ABSTRACT

STOFOROS, GEORGE NIKOLAOS. Acoustic Enhancement of Continuous Flow Cooling. (Under the direction of Dr. Brian E. Farkas.)

Advanced heating technologies, continuous flow microwave (MW) systems, and ohmic and radiofrequency heating technologies have enabled food processors to maximize product quality during continuous thermal processing of highly viscous and multiphase food products. During the cooling cycle of thermal processes, laminar flow and low conductivity of highly viscous foods lead to a wide temperature distribution within the product, resulting in unequal cooling and product quality degradation. The hypothesis of this study was that in a cooling system consisting of a series of coolers, minimization of product temperature distribution at the entrance of each cooler could enhance cooling by improving radial mixing.

Numerical simulation, using the mathematical solution of Graetz problem and its computer implementation in a code written on Maple 16, was used to calculate the temperature distribution and the bulk product temperature of highly viscous foods, such as applesauce during cooling, and the influence of temperature homogenization as a method of enhancement of total cooling process. Results of numerical simulation showed a significant improvement of 61.4% on total cooling time with the application of thermal mixing during applesauce cooling.

Experimentally, a complete continuous flow thermal system, consisting of a continuous flow microwave (MW) system (formed from a series of home MW ovens), a cooler, and a mixing unit (consisting of a 180° bend pipe and an acoustic vibrator) was built to study thermal mixing of highly viscous foods. Acoustics were used to impose transversal vibration motion on a 180° bend pipe placed right after the cooler and to generate chaotic
fluid motion on various highly viscous foods, thus achieving thermal mixing. The effects of different acoustic frequencies (0-100 Hz), amplitudes, and protocols of vibration on radial temperature distribution during cooling of highly viscous foods, such as sweet potato puree, banana puree, applesauce, and canned and aseptically processed cheese sauce, were studied. Through application of continuous vibration at the maximum level of volume of amplifier and at 20 Hz, the resonance frequency of vibration of the mixing unit resulted in a uniform radial temperature distribution (with a temperature difference of 0-4 °C between the product at the center of the pipe and the product close to the wall) for all the products except canned processed cheese sauce, which presented a temperature difference of 6-8 °C.

Shear-thinning flow behavior, non-slip wall boundary conditions, and the magnitude of the initial temperature difference between the product temperature at center and close to the wall, before the application of vibration, effected the final uniformity of temperature for all the products. Initial temperature difference, and therefore the degree of viscosity ratio inside the product, was the most important factor influencing the uniformity of temperature at the exit of mixing unit. A critical value of 12-15 °C was found; above this value, a rather wide temperature distribution (5-12 °C) remained for all the tested products.

The results from this study proposed an efficient method of thermal mixing for enhancement of continuous flow cooling and retaining the maximum integrity of food particles. Transversal acoustic vibration proved to be an efficient mechanism for homogenizing the temperature profile and enhancing the cooling process of food products; however, rheological parameters of food materials, such as shear-thinning flow behavior, non-slip wall boundary conditions, and the degree of viscosity ratio inside the product, need to be addressed for industrial application of this method.
© Copyright 2014 George Nikolaos Stoferos

All Rights Reserved
Acoustic Enhancement of Continuous Flow Cooling

by
George N. Stoforos

A thesis submitted to the Graduate Faculty of
North Carolina State University
in partial fulfillment of the
requirements for the degree of
Master of Science

Food Science

Raleigh, North Carolina

2014

APPROVED BY:

Dr. Josip Simunovic
Dr. K.P. Sandeep

Dr. Brian E. Farkas
(Chair of Advisory Committee)
DEDICATION

To my family.
George Stoforos was born on February 27, 1986 in Sacramento, CA, and he grew up in Lamia, Greece. He graduated from the School of Chemical Engineering of the National Technical University of Athens (NTUA), in November of 2008 (a five-year program). Under the supervision of Dr. Taoukis, his thesis for the program focuses on Kinetic studies of the effect of high hydrostatic pressure and temperature on the pectinmethylesterase and polyphenoloxidase of strawberry juice. From February of 2009 until January of 2010, he served in the Greek Army. He then worked for one and a half years as a coach for a swimming team at his home town of Lamia (a hobby aroused by his career of over fifteen years as a nationally competitive swimmer and as a five-year member of the national team). From October 2011 until April 2012, he worked as an intern on the Laboratory of Food Chemistry and Technology of NTUA, where he studied the effect of high hydrostatic pressure and temperature on the destruction kinetics and structural changes of fruit enzymes. During the summer of 2012, he worked in the United States as an intern at Del's Lemonade & Refreshments, Inc., in Cranston, RI. In August 2012, he began his Master of Science degree program in the Department of Food Science at North Carolina State University under the supervision of Dr. B.E. Farkas and Dr. J. Simunovic, during which he also worked as a graduate research assistant. His research focuses on the acoustic enhancement of continuous flow cooling of multiphase food products.
ACKNOWLEDGMENTS

First of all, I would like to take this opportunity to thank my major advisors, Dr. Brian E. Farkas and Dr. Josip Simunovic, for their guidance, motivation, support, and encouragement throughout this work.

Special thanks to Michael Bumgardner for his suggestions and assistance with the experiment setup. He was always available for timely help, which is greatly appreciated.

I would like to acknowledge the Center for Advanced Processing and Packaging Studies (CAPPS) for their financial support.

Finally, thanks to all former and current students of the department who have helped me in this work: Ryan Dowdy, Andrew Kaufmann, Mehmet Kemal, An Truong, and Nihat Yavuz.
# TABLE OF CONTENTS

LIST OF TABLES .......................................................................................... x

LIST OF FIGURES ....................................................................................... xi

CHAPTER 1: Introduction ............................................................................. 1

CHAPTER 2: Review of literature .................................................................. 6

2.1 Introduction ............................................................................................. 6

2.1.1 Mechanism of mixing .......................................................................... 6

2.1.1.1 Reynolds and Péclet number ......................................................... 7

2.1.1.2 Mixing of viscous fluids ............................................................... 8

2.1.2 Kinematics of fluid mixing ................................................................. 10

2.1.3 Chaotic advection ............................................................................... 12

2.1.3.1 Major points and tools for chaotic flow study ......................... 13

2.1.3.2 Mapping of fluid particles trajectories ..................................... 18

2.1.4 Studies on chaotic advection ............................................................... 19

2.1.4.1 Studies on chaotic advection in 2-D time-periodic flows .......... 19

2.1.4.2 Studies on chaotic advection in 3-D flows ................................. 21

2.1.4.3 Studies on chaotic advection in 3-D open duct flows ............ 23

2.1.5 Chaotic advection and heat transfer ............................................... 47

2.1.6 Chaotic flow produced by pipe vibration on continuous flows .......... 53

2.1.6.1 Flow enhancement of viscous fluids with pipe vibration .......... 54

2.1.6.2 Vibration effects on radial mixing; temperature uniformity ...... 56
2.2 Acoustics ............................................................................................................ 57
  2.2.1 Acoustic wave .............................................................................................. 57
  2.2.2 Acoustic vibration ....................................................................................... 58
  2.2.3 Acoustic cavitation ..................................................................................... 59
    2.2.3.1 Resonance frequency for cavitation .................................................... 61
    2.2.3.2 Jet formation and shock wave from acoustic cavitation ................. 62
  2.2.4 Acoustic wave and boundary layer ............................................................ 64
    2.2.4.1 Acoustic streaming and turbulence flow ........................................... 64
    2.2.4.2 Acoustic cavitation and turbulence flow ........................................... 65
  2.2.5 Heat transfer enhancement by acoustic streaming and acoustic
  cavitation .............................................................................................................. 66
    2.2.5.1 Heat transfer enhancement by ultrasound vibration ....................... 67
    2.2.5.2 Heat transfer enhancement by ultrasound in heat exchangers ... 70
  2.2.6 Ultrasound in viscous products ................................................................. 76
  2.2.7 Ultrasound in food processing ................................................................. 77
    2.2.7.1 Non-thermal and thermal treatment with ultrasound ................. 78
    2.2.7.2 Quality parameters of food products after ultrasound ............... 79
  2.2.8 Acoustic reactors ....................................................................................... 81
    2.2.8.1 Heat transfer enhancement using a sonitube-tube heat
    exchanger ......................................................................................................... 90
  2.3 Conclusions .................................................................................................. 91
Nomenclature ....................................................................................................... 93
CHAPTER 3: Mathematical solution and its computer implementation for temperature predictions during continuous flow cooling of non-Newtonian products...

