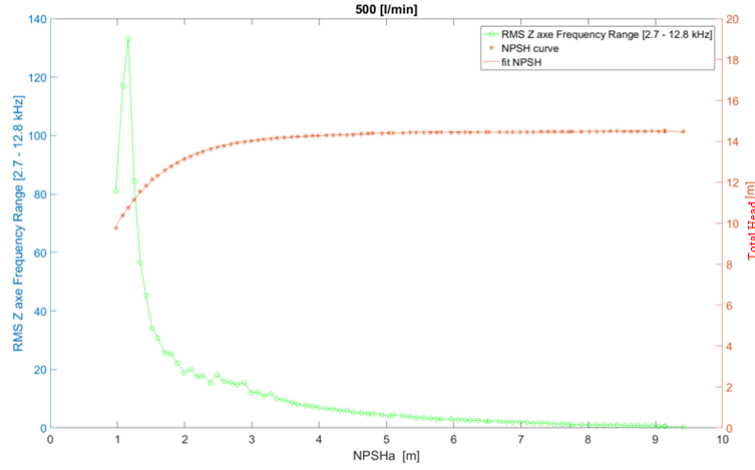


Figure 4.43: RMS trend in the Z axis between 2.7 and 12.8 kHz (500 l/min 1600 ppm)

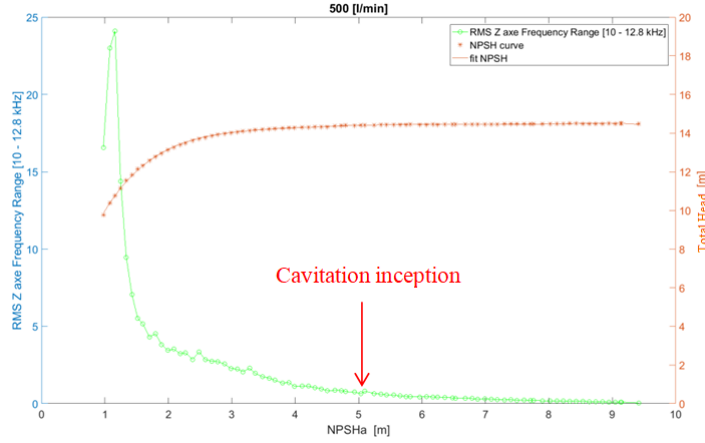


Figure 4.44: RMS in frequency range 10 to 12.8 kHz Z axis (500 l/min)

cavitating flow structures in the impeller blades, as described in [82]. Vibration energy in the frequency range between 500 Hz and 10 kHz does not show particular slope changes before the decreasing of the total head of the pump, how it would instead be necessary for detecting cavitation inception.

Spectral Entropy Spectral Entropy is a statistical parameter used in condition monitoring for evaluating different kinds of faults as, for example, in rolling elements of bearings or valves spring failures [93].

Considering a general continuous random variable α that has a probability density function $p(\alpha)$, the so called information entropy is the predecessor statistical parameter of the Spectral Entropy and it is defined by Eq. 4.37 [93]:

$$H(\alpha) = - \int p(\alpha) \log_2(p(\alpha)) d\alpha \quad (4.37)$$

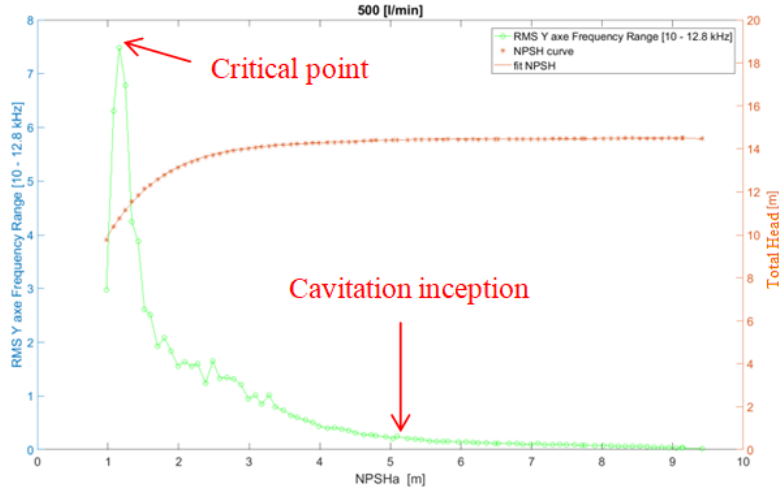


Figure 4.45: RMS in frequency range 10 to 12.8 kHz Y axis (500 l/min)

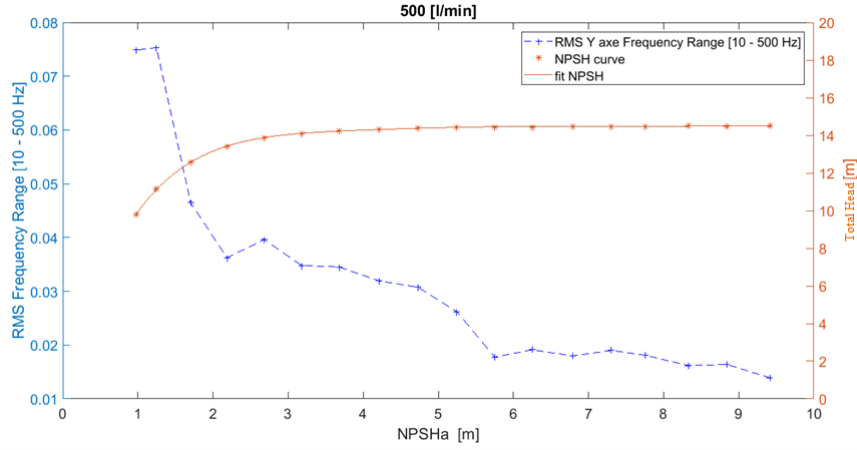


Figure 4.46: RMS in frequency range 10 to 500Hz (500 l/min)

Spectral Entropy is used for measuring the distribution of frequencies. Defining a discrete signal $x(i)$ and the Fourier Transform of the signal $X(i)$, with $i = 1, 2, \dots, N$, the Spectral Entropy is calculated with Eq. 4.38:

$$SE = - \sum_{i=1}^N p_i \cdot \log_2(p_i) \quad (4.38)$$

The Normalised Spectral Entropy is defined by Eq. 4.39:

$$SE = \frac{-(\sum_{i=1}^N p_i \cdot \log_2(p_i))}{\log_2(N)} \quad (4.39)$$

where the probability is calculated by Eq.4.40:

$$p_i = \frac{X(i)}{\sum_{j=1}^N X(j)} \quad (4.40)$$

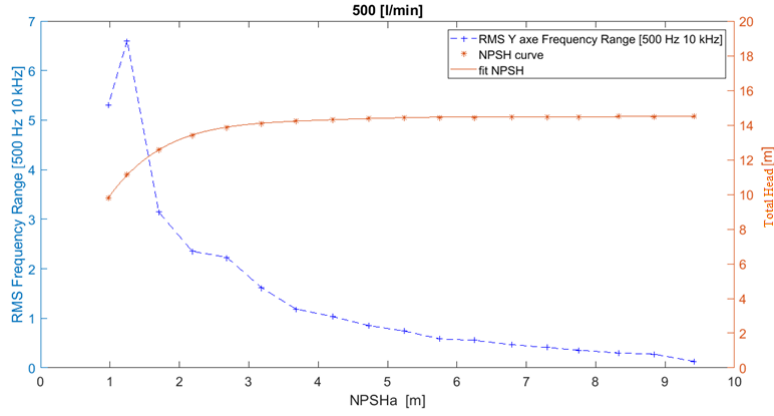


Figure 4.47: RMS in frequency range 500Hz to 10 kHz (500 l/min)

The number of frequency components is N and the frequency percentage in the whole spectrum is represented with p_i .

$$\sum_{i=1}^N p_i = 1 \quad (4.41)$$

Vibration signal spectral structure is evaluated with spectral entropy and it has high values when the amplitude distribution is flat, instead it assumes values tending to zero when amplitudes concentrate to few frequency components.

In Fig.4.48 it is represented the Spectral Entropy calculated at 500 l/min for the accelerometer positioned in the X axis and related to the pressure difference among the pump delivery and suction. It shows that spectral entropy increases constantly on the whole test.

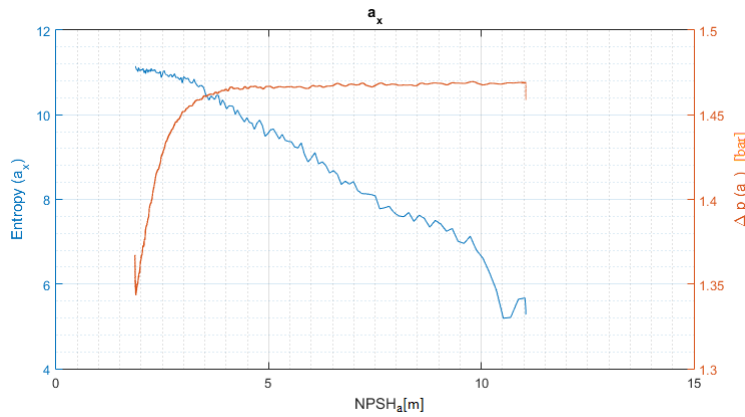


Figure 4.48: Spectral Entropy (400 l/min)

This parameter is not useful for detecting cavitation inception because it does not show pronounced changes of slope during the NPSHa decrease.

Spectral Crest Factor Spectral Crest Factor is evaluated calculating the FFT every second, finding the maximum values of the acceleration in the frequency domain and dividing it by the RMS value in the frequency domain.

In Fig.4.49 it is represented the Spectral Crest Factor calculated at 500 l/min for the accelerometer positioned in the X axis and related to the pressure difference among the pump delivery and suction. It shows that Spectral Crest Factor decreases on the whole test.

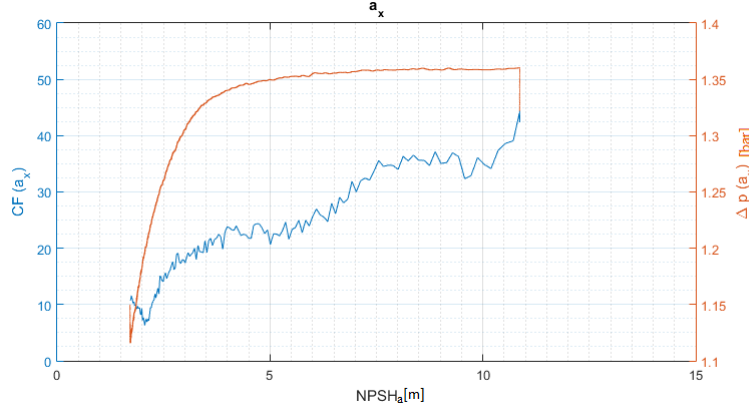


Figure 4.49: Crest Factor in frequency domain

Also this parameter is not useful for detecting cavitation inception because it does not show pronounced slope changes during the NPSHa decrease.

4.5.3 Time Frequency Analysis

Time Frequency Analysis can represent the energy and intensity of a signal simultaneously in time and frequency and is particularly used in condition monitoring of a machine. Considering a non stationary transient signal, it is important to highlight the characteristics of the signal during the time within all frequency scales, because in a non stationary signal its structure changes intrinsically as a function of time [59] [94]. There are different methods for analyzing the signal with time-frequency, as the Short Time Fourier Transform and the Continuous Wavelets Analysis.

Short Time Fourier Transform (STFT) STFT is a technique that uses a windowed Fourier Transform to determine at what frequency an event occurs [59]. STFT has the disadvantage of being a compromise between precision and scale so that, for a correct application, several tests are needed with different window lengths. Briefly, it is impossible to achieve a fine resolution in both frequency and time domains, due to the Uncertainty Principle (UP). In particular, UP states that more precisely it is specified when an event comes out, less specifically it is possible to define what frequencies are involved [77]. This concept can be summarized with Eq. 4.42:

$$\Delta w \cdot \Delta t \geq \frac{1}{2} \quad (4.42)$$

where Δw is the bandwidth and Δt is the time width and the product of the two is never less than 0.5.

In general the well known Fourier Analysis provides the decomposition of a time history in the frequency domain of the signal $x(t)$ with Eq. 4.44:

$$x(t) = \int e^{j\omega t} X(\omega) d\omega \quad (4.43)$$

$$X(\omega) = \frac{1}{2\pi} \int x(t) e^{-j\omega t} dt \quad (4.44)$$

For a window $h(t)$ centered at the time t , the spectrum of $x(t')h(t-t')$ is calculated and the STFT is given by Eq. 4.45:

$$S_t(t, \omega) = \frac{1}{\sqrt{2\pi}} \int e^{-j\omega t'} x(t') h(t-t') dt' \quad (4.45)$$

The spectrogram or energy density spectrum represents the energy density at t and ω and it is calculated with Eq. 4.46:

$$P_S(t, \omega) = |S_t(\omega)|^2 \quad (4.46)$$

Various kind of windows permit to control the relative weight imposed at a signal [95]. The selection of the window depends on the application because different windows can be used for estimating different properties of the signal. Generally speaking, as a compact window in the time domain achieves an higher resolution in time, a compact window in the frequency domain permits an high frequency resolution.

The graphic of spectrogram and STFT is usually three dimensional and the intensity is represented by different color scales. Basically the effectiveness of the spectrogram depends on the functional form of the window but the estimated properties have to be not influenced by the detail of the window.

A plot of the STFT at 500 l/min for the accelerometer positioned in the y direction is represented in Fig.4.50, which shows group of frequencies that are in strong correlation with the cavitation development process.

Analysing the spectrogram of the signal in the radial direction Z, during the NPSHa decrease from normal conditions to cavitation, it is possible to see at the BPF (287 Hz) and at the associated second and third harmonic that these frequencies increase slightly when cavitation starts (Fig. 4.51, Fig. 4.52 and Fig. 4.53).

Furthermore, in [85] author states that vibration amplitude at the first BPF presents a decreasing trend in cavitation conditions. Considering vibration in the Y

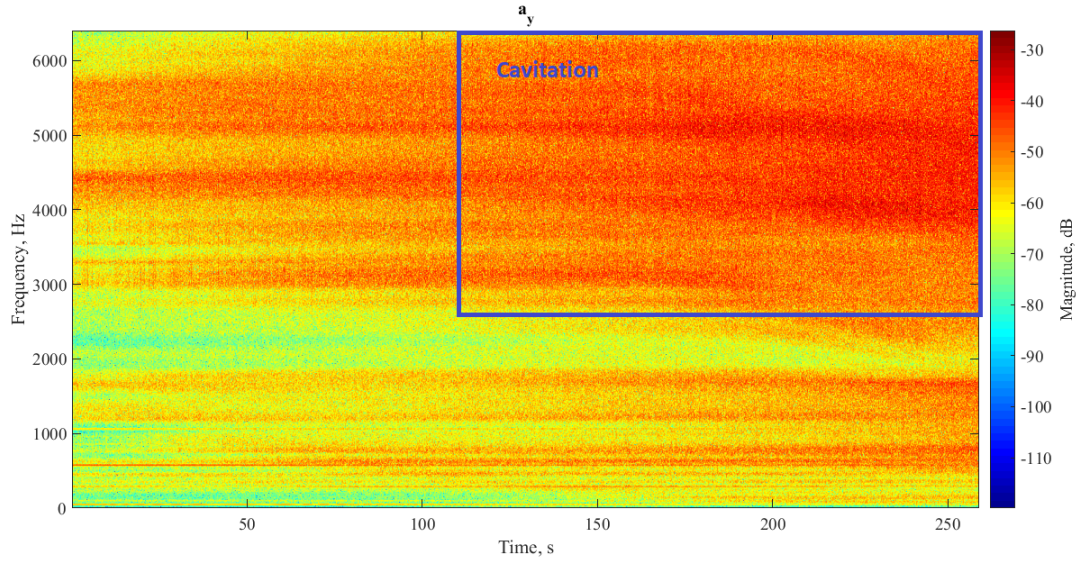


Figure 4.50: STFT 500 l/min for the y accelerometer of the pump

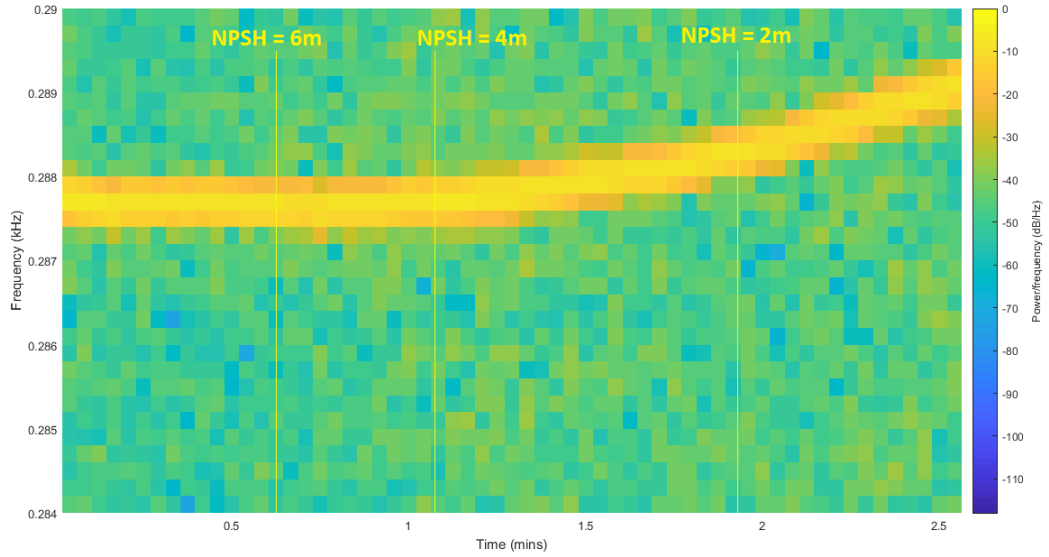


Figure 4.51: Spectrogram in radial Z axis around the first harmonic of the BPF

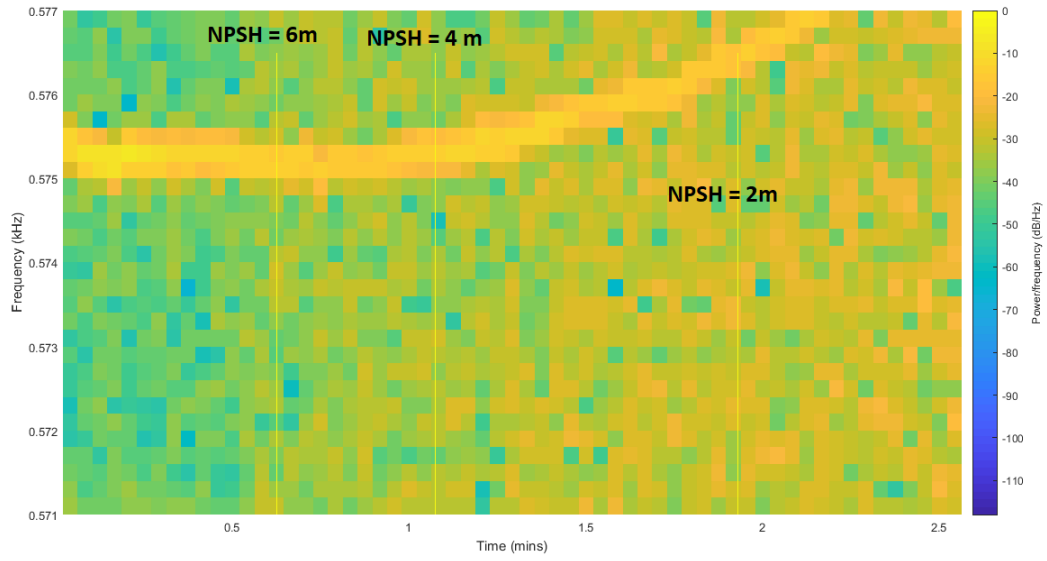


Figure 4.52: Spectrogram in radial Z axis around the second harmonic of the BPF

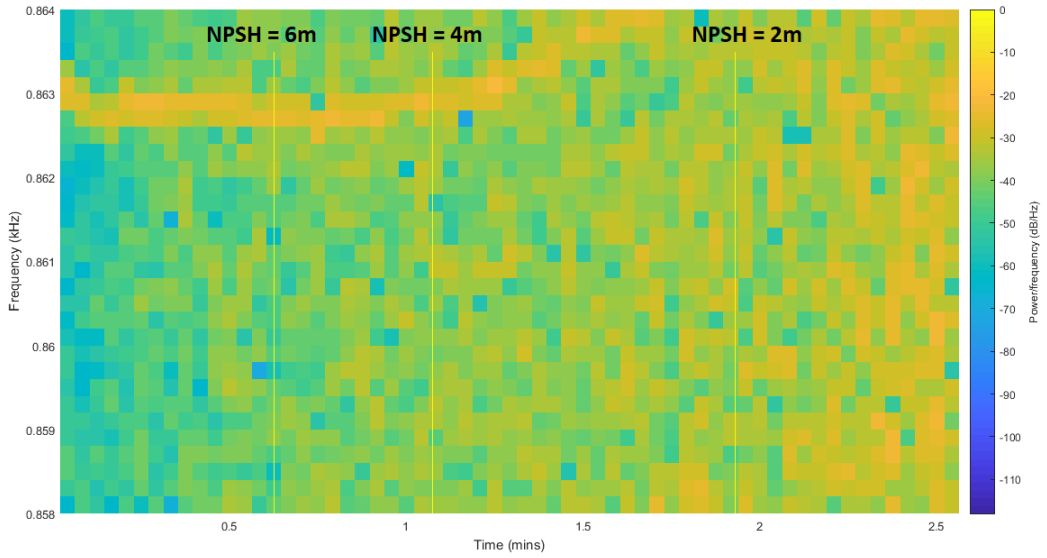


Figure 4.53: Spectrogram in radial Z axis around the third harmonic of the BPF

axis, in our experimental analysis the result is presented in Fig.4.54 .

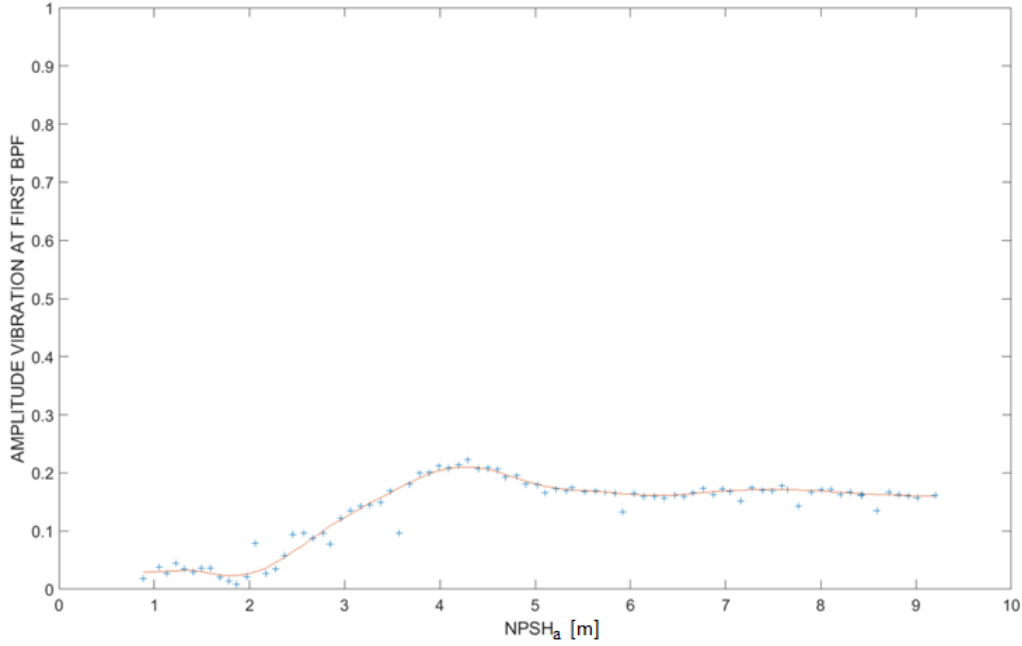


Figure 4.54: Vibration amplitude at 287 Hz

Spectral Kurtosis Kurtosis, in time domain, is a statistical parameter based on fourth moment of the signal and normalized by the square of the second moment minus three, because it tends to zero for a stationary signal and gaussian noise while it increases for impulsive signals [96]. Spectral Kurtosis is calculated using the Kurtosis at every frequency line in a time frequency diagram. The Spectral Kurtosis was defined in terms of Short Time Fourier Trasform by Antoni [97], using a sliding short time window (Hanning window) along the record in overlapping steps [96]. The spectra for each frequency line is calculated in the time direction.