3.1 Introduction ........................................................................................................................................ 111

3.2 Mathematical solution .......................................................................................................................... 113

3.2.1 Graetz problem for non-Newtonian fluids with Dirichlet boundary conditions .......................................................... 115

3.2.2 Solution of Graetz problem for non-Newtonian fluids ................................................................. 119

3.2.3 Temperature distribution solution ............................................................................................... 123

3.2.4 Bulk temperature at the end of cooling unit .................................................................................... 124

3.3 Compute modeling on Maple of temperature distribution solution and bulk temperature at the exit of cooling unit ........................................................................................................................................ 124

3.3.1 Comparison of computer model results with values from literature........................................... 126

3.3.2 Comparison of computer model results with experimental results ............................................. 127

3.3.3 Numerical simulations of thermal mixing effects in enhancement of continuous flow cooling ........................................................................................................................................ 129

3.4 Conclusions ......................................................................................................................................... 131

NOMENCLATURE .................................................................................................................................... 132

REFERENCES .......................................................................................................................................... 134

CHAPTER 4: Thermal mixing via acoustic vibration during continuous flow cooling of sweet potato puree ................................................................................................................................. 136

4.1 Introduction ......................................................................................................................................... 137
4.2 Materials and methods .................................................. 141

4.2.1 Tested Materials .................................................. 143

4.2.2 Experimental set up .................................................. 143

4.2.3 Experimental methods ............................................. 152

4.3 Results and discussion ................................................ 156

4.3.1 Time-temperature results during heating ..................... 156

4.3.2 Vibration effects on temperature distribution ............... 157

4.4 Conclusions ............................................................ 188

Acknowledgments .......................................................... 189

NOMENCLATURE ........................................................... 190

REFERENCES ............................................................... 192

CHAPTER 5: Acoustic enhancement of continuous flow cooling of viscous and
multiphase food products .................................................. 196

5.1 Introduction ........................................................... 197

5.2 Materials and methods ............................................... 199

5.2.1 Tested Materials .................................................. 199

5.2.2 Experimental set up ................................................ 199

5.2.3 Experimental methods ............................................. 200

5.3 Results ................................................................. 203

5.3.1 Banana puree ....................................................... 203

5.3.2 Applesauce .......................................................... 214

5.3.3 Cheese sauce (canned processed) ............................. 224
5.3.4 Cheese sauce (aseptically processed) .................................................. 234
5.3.5 Sweet potato puree ............................................................................. 247
5.4 Conclusions ......................................................................................... 256
Acknowledgments .................................................................................... 258
NOMENCLATURE ..................................................................................... 259
REFERENCES ............................................................................................ 260
CHAPTER 6: Conclusions .......................................................................... 263
CHAPTER 7: Recommendations for future work ....................................... 268
REFERENCES ............................................................................................ 270
LIST OF TABLES

Chapter 3: Mathematical solution and its computer implementation for temperature predictions during continuous flow cooling of non-Newtonian products

Table 3.1: Comparison of non-dimensional bulk temperature $\theta_b$ values from this work with these from literature; for different values of flow behavior index, $n$, and different values non-dimension distance variable $z$. .................. 127

Chapter 5: Acoustic enhancement of continuous flow cooling of viscous and multiphase food products

Table 5.1: Flow consistency index (K) and flow behavior index (n) values for banana puree at different temperatures calculated through Eq. 5.1. ......................... 212

Table 5.2: Flow consistency index and flow behavior index values for applesauce at different temperatures calculated through Eq. 5.1. .............................. 223

Table 5.3: Flow consistency index and flow behavior index values for canned processed cheese sauce at different temperatures as calculated through Eq. 5.1. ........................................................................................................... 233

Table 5.4: Flow consistency index and flow behavior index values for aseptically processed cheese sauce at different temperatures as calculated through Eq. 5.1. ........................................................................................................... 246

Table 5.5: Flow consistency index and flow behavior index values for sweet potato puree at different temperatures calculated through Eq. 5.1......................... 255
LIST OF FIGURES

CHAPTER 2: Review of literature

Figure 2.1. Relation of Peclet number in low Reynolds number laminar flow (Nguyen and Wu, 2005)........................................................................................................................................ 9

Figure 2.2. Periodic stretching and folding; exponential increase of interfacial area (Cullen, 2009).......................................................................................................................... 14

Figure 2.3. Chaotic region close to a hyperbolic point (Leong and Ottino, 1989). ........... 17

Figure 2.4. (a) Picture of hyperbolic point with a transverse homoclinic intersection and an elliptic point at the center of an island, (b) the hyperbolic point moves periodically in a closed orbit, and its stable and unstable manifolds create a transverse homoclinic point (intersection), (c) chaotic region close to an a hyperbolic point (Ottino, 1990).......................................................................................................................... 17

Figure 2.5. (a) Experimental results of mixing of India ink in sugar syrup using a stirring rod, (b) Pionceré section showing the rod's trajectory, with the islands located close to the fixed wall of agitator (Thiffeault et al. 2011).... 19

Figure 2.6. Coordinates of experimental aperture of Fountain et al. (2000) experiment in a batch tank 3-D flow........................................................................................................... 22

Figure 2.7. Typical structure of helical static mixers; Periodic reorientation of fluid elements results on enhancement of radial mixing (Stamixco-mex, 2014). .. 24

Figure 2.8. Different types of static mixer design options. From left: vortex mixer (type KVM), corrugated plate (type SMV), wall-mounted vanes (type SMF),
cross-bar (type SMX), helical-twist (type KHT), cross-bar (type SMXL) (Etchells III and Meyer, 2004).  

Figure 2.9. Centrifugal forces create Dean roll-cells, causing periodic reorientation and mixing to main axial flow (Changy et al., 2000).  

Figure 2.10. Schematic configuration model of PPM (Khakhar et al., 1987).  

Figure 2.11. Schematic configuration of EHAM, with the two eccentric cylinders rotating with reverse cyclic direction (Mota et al., 2008).  

Figure 2.12. Schematic experimental equipment for Kusch and Ottino (1992) experiments.  

Figure 2.13. Experimental apparatus of EHAM and PPM. The inner cylinder for EHAM was a stainless steel rod with a diameter of 2.93 cm and length of 135 cm; for PPM the inside cross sectional obstacles were made of thin Pyrex plates (0.32 cm x 5.08 cm x 8.26 cm) (Kusch and Ottino, 1992).  

Figure 2.14. (a) Schematic demonstration of experimental mixing with micro-mixer working with time periodic pressure perturbation, (b) Pressure driven sinusoidal perturbation mixer with spatial periodic mixing from different channels (Niu and Lee, 2003).  

Figure 2.15. Schematic showing of experimental mixing with electrokineomatical forces perturbation created from electrical field (Lee et al., 2001).  

Figure 2.16. Rotated Arc Mixer (RAM) (Metcalf et al., 2006).  

Figure 2.17. Chaotic mixing of viscous glycerine-water mixture, with RAM (Metcalf and Lester, 2009).
Figure 2.18. Cross-sections surface view and geometric parameters of inside cylinder of RAM. Where \( R \) is the inner radius of the inner stationary cylinder, \( H \) is the window’s axial length, \( \Delta \) is the angle of the window opening, and \( \Theta \) is the angle by which each window is offset from its downstream neighbour (Metcalfe et al., 2006).

Figure 2.19. Configuration of two different oscillatory flow mixers (a) ring-shaped baffled tube and (b) meso-tube (Zheng and Mackle, 2008).

Figure 2.20. (a) and (b) Oscillatory flow, (c) regular no chaotic flow for oscillatory flow in a smooth wall column, (d) and (e) chaotic flow in an oscillatory flow in a baffled column (Ni et al., 2003).

Figure 2.21. Schematic demonstration of synthetic jet mixer (Xia and Zhong, 2013).