Antoni [98] [97] has introduced the Spectral Kurtosis based on the Wold-Cramer decomposition of a non-stationary process $Y(t)$ as the output of a causal, linear and time varying system [99] with Eq. 4.47:

$$Y(t) = \int_{-\infty}^{+\infty} e^{j2\pi ft} H(t, f) dX(f) \quad (4.47)$$

where $H(t, f)$ is the time varying transfer function of the process $Y(t)$ at a frequency f and the $dX(f)$ is an orthogonal spectral process of unit variance [97]. In particular, Antoni explained [97] that usually the time varying transfer function is considered deterministic and due to this he introduces the complex envelope in terms of the outcome from the random variable w , with the following formulation: $H(t, f, w)$. Antony proposed the concept of Conditionally Nonstationary Process, considered as the processes generally stationary but non stationary for any partic-

ular outcome w [97]. The author explained that $H(t, f, w)$ is time stationary but independent in the spectral process $dX(f)$. The author states also emphasis that a non-stationary process can be translated in Conditional Nonstationary by randomisation of the time datum.

Introducing the spectral statistics for the second order moments, the general formulation is presented in Eq. 4.48:

$$S_{2nY}(f) = E\{|H(t, f)dX(f)|^{2n}\} \quad (4.48)$$

where it can be physically interpreted as measuring the time average of $|H(t, f)|^2$ at each frequency f or, also, it can be seen as a measure of how much energy is fluctuating in the time direction [98].

Considering the case $2n = 2$, the spectral moment of Eq. 4.49 is the Power Spectral Density of $Y(t)$ [98].

$$S_{2Y}(f) = E\{|H(t, f)|^2\} \cdot \sigma_X^2 \quad (4.49)$$

The spectral Kurtosis for Conditionally Non stationary Process is represented with Eq. 4.50 [98] and it is physically considered as a measure of temporal dispersion of the time frequency energy distribution.

$$K_Y(f) = \frac{S_{4Y}(f)}{S_{2Y}^2(f)} - 2 \quad (4.50)$$

Spectral Kurtosis at a given frequency measures the peakdness of the squared envelope $|H(t, f)dX(f)|^2$ and it is influenced by non-stationary patterns in the signal, indicating at which frequencies these patterns appear [98].

When a fault occurs in a machinery it causes a series of impacts [97] such that the measured vibrations are non-stationary.

The Spectral Kurtosis estimator used for analysing the cavitation signal from the accelerometers is based on the STFT [97].

The Short Time Fourier Transform in a discrete form for a process $Y(n)$ is presented in Eq. 4.51:

$$Y_w(kP, f) = \sum_{n=-\infty}^{\infty} Y(n)w(n - kP)e^{-j2\pi nf} \quad (4.51)$$

Where $w(n)$ is the analysis window with a length N_w , P represents the time step in the analysis.

For applying this parameter, the analysis has to fulfill certain constraints such that Spectral Kurtosis assumes high values when the signal is transient and zero when the signal is stationary (Gaussian). In particular the window length is selected with the following constraints [97]:

- $H(n, f)$ needs to have a slow temporal variation in n with respect to the window length N_w . The interval considered in the analysis window covers a signal that is quasi stationary;
- $H(n, f)$ needs to have a slow frequency variation in f as compared to the spectral bandwidth $w(n)$. This condition states that $H(n, f)$ has to be enough fast that no information is lost.

In Fig. 4.55 the Power Spectrum and the Spectral Kurtosis are represented at 500 l/min when the pump is working in normal conditions, instead in Fig. 4.56 they are represented at the same flow rate when the pump is going from normal to cavitation conditions.

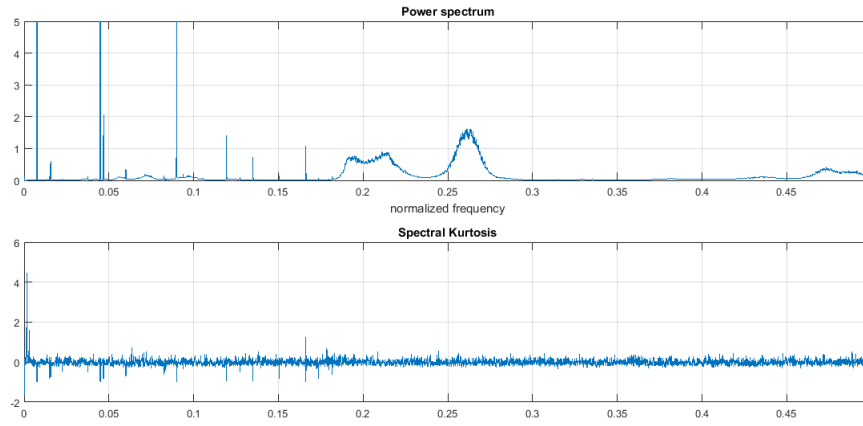


Figure 4.55: Power Spectrum calculated with the Welch method and Spectral Kurtosis at 500 l/min with the pump working in normal conditions without cavitation

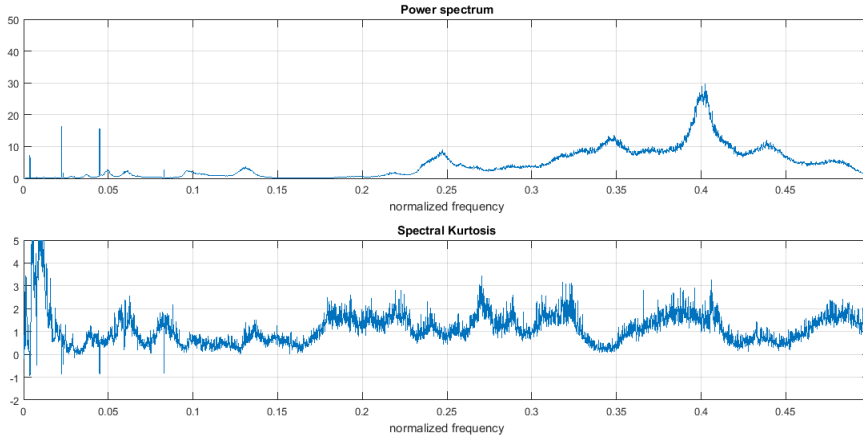


Figure 4.56: Power Spectrum calculated with the Welch method and Spectral Kurtosis at 500 l/min with the pump working from normal conditions to cavitation

The spectral Kurtosis shows that the impulsiveness of the signal in cavitation conditions, Fig.4.56, involve most of the frequencies of the signal compared to the

pump working in normal conditions, Fig.4.55. This parameter could be useful for detecting cavitation inception.

4.6 Statistical parameters selection for cavitation analysis

Cavitation, as introduced before is an unsteady phenomenon, that produces pressure pulses and oscillations. Pressure oscillations are related to the bubble dynamics, instead the pressure pulses are generated by the bubble collapses. Bubble collapses generate vibrations and acoustic noise that propagates through the liquid medium and the mechanical system. As introduced in the previous section, accelerometer sensors allow the detection of cavitation inception from the vibrations generated by the first bubbles collapse.

Pump cavitation presents different behaviors depending on the pump operating conditions, hydraulic design, type of cavitation and its detection is not simple. Vibration signals are affected by the distinct cavitation behaviors and some statistical parameters of the signal can be useful only at certain conditions.

The aim of the analysis presented in this chapters is to select a set of variables and sensor positions that can be used for developing a sensor and an algorithm for monitoring cavitation inception in centrifugal pumps. Different statistical approaches on the signal are presented in literature for detecting cavitation inception as presented in [72–74, 100–102].

Root Mean Square (RMS) analysis is extensively used in condition monitoring to indicate the energy content of a signal [81]. It permits to understand if the intensification of the vibration signal is a resulting of the cavitation process and if it shows similar development trends in all axes. In the experimental analysis, accelerometers have been positioned in the surface of the volute as presented in chapter 4.3. In [84], author state that if just one sensor is available, for developing a cavitation control system on a pump, it should be positioned in the radial direction Y. In [103], authors state that the best accelerometer position is, instead, in axis with the pump suction. Considering our experimental analysis, the RMS of vibration signals in different accelerometer positions (X, Y and Z axes) explains the behavior of pump vibrations during the NPSHa decreases. A resulting relation between the RMS of vibrations and the total head performance of the pump is presented in Fig. 4.57, Fig. 4.58 and Fig. 4.59. The signals computation have been made in all frequency range for six fluid flows [300 - 400 - 500 - 600 - 700 - 800 l/min].

The evolution of the RMS signals in X, Y and Z directions show similar trends with the development of cavitation while the intensification process differs in the three positions. Accelerometer Z reaches higher RMS values because it is closer to the region of cavitation bubbles collapse but it is also more influenced by the tongue vicinity in comparison to the other two measuring points. In particular, the blades of the impeller create in the region of the volute tongue a periodically varying velocity and pressure field, as explained in [57].

It is possible to state that the signal in the Z direction (radial) is preferable for performing the analysis on cavitation inception because vibrations are more intense, instead the analysis in the axial X direction is not essential, as will be analyzed and explained in the following considering the frequencies excited by cavitation inception.

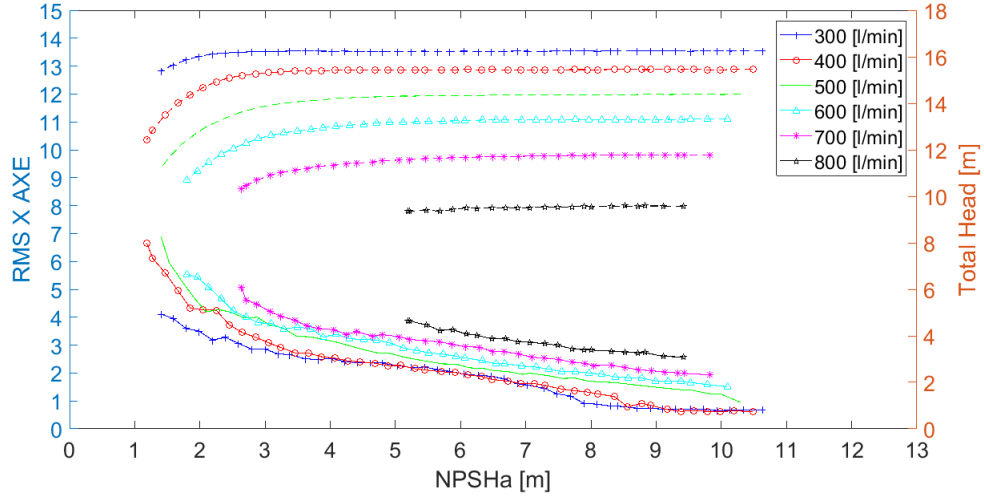


Figure 4.57: RMS in X axis, all frequency range, test with water

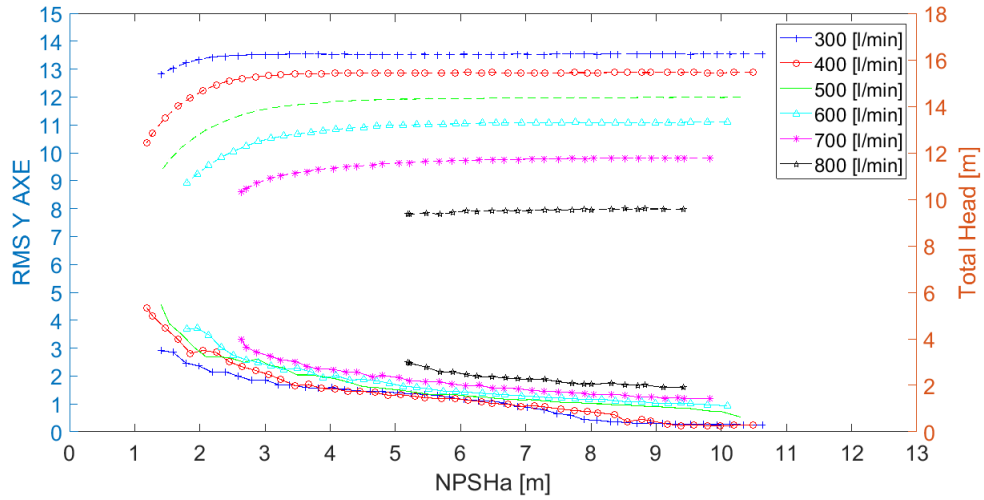


Figure 4.58: RMS in Y axis, all frequency range, test with water

As explained in the analysis of the power spectrum (chapter 4.5.2), the frequency range considered for detecting cavitation is in the range between 10 and 12.8 kHz. Considering now the RMS only in this frequency range (Fig.4.60, Fig.4.61 and Fig.4.62), it is confirmed that the accelerometer positioned in the X axis is useless at all the fluid flow ranges, because it doesn't show slope changes in the RMS but there is a not significant linear trend from non-cavitation to cavitation regime. Instead, the Z axis is preferable because it shows higher intensity and slope variations

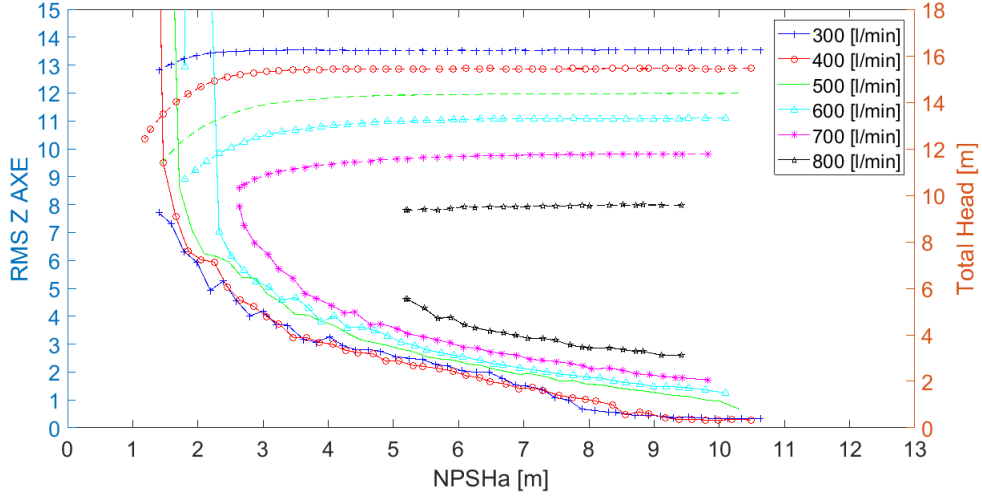


Figure 4.59: RMS in Z axis, all frequency range, test with water

in the signal.

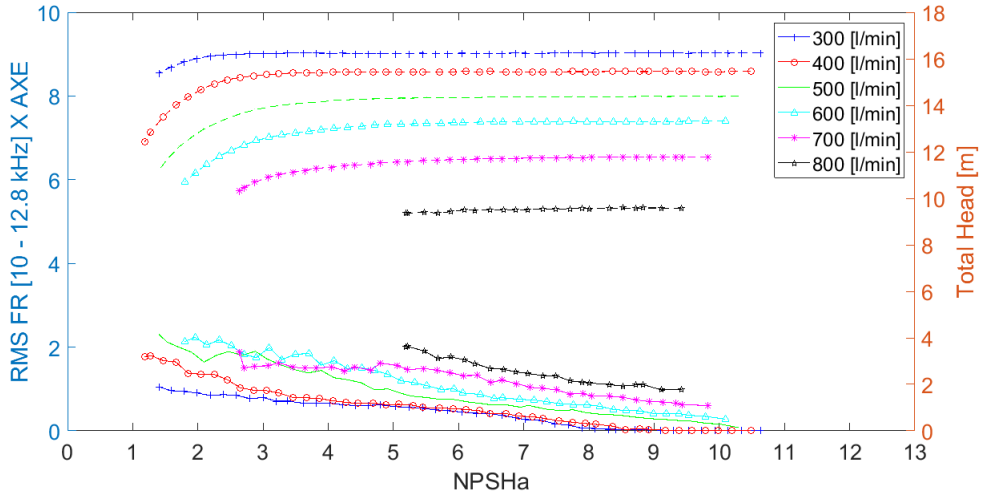


Figure 4.60: RMS in X axis in the frequency range [10 - 12.8 kHz], tests with water

In conclusion, the radial Z axis is the best position for an accelerometer having the aim of developing a cavitation control system for pumps used in professional appliances.

A selection process has been then performed in order to select, among the different signal analysis techniques presented in chapter 4.5, the most sensible statistical parameters for detecting cavitation inception because of their ability to provide a reliable physical description of the characteristics of the time domain vibration data. These parameters will be used for comparing the impact on cavitation of different polymers concentrations in water polymers solutions and, as future step of the research, for developing the previously cited control system.

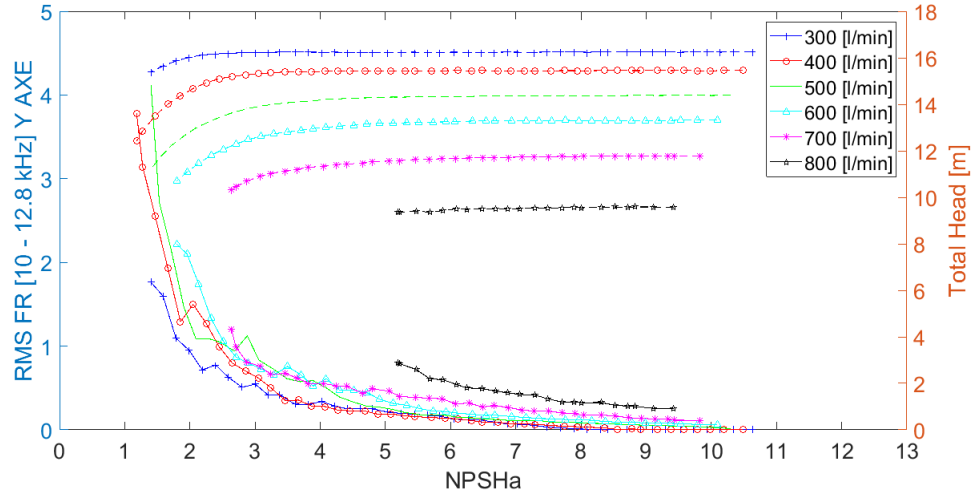


Figure 4.61: RMS in Y axis in the frequency range [10 - 12.8 kHz], tests with water

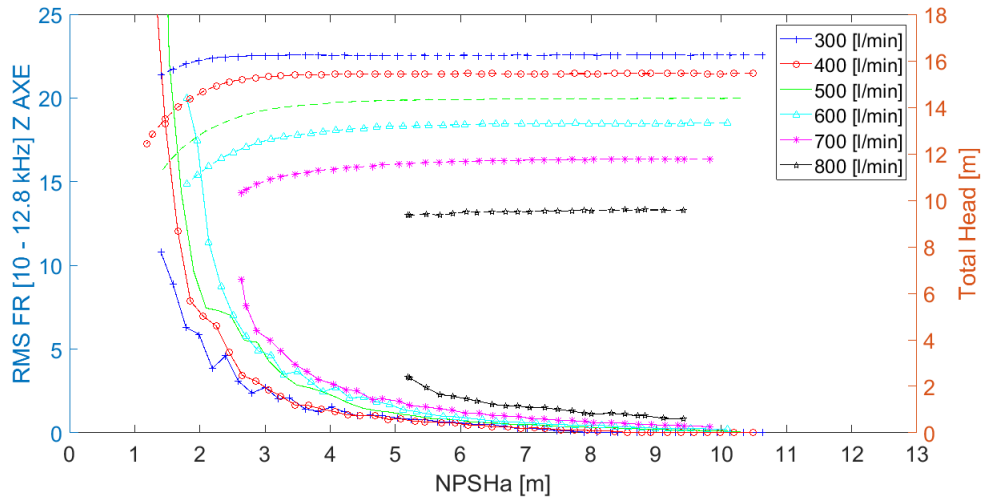


Figure 4.62: RMS in Z axis in the frequency range [10 - 12.8 kHz], tests with water

The following parameters have been selected:

- max peak values;
- max peak values in the cavitation frequency range;
- Kurtosis;
- Kurtosis in the cavitation frequency range;
- RMS;
- RMS in the cavitation frequency range;
- RMS of both radial accelerometers in the cavitation frequency range.

These results are confirmed also in [103], [74] [104] where it is shown that RMS, Kurtosis and peak values are the best time domain parameters for detecting cavitation. Another parameter useful for developing a cavitation control system, as presented in paragraph 4.5.3, is the frequency increment with respect to BPF at which a peak of vibration amplitude appears.

4.7 Pump performance with polymers solutions

As previously said in paragraph 4.2, the actual trend for the cleaning process in professional appliances brings to the use of solutions of water and highly concentrated chemistry which has significant impacts on pump operating behaviors. In particular, polymers could have an influence on the cavitation inception [39] and on pump performance [105–109], the last effect being mainly due to drag reduction, in particular at high Reynolds numbers, that can be enhanced by the presence not only of polymers [110], but also of surfactants. Drag reducing effect was discovered by Toms in 1948 [111] stating that a small concentration of poly(methyl methacrylate), approximately 10 ppm, showed a substantial reduction of the friction factor, even if the viscosity and density of the polymer solution only slightly diverged from those of the pure solvent. When polymers are added into turbulent flows, they are subjected to local flow conditions that induce flow orientation, chain stretching and relaxation. The result of these structural changes is evident as an intrinsic elastic stress, which alters the flow field [112]. In particular, the dynamics of the near wall turbulent structure influences the momentum transfer into the wall that, in macroscopic scale, is seen as a reduction of the friction factor [110]. The last one is a dimensionless pressure gradient that, for a given plant, is a function of only the Reynolds number for fully developed flow of Newtonian fluids. Good drag reducing agents are characterised by long and flexible chain backbone. Considering two distinct polymers having the same molecular weight and configuration, linear polymers will be more effective than branched polymers. Furthermore, polymers formed by low molecular weight monomers will have a greater drag reduction [111]. The object of this part of the thesis, as a part of the wider research regarding the overall analysis of pumps operating with detergent solutions and the monitoring of cavitation inception, is to analyse the rheology of water - Polyox WSR 301 solutions (considered as a representative component present in a detergent solution) and to evaluate their effects at different concentrations on the performance and the cavitation induced vibration of a pump for professional appliances.

4.7.1 Fluid solutions characterization

A polymer is a high mass macromolecule, composed by a certain number of repeating units (monomers) which form chains, whose length is proportional to the number of monomers present in the macromolecule [110]. Polymers normally form random coils. A coil stretch transition happens to a polymer chain if an external force is applied and a critical value is exceeded such that the polymer will deform proportionally to the strength applied [110]. Depending on the concentration of polymers in the solvent, it is possible to define two solution regimes: diluted and concentrated. In a diluted polymer solution, intramolecular interactions control the conformation

of the individual chains while the interactions between coils are negligible, i.e. there is no overlapping between the chains. Instead, in a concentrated regime every chain influences the nearest ones, so that there is an overlapping phenomenon between the polymer coils. Going from one regime to the other, the viscosity of the solutions varies according to different law scale. Doubling the concentration, in a diluted solution the viscosity increases its value quite proportionally instead, at concentrated regimes, viscosity increases ten or more times over. The solutions of WSR 301 used in this work have been tested with a modular rheometer platform (Thermo Scientific Haake Mars Rheometers) [113] with the aim of identify the concentration values at which they switch from the diluted to the concentrated regime. In Fig. 4.63 - 4.64 a bi-logarithmic representation of the obtained results are shown in terms of viscosity as a function of the measured shear stress 4.63 and the applied shear rate 4.64.

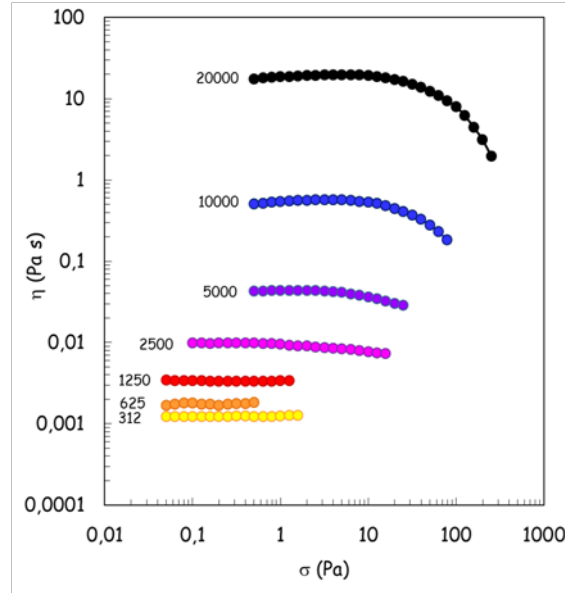


Figure 4.63: Experimentally measured shear stress-viscosity of different WSR 301 solutions

The solutions have been obtained doubling the dilution degree starting from a maximum concentration of 20000 ppm, so the successive considered values have been 10000, 5000, 2500, 1250, 625 and 312 ppm, respectively. The rehometer tests can cover only laminar flow conditions and, as long as viscosity is practically constant, the tested fluid behaves as a Newtonian fluid. From 312 ppm up to 1250 ppm, the solutions of Polyox WSR 301 meet this requirement, while from 2500 ppm up, the behavior of the solutions appear to be of the so-called shear thinning type. The diluted and concentrated regimes can be better recognized using these results to define the specific viscosity, Eq.4.52, where η_0 is the viscosity of the polymer solution and η_s is the viscosity of the pure solvent (water in this case).

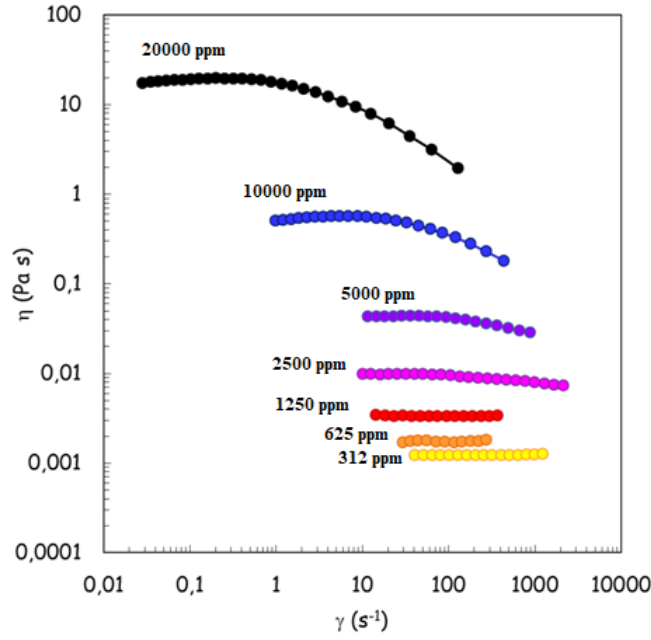


Figure 4.64: Experimentally measured shear rate- viscosity of different WSR 301 solutions

$$\eta_{sp,0} = \frac{(\eta_0 - \eta_s)}{\eta_s} \quad (4.52)$$

Fig. 4.65 shows the trend of η_0 and Fig. 4.66 that of $\eta_{sp,0}$ at increasing concentration values. The last one highlights the overlap concentration, c^* , defined as the upper limit between diluted regime and concentrated regime. The corresponding power laws of viscosity are, for the diluted regime, $\eta_{sp,0}c^{1.3}$ and, for the concentrated, $\eta_{sp,0}c^{5.1}$. As previously said these results, useful to classify the solutions used in the successive tests on the rig, have been obtained in laminar regimes, while it is well known that pumps usually works at high Reynolds number, in turbulent regimes. The effects on drag coefficient of polymer concentration is more evident in such regimes, as shown in Fig. 4.67 [111], which highlights the behavior of friction factor of water with or without a small amount of poly(ethylene oxide) in dependency of the Reynolds number.

4.7.2 Influence of polymers on pump performance

Tests with Polyox WSR 301 have been carried on at six concentrations falling in the field of diluted solutions, i.e. 100, 200, 400, 800, 1600 and 3200 ppm, and two concentrations around the lower limit of concentrated solutions, namely 6400 and 7000 ppm. Total head, absorbed power, efficiency and NPSHr of the pump obtained with pure water and with the polymers solutions are reported in Figs. 4.68, 4.69,

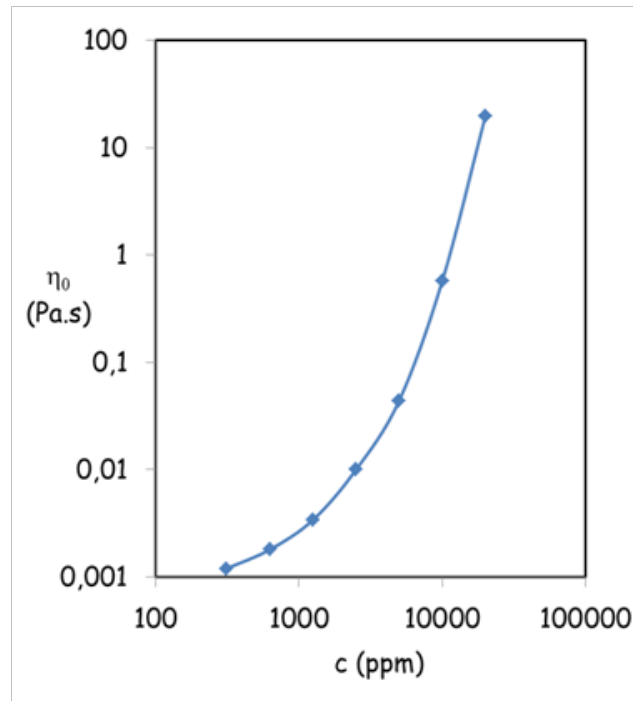


Figure 4.65: Trend of the viscosity solution

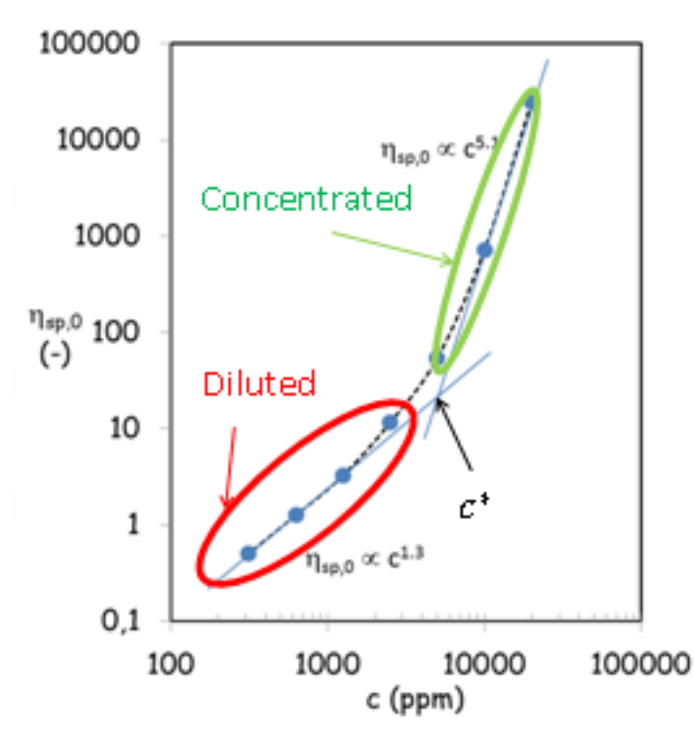


Figure 4.66: Trend of the specific viscosity in dependency of the solute concentration

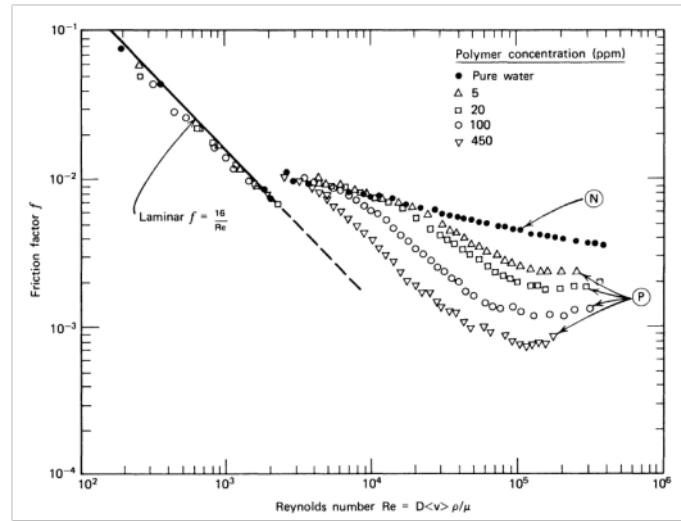


Figure 4.67: Friction factor for dilute aqueous solutions of poly(ethylene oxide)[12]

4.70 and 4.71 respectively.

The results obtained with all the tested fluids for the various performance parameters are given, instead, in terms of percentage of normalized differences, i.e. of difference between the value obtained with the considered solution and that obtained with pure water divided by the last one, expressed in %. Such percentages are calculated for the mass flow rate of maximum efficiency with water (Best Efficiency Point – BEP) and for that of maximum total head (Maximum Head Point – MHP). The symbols used for the various parameters are H, P, E and C for total head, absorbed power, efficiency and NPSHr, respectively. Figs. 4.72, 4.73, 4.74, and 4.75 report the values of H at MHP and of P, E and C at BEP as a function of the concentration of polymer, on a logarithmic scale.

The results of the experimental analysis show that the presence of polymers leads to an improvement of total head, relatively large especially at high concentrations, and to an almost equal trend for the absorbed power. As a result, efficiency shows quite limited variations, with detectable improvements only at the lowest concentrations. These results are as a whole aligned with the literature experimental trends, that refer almost at all to very low polymer concentrations. In [106] literature results are discussed showing that polymeric additives yields higher head (at a given flow rate), higher efficiency and lower required power. The author found, in his experimental analysis on the centrifugal pump of a waterjet, that with only 10 ppm of polymer concentration there was a 7% head increase and a 9% efficiency increase and that, generally, efficiency increases more than head, especially toward higher rotational speeds. This can be due to a reduction in some internal losses of the pump, as disk and casing friction [106–108], which affect more the overall efficiency of the pump than the energy usefully transferred to the fluid. In [114] efficiency im-

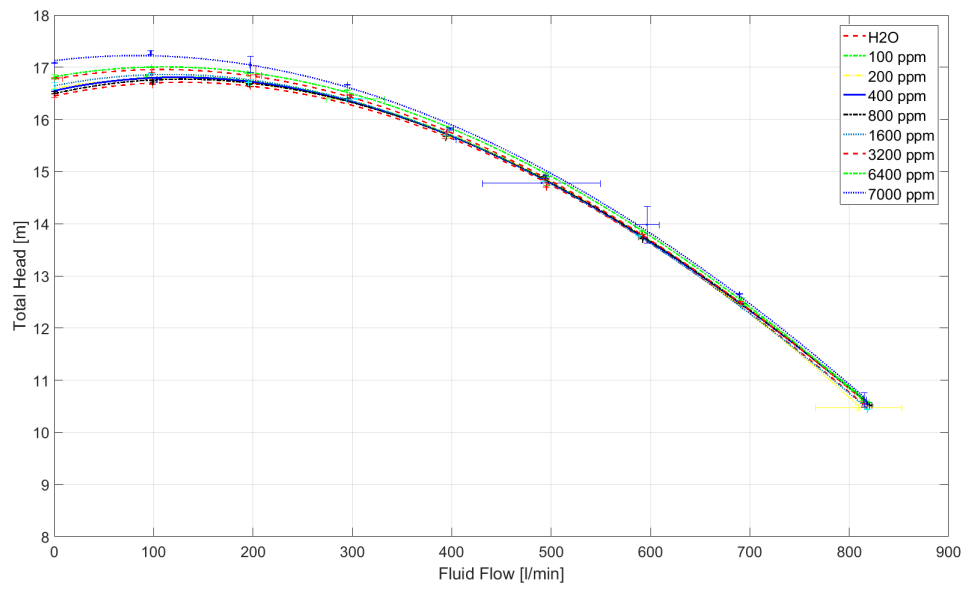


Figure 4.68: Total head at different concentrations (bar represents the error on the averaged of three samples)

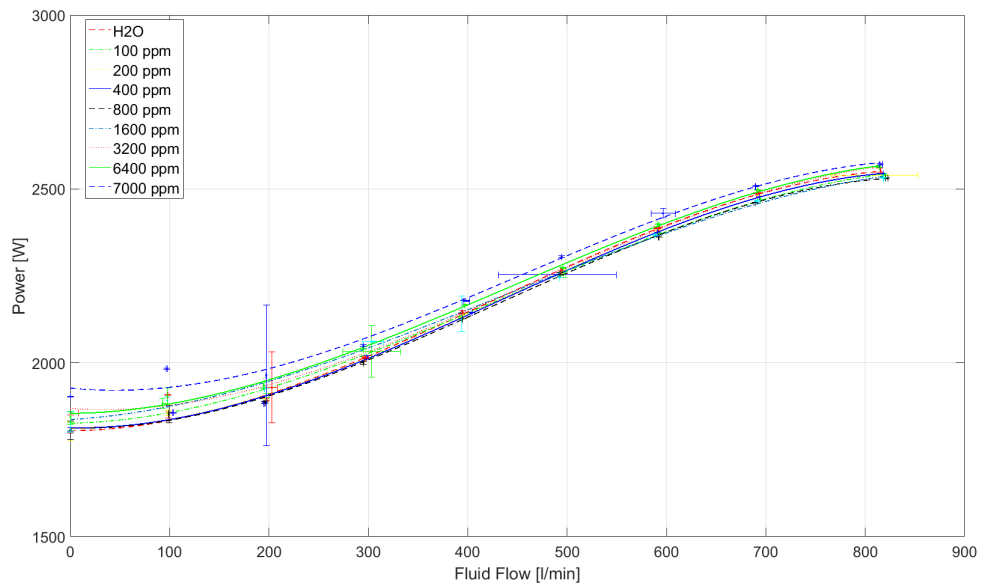


Figure 4.69: Power at different concentrations (bar represents the error on the averaged of three samples)

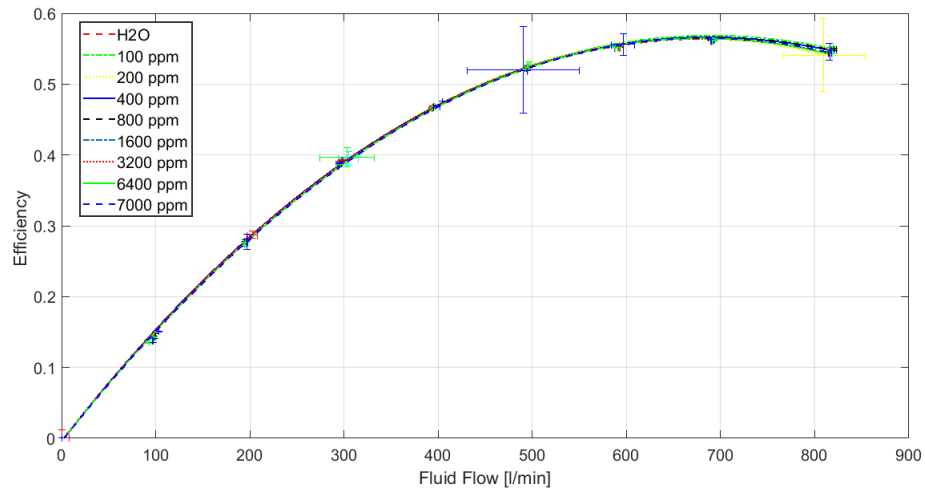


Figure 4.70: Efficiency at different concentrations (bar represents the error on the averaged of three samples)

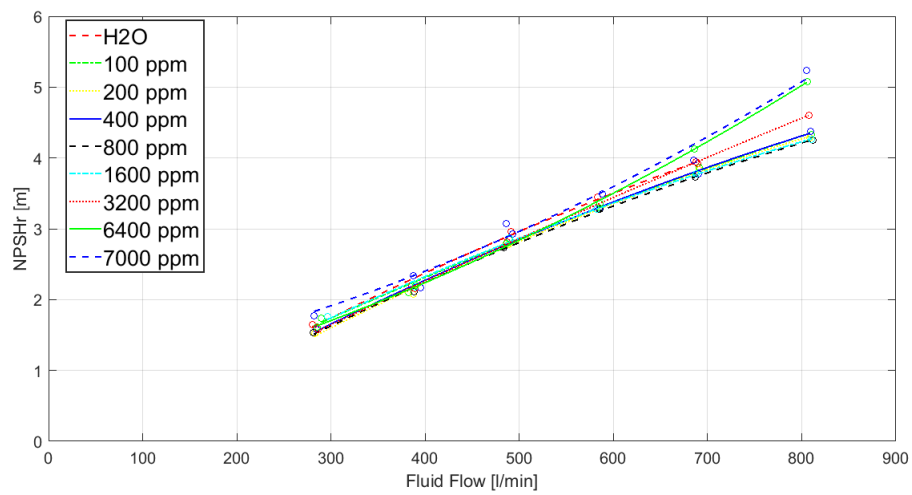


Figure 4.71: NPSHr at different concentrations

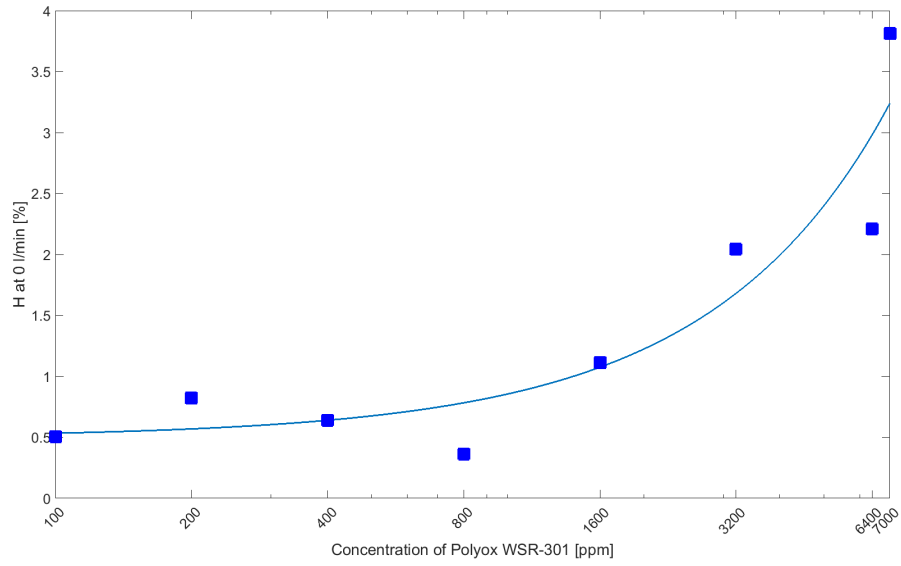


Figure 4.72: Relative increment of total head with solutions, H, at MHP

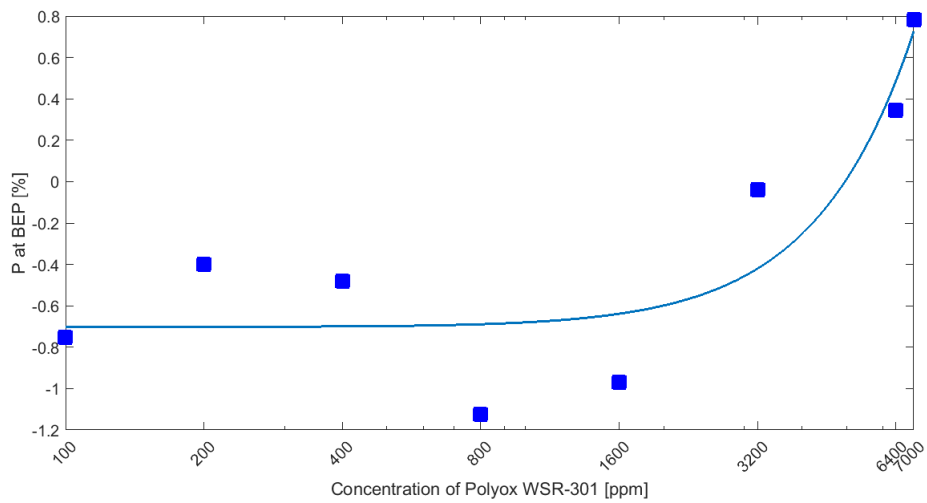


Figure 4.73: Relative increment of power with solutions, P, at BEP

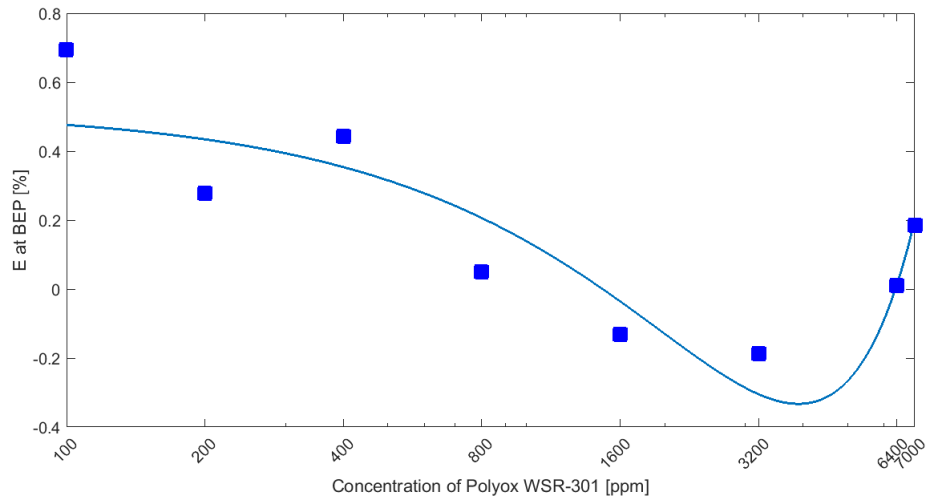


Figure 4.74: Relative increment of efficiency with solutions, E, at BEP

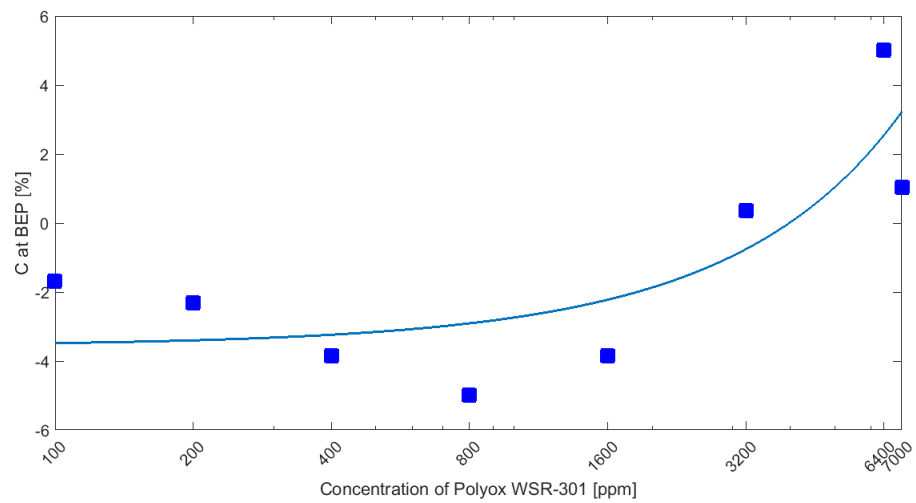


Figure 4.75: Relative increment of NPSHr with solutions, C, at BEP

provements were between 6% and 9%. In [107], with a 20 ppm solution, maximum efficiency was increased of 12% while the head coefficient remained almost the same, when compared with the pure water values. At speeds higher than the design one, polymer should become less effective in presence of stronger viscous dissipation due to high turbulence levels in the blade channels, but its effectiveness could remain in reducing the clearance viscous dissipation, leading to more significant efficiency increment, as reported in [106]. Results regarding the NPSHr values are difficult to interpret because the analyzed experimental data are too few and they do not show a clear trend and also because the authors have not been able to find comparative data in literature.