Figure 2.22. Pionceré section results for RAM, with a shear-thinning fluid with flow behavior index, \( n \), of 0.29. On the left side, the Pionceré section with the usual aperture used with RAM, with a big unmixed island formation. On the right Pionceré section with an extra offset window in RAM to get full mixing (Metcalfe et al., 2006).

Figure 2.23. (a) Picture of Elongational Pin Mixer (EPM), (b) elongation flow crated in EPM (Rauwendaal, 2014).

Figure 2.24. Schematic illustration of two-rod mixer with alternative rotation of two inner rods (El Omari and Le Guer, 2010 a).

Figure 2.25. Multiple 90° pipe bends with different periodic switching angle; (a) 45°, (b) 180°, (c) 90°, (d) 0° (Yamagishi et al., 2007).
Figure 2.26. (a) Chaotic configuration heat exchanger, with 33 90° bend pipes and a 90° periodic switch angle between the bends (Changy et al., 2000); (b) Chaotic configuration heat exchanger, with four 90° bend pipes with a cubic cross section and a 90° periodic switch angle between the bends (Peerhossaini et al., 1993).

Figure 2.27. Helical configuration heat exchanger, with thirty-three 90° bend pipes and a 0° periodic switch angle between the bends (Changy et al., 2000).

Figure 2.28. Types of vibration: transversal, longitudinal, and rotational.

Figure 2.29. Experimental set up of acoustic transverse vibration in a capillary tube for studies on flow enhancement on Newtonian fluids (water, silicone, oil) and human blood (Shin et al., 2003).

Figure 2.30. Description of microbubble formation, growth, and collapse (Leong et al., 2011).

Figure 2.31. Illustration of jet formation (Lauterborn and Claus-Dieter Ohl, 1997).

Figure 2.32. Picture of acoustic streaming generated from a probe horn (Monnier et al., 1999).

Figure 2.33. Enhancement of convection heat transfer by acoustic streaming (Legay et al., 2011).

Figure 2.34. Enhancement of convection heat transfer by acoustic cavitation (Legay et al., 2011).

Figure 2.35. Shell-and-tube structure of Yao et al. (2010).
Figure 2.36. Shell-and-tube (sonitube ultrasound reactor) structure of Gondrexon et al. (2010). ................................................................. 73

Figure 2.37. Tube in tube heat exchanger, (a) Legay et al. (2012 a), (b) Legay et al. (2012 b), all the dimensions in mm.............................................................. 74

Figure 2.38. Illustration of ultrasound effects: heat transfer enhancement inside the heat exchanger (Legay et al., 2012 b)........................................................................ 75

Figure 2.39. Image of Horn, Ultrasound reactor at 20 kHz (Faid et al., 1998 b)................. 83

Figure 2.40. Image of Cup-Horn, Ultrasound reactor at 20 kHz (Faid et al., 1998 b)....... 84

Figure 2.41. Image of stainless steel resonating tube at 20 kHz (with inner diameter at 42 mm), where C: transducer, B: booster, M: modular unit (Faid et al., 1998 b)........................................................................ 85

Figure 2.42. Image of the collar of Modular unit of Sonitube (Vaxelaire, 1995).............. 86

Figure 2.43. Experimental set up for investigation of ultrasound power of Sonitube by studying the thermal effects ($\Delta T$) (Faid et al., 1998 a). ........................................ 88

Figure 2.44. Thermal effect of ultrasound power by Sonitube, at the center $Y=0$mm and at the radius of Resonator $Y=13$mm. Where $T_{eq}$ equilibrium temperature, $T_0$ is the initial temperature of the medium, and $X$ is the axial length (Faid et al., 1998 a). .................................................................................. 88

Figure 2.45. Structure of Heat exchanger with outer tube and reactor a titanium Sonitube at 35 kHz (Legay et al., 2012 b)...................................................... 89

Chapter 3: Mathematical solution and its computer implementation for temperature predictions during continuous flow cooling of non-Newtonian products
Figure 3.1. Laminar fully developed flow of viscous product inside the cooler with constant wall temperature ................................................................. 116

Figure 3.2. Picture of input values of constant fluid properties thermal conductivity \((k)\), density of the food \((\rho)\), specific heat \((cp)\), and flow behavior index \((n)\); parameters of the cooling unit, such as the radius \((R)\), and the length \((L)\) of the tube of the cooling section; average volumetric flow rate \((V)\) and the mean velocity \((Um)\), the initial uniform temperature value \((Ti)\) and the constant wall temperature \((Tw)\), and Peclect number \((Pe)\) calculation .......... 125

Figure 3.3. Radial temperature distribution curve from computer model and bulk temperature, \(T_{bulk}\), calculation using the final temperature distribution function, \(T_{func}\), estimated from the mathematical solution of Graetz problem and the computer model .......................................................... 126

Figure 3.4. Comparison of the radial temperature distribution curve at the exit of the cooling section, predicted from computer model, with experimental temperature measurements at three different cross sectional points ............. 129

CHAPTER 4: Thermal mixing via acoustic vibration during continuous flow cooling of sweet potato puree

Figure 4.1. Time temperature history of three radial cross-sectional points of the pipe (center, intermediate, wall) of sweet potato puree during heating (until 1700 s) with a continuous flow MW unit (pilot-scale 5 kW, 915 MHz system; Model number P09Y5KA02, Industrial Microwave Systems, Morrisville, NC) and cooling with shell in tube heat exchanger .................... 142
Figure 4.2. a) Schematic diagram and b) picture of experimental apparatus of continuous flow thermal processing system...................................................... 144

Figure 4.3. Figure of continuous flow MW system. ................................................................. 146

Figure 4.4. Pictures of two MW ovens from the first row of MW series continuously connected with polypropylene pipe. ................................................................. 147

Figure 4.5. Picture a) of aluminum base of Buttkicker LFE and b) Buttkicker LFE placed at the mounted at the springer of the base with the U-turn elbow located at the top of Buttkicker LFE. ................................................................. 151

Figure 4.6. a) Schematic diagram of mixing unit, b) and c) Picture of mixing unit with the SlamStick™ accelerometer at the top of the U-turn elbow, which was wrapped at the top of the Buttkicker LFE. ................................................................. 152

Figure 4.7. Time-temperature history outlet of the MW system. ............................................ 156

Figure 4.8. 1-D FFT plot, showing acceleration (using as unit Gs) magnitude of vibration of different tested frequencies, at a range of 10-100 Hz. ............... 157

Figure 4.9. Time-temperature history during vibration at different amplitudes at a) the inlet and b) the outlet of the mixing unit, at three different points of the pipe: at the center, at the wall and at an intermediate point. The dashed lines represent the beginning and the solid lines represent the end of vibration at each different volume (50, 75, and 100%) of amplifier. Note that the average time it took fluid particle to move from the inlet to the outlet of U-turn was 24 sec.................................................. 159
Figure 4.10. Comparison of temperatures of center (T_c) and wall (T_w) elements at the inlet and the outlet of the mixing unit. The dashed lines represent the beginning and the solid lines represent the end of vibration at each different volume (50, 75, and 100%) of amplifier. .......................... 160

Figure 4.11. 1-D FFT plots, during vibration at 20 Hz, a) at 100% volume of amplifier, b) at 75% volume of amplifier and c) at 50% volume of amplifier. ............... 162

Figure 4.12. Time-temperature history at a) the inlet and b) the outlet of the mixing unit, at three different cross sectional points of the pipe (at the center, at the wall and at an intermediate point) during continuous vibration of 60 s at different frequencies of 16 Hz, 20 Hz, 24 Hz, 30 Hz, and 40 Hz. .............. 166

Figure 4.13. Comparison of temperatures between T_c and T_w elements at the inlet and the outlet of the mixing unit during continuous vibration of 60 s at different frequencies of 16 Hz, 20 Hz, 24 Hz, 30 Hz, and 40 Hz. ............... 167

Figure 4.14. Time-temperature history during vibration of 60 s at different frequencies, 15 Hz, 20 Hz, and 25 Hz, followed by 120 s of vibration at 20 Hz, a) the inlet and b) the outlet of the mixing unit, at three different cross sectional points of the pipe: at the center, at the wall and at an intermediate point. ...... 170