4.8 Analysis of cavitation induced vibrations with polymers solutions

As explained in the previous paragraph, the cavitation behaviour and the generated mechanical vibrations have been experimentally tested with different Polyox WSR 301 solutions at different concentrations, and compared with the analogous data obtained with pure water. The considered concentrations are 100 - 200 - 400 - 800 - 1600 - 3200 - 6400 - 7000 ppm. The tests have been repeated three times for every concentration to evaluate the repeatability of the results. The experimental data have been then analyzed in terms of the statistical parameters presented in paragraph 4.6.

Fig. 4.76 and Fig. 4.77 show the values of RMS in the Z and Y axes, respectively, at all frequency range, at the fixed fluid flow of 500 l/min and for every concentration of Polyox WSR 301: clear differences among pump vibrations measured at the various concentrations cannot be observed, there are only two out-liner, one at 7000 ppm and one at 400 ppm.

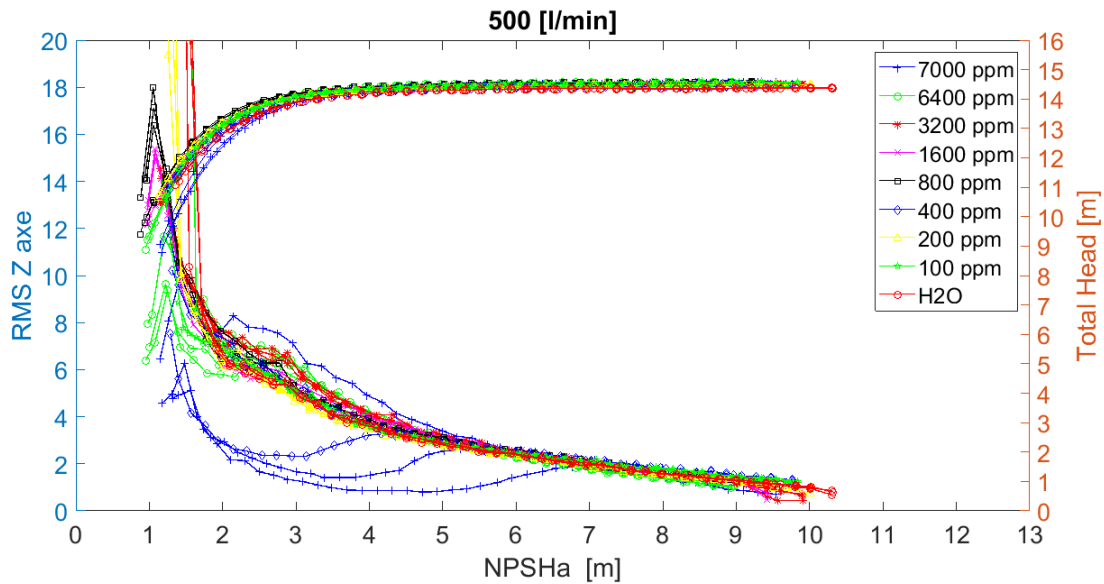


Figure 4.76: RMS at the Z axis

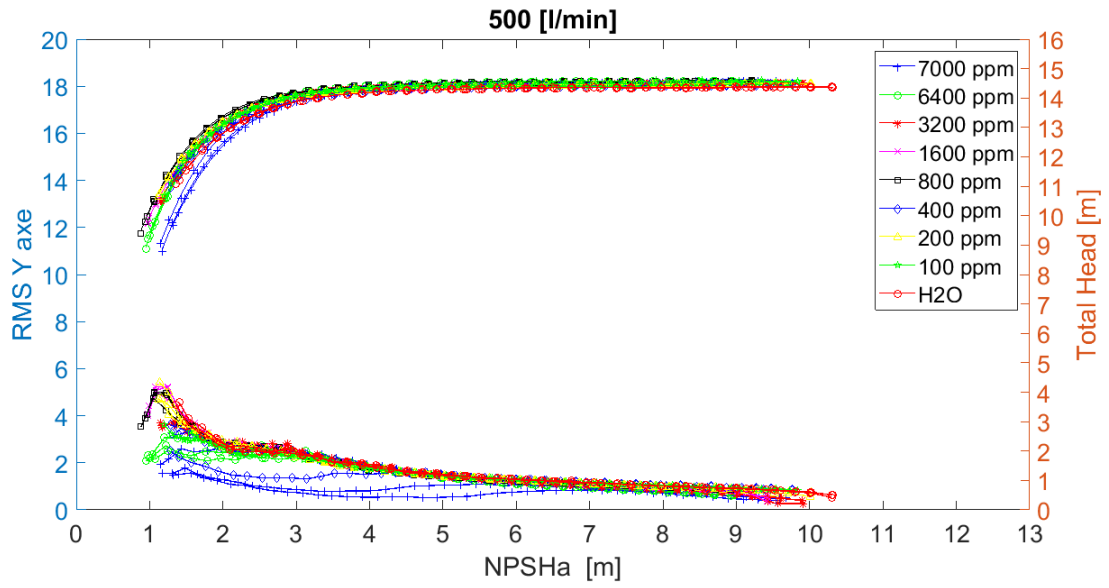


Figure 4.77: RMS at Y axis

Things change if the analysis is carried on in the cavitation frequency range, from 10 to 12.8 kHz, and considering only the Z axis, according to the conclusions of paragraph 4.6. Fig. 4.78 reports the results obtained with all the tests performed at 500 l/min: it shows different trends of the RMS at different concentration values of Polyox WSR 301. Results become even more significant if the mean of RMS values obtained with the three repeated tests are taken into account, as shown in Fig.4.79. In particular, it can be observed that in cavitation conditions mean RMS values increase when NPSHa decreases, but the root mean square of vibration at a fixed value of NPSHa is lowered by the increase of polymer concentration.

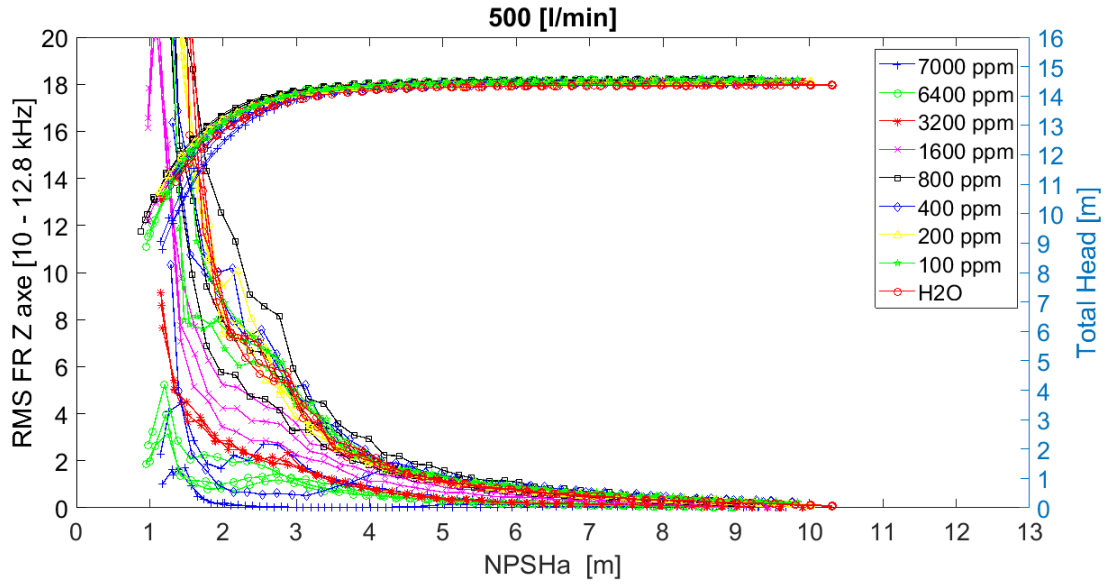


Figure 4.78: RMS in cavitation frequency range [10 - 12.8 kHz] at 500 l/min for all the test acquired in the Z axis for all the concentrations

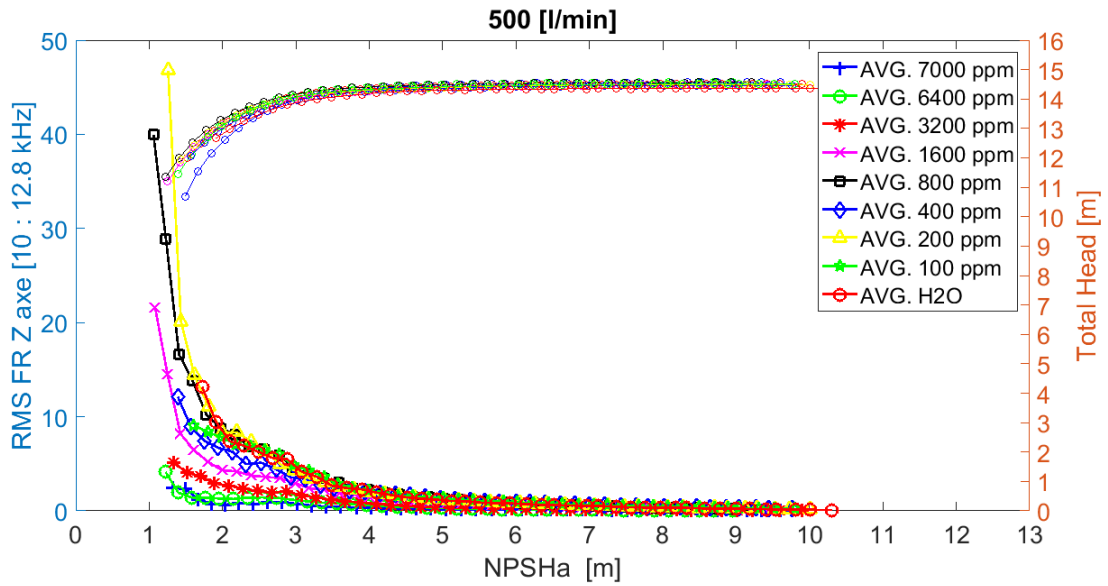


Figure 4.79: Averaged RMS of three test in cavitation frequency range [10 - 12.8 kHz] at 500 l/min acquired in the Z axis for all the concentrations

Figs. 4.80, 4.81, 4.82, 4.83, 4.84, 4.85 show the same curves of mean RMS as a function of NPSHa, obtained at all the considered values of mass flow rate. They confirm the correlation between RMS of vibrations and concentration values observed at 500 l/min.

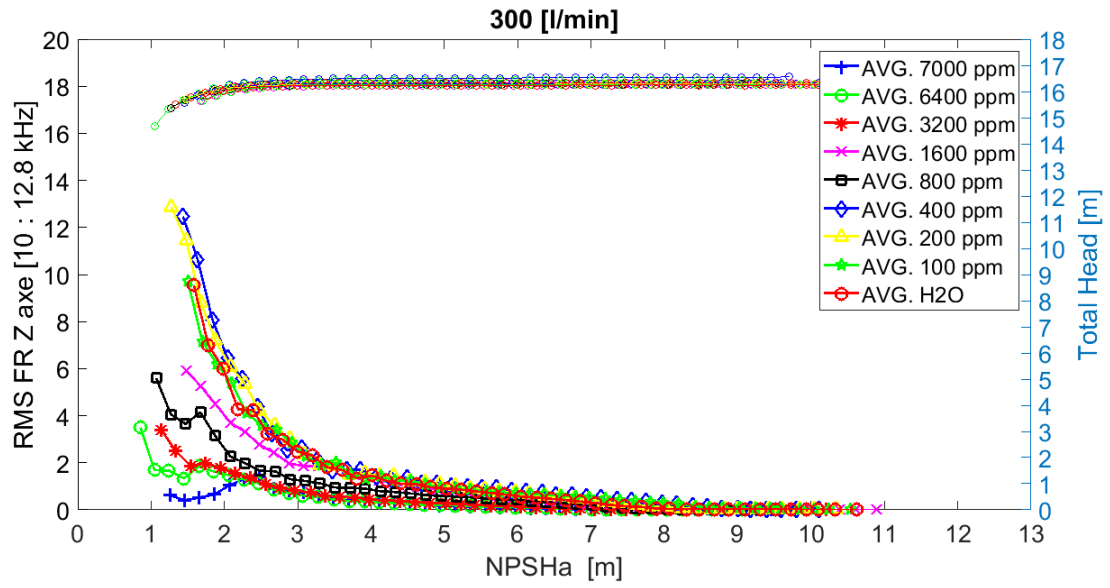


Figure 4.80: Averaged RMS in Z axis in cavitation frequency range [10 - 12.8 kHz] at 300 l/min for all the concentrations

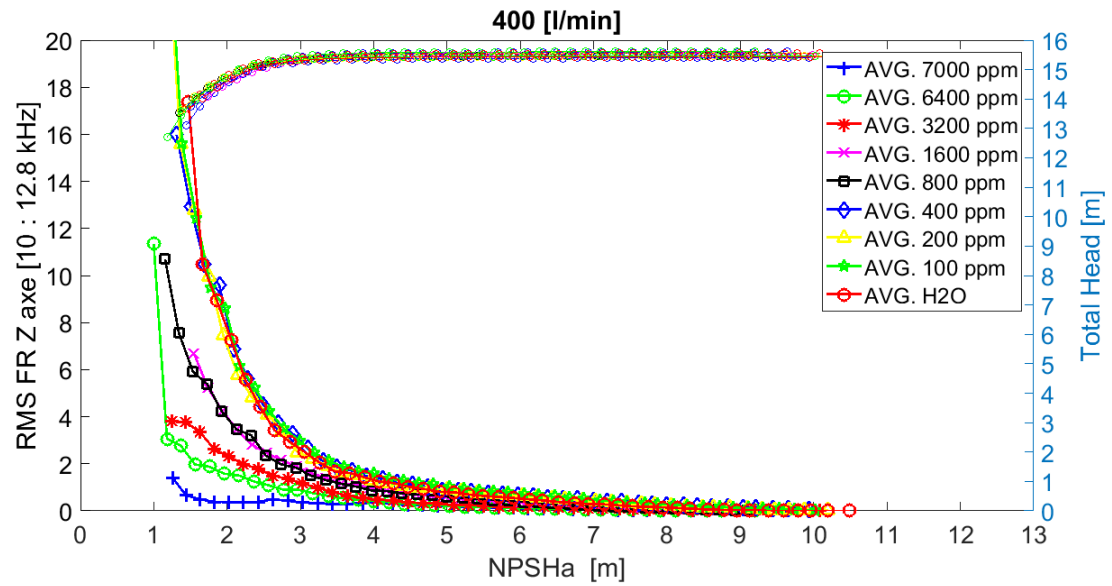


Figure 4.81: Averaged RMS in Z axis in cavitation frequency range [10 - 12.8 kHz] at 400 l/min for all the concentrations

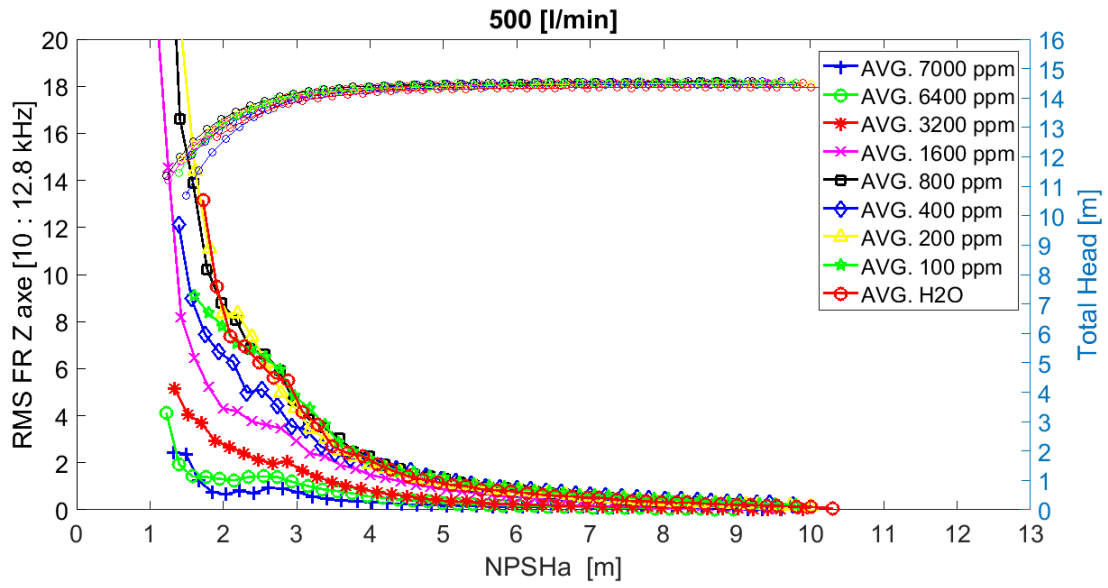


Figure 4.82: Averaged RMS in Z axis in cavitation frequency range [10 - 12.8 kHz] at 500 l/min for all the concentrations

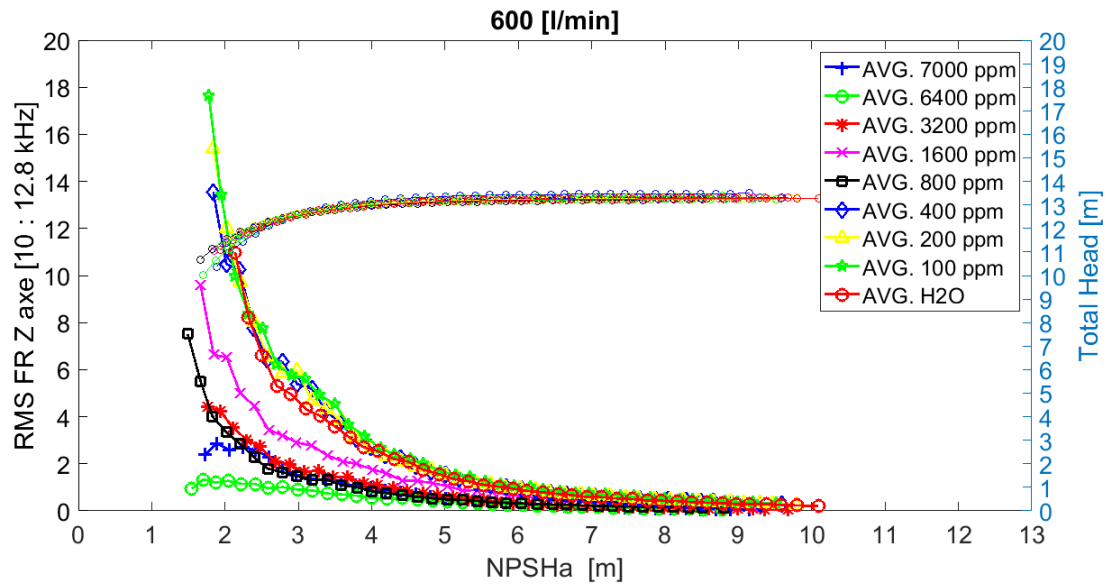


Figure 4.83: Averaged RMS in Z axis in cavitation frequency range [10 - 12.8 kHz] at 600 l/min for all the concentrations

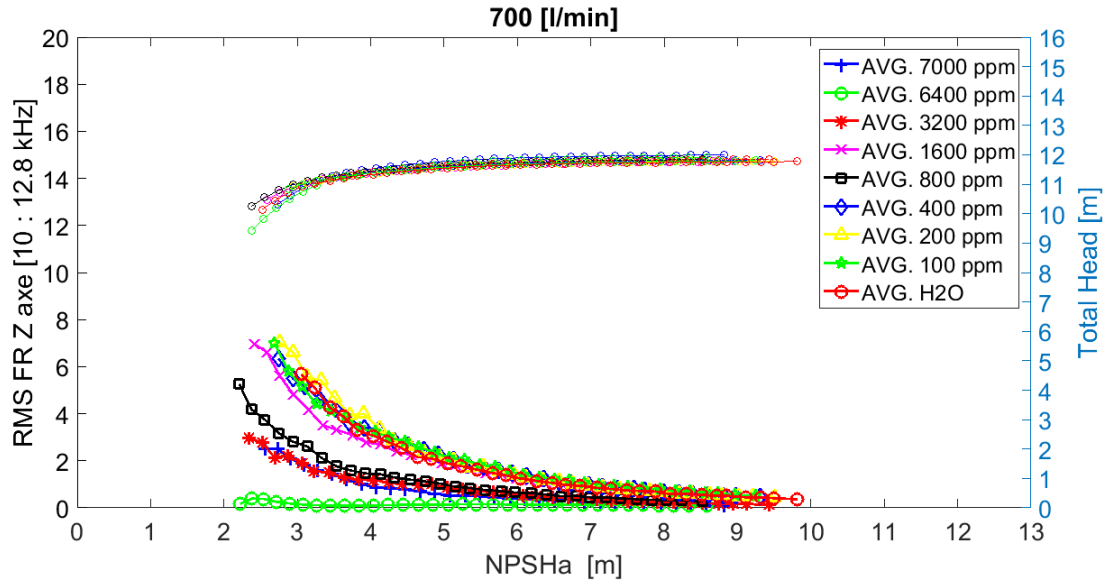


Figure 4.84: Averaged RMS in Z axis in cavitation frequency range [10 - 12.8 kHz] at 700 l/min for all the concentrations

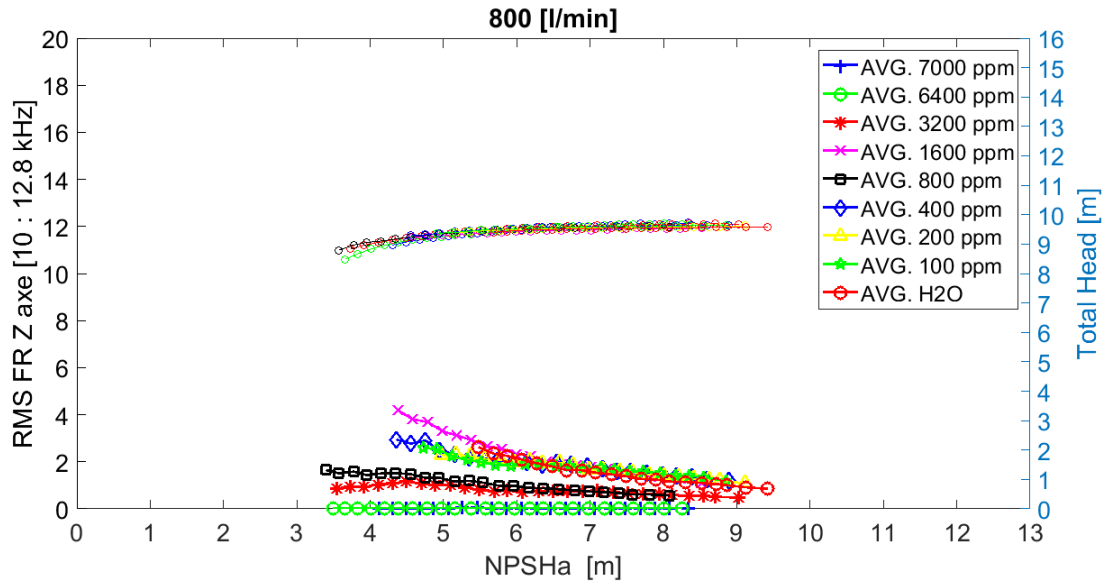


Figure 4.85: Averaged RMS in Z axis in cavitation frequency range [10 - 12.8 kHz] at 800 l/min for all the concentrations

Figs. 4.86, 4.87, 4.88, 4.89, 4.90, 4.91 show the mean peak values curves calculated in the frequency range between 10 and 12.8 kHz as a function of NPSHa, obtained at all the considered values of mass flow rate. Also in this case, it can be observed that in cavitation conditions the values in the frequency range between 10 and 12.8 kHz of the considered parameter increase when NPSHa decreases, but the peak values of vibration at a fixed value of NPSHa is lowered by the increase

of polymer concentration. The reduction of the acceleration peak values could be explained by the effect of polymers on cavitation. They suppress cavitation reducing the fine bubble cloud.