Figure 4.15. Comparison of temperature differences between T_c and wall T_w elements at the inlet and the outlet of the mixing unit during vibration of 60 s at different frequencies, 15 Hz, 20 Hz, and 25 Hz, followed by 120 s vibration at 20 Hz.  ............................................................................................................. 171
Figure 4.16. Time-temperature history a) the inlet and b) the outlet of the mixing unit, at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during vibration of 120 s at different frequencies, 16 Hz, 18 Hz, and 20 Hz, and at different time-periods of cooling. ........................................................................................................ 172

Figure 4.17. Comparison of temperature differences between T_c and T_w elements at the inlet and the outlet of the mixing unit during vibration of 120 s at different frequencies, 16 Hz, 18 Hz, and 20 Hz, and at different time-periods of cooling. ........................................................................................................ 173

Figure 4.18. Time-temperature history a) the inlet and b) the outlet of the mixing unit, at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during vibration of 120 s at 20 Hz and at different time-periods of cooling. ........................................................................................................ 174

Figure 4.19. Comparison of temperature differences between T_c and T_w elements at the inlet and the outlet of the mixing unit, during vibration of 120 s at 20 Hz, at different time-periods of cooling. ........................................................................................................ 175

Figure 4.20. Time-temperature history a) the inlet and b) the outlet of the mixing unit, at three different cross sectional points of the pipe (at the center, at the wall and at an intermediate point) during vibration at 20 Hz of 60 s at the start of the experiment and 120 s at the end of the experiment, with an intermediate tested vibration protocol of 75 s combining 30 s of vibration at 20 Hz and 25 Hz, with 5 s of non-vibration period in between. .............. 177
Figure 4.21. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit, during vibration at 20 Hz of 60 s at the start of the experiment and 120 s at the end of the experiment, with an intermediate tested vibration protocol of 75 s combining 30 s of vibration at 20 Hz and 25 Hz, with 5 s of non-vibration period in between. .......................... 178

Figure 4.22. Time-temperature history a) the inlet and b) the outlet of the mixing unit, at three different cross sectional points of the pipe (at the center, at the wall and at an intermediate point) during vibration of two different protocols of vibration of 90 s and 91 s, respectively. The first one with 7 s vibration followed by 3 s of non-vibration tested at two different frequencies at 20 Hz and 25 Hz, and the second with 10 s vibration at 20 Hz followed by 3 s of non-vibration. ...................................................... 179

Figure 4.23. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during vibration of two different protocols of vibration of 90 s and 91 s, respectively. The first one with 7 s vibration followed by 3 s of non-vibration tested at two different frequencies at 20 Hz and 25 Hz, and the second with 10 s vibration at 20 Hz followed by 3 s of non-vibration. ...................................................... 180

Figure 4.24. Time-temperature history during continuous vibration of 4 min at different frequencies, 16 Hz, 20 Hz, 24 Hz and 28 Hz with a duration of 60 s at each frequency a) the inlet and b) the outlet of the mixing unit, at three
different cross sectional points of the pipe: at the center, at the wall, and at an intermediate point.......................................................................................................................... 182

Figure 4.25. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 4 min, with 60s min vibration at frequencies of 16 Hz, 20 Hz, 24 Hz, and 28 Hz. ......................................................................................................................................................... 183

Figure 4.26. 1-D FFT plot for vibrations tests of 60s at a) 15 Hz, b) 20 Hz, and c) 24 Hz ......................................................................................................................................................... 185

Chapter 5: Acoustic enhancement of continuous flow cooling of viscous and multiphase food products

Figure 5.1. Picture of mixing system during resonance frequency tests, with two SlamStick™ devices: one at the top of U-turn elbow and the other at middle of the flexible tube close to the inlet of mixing unit. ......................... 202

Figure 5.2. Time-temperature history at the exit of the MW system for banana puree. ..... 203

Figure 5.3. Effect of the acoustic frequency on the acceleration (using as unit Gs) magnitude of vibration of different tested frequencies, as determined from 1-D FFT plot, for banana puree. ................................................................. 204

Figure 5.4. Time-temperature history for banana puree at a) the inlet and b) the outlet of the mixing unit, at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 120 s at different 20 Hz. The dashed lines represent the beginning, and the solid lines represent the end of vibration. Note that the
average time it took for the fluid particle to move from the inlet to the outlet of the U-turn was 24 sec. ................................................................. 207

Figure 5.5. Comparison of temperature differences between center \( T_c \) and wall \( T_w \) at the inlet and the outlet of the mixing unit during continuous vibration of 120 s at 20 Hz for banana puree. ......................................................... 208

Figure 5.6. Time-temperature history for banana puree at a) the inlet and b) the outlet of the mixing unit, at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 120 s at different frequencies of 20 Hz. ........................................... 209

Figure 5.7. Comparison of temperature differences between \( T_c \) and \( T_w \) elements at the inlet and the outlet of the mixing unit during continuous vibration of 120 s at 20 Hz, for banana puree................................................................. 210

Figure 5.8. Shear stress vs. shear rate relationship of banana puree at different temperatures. Symbols refer to experimental data and lines refer to simulated values through Eq. 5.1................................................................. 211

Figure 5.9. Effect of temperature on the apparent viscosities of banana puree at different shear rates................................................................. 213

Figure 5.10. Time-temperature history at the exit of the MW system for applesauce........ 214

Figure 5.11. Effect of the acoustic frequency on the acceleration magnitude of vibration of the mixing system with applesauce as the tested product, as determined by the 1-D FFT plot, at a) 50% and b) 25% of the level of the volume of amplifier. ...................................................................................... 216
Figure 5.12. Time-temperature history for applesauce at a) the inlet and b) the outlet of the mixing unit at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 120 s at 60 Hz and 40 Hz. Note that the average time it took a fluid particle to move from the inlet to the outlet of U-turn was 32 sec. .

Figure 5.13. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 120 s at 60 and 40 Hz for applesauce.

Figure 5.14. Time-temperature history for applesauce at a) the inlet and b) the outlet of the mixing unit at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 120 s at 20 Hz and 24 Hz.

Figure 5.15. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 120 s at 20 Hz and 24 Hz, for applesauce.

Figure 5.16. Shear stress vs. shear rate relationship of canned processed applesauce at different temperatures. Symbols refer to experimental data and lines refer to simulated values calculated through Eq. 5.1.

Figure 5.17. Effect of temperature on the apparent viscosities of applesauce at different shear rates.

Figure 5.18. Time-temperature history at the exit of the MW system for canned processed cheese sauce.
Figure 5.19. Effect of the acoustic frequency on the acceleration magnitude of vibration of different tested frequencies, as determined from 1-D FFT plot, for different series of experiments with canned processed cheese sauce. .......................... 226

Figure 5.20. Time-temperature history for canned processed cheese sauce at a) the inlet and b) the outlet of the mixing unit at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 60 s at 16 Hz, 20 Hz, and 25 Hz, and 120 s at 20 Hz. Note that the average time it took the fluid particle to move from the inlet to the outlet of U-turn was 28 sec. .......................................................... 228

Figure 5.21. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 60 s at 16 Hz, 20 Hz, and 25 Hz and 120 s at 20 Hz for canned processed cheese sauce.............................................................................................................. 229

Figure 5.22. Time-temperature history for canned processed cheese sauce at a) the inlet and b) the outlet of the mixing unit at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 120 s at different 20 Hz. ....................................................... 230

Figure 5.23. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 120 s at 20 Hz for canned processed cheese sauce............................................................... 231
Figure 5.24. Shear stress vs. shear rate relationship of canned processed cheese sauce at different temperatures. Symbols refer to experimental data and lines refer to simulated values calculated through Eq. 5.1. ........................................ 232

Figure 5.25. Effect of temperature on the apparent viscosities of canned processed cheese sauce at different shear rates..................................................... 234

Figure 5.26. Time-temperature history at the exit of the MW system for aseptically processed cheese sauce................................................................. 235

Figure 5.27. Effect of the acoustic frequency on the acceleration magnitude of vibration of different tested frequencies at the U-turn elbow, as determined from 1-D FFT plot, for different series of experiments with aseptically processed cheese sauce............................................................. 236

Figure 5.28. Effect of the acoustic frequency on the acceleration magnitude of vibration of different tested frequencies at the flexible tube at the inlet of the mixing unit, as determined by 1-D FFT plot, for different series of experiments with aseptically processed cheese sauce. ...................................................... 237

Figure 5.29. Time-temperature history for aseptically processed cheese sauce at a) the inlet and b) the outlet of the mixing unit at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 60 s at 60 Hz and 20 Hz and 120 s at 20 Hz. Note that the average time it took the fluid particle to move from the inlet to the outlet of U-turn was 24 sec. ...................................................... 239
Figure 5.30. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 60 s at 60 Hz and 20 Hz and 120 s at 20 Hz for aseptically processed cheese sauce.