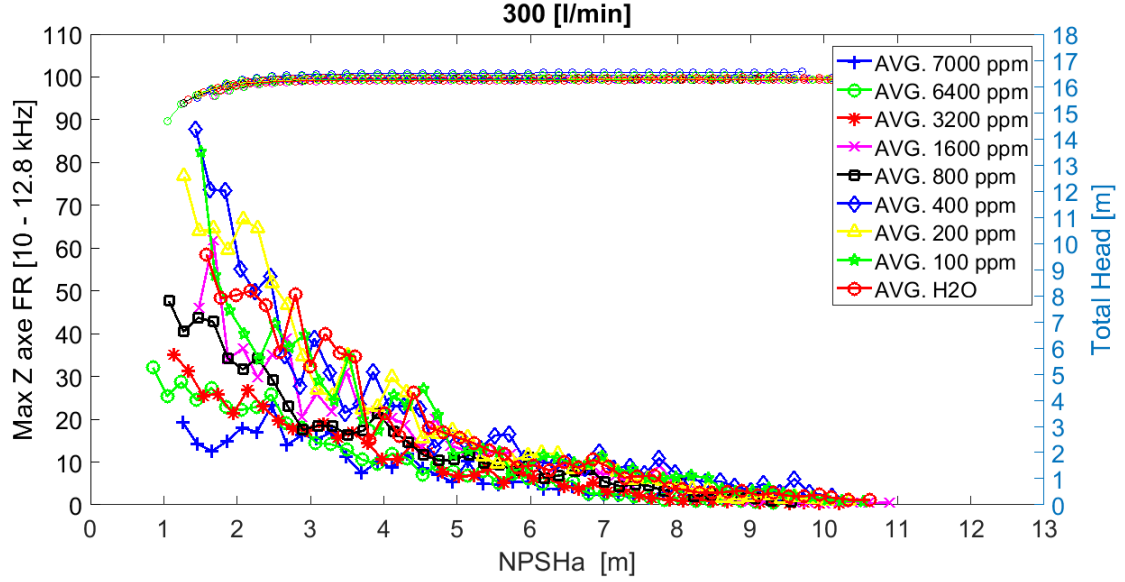


Figure 4.86: Averaged max accelerations in Z axis in cavitation frequency range [10 - 12.8 kHz] at 300 l/min for all the concentrations

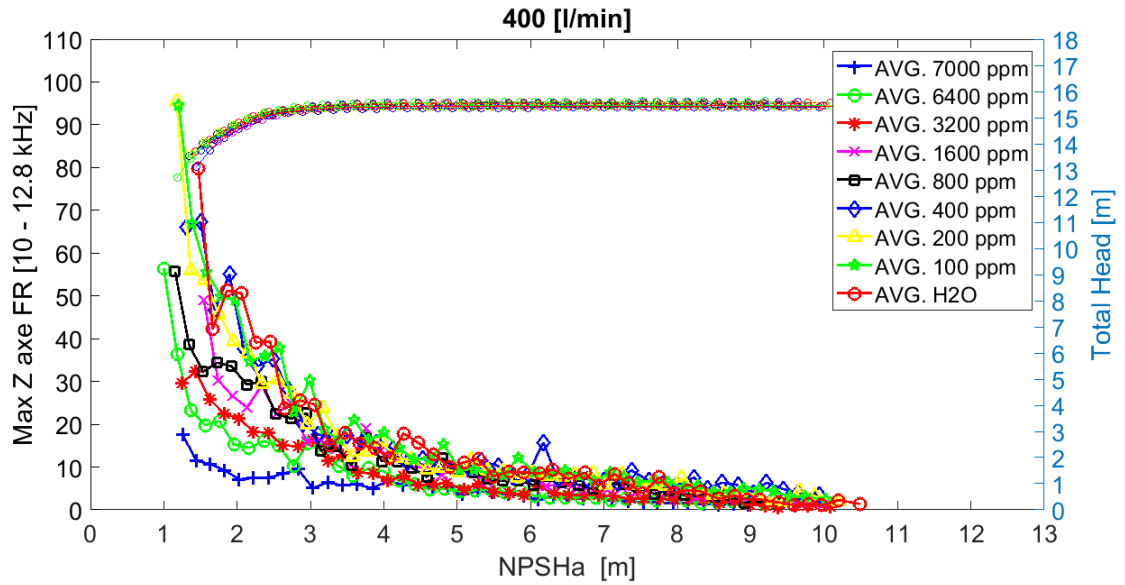


Figure 4.87: Averaged max accelerations in Z axis in cavitation frequency range [10 - 12.8 kHz] at 400 l/min for all the concentrations

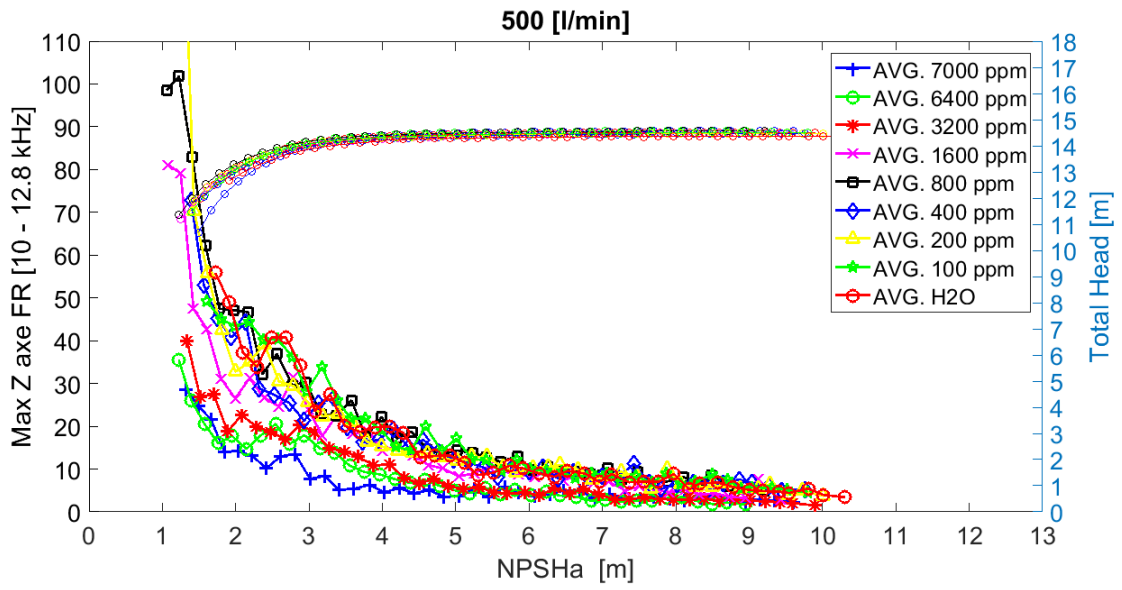


Figure 4.88: Averaged max accelerations in Z axis in cavitation frequency range [10 - 12.8 kHz] at 500 l/min for all the concentrations

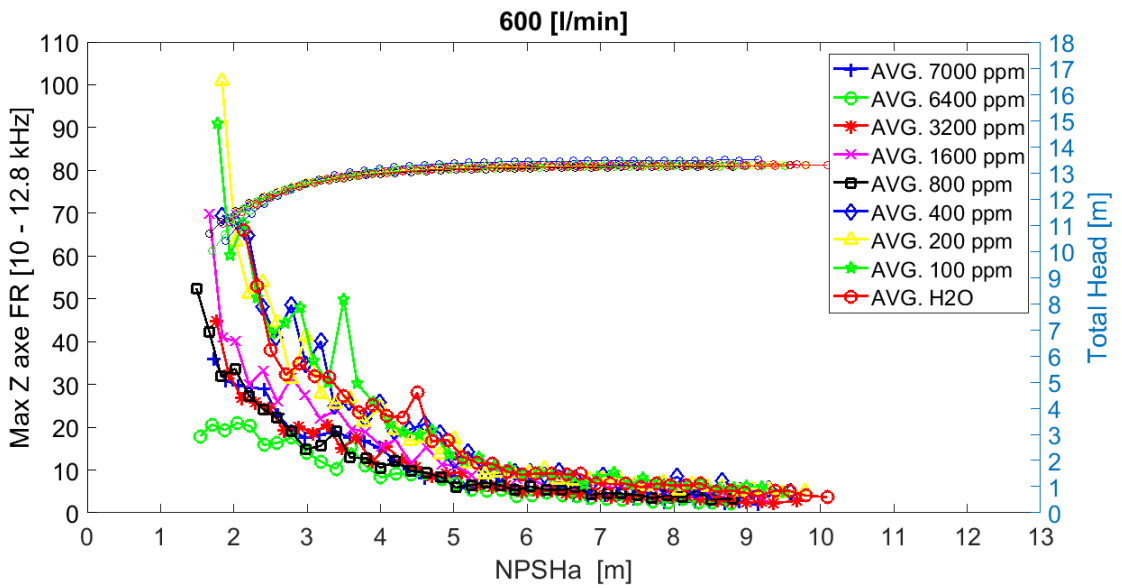


Figure 4.89: Averaged max accelerations in Z axis in cavitation frequency range [10 - 12.8 kHz] at 600 l/min for all the concentrations

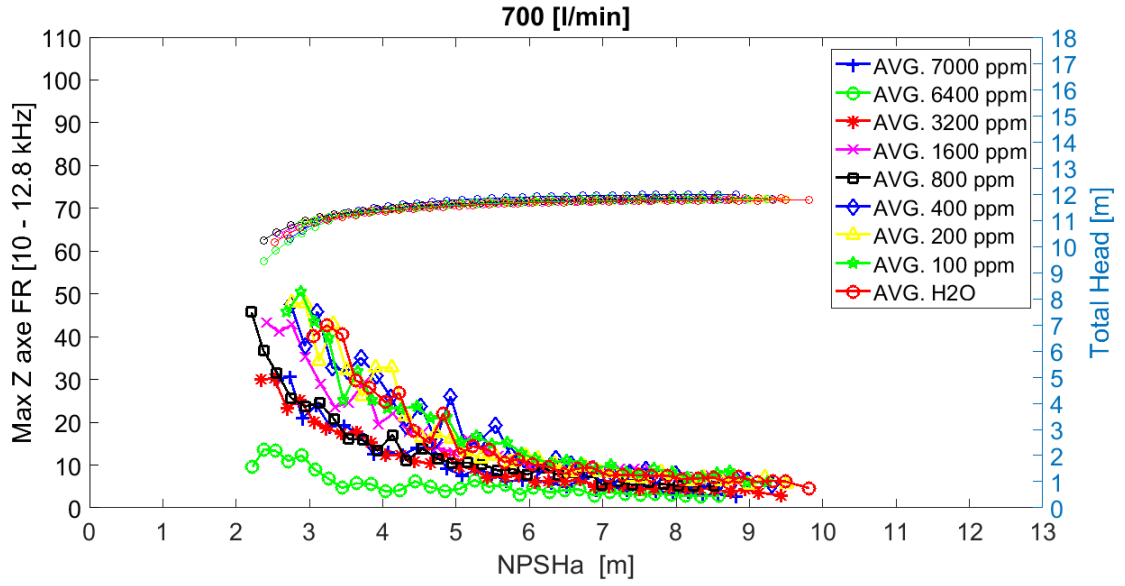


Figure 4.90: Averaged max accelerations in Z axis in cavitation frequency range [10 - 12.8 kHz] at 700 l/min for all the concentrations

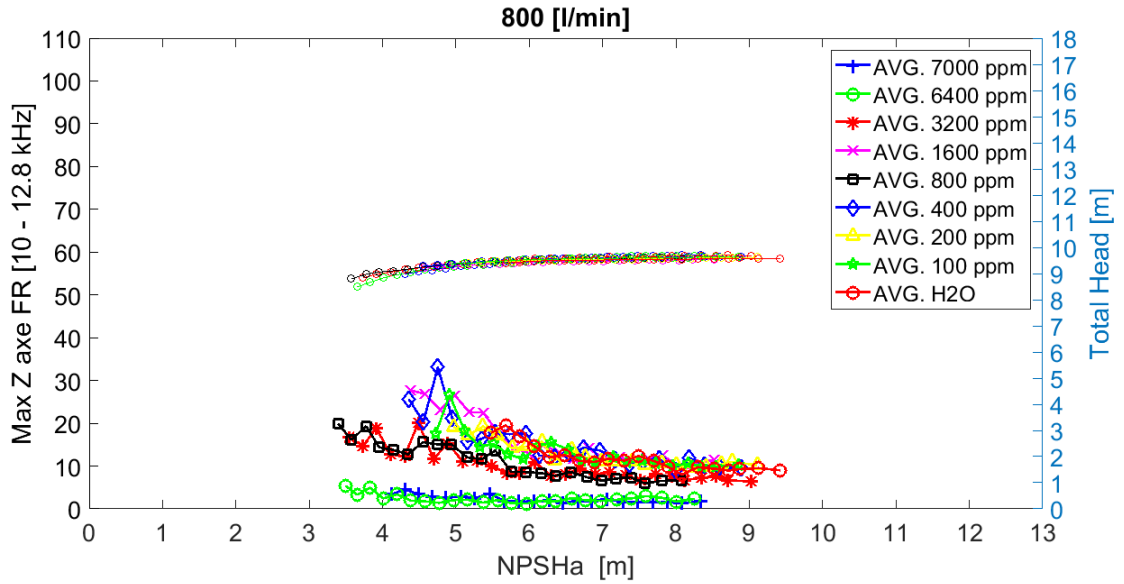


Figure 4.91: Averaged max accelerations in Z axis in cavitation frequency range [10 - 12.8 kHz] at 800 l/min for all the concentrations

The trend among different concentrations and vibration levels for both RMS and peak values in the cavitation frequency range between 10 and 12.8 kHz confirms the results found in literature as [115], [116], [117] where it is reported that a small quantity of polymers additives suppress inception and development of cavitation, changes the shape of the cavitation bubbles and decreases the energy of the cavitation shock pressures drastically. In literature there are several studies that deal with the ef-

fects of polymers on cavitation erosion [117], [43], [116], where authors report that polymers solution substantially mitigate cavitation damage, increasing the liquid's cavitation threshold stress [43]. Authors in [43] tested different polymer concentrations in a cavitation erosion test rig analyzing the weight loss of the samples due to cavitation erosion. Authors in [43], state that the samples subjected to cavitation in a polymer solution (1 % concentration) present one order of magnitude less damage than in the case of water. Authors show that the addition of polymers suppresses the cavitation damage in polymer aqueous solution because it reduces the maximum pressure inside the bubble at its minimum volume. Furthermore, in papers [116], [43] authors explain that a polymer solution decreases the nuclei population with the increasing polymer concentration. The cavitation threshold and erosion effect are affected by the decreased nuclei population. A collapsing bubble in a polymer solution has lower compression compared to a bubble positioned in water. Polymers affect the extensional viscosity dissipating energy during the collapse of the bubble. Authors in [43] observed a reduction of the individual craters and damaged area of the samples with an increasing quantity of polymers in a solution. Polymer solutions diminish the amplitude of the shock waves because the bubble content is less compressed.

The impact of Polyox WSR 301 on cavitation is less visible if the signal peak values are considered not only in the cavitation frequency range, just considered, but in all the frequency range as can be seen in in Figs. 4.92, 4.93, 4.94, 4.95, 4.96, 4.97, where they do not show clear trends among different concentrations.

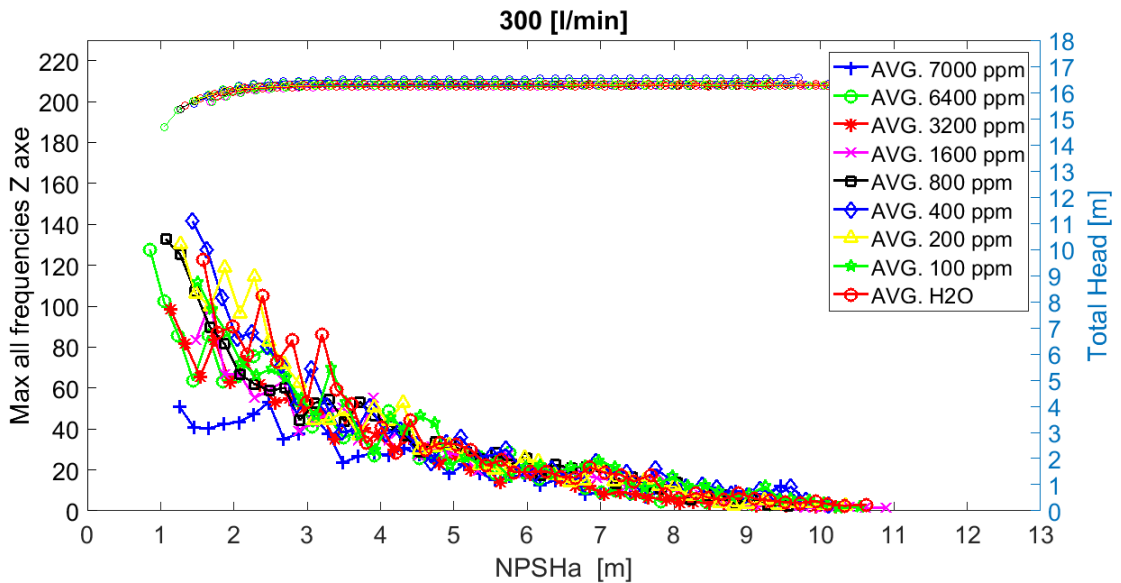


Figure 4.92: Averaged max values in Z axis in all frequency range at 300 l/min for all the concentrations

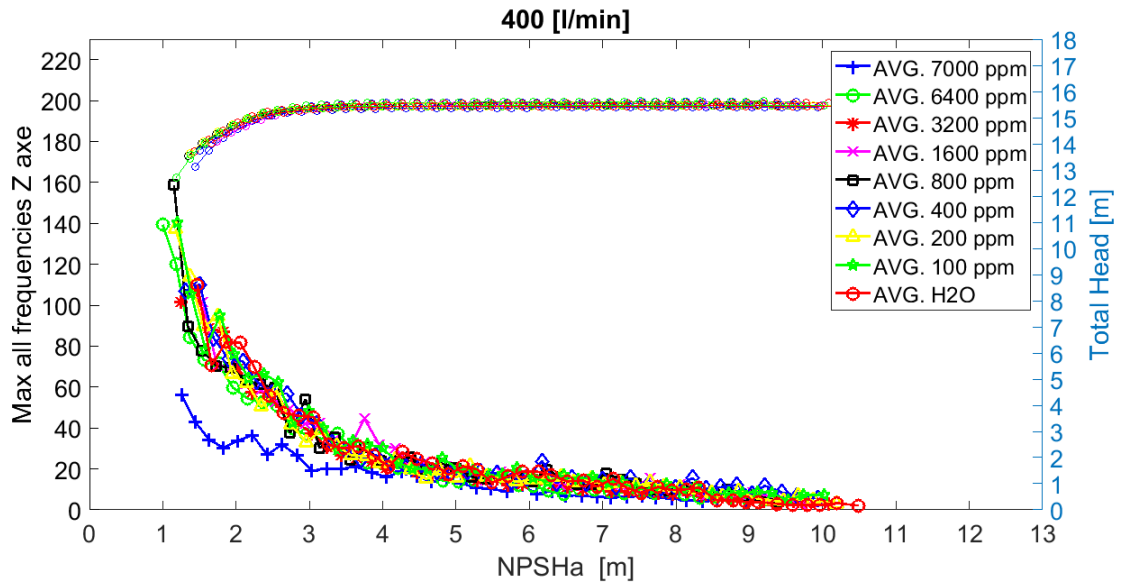


Figure 4.93: Averaged max values in Z axis in all frequency range at 400 l/min for all the concentrations

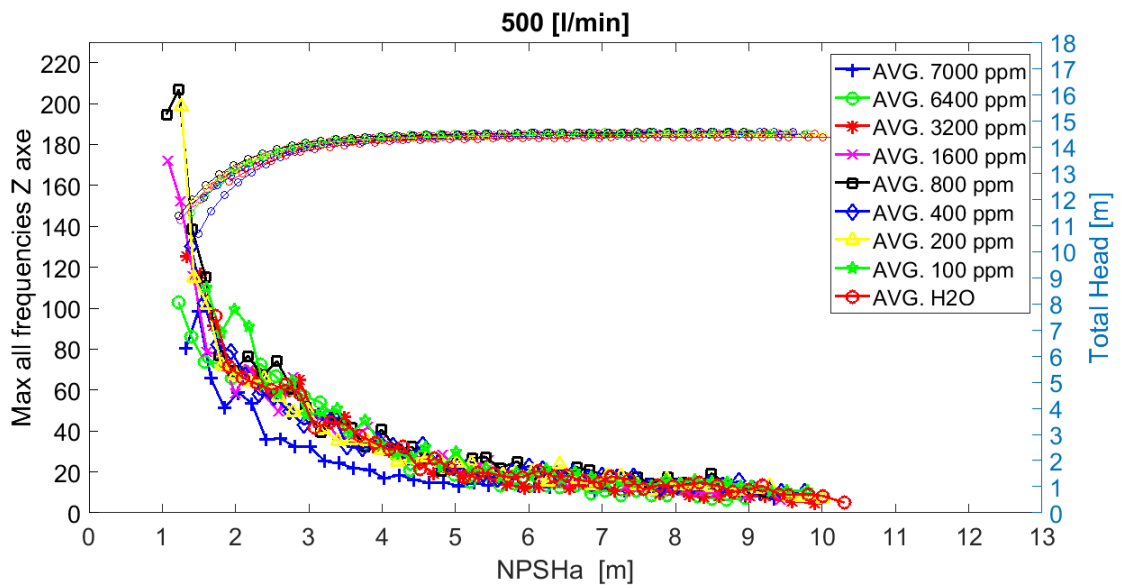


Figure 4.94: Averaged max values in Z axis in all frequency range at 500 l/min for all the concentrations

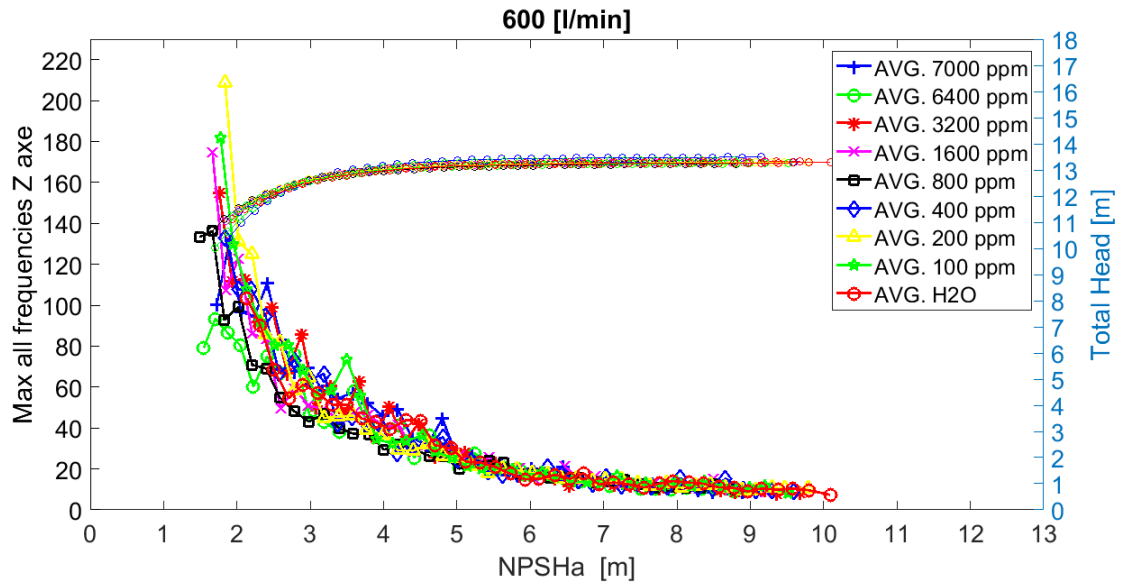


Figure 4.95: Averaged max values in Z axis in all frequency range at 600 l/min for all the concentrations

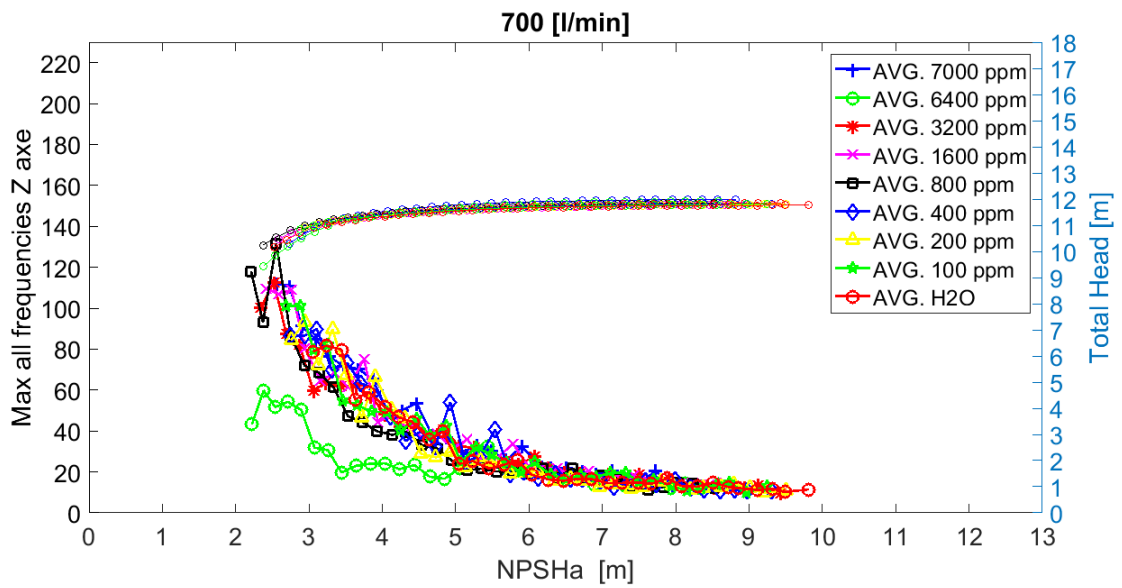


Figure 4.96: Averaged max values in Z axis in all frequency range at 700 l/min for all the concentrations

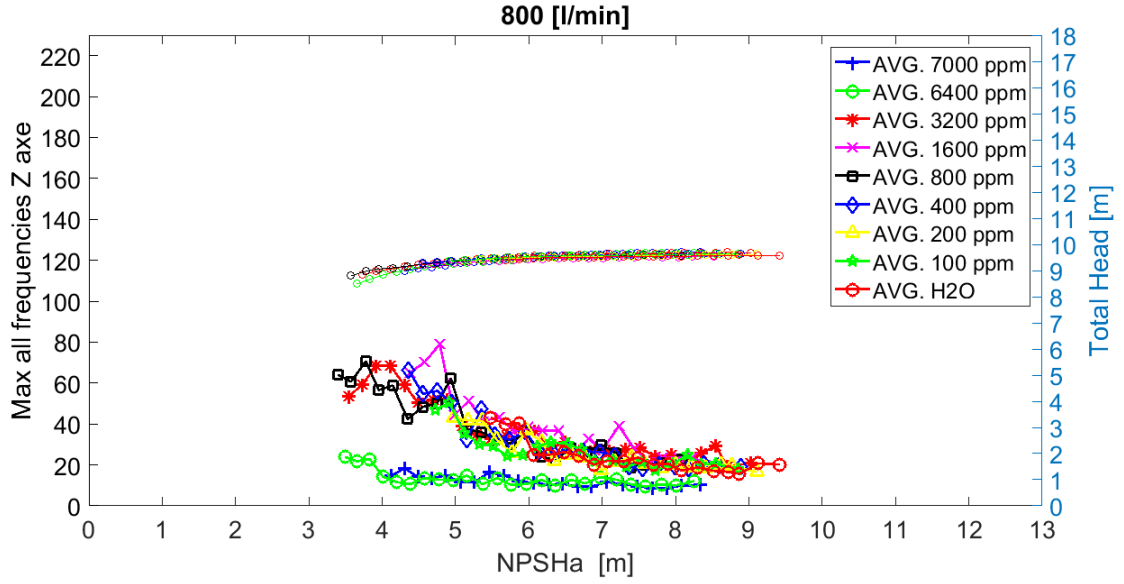


Figure 4.97: Averaged max values in Z axis in all frequency range at 800 l/min for all the concentrations

Figs. 4.98 , 4.99, 4.100 , 4.101 , 4.102 , 4.103 show the mean kurtosis value curves calculated in the frequency range between 10 and 12.8 kHz as a function of NPSHa, obtained at all the considered values of mass flow rate.

In particular, it can be observed that in cavitation conditions, at flow rates from 600 l/min to 800 l/min, mean Kurtosis values increase when NPSHa decreases, but at a fixed value of NPSHa they do not show a trend with the increase of polymer concentration similar to that observed in the cases of RMS and peak values. Lower fluid flows, between 300 and 400 l/min, show the strong impulsive characteristic of the vibrations generated because the fluid flow does not smoothly match the solid boundaries inside the pump. Generally, the averaged amplitude of the pressure fluctuations is higher at low fluid flows than at both the design flow and high fluid flow conditions [118]. In [119] authors show that at low flow rates, an unsteady backflow is believed as a source that disturbs the straight flow field, strengthening the magnitude of unsteady pressure fluctuations. This effect is generated by flow separation, recirculation and increased fluid structure interactions that are independent from cavitation: for this reason, Kurtosis in the frequency range between 10 and 12.8 kHz is not reliable for detecting cavitation at low fluid flows. It could be used only for fluid flows from 600 l/min to 800 l/min.

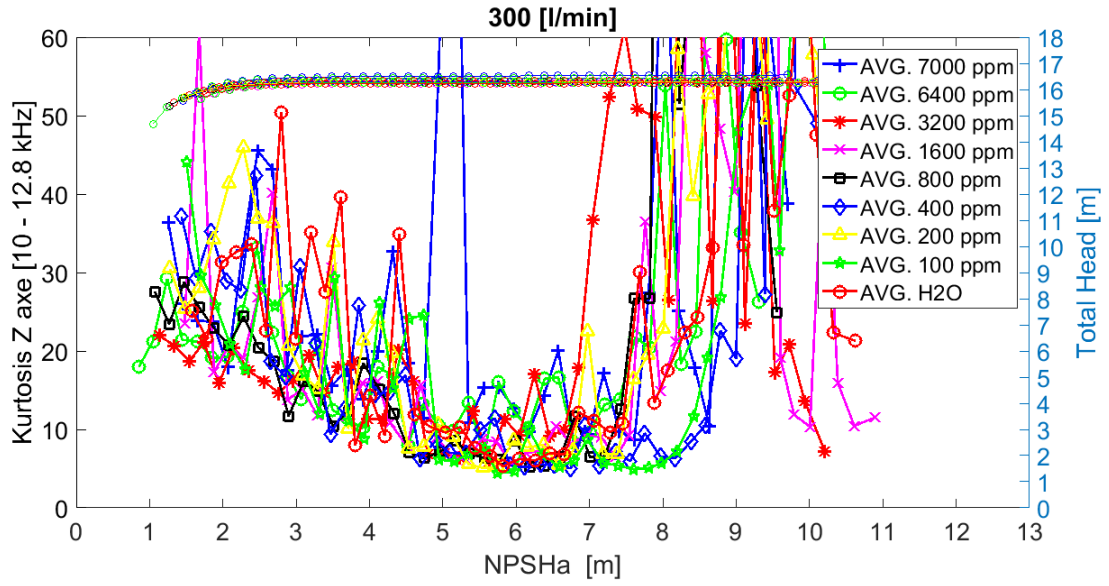


Figure 4.98: Averaged Kurtosis in Z axis in cavitation frequency range [10 - 12.8 kHz] at 300 l/min for all the concentrations

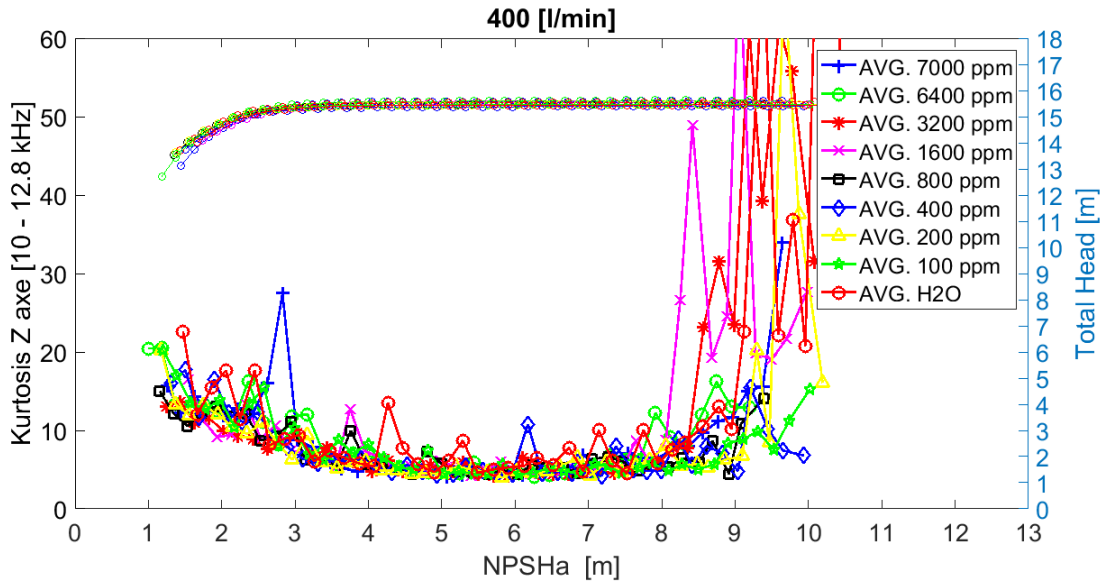


Figure 4.99: Averaged Kurtosis in Z axis in cavitation frequency range [10 - 12.8 kHz] at 400 l/min for all the concentrations

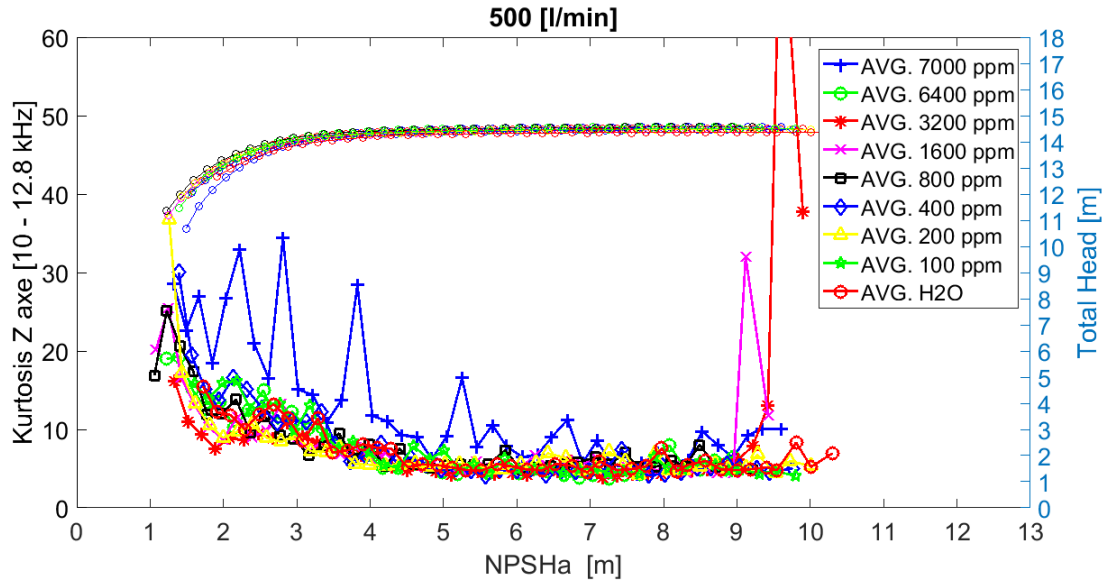


Figure 4.100: Averaged Kurtosis in Z axis in cavitation frequency range [10 - 12.8 kHz] at 500 l/min for all the concentrations

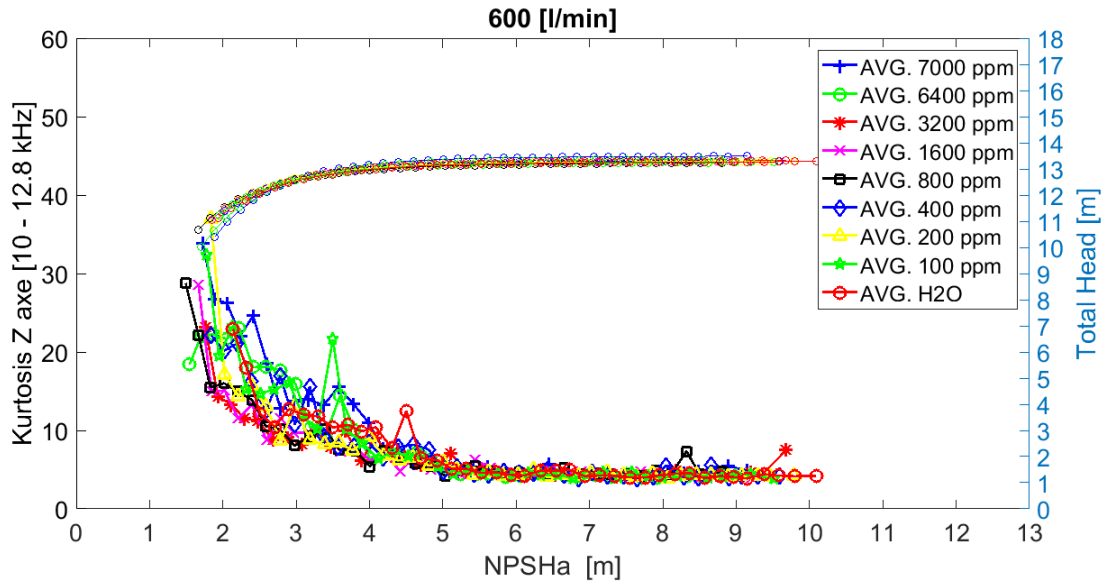


Figure 4.101: Averaged Kurtosis in Z axis in cavitation frequency range [10 - 12.8 kHz] at 600 l/min for all the concentrations

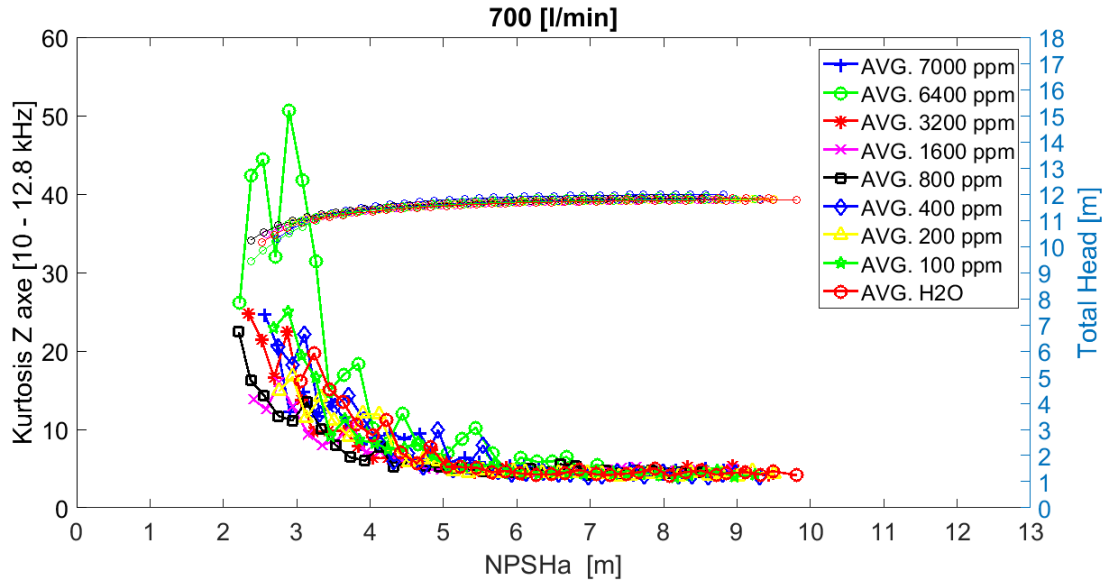


Figure 4.102: Averaged Kurtosis in Z axis in cavitation frequency range [10 - 12.8 kHz] at 700 l/min for all the concentrations

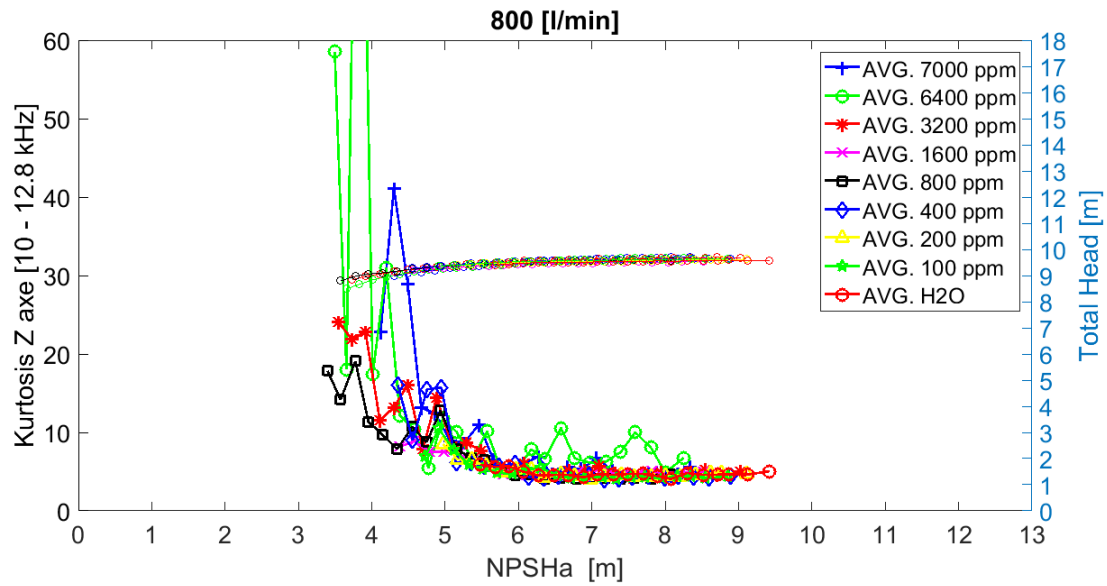


Figure 4.103: Averaged Kurtosis in Z axis in cavitation frequency range [10 - 12.8 kHz] at 800 l/min for all the concentrations

Figs. 4.104 , 4.105, 4.106 , 4.107 , 4.108 , 4.109 show the mean Kurtosis value curves calculated in all the frequency range as a function of NPSHa, obtained at all the considered values of mass flow rate. The same impulsive effect due to unstable fluid flows in the pump, shown in the previous case, is also present in the Kurtosis calculated in all the frequency range. Kurtosis does not present a clear trend among different concentrations of Polyox WSR 301.