Figure 5.31. Time-temperature history for aseptically processed cheese sauce at a) the inlet and b) the outlet of the mixing unit at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 60 s at 60 Hz and 20 Hz and 120 s at 20 Hz.

Figure 5.32. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 60 s at 60 Hz and 20 Hz for aseptically processed cheese sauce.

Figure 5.33. Time-temperature history for aseptically processed cheese sauce at a) the inlet and b) the outlet of the mixing unit at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 60 s at 60 Hz and 20 Hz and 120s at 20 Hz.

Figure 5.34. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 60 s at 60 Hz and 20 Hz and 120s at 20 Hz for aseptically processed cheese sauce.
Figure 5.35. Shear stress vs. shear rate relationship of aseptically processed cheese sauce at different temperatures. Symbols refer to experimental data and lines refer to simulated values calculated through Eq. 5.1.

Figure 5.36. Effect of temperature on the apparent viscosities of aseptically processed cheese sauce at different shear rates.

Figure 5.37. Effect of the acoustic frequency on the acceleration magnitude of vibration of the mixing system with sweet potato puree as the tested product, as determined from the 1-D FFT plot.

Figure 5.38. Time-temperature history of sweet potato puree at a) the inlet and b) the outlet of the mixing unit at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 120 s at different 20 Hz. Note that the average time it took the fluid particle to move from the inlet to the outlet of U-turn was 24 sec.

Figure 5.39. Comparison of temperature differences between T_c and T_w elements at the inlet and the outlet of the mixing unit during continuous vibration of 120 s at 20 Hz for sweet potato puree.

Figure 5.40. Time-temperature history for sweet potato puree at a) the inlet and b) the outlet of the mixing unit at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 120 s at different 20 Hz.
Figure 5.41. Comparison of temperature differences between \( T_c \) and \( T_w \) elements at the inlet and the outlet of the mixing unit during continuous vibration of 120 s at 20 Hz for sweet potato puree. ................................................................. 253

Figure 5.42. Shear stress vs. shear rate relationship of sweet potato puree at different temperatures. Symbols refer to experimental data and lines refer to simulated values calculated through Eq. 5.1. ................................................................. 254

Figure 5.43. Effect of temperature on the apparent viscosities of sweet potato puree at different shear rates. ........................................................................................................ 256
CHAPTER 1: Introduction

Continuous flow thermal processing (heating, holding, and cooling) of highly viscous and multiphase food products, such as fruit and vegetable purees, soups, and typical dairy products, remains a major challenge for the food industry (Barigou et al., 1998). Laminar flow and low conductivity, characteristics of highly viscous foods, lead to slow heat transfer and a wide temperature distribution inside the product, increasing the total processing time and cost and reducing the final food product quality (Barigou et al., 1998).

The problem of low conductivity of viscous multiphase products has been addressed during the heating cycle of continuous thermal processing, utilizing a variety of advanced heating technologies, such as continuous flow microwave (MW) systems, and ohmic and radiofrequency heating technologies (Coronel et al., 2005; Steed et al., 2008; Cullen et al., 2012). Advanced heating technologies provide rapid volumetric heating and enhance the thermal process of viscous food products by increasing the temperature levels of sterilization, minimizing hold time exposure, and maximizing the final food product quality (Coronel et al., 2005; Steed et al., 2008; Cullen et al., 2012).

Due to lack of advanced cooling technologies, cooling depends on slower convection heat transfer methods. During the cooling cycle of thermal processes, laminar flow, non-Newtonian flow behaviour (characteristic for the most of the processed foods (Tokisangyo, 2014)), and low conductivity of highly viscous foods lead to a wide radial temperature distribution within the product, resulting in unequal cooling treatment and degradation of final food quality. Production systems, which utilize advanced thermal methods, could
benefit from operational enhancements of the cooling stage of the process, for both existing and future installations. Unfortunately, predicted and anticipated developments of volumetric cooling technologies, such as magnetic field cooling (Sarlah et al., 2006; Kawanami et al., 2011) have often been slow and expensive in their progress to large scale commercial applications.

Continuous flow cooling processes of highly viscous foods can be enhanced significantly by improving radial mixing, which can be achieved by turbulent flow conditions or chaotic advection (Metcalfe and Lester, 2009; Eesa and Barigou, 2010; Saatdjian, et al., 2011). In highly viscous food products, turbulence mixing is unfeasible or requires too much energy due to the dominated viscous forces (Saatdjian, et al., 2011). Mixing in the laminar regime can be enhanced through the mixing technique of chaotic advection. Chaotic advection is a mixing technique in the laminar flow regime where, through passive (by periodically changing the geometry of flow) and active (applying periodic non-linear forces) methods, chaotic fluid motion is generated. Chaotic advection generates chaotic trajectories and breaks the mixing barriers—the streamlines where the fluid elements flow in the laminar flow (Aref, 1984; Ottino, 1989; Aref, 2002; Cullen, 2009). Chaotic advection is a method to achieve mixing in very viscous materials, like food purees, soups, etc., through periodic stretching and folding of fluid elements, which results in an exponential increase on interfacial area and pattern with high gradients (density, pressure, temperature, etc.) to facilitate chaotic mixing (a combination of chaotic advection and diffusion) (Ottino, 1989; Arref, 2002; Cullen, 2009). The degree of stretching is an important factor during mixing with chaotic advection to achieve global chaotic flow due to the breakdown of flow
symmetries inside the flow, which can work as barriers during mixing (Ottino, 1989; Kusch and Ottino, 1992; Metcalfe et al., 2006; Singh et al., 2008).

In continuous flow thermal treatment of viscous and multiphase food products, radial mixing via chaotic advection has been studied and used as a method for heat transfer enhancement and thermal mixing (temperature profile homogenization) with the most characteristic examples in the food industry: scraped surface heat exchangers and static mixers, respectively (Carlson, 1991; Acharya et al., 1992; Acharya et al., 2001; Etchells III and Meyer, 2004; Metcalfe and Lester, 2009; Lester et al., 2009; Eesa and Barigou, 2010; El Omari and Le Guer, 2010 a, El Omari and Le Guer, 2010 b; Saatdjian et al., 2011; Le Guer and El Omari, 2012). Although these heat exchangers and inline mixers enhance the total thermal process, the problem of reduction of final food product quality remains through the damage to solid food particles. In food industry, most liquid-solid mixtures of highly viscous processed foods contain a high concentration of solid particles; these solid particles can be damaged from the high shear imposed by blades of scraped surface heat exchangers and the large pressure drop created by the physical obstacles-baffles of static mixers, which block the solid-liquid mixture flow and clog the solid particles. Combining active and passive techniques to generate chaotic advection in an open duct flow system could reduce the shear and the pressure drop and benefit the food process by maximizing the integrity of food particles and, thus, the final food product quality.

Two techniques that have been studied as active methods for generating chaotic advection in open duct flows are acoustic and sinusoidal transversal mechanical vibration. Acoustic vibration at low acoustic frequencies (of about 60 Hz) is a method industrially used
for batch mixing of viscous materials (Resodyn, 2014). Additionally, Computational Fluid Dynamics (CFD) and experimental studies have shown that mechanical vibration at low acoustic frequencies (0-75 Hz) can produce mixing and improve heat transfer of viscous products (Khalaf and Sastry, 1996; Eesa and Barigou, 2010). Also, bent or curved pipes, via centrifugal forces, generate Dean flow and have been used in a passive technique of chaotic advection, with less pressure drop than static mixers, and for heat transfer enhancement through shell and tube heat exchangers with chaotic advection configuration—in which the shell consists of periodically connected bend pipes (Changy et al., 2000).