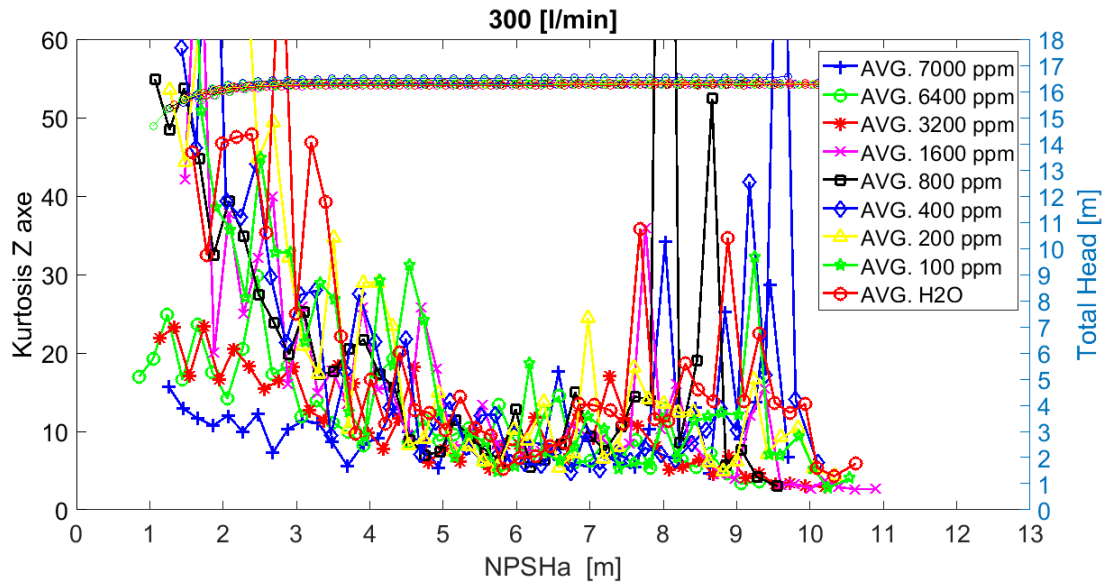


Figure 4.104: Averaged Kurtosis in Z axis in all frequency range at 300 l/min for all the concentrations

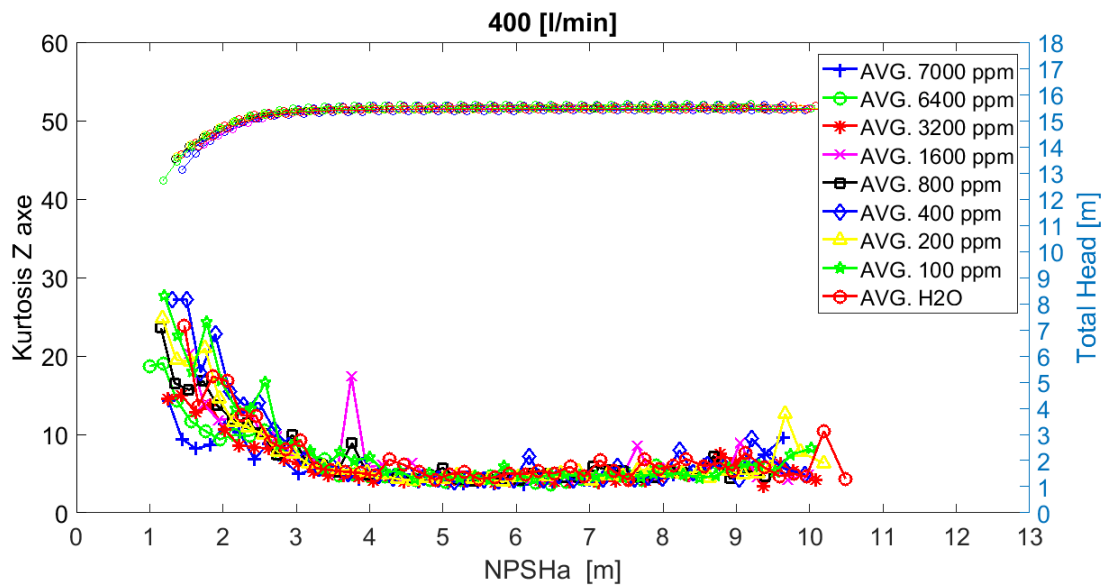


Figure 4.105: Averaged Kurtosis in Z axis in all frequency range at 400 l/min for all the concentrations

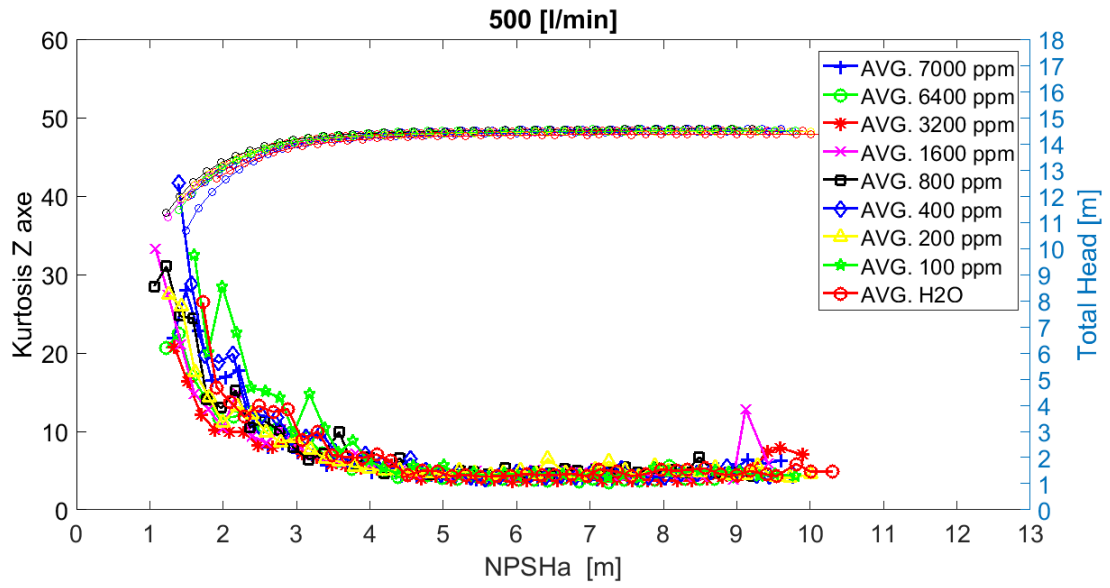


Figure 4.106: Averaged Kurtosis in Z axis in all frequency range at 500 l/min for all the concentrations

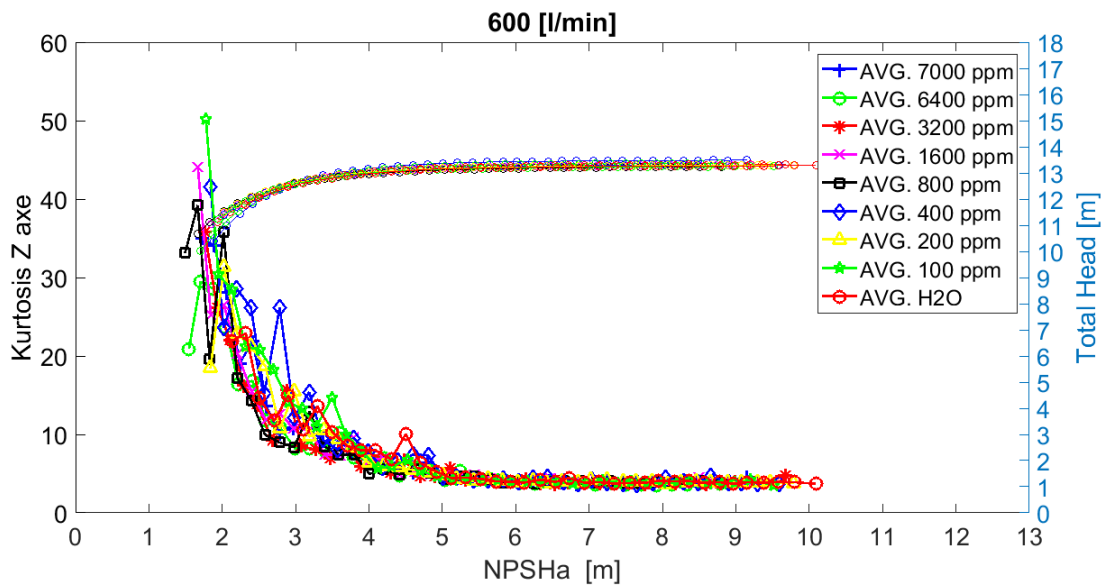


Figure 4.107: Averaged Kurtosis in Z axis in all frequency range at 600 l/min for all the concentrations

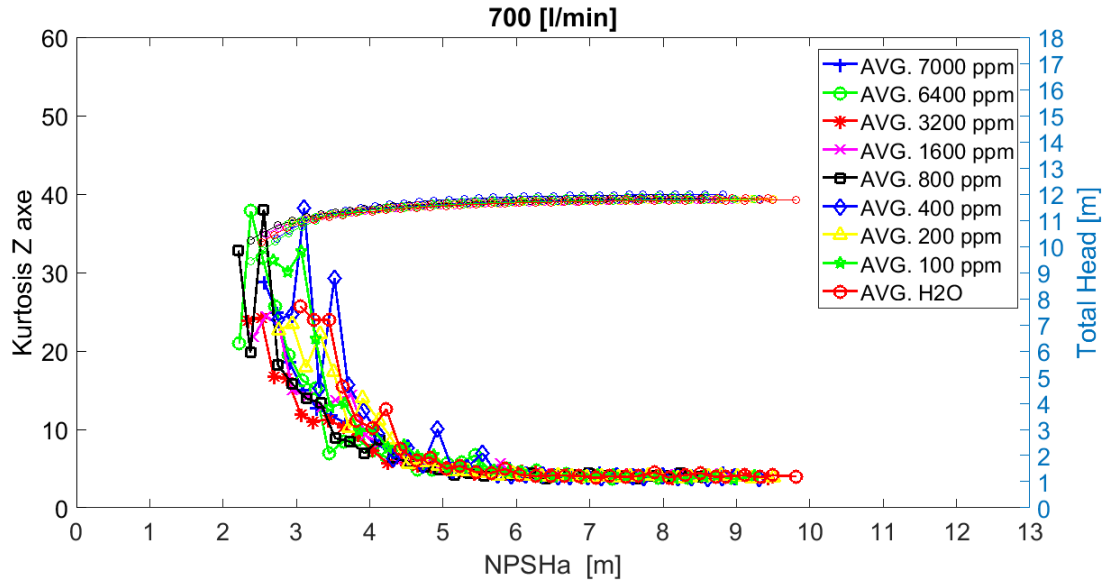


Figure 4.108: Averaged Kurtosis in Z axis in all frequency range at 700 l/min for all the concentrations

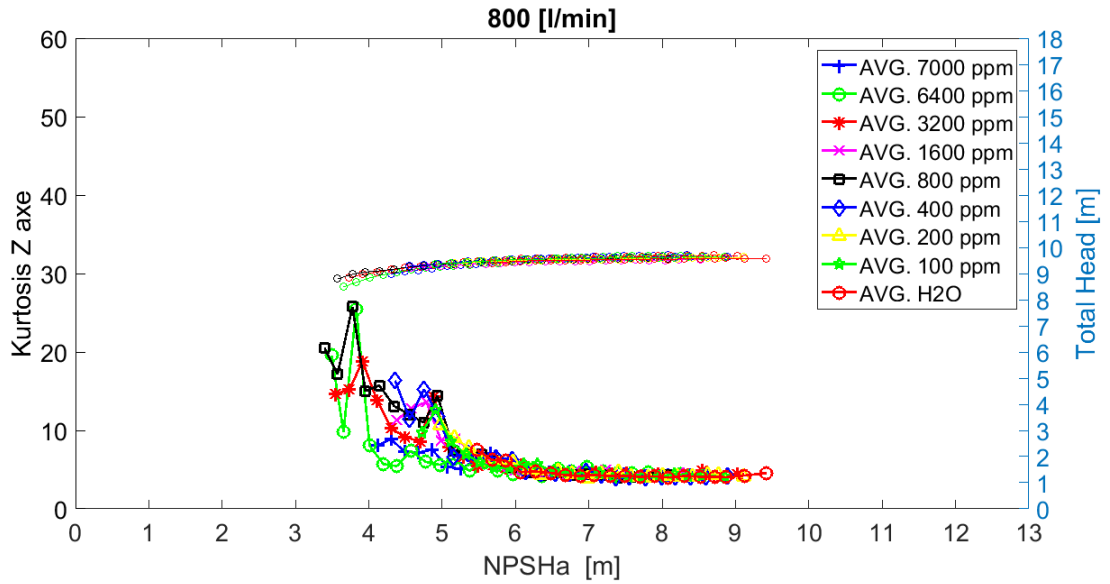


Figure 4.109: Averaged Kurtosis in Z axis in all frequency range at 800 l/min for all the concentrations

The Kurtosis of the signal in all frequency range is a good feature for developing a cavitation control system, in particular at high fluid flows (from 500 to 800 l/min), because it assumes an almost constant value, equal to 5.8, before cavitation inception. When cavitation appears, the Kurtosis of the signal starts to fluctuate and increases in its value, showing the characteristics impulsiveness due to the cavitation bubble collapses. Instead, at low fluid flows (in particular at 300 l/min) the Kurtosis

value is fluctuating in normal operating conditions, due to the high pressure fluctuations of the pump that generate spikiness vibrations.

Another parameter for cavitation analysis considered for a triaxial accelerometer, the total vibration energy is presented in [81]. We have modified the parameter considering only two accelerometer positions (the two radial directions of the volute) and analyzing the total energy with Eq. 4.53.

$$E = \sqrt{\frac{RMS_Y^2 + RMS_Z^2}{2}} \quad (4.53)$$

This parameter shows a clear ramp up when cavitation is fully developed. The results of such an analysis are presented from Fig. 4.110 to Fig. 4.115. Also the total vibration energy presents a clear trend of vibration levels at various polymer concentrations.

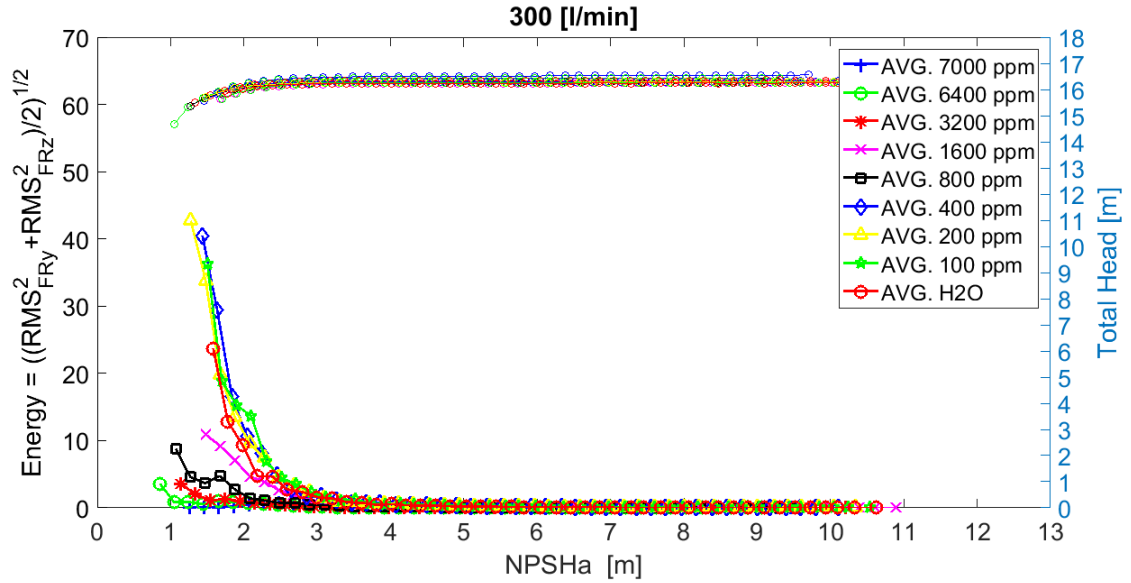


Figure 4.110: Averaged Energy in cavitation frequency range [10 - 12.8 kHz] at 300 l/min for all the concentrations

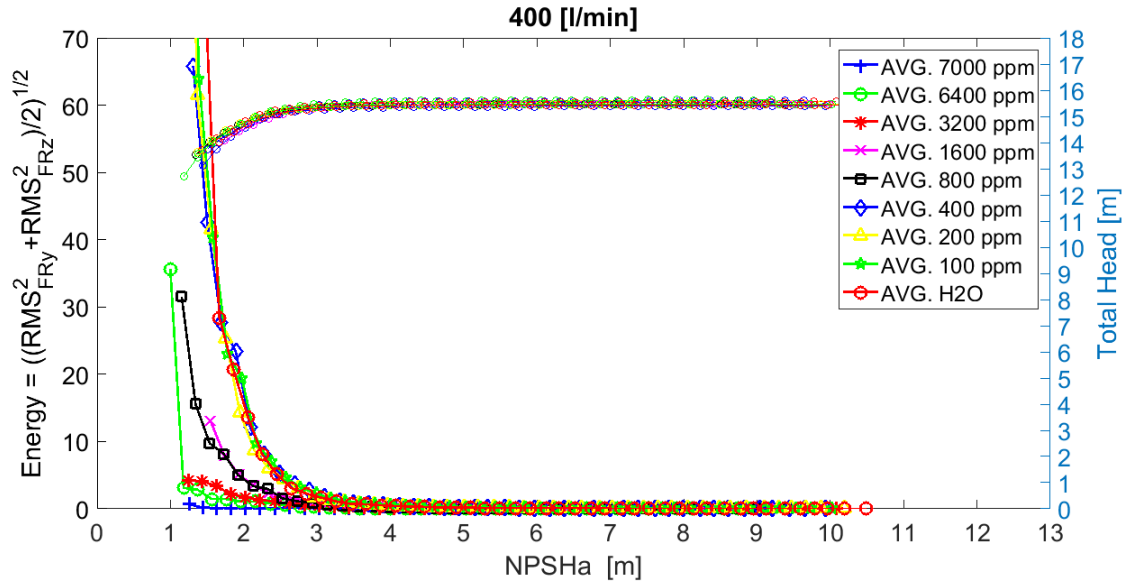


Figure 4.111: Averaged Energy in cavitation frequency range [10 - 12.8 kHz] at 400 l/min for all the concentrations

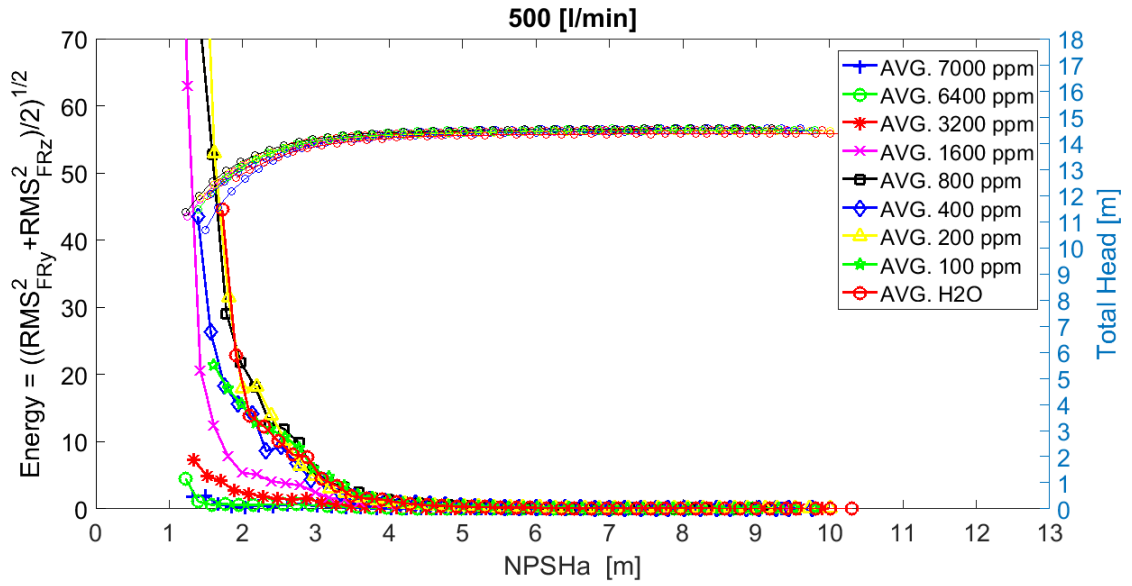


Figure 4.112: Averaged Energy in cavitation frequency range [10 - 12.8 kHz] at 500 l/min for all the concentrations

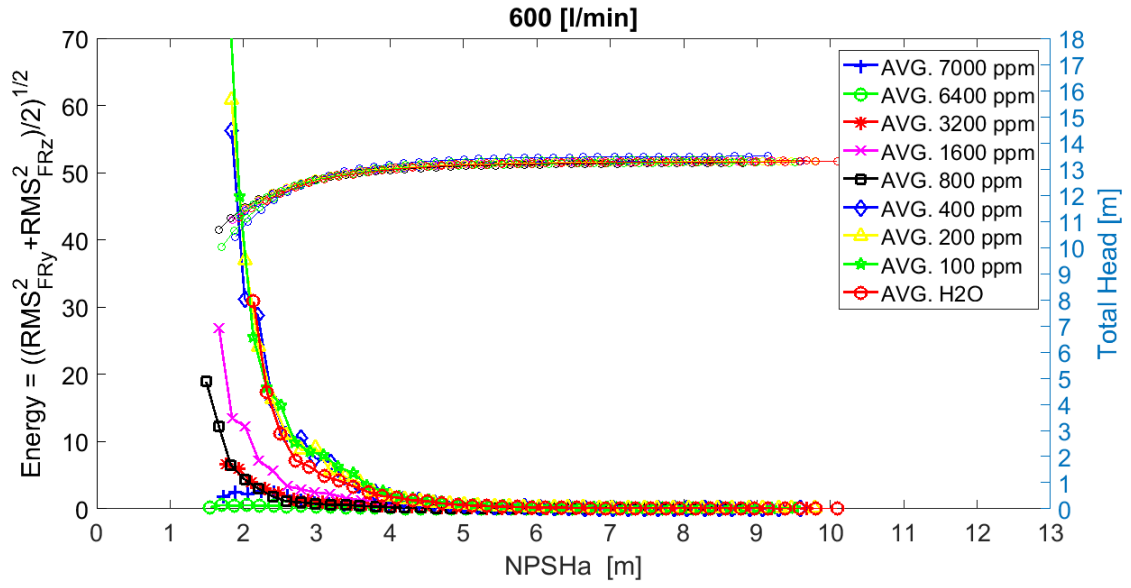


Figure 4.113: Averaged Energy in cavitation frequency range [10 - 12.8 kHz] at 600 l/min for all the concentrations