In this study, improvement of temperature uniformity through thermal mixing during cooling was tested as a method for enhancing continuous flow cooling of viscous and multiphase food products. At first, the mathematical solution, and its computer implementation, was used to estimate the radial temperature distribution and the bulk temperature of highly viscous foods, such as applesauce at the exit of the cooling section (cooling time or cooling unit) of a bulk sterilization system, in order to establish the benefits of thermal mixing and temperature profile homogenization on the enhancement of the continuous flow cooling process. Additionally, a new mixing unit was designed and built. It combined active and passive techniques to generate chaotic advection in an open duct flow system in attempt to achieve a reduced shear and pressure drop and to retain the maximum integrity of food particles. Acoustics were used to impose transversal sinusoidal vibration into a 180° bend pipe in order to generate chaotic advection and enhance thermal mixing during continuous flow cooling process of highly viscous food products, such as sweet potato puree, applesauce, banana puree, and cheese sauce. A complete continuous flow thermal
system, consisting of a continuous flow microwave (MW) system (formed from a series of home MW ovens), a cooler, and the mixing unit, was built and used for studies on thermal homogenization of food products during the cooling cycle. The methodology and parameters of vibration, such as frequency, amplitude, and protocol of vibration used for the experiments, were first determined experimentally on studies with sweet potato puree.
CHAPTER 2: Review of literature

2.1 Introduction

Advanced heating technologies, such as ohmic heating, continuous flow microwave, and radio frequency thermal treatment systems (Coronel et al., 2005; Steed et al., 2008), have enabled food processors to maximize product quality during continuous thermal processing of viscous and multiphase food products, such as typical dairy products, fruit and vegetable purees and soups. During the cooling cycle of thermal processes, laminar flow and low conductivity of viscous foods lead to a wide radial temperature distribution within the product, resulting in unequal cooling and degradation of final food quality. The enhancement of mixing of high viscous food products during continuous flow cooling could result in enhancement of the total cooling process, which will help maximize the final product quality.

This chapter aims to provide a review of the mechanism of mixing and convectional heat transfer of viscous materials in the laminar flow regime. Secondly, this chapter contains a review on how mixing and heat transfer can be utilized from the mechanisms of vibration and acoustics.

2.1.1 Mechanism of mixing

In continuous flow thermal processing of viscous and multiphase food products, like typical dairy products, fruit and vegetable purees, laminar flow and low conductivity lead to a non-uniform velocity profile and a wide radial distribution of product temperature (Barigou et al., 1998). Thus, thermal treatment is not equivalent for all the parts of the food product during the processing, which results in a wide variation of product sterility and quality.
Radial heat transfer and uniformity of temperature distribution can be enhanced by radial mixing, which can be achieved by turbulent flow conditions or chaotic flow (Eesa and Barigou, 2010).

2.1.1.1 Reynolds and Péclet number

To understand and improve the radial mixing and transport mechanisms, it is important to introduce two dimensionless numbers: Reynolds (Re) and Péclet (Pe) numbers.

The Reynolds number refers to the ratio of inertial to viscous forces in a fluid (Equation 2.1) and determines the flow, laminar, or turbulent regime (Singh and Heldman, 2009).

\[ \text{Re} = \frac{\rho d \bar{u}}{\mu} \] (2.1)

Where \( \rho \) (k·g/m\(^3\)) is the density of fluid, \( d \) (m) is the diameter of the pipe, \( \bar{u} \) (m/s) is the average fluid velocity, and \( \mu \) (Pa/s) is the viscosity of fluid.

Turbulence flow exists above a critical value of Reynolds number; for pipe flow, this value equals to 2100. The flow remains laminar under the critical Reynolds number (Singh and Heldman, 2009). Mixing in turbulent flow is much faster than laminar mixing due to interlayer convection. Therefore, laminar mixing is more challenging at low Reynolds numbers, which is the characteristic transport and mixing regime of viscous and multiphase foods.

Péclet number (Pe) (Equation 2.2) quantifies the ratio between advection and diffusion, where a Pe→0, means the diffusion is rapid in a very slow flow, while Pe→∞ refers to a very slow diffusion in a fast flow.
\[ Pe = \frac{UL}{D} \]  
(2.2)

Where \( U \) (m/s) is the characteristic velocity of fluid, \( L \) (m) is a characteristic length of the flow, and \( D \) is the diffusivity of either heat (thermal diffusivity) or concentration (diffusivity). The equation for \( Pe \), for a pipe flow heat transfer problem, is given from the following equation:

\[ Pe = \frac{R\rho c_p u_m}{\kappa} \]  
(2.3)

Where \( c_p \) (J/kg·°C) is specific heat of fluid, \( k \) (°C·m/W) is the thermal conductivity of fluid; \( R \) (m) is the radius of the pipe, and \( u_m \) (m/s) is the average velocity of the flow.

### 2.1.1.2 Mixing of viscous fluids

Mass and thermal mixing in laminar flow is very slow. There are two types for higher rate of mixing: turbulence and chaotic mixing. The efficiency of turbulence mixing is related with the rate of turbulent kinetic energy dissipation due to viscosity (Cullen, 2009). In high viscous products, like food pastes, the rate of kinetic energy dissipation is very high due to the high viscous forces, and the flow remains laminar (Cullen, 2009). The mixing of such viscous food products is associated with a laminar mixing process known as chaotic advection, where the term advection means that fluid particles always follow the flow streamlines and their inertial could be neglected (Aref, 1984; Cullen, 2009; Metcalfe and Lester, 2009; Eesa and Barigou, 2010; Le Guer and El Omari, 2012).

Due to laminar flow (low Reynolds) and low conductivity (high Péclet number) characteristics, efficient mixing and heat transfer for highly viscous materials, like food pastes, can be achieved with the mechanism of chaotic advection; the technique in which the
velocity field is characterized form complex trajectories and the mixing depends on advection (Figure 2.1).

Figure 2.1. Relation of Peclet number in low Reynolds number laminar flow (Nguyen and Wu, 2005).
2.1.2 Kinematics of fluid mixing

The mixing of fluids in laminar flow can be studied through dynamical system theory (Ottino, 1989; Cullen, 2009). Fluid flow can be expressed with two different approaches: Eulerian or Lagrangian point of view (Metacalfe, 2010; Cullen, 2009). In Eulerian approach, changes on fluid properties, such as velocity, pressure, and/or temperature (which vary in time), are studied in a selected fixed frame (or point) of the flow. In Lagranian approach, the flow properties of particular fluid particles are studied and tracked, where the term fluid particle refers to a small coherent region of fluid with neglected dimension (Cullen, 2009; Metacalfe, 2010).

Fluid mixing in laminar flow is studied using the Langarian approach in dynamical systems (Aref, 1984). Dynamical system is used for studying the motion and mixing of the trajectories of fluid particles. The term trajectory refers to the orbit or path that fluid particles follows in the fluid motion. The study of the trajectory of its fluid particle is given by the kinematic equation or advection equation (Aref, 1984; Metclafe, 2010):

\[
\frac{dX}{dt} = V(X,t)
\]  

(2.4)

Where equation 2.4 is the equation of motion of a fluid particle, which has been moved from its initial position X via the applied velocity field V. In a different initial position, X refers to a different fluid particle (Metclafe, 2010).

For the study of fluid motion and mixing, except from dynamical systems, another important subject of study is mass conservation, which is given in the continuity equation (Metclafe, 2010):
\[
\frac{d\rho}{dt} + \nabla(\rho V) = 0 \quad (2.5)
\]

Where \( \rho \) is the density of the fluid, \( V \) is the velocity profile.

When the density of the fluid remains constant, the flow of this flow is known as \textit{incompressible flow}. During incompressible flow, the continuity equation 2.5 becomes (Metclafe, 2010):

\[
\nabla V = 0 \quad (2.6)
\]

For the characteristic two-dimensional steady laminar flow for viscous food products, Eq. 2.6 is converted to (Metclafe, 2010):

\[
\frac{\partial V_x}{\partial x} + \frac{\partial V_y}{\partial y} = 0 \quad (2.7)
\]

The solution of partial differential Eq. 2.7 can be satisfied by any scalar stream function \( \Psi(x, y) \), with the derivatives of function \( \Psi \) to be (Metclafe, 2010):

\[
\frac{\partial \Psi}{\partial y} = V_x \text{ and } \frac{\partial \Psi}{\partial x} = -V_y \quad (2.8)
\]

When scalar function \( \Psi \) has a constant value, the derivative of derivatives of function \( \Psi \) is equal with zero and is given from the following equation (Metclafe, 2010):

\[
d\Psi = \frac{\partial \Psi}{\partial x} dx + \frac{\partial \Psi}{\partial y} dy = 0 \quad (2.9)
\]

Combination of Eq. 2.8 and 2.9 gives:

\[
\frac{dx}{V_x} = \frac{dy}{V_y} \quad (2.10)
\]
Eq. 2.10 in dynamical systems is associated to *Hamiltonian system*, which defines a vector field whose flows are the trajectories of fluid particles (Aref, 1984; Ottino, 1989).

The conclusion from kinematic and continuity equation study in laminar flow can be summarized as follows: for a two-dimensional (2-D) steady, incompressible flow, the Hamiltonian dynamical system is *integrable*, which means the stream scalar function $\Psi$ of dynamical system is constant, and therefore, the trajectories of fluid particles are moving inside lines—close streamlines. (Aref, 1984; Cullen, 2009; Metacalfe, 2010). According to chaos theory, for a dynamical system to be *unintegrable* it needs to have at least three degree of freedom, where the term degree of freedom is related to the dimensions of the flow. Therefore, chaos is almost universal in 2-D time periodic flows and three-dimensional (3-D) flows, where the degree of freedom is higher or equal to three degrees of freedom ($\geq 3$), and results in the unintegrable Hamiltonian system. In the unintegrable Hamiltonian system, the characteristic closed trajectories of laminar flow are broken down, and the fluid particle trajectories are chaotic. If the dynamical system governing trajectories is chaotic, the integration of a trajectory is extremely sensitive to initial conditions, and neighboring points separate exponentially with time, which enhances the transport and mixing of fluid particles. This phenomenon is called *chaotic advection* (Aref, 1984; Ottino, 1989).

### 2.1.3 Chaotic advection

Chaotic advection, or Lagrangian chaos, a term introduced from Aref (1984), is the phenomenon in which the characteristic streamline trajectories of steady laminar flow of viscous materials are transformed to extremely complex fluid trajectories through a passive (change of flow geometry) or active mixing method (creation of a secondary flow) (Aref,

In laminar flow, a 2-D steady (time independent) flow, and in small scale eddies in turbulence, viscous forces are dominated. Thus, fluid particles flow within streamlines, therefore limiting mixing (Franjione and Ottino, 1991). Though mixing in laminar field can be exhibited through chaotic advection, the phenomenon, where passive particles affected by a periodic velocity field, exhibits chaotic trajectories (Cullen, 2009). Chaotic mixing exists when highly complicated trajectories are observed in the Lagranian frame (time dependent trajectories). Chaotic advection is a method for achieving mixing in very viscous materials, like food purees, soups, etc., where turbulence flow cannot be achieved. Chaotic advection can be achieved in 2-D flows, with the application of non-linear periodic forces into the fluid, creating a 2-D time periodic (time dependent) flow, and in 3-D flows. Chaotic advection applies simple non-turbulence flows (chaotic flow) to create very efficient mixing, due to periodic (n= 2,3,4...) stretching and folding of fluid elements (Figure 2.2), which results in an exponential increase in interfacial area and pattern with high gradients of concentration (Ottino, 1989; Arref, 2002; Cullen, 2009). The stretching rate is as the most common factor that determines the effectiveness of chaotic advection on mixing (Leong and Ottino, 1989).

2.1.3.1 Major points and tools for chaotic flow study

Theory of chaotic mixing relates with coexistence of order and disorder or symmetry and chaos regions of fluid elements (Ottino et al., 1992). Chaotic advection is not irregular; periodic changes to the flow create well mixed regions, chaotic regions,
surrounded by unmixed regions—*islands* also known as known as KAM (from the theorem of Kolmogorov-Arnold-Moser about the persistence of quasi-periodic motions in dynamical systems) islands or *KAM-tori* (Metcalfe, 2010). While in chaotic regions the mixing is very rapid, islands work as barriers that block the mixing and transport of the flow (Ottino, 1989; Metcalfe, 2010). The largest coherent regions of unmixed materials are located in the symmetries of the flow. Flow symmetries in laminar low Reynolds number flow (Stoke Flow) exist because of the characteristic parabolic velocity profile on the cross section; the poorly mixed area in a laminar pipe flow is located close to the symmetry line of the flow, at
the center of the pipe (Ottino, 1989; Franjione et al., 1989; Metaclafe et al., 2006; Metcalfe, 2010).

Periodic reorientation of fluid elements is the key factor in chaotic advection. Since the velocity field is periodic, there are some fluid particles that return to their initial point. The initial fluid element corresponds with a \textit{periodic point}, and a sequence of periodic points gives a \textit{periodic orbit}. The order of the points is determined by the number of periods the fluid needs to return to the initial position; for example, if the fluid element returns after one period, it is called \textit{point of period one}; if returns after two periods, it is called \textit{point of two periods}; etc. (Otinno, 1989; Leong and Ottino, 1989). The character and structure of periodic points in the flow can provide a better understanding of chaotic advection flow regions. The periodic points are categorized into \textit{hyperbolic} and \textit{elliptic}, with \textit{hyperbolic periodic orbits} and \textit{elliptic periodic orbits} respectively. Hyperbolic periodic points and orbits provide mechanisms for the contraction and expansion (stretching and folding) of fluid elements. A small chaotic region is created around them, which could result in efficient global mixing (Wiggins and Ottino, 2004). Stable and unstable manifolds and orbits are formed close to hyperbolic points and orbits, which work as barriers for mixing. However, a transverse intersection of stable and unstable manifolds (unstable manifolds cannot intersect with other unstable manifolds) of the same hyperbolic periodic point or orbit creates a \textit{transverse homoclinic intersection}, where the transverse intersection of stable and unstable manifolds of different hyperbolic points or orbits creates a \textit{transverse heteroclinic intersection}. The mixed areas of flow are created from these intersections (Figure 2.3; Figure 2.4). On the other hand, according to KAM theory, elliptic points are the center of unmixed regions, or islands, where
the stretching mechanism of chaotic mixing is very low (Figure 2.4) (Otinno, 1989; Leong and Ottino, 1989; Wiggins and Ottino, 2004). Mathematically, Lyapunov exponents are used to describe the type of region as chaotic or KAM tori. Lyapunov exponents are a mathematical expression of the average exponential rates of stretching and folding of fluid particles orbits, and they can provide a numerical method for quantifying the intensity of chaotic transport and mixing in the flow. The Hamiltonian system can have two groups of Lyapunov exponents: zero and positive or negative. Zero Lyapunov exponents are associated with KAM tori regions and elliptic periodic orbits, while chaotic and hyperbolic periodic orbits are positive or negative (Wiggins and Ottino, 2004).

Birufication of islands is a very important factor for good mixing. Birufication can occur by changing the geometry of the flow and the symmetries of the flow. Birufication takes place when an elliptic point of island changes to a hyperbolic point; this can occur by changing parameters of mixing, such as extending the time of mixing. The transport of fluid in birufication islands is dominated by regions called cantori. Cantori are broken up KAM tori or KAM islands and are unmixed regions, like KAM tori. The difference between them and cantori is that fluid can pass from one side to the other of cantori, which is not possible in the KAM islands (Otinno, 1989; Leong and Ottino, 1989; Metcalfe, 2010). Birufication of islands results in a well-mixed area surrounded by cantori, and efficiency of mixing depends on the collapse of KAM island and cantori structures inside the flow.
Figure 2.3. Chaotic region close to a hyperbolic point (Leong and Ottino, 1989).

Figure 2.4. (a) Picture of hyperbolic point with a transverse homoclinic intersection and an elliptic point at the center of an island, (b) the hyperbolic point moves periodically in a closed orbit, and its stable and unstable manifolds create a transverse homoclinic point (intersection), (c) chaotic region close to an a hyperbolic point (Ottino, 1990).
2.1.3.2 Mapping of fluid particles trajectories

In chaotic mixing, study is very important for understanding and illustrating the efficient chaotic and island regions during mixing. For this reason a specific class of map, the *Pionceré map*, is used to demonstrate the paths of fluid trajectories during advection.