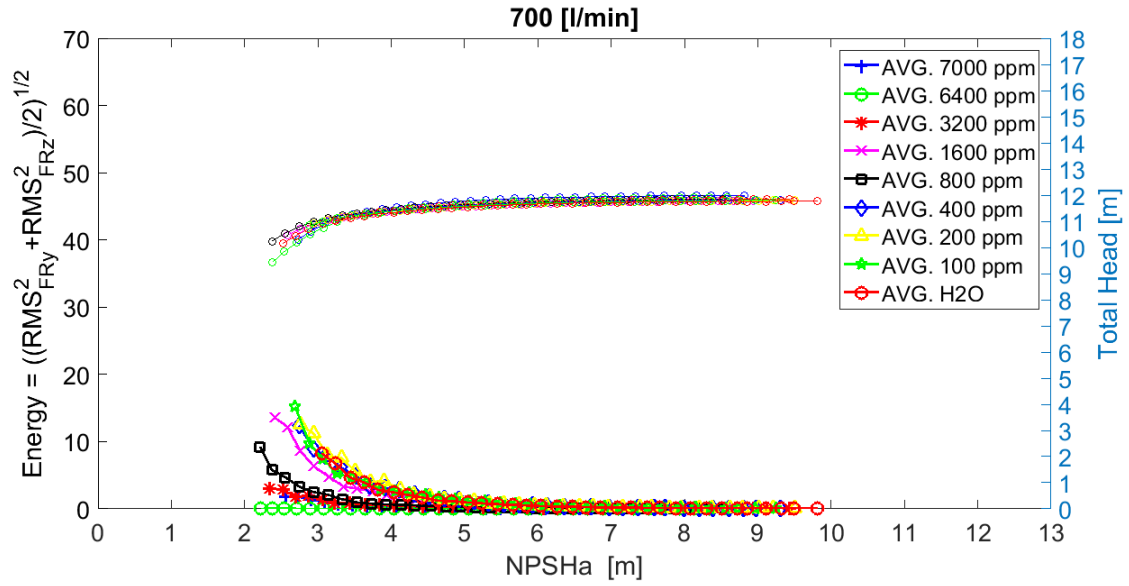


Figure 4.114: Averaged Energy in cavitation frequency range [10 - 12.8 kHz] at 700 l/min for all the concentrations

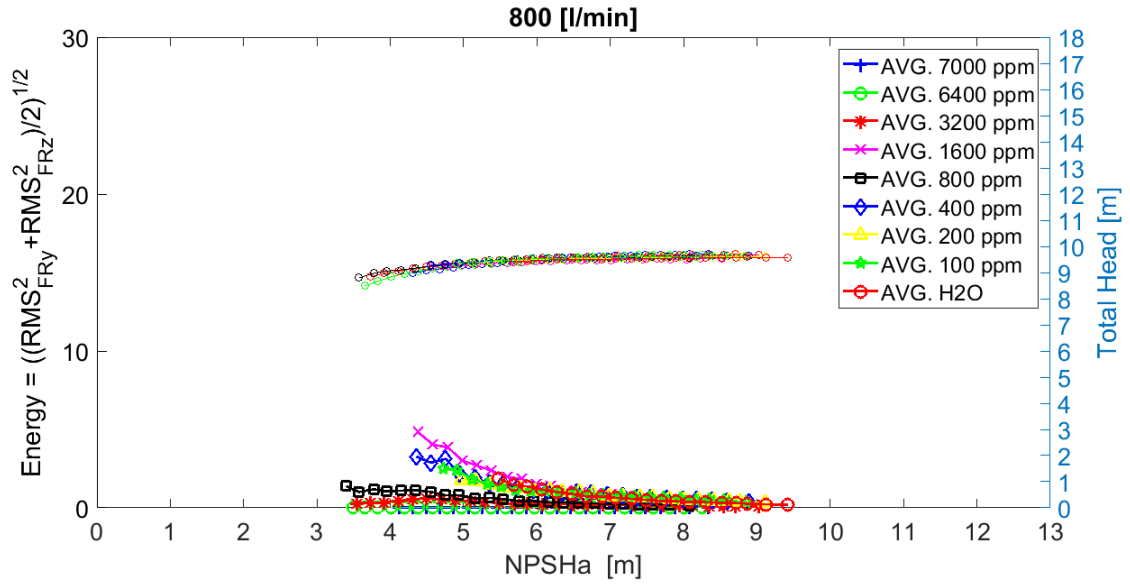


Figure 4.115: Averaged Energy in cavitation frequency range [10 - 12.8 kHz] at 800 l/min for all the concentrations

For having a better understanding of how the RMS values at different concentrations differ in non cavitation and developed cavitation conditions, only the values at 3 % total head decrease at all the considered flow rates are presented in Figs.4.116,4.117, for all the WSR 301 concentrations and with reference to the Y and Z axis respectively.

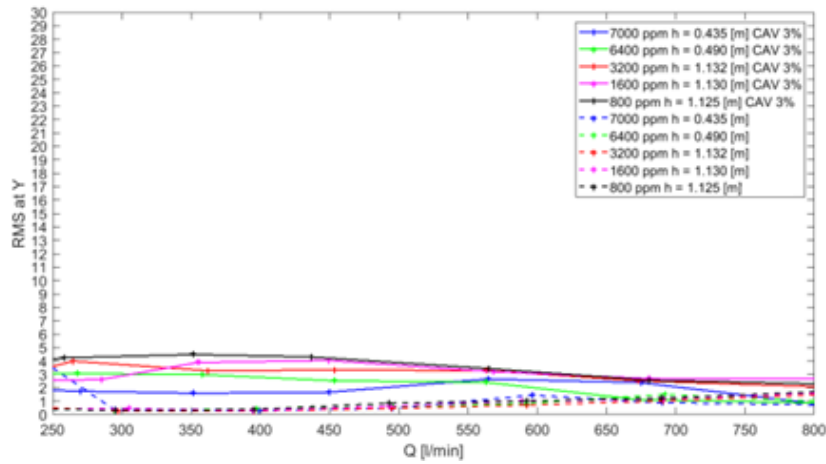


Figure 4.116: Evaluation of the RMS in non cavitation and developed cavitation in Y axis at different fluid flows and different concentrations of WSR 301

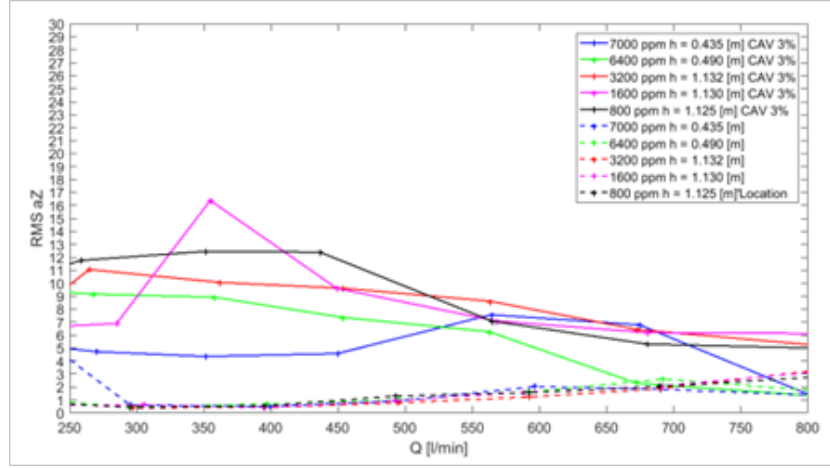


Figure 4.117: Evaluation of the RMS in non cavitation and developed cavitation in Z axis at different fluid flows and different concentrations of WSR 301

Figs. 4.116 ,4.117show a shift in the RMS values in cavitation regime among different concentrations, while in non cavitation regime the RMS at different concentrations assumes the same values. In particular, at central fluid flows, the RMS values in cavitation regime increases from low RMS values at high concentrations to higher values at low concentrations. It is also possible to state that in a range between 300 l/min to 600 l/min the family of curves with and without cavitation are well distinct such that it is possible to set a threshold value for a control system.

4.8.1 Concluding remarks

In the present chapter an investigation of the possible influences of detergent component solutions on pump performances and cavitation is presented. The first step in the analysis of the performance in terms of total head, efficiency and cavitation requirements with water-detergent solutions has been a series of tests using some diluted and one concentrated solution of a polymer, the Polyox WSR 301, representative of a class of substances that are usually found in the composition of detergents. The definition of diluted or concentrated solution has been based on preliminary rheological tests carried on in laminar conditions for a systematic series of samples of various polymer concentration. The results are as a whole in accordance with the not very numerous comparable data found in literature: at low concentration, the presence of polymers leads to a small improvement of efficiency, especially at high flow rates, and to a less evident increment of total head. This can be due to a reduction in some internal losses of the pump, as disk friction, that are substantially independent from the exchange of energy between machine and fluid in the blade channels. At higher concentrations of the solution, such advantages seem to reduce in terms of efficiency, instead total head increases, but, of course, other investigations are needed. The results obtained as far as the variation of cavitation

margin is concerned are, at the moment, quite doubtful, because to the knowledge of the authors, there are no comparative data available in literature.

Cavitation behavior has been also experimentally investigated by means of vibration measurements in order to characterize the effect of different concentrations of polymers solutions on vibration data, considering in particular the energy levels (RMS) and peak values at the cavitation frequency range. There is a good correlation among different concentrations of Polyox WSR 301 and the vibration levels at cavitation conditions, in particular increasing the Polyox WSR 301 concentration, there is a considerable reduction in the energy and peak values level of the pump. This phenomena could be explained with the results present in literature where is reported that a polymer solution suppresses inception and development of cavitation, changes the shape of the cavitation bubbles, decreases the energy of the cavitation shock pressures and reduces the nuclei population. The future steps of this analysis will be used to develop reliable control and monitoring strategies to ensure the safe operation of professional appliances.

4.9 Conclusions

At the beginning of this chapter is presented a first bibliographic analysis on the interactions among detergent components on pump performance in particular cavitation. It emerges that certain detergent components could affect cavitation due to the content of surfactants - polymers in the solution, which alters rheology, surface tension, viscosity and vapor pressure with respect to the case of pure water. Furthermore, in literature there are only few studies relating cavitation inception in the presence of liquid detergents. This result has pushed the need of designing and constructing a test rig according to the International Standard 9906:2012 that allows measuring the characteristic curves and NPSHr of different pumps, with various fluids (e.g. water - polymers solution and detergent at different concentrations). The pump mounted on the test rig has been instrumented also with three accelerometers covering all three axes along the pump volute and a microphone.

A theoretical explanation on the noise and vibrations in pumps with a list of the different noise generating mechanism has been also presented.

Furthermore, vibration and noise signal analysis has been classified according to three different domains: time domain, frequency domain and time frequency domain analysis. Vibration and noise signals have been analyzed with the different statistical approaches presented. In particular, a deep investigation on the frequencies involved in the cavitation inception and development has been presented. The first screening among different statistical parameters has resulted in a selection of a set of parameters and sensor positions that can be used for developing a sensor and an algorithm for monitoring cavitation inception in centrifugal pumps.

A general representative component present in detergents has been selected in order to investigate its effects on the pump performances and cavitation inception. The selected component is the Polyox WSR 301. The rheology of water - Polyox WSR 301 solutions has been evaluated in order to define at which concentration the solution goes from the diluted regime to the concentrated regime. Afterwards, the effect of different concentrations of Polyox WSR 301 on pump performances has been presented.

The results of the experimental analysis shows that at low concentration, the presence of polymers leads to a relatively small improvement of efficiency, especially at high flow rates, and to a slightly less evident increment of total head. At higher concentrations of the solution, such advantages seem to reduce in terms of efficiency, instead total head increases also of the four percent at zero flow rate in concentrated regime.

Cavitation behavior has been also experimentally investigated by means of vibration measurements in order to characterize the effect of different concentrations of polymers solutions on pump vibration. For performing the analysis, the energy

levels (RMS) and peak values at the cavitation frequency range between 10 and 12.8 kHz has been considered. There is a clear correlation among different concentrations of Polyox WSR 301 and the vibration levels at cavitation conditions, in particular as the concentration of Polyox WSR 301 increases, the RMS and peak levels decrease.

This phenomena has been explained with a literature analysis presented at the end of the chapter. In literature is reported that a polymer solution suppresses inception and development of cavitation, changes the shape of the cavitation bubbles, decreases the energy of the cavitation shock pressures and reduces the nuclei population.

All this analysis will be used to develop reliable control and monitoring strategies to ensure safe operations against cavitation in professional appliances.

Conclusions

In the field of developing energy efficiency appliances, in order to reduce energy consumption, an analysis on different test procedures has been made for the qualification of energy efficiency of professional ovens. The standards review conducted in Chapter 1 of the present thesis revealed that the energy efficiency yields calculated with the test procedures EFCM, ENAK and ASTM show discrepancies. A methodology based on the first principle of thermodynamics has been developed and it is based on balance of fluxes entering and going out from the oven control volume in different cooking modes as the convection mode and steam mode. In this thesis particular attention has been reserved for two of these technical solutions most influencing the energetic performances of the oven, i.e. the thermal insulation and the oven washing system presented respectively in Chapter 2 and Chapter 3.

In order to study the thermal insulation of the oven, an appropriate design strategy has been developed a priori and then experimentally validated on a prototype. A professional oven during the cooking process goes through a sequence of unsteady phases (cavity heating-up, food introduction and extraction, switching from one cooking mode to another) interspersed with steady cooking phases. Therefore, for analyzing the thermodynamic behavior, a special theoretical dynamic model has been developed and presented in Chapter 2. The model is based on the lumped capacitance method and is capable of predicting the thermodynamic performances with low computational efforts compared to other numerical techniques (i.e. CFD). The oven, physically presents two distinct zones: the power zone and the cooking cavity. The numerical model can predict the thermodynamic behavior of both the zones, permitting an evaluation of the overall energy performances of the device. All heat transfer mechanisms are considered and simulated. An estimation of the convection parameters have been required with a tuning procedure to find their averaged values. The contribution of radiation heat transfer has been considered with the net radiation method for encloses. The results have been compared with a set of experimental data showing a good agreement in both the transitory and the stationary operating phases. The future activities for this model will be a comparison with more experimental data at different cooking modes, an optimization study to find correlations

of the tuning coefficients according to different operating conditions, introduction of the hygrometric balance in a combined oven (presence of steam produced by a dedicated boiler) and the simulation of the food thermodynamic behavior inside the oven.

A dedicated analysis for developing an efficient oven washing system in terms of water, energy and detergent consumption has been developed and presented in Chapter 3. The first comparison among closed and open washing systems shows that the closed solutions, selected as benchmarks, present lower water, energy and detergent consumption. A first numerical model of the oven washing system has been developed in order to detect the operating point of the pump and for calibrating the pressure drop orifices in the oven washing plant. The results have been compared with experimental measures on an oven prototype washing plant. A second numerical model (CFD) has been developed in order to analyze a new possible design for the suction duct of the oven washing plant. The analysis has been carried out with the aim of comparing the air mass flow rate of the suction duct with and without the pipes for water introduction and solution recirculation. It emerges that the pipes influence very lowly the air mass flow rate of the duct. During the development of the oven washing plant, an investigation of the pump operating conditions shows possible risks of cavitation inception for the pump.

The complex environment to which a pump is subjected due to the presence of food residues and in particular detergents, has pushed the need of studying deeply the behavior of a pump, specially in cavitation conditions. In literature, the cavitation phenomena in Newtonian fluids are well known, particularly as far as pure water is concerned, but there are not clear studies on the effect of detergents. The wide range of variables affecting the phenomenon has led to the development and construction of a test rig for testing centrifugal pumps with aqueous solutions representative of those used in the professional appliances sector. A component, Polyox WSR 301, has been selected as representative of a class of substances that are usually found in the composition of detergents. Preliminary rheological tests have been carried out in laminar conditions on a systematic series of samples of various polymer (Polyox WSR 301) concentrations dissolved in water. Furthermore, measurements of the pump characteristic curves, vibration and noise levels have been conducted on the test rig and reported in Chapter 4. They have allowed to find that polymers solutions affect pump performances and reduce the vibrations induced by cavitation.

In conclusion, this thesis would contribute to develop a comprehensive methodology for the improvement of energy efficiency in professional appliances. In particular, the research on pump cavitation will be continued with the development of a control

system capable of detecting and avoiding cavitation in centrifugal pumps used in energy optimized closed washing systems.

Nomenclature

Chapter 1 Nomenclature

$\dot{m}$	mass flow [Kg/s]
A	area [m ²]
B	position of
c_k	specific heat, k stands for individual food component (i), load (load) or tray (tray)[W/m ² °C]
E_k	energy, k stands for energy lost through the opening of the door (door), steam exiting (vent), wall (wall), drain of liquids (liq) or for energy adsorbed by the Gastronorm trays and lids (GN) (Lid), auxiliary (aux) or energy introduced from the water entering (w), electricity (el) or energy adsorbed by the load (load) [kJ]
h	specific enthalpy [kJ/kg]
m_k	mass, k stands for trays, water inlet, loss or gain in the load, liquid discharge and condensed [kg]
m_{cv}	mass inside the control volume [Kg]
m_{load}	mass at the end of the cooking process [kg]
Q_{cv}	net amount of heat supplied to the control volume [kJ]
U_{cv}	internal energy [kJ]
W_{cv}	net amount of energy transferred as work [kJ]
x_i	mass fraction of the food component i [kg/kg]
ΔT_{load}	temperature difference reached by the load [°C]
ΔT_{tray}	temperature difference reached by trays [°C]
η_i	efficiency [%]

λ latent heat of evaporation [kJ/kg]

Chapter 2 Nomenclature

$\bar{Pr}$ averaged Prandtl number

$\dot{m}_{fan}$ fan mass flow [kg/s]

A area [m²]

C thermal capacity [J/K]

c specific heat capacity [J/kgK]

D heater tube characteristic length [m]

F view factor

h convective coefficient [W/m²K]

L wall characteristic length [m]

N number of walls

Nu Nusselt number

P_{el} electric power [W]

P_{rad} radiated power [W]

Pr Prandtl number

R thermal resistance [Km²/W]

R_g overall glass thermal resistance [Km²/W]

Re Reynolds number

T temperature [K]

t time [s]

V volume [m³]

δ_{ij} Kronecker Delta

Δx spatial step [m]

ϵ radiation emissivity

λ thermal conductivity [W/mK]

ρ	density [kg/m ³]
σ	Boltzmann constant [J/m ² sK ⁴]

Chapter 2 Subscripts

C	cooking zone
e	external ambient
g	glass
$g1$	internal glass
$g2$	external glass
mC	coocking zone thermal masses
mP	power zone thermal masses
n	timestep
P	power zone
$R1$	resistor external node
$R2$	resistor internal node
rad	radiation
s	wall superficial internal node
w_i	i-th wall
w_i1	external superficial node
w_j	j-th wall
$cond$	conductive
$conv$	convective

Chapter 3 Nomenclature

$\dot{Q}_e$	heat exchanged with the enviroment
λ	Darcy friction factor
ρ	mean density[kg/m ³]
A	area[m ²]

d	pipe diameter [m]
F_F	Fanning friction factor
g	gravity constant [m/s ²]
l	pipe length [m]
p	mean pressure [Pa]
T	mean temperature [°C]
t	time[s]
u	specific internal energy[kJ/kg]
v	mean velocity [m/s]
x	independent spatial coordinate along the fluid direction
z	height over ground [m]
Δp	pipe pressure difference due to static head [Pa]
ζ	loss factor

Chapter 4 Nomenclature

$\alpha(t)$	random variable
$\dot{\gamma}$	shear rate [s ⁻¹]
η	viscosity [Pa s]
$\frac{k}{N}$	frequency of the synusoidal wave (cycles per time unit) where 0<k<N
μ_α	mean
ρ	density [Kg/m ³]
σ	shear stress [Pa]
σ_α	variance
τ	time lag
BEP	Best Efficiency Point
$BPF = f_p$	Blade Passing Frequency
C	relative increment of NPSHr with c [%]

c	concentration [ppm]
c^*	overlap concentration [ppm]
$C_{\alpha\beta}(t_1, t_2)$	cross co variance function
CF	crest factor
dt	time interval
$dX(f)$	orthogonal spectral process of unit variance
E	relative increment of efficiency with c [%]
$E[]$	expected value
$E[(\alpha - \mu_\alpha)^k] = M_k$	generalized form of the moment of the probability density function
$E[\alpha]$	first statistical moment of the probability density function
$E[\alpha^2]$	second statistical moment of the probability density function
$E[\alpha^k] = M'_k$	generalized form of the moment of the probability density function
$e^{-j(\frac{2\pi}{N})nk}$	Euler's formula $e^{-j(\frac{2\pi}{N})nk} = \cos((\frac{2\pi}{N})nk) + j \sin((\frac{2\pi}{N})nk)$
f	frequency
f_i	probability of occurrence of a variable
g	gravitational acceleration 9.807[m/s ²]
H	relative increment of H_T with c [%]
$h(t)$	window centered at the time t
$H(t, f)$	time varying transfer function of the process Y(t)
$H(t, f, w)$	complex envelope in terms of the outcome coming out from the random variable w
$H(\alpha)$	information entropy
H_T	total head [m]
I	current [A]
$K_Y(f)$	spectral kurtosis
$Kurt$	kurtosis

M	total number of spectra
m_k	moles, k stands for water (H ₂ O), potassium caprate (PC)
MHP	Maximum Head Point
MM_{PC}	molar mass of potassium caprate (PC)
N	Time unit
N	total number of experiments
n	armonic considered
n_i	occurred times of the variable
N_w	length of the analysis window
Ni	number of symbols
$NPSH_k$	Net Positive Suction Head, k stands for available (a); required (r)
P	relative increment of power with c [%]
P	time step in the STFT analysis
p	pressure [bar]
$P(\alpha)$	probability distribution function
$p(\alpha)$	probability density function
p_i	probability of a symbol
p_k	pressure, k stands for undisturbed liquid at distance of the object (0), critical (c), vapour (v)
p_s	pressure at pump suction [bar]
$P_S(t, w)$	Spectrogram or energy density spectrum
p_v	vapour pressure [bar]
p_{l0}	ambient or mean liquid pressure
p_{tk}	threshold pressure amplitude, k stands for oscillation (v), for a nucleus to grow by rectified diffusion (d)
$Peak$	peak level
PSD	Power Spectral Density

Q	fluid flow [m ³ /s]
$R_{\alpha\alpha}$	autocorrelation
RF	rotational frequency
rpm	rotational velocity [rpm]
S	surface tension
$S_t(t, w)$	Short Time Fourier Trasform
$S_{2nY}(f)$	spectral statistics for the second order moment
$S_{4Y}(f)$	spectral statistics for the fourth order moment
$S_{\alpha\alpha}$	power spectral density function of a process (foward Fourier transform of the autocorrelation function)
SE	Spectral Entropy
Skw	skwness
t	time
TE	total energy
u_d	fluid velocity delivery [m/s]
u_s	fluid velocity suction [m/s]
V	tension [V]
Var	variance:second moment about the mean
w	angular speed $w=2 \pi f$
$w(n)$	analysis window with a length N_w
w_{PC}	weight of potassium caprate (PC)
$X(i)$	Fourier Trasform of the signal
$x(n)$	n-th complex number
x_a	molar fraction of the solvent
X_k	independent random variable with its own probability distribution ($k = 1,..N$)
x_n	frequency amplitude
$Y(t)$	non-stationary process

$Y_w(kP, f)$	STFT for a process $Y(n)$
Z	number of pump events per revolution
α_{RMS}	root mean square
Δp	pressure difference (delivery-aspiration) [bar]
Δt	time width
Δw	bandwidth
Δz	height difference (delivery-aspiration) [m]
η_0	viscosity polymer solution [Pa s]
η_s	viscosity pure solvent [Pa s]
$\eta_{(sp,0)}$	specific viscosity
μ_n	mean value of the amplitude over all the spectra
ρ	Density of the liquid
σ_i	Incipient cavitation number
C_k	concentration of gas, k stands for actual (∞), <i>saturation</i> (0)
p_A^k	Vapour tension, k stands for partial of the solvent (p), pure solvent (ps)
R_k	radius, k stands for bubble (b), initial bubble (n)
v_0	relative velocity between immersed object and surrounding liquid

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