The Pionceré map is a very useful mathematical tool that provides a clear picture of mixed and unmixed regions for 2-D periodic flows, which also can be extended to 3-D flows (Metcalfè, 2010). First, the diametral plane of study is defined as the plane section of the Poincarè map, *Pionceré section*. Pionceré sections illustrate the asymmetrical timed topology of a small number of fluid particles after every period. Sections are designed by following the trajectories of the chosen fluid particles as they pass through the studied plane. The trajectories are tracked for a large number of periods, and a dot is designed in the plane every time the orbits of fluid particles cross the plane (Metcalfè, 2010). In the Pionceré map, chaotic regions are illustrated as areas filled with evenly spread points (dots), while unmixed areas are regions with no points (Figure 2.5). For continuous duct flows, making a Pionceré map is more complicated because the plane, which is studied, is displaced in space due to continuous flow. For this reason, dots of trajectories are pictured at the end of each mixing section of duct and are then translated to a common plane map (Metcalfè, 2010).
2.1.4 Studies on chaotic advection

In 1984, Aref's initial paper introduced the term chaotic advection and the prospect of studying dynamical systems to achieve mixing in laminar viscous flows. After this paper, an increased number of numerical and experimental research projects have studied the effects of chaotic advection on closed (batch) or open (continuous) 2-D, time dependent, and 3-D flows (Wiggins and Ottino, 2004).

2.1.4.1 Studies on chaotic advection in 2-D time-periodic flows

A large amount of numerical and experimental research has focused on chaotic fluid motion produced from 2-D-time periodic flows based on Hamiltonian dynamic systems. One of the major works, as mentioned above, was that of Aref (1884), who studied numerically
the fluid particles motion from 2-D-time periodic blinking vortex flow produced by two alternative rotating agitators in a tank. *Aref-blinking-vortex flow* generates periodic stretch and fold of fluid elements, which leads to mixing (Khakhar et al., 1986). The stretching and folding mechanism depends on the period of stirring of the two agitators with the existence of an optimum frequency of stirring, which maximizes mixing (Khakhar et al., 1986). The next studies on 2-D-time periodic flows focused on two types of flows: cavity flow and journal bearing flow.

Experimental and computational studies on laminar mixing on 2-D cavity flows have proven the existence of chaotic flow by the periodic oscillation of a cavity's boundary walls (Chien et al., 1986; Leong and Ottino, 1989). Experiments showed that alternative periodic motion of boundaries could lead to complete deformation of a color dye through the mechanism of chaotic advection. As in the case of Aref-blinking-vortex flow and for cavity flow, the maximum mixing efficiency for a given time was observed for oscillation of the walls at an optimum frequency (Chien et al., 1986; Leong and Ottino, 1989). Leong and Ottino (1989) tested the influence of continuous versus discontinuous (i.e., 20s with alternate oscillation of top and bottom cavity's wall followed by 5 s of pause) protocol of wall oscillation. For discontinuous protocol oscillation, large islands were formed close to the center of the cavity; small islands were observed with the continuous oscillation protocol.

The second type of flow studied for the case of 2-D-time periodic flows and chaotic advection was the case of journal bearing flow, the periodic flow created between two eccentric cylinders (Chaikan et al., 1986; Swanson and Ottino, 1990). Chaikan et al. (1986) did an experimental and computational study on chaotic particle motion in time-periodic flow
between eccentric rotating cylinders. They observed a very clear correlation between the computer simulation results for the shape and location of islands and the experimental results. They observed a generation of chaotic advection, from journal bearing flow in very low Reynolds number (Re≈0.1). Another study with two eccentric rotating cylinders, using a discontinuous velocity protocol was that of Swanson and Ottino (1990). In this work, the authors attempted to provide a computational way to describe the experimental results with accuracy. Using the Pionceré section, they were able to match the unmixed regions with experimental results. However, the Pionceré section did not give good information about the rate of mixing, which was related to the degree of stretching of fluid particles. For this reason, Swanson and Ottino (1990) proposed the use of stretching plots (similar to Lyapunov exponents) to illustrate the dye spreading rate. Stretching plots are a mathematical method proposed to evaluate the degree of stretching of fluid particles, understanding its particle's initial position and velocity field. Solomon et al. (1994) studied experimentally chaotic and turbulent flows in a rotating annular tank creating annular journal bearing flows. They concluded that periodic annular flow generated chaotic advection. They also introduced the mathematical method of Lévy flights to describe the irregular growth of particle displacement with time. Solomon et al. (1994) concluded that the particle displacement in a chaotic motion was better described by the length and time probability power law function in comparison to the exponential function.

2.1.4.2 Studies on chaotic advection in 3-D flows

Chaotic advection in 3-D flows can exit in closed, batch type systems, such as stirred tanks (Figure 2.6) (Fountain et al., 2000), or in continuous flow systems, such as open duct
flows (Kusch and Ottino, 1992; Metcalfe et al., 2006; Le Guer and El Omari, 2012). Chaotic advection in batch type systems usually works by stirring fluid in a batch agitator, where fluid mixing is induced by stretching and folding fluid elements (Figure 2.6) (Fountain et al., 2000). Good mixing in a batch system depends on the collapsing of island regions, which is a function of time (\( t \to \infty \) good mixing) (Aref et al., 2014).

Figure 2.6. Coordinates of experimental aperture of Fountain et al. (2000) experiment in a batch tank 3-D flow.
Most studies on chaotic advection in 3-D flows look into mixing in open duct flows, which this review will also be focused on. Mixing in open duct (steady) 3-D flows is connected with many important industrial problems, such as mixing, transport, and heat transfer of viscous materials—like viscous food products. Like in 2-D-time periodic flows and in 3-D steady flows, periodic stretching and folding of fluid elements is the technique used for mixing on continuous flow systems. In contrast to closed systems, continuous flow systems are related with transient dynamical systems, and therefore, the chaotic advection mechanism is focused more on partial mixing techniques (Aref et al., 2014). The term *partially mixing mechanism of chaotic advection* means the efficiently mixed area in a continuous flow system is in the transversal direction of unstable manifolds, while in closed systems of chaotic mixing, the well-mixed area is across all directions close to unstable manifolds (Aref et al., 2014).

**2.1.4.3 Studies on chaotic advection in 3-D open duct flows**

A number of studies on chaotic advection in 3-D open duct flows have been studied and designed, and different types of mixers (micromixers or macromixers), passive or active, have been built for mixing of viscous materials (Wiggins and Ottino, 2004; Nguyen and Wu, 2005).

*Passive chaotic advection mixers*

The basic concept of chaotic advection passive mixers is the increase of the interfacial area and, thus, improvement of diffusion and radial mixing (Nguyen and Wu, 2005). Passive mixers are designed to enhance radial mixing by modifying the geometry of axial channel flow in a way that achieves stretching, folding, and mixing of fluid elements.
One-way to change the flow geometry is to introduce obstacles in the open duct flow and force fluid to merge and mix (Lin et al., 2003; Nguyen and Wu, 2005). The other way to create mixing with chaotic advection passive mixers is with a split-and-join mixer structure, which splits and then joins and mixes the main axial flow (Nguyen and Wu, 2005; Xia et al., 2005). One the most used and characteristic passive chaotic advection mixer is static mixers, which use geometry reorientation of flow to create efficient radial mixing (Etchells III and Meyer, 2004; Cullen, 2009).

**Static mixers**

Static mixers are used as *thermal homogenizers* and *flow inverters*. Static mixers have been used since early 1950s to enhance the temperature homogeneity in thermal processing of polymer and viscous materials (Etchells III and Meyer, 2004). Static mixers can be described as a series of flow reorientation elements (obstacles) placed on the cross section of straight pipe. The flow reorientation elements are designed in that way so as to periodically split and rejoin fluid and cause mixing based on chaotic advection (Figure 2.7) (Etchells III and Meyer, 2004; Cullen, 2009).

![Figure 2.7. Typical structure of helical static mixers; Periodic reorientation of fluid elements results on enhancement of radial mixing (Stamixco-mex, 2014).](image)
There are many different static mixers available for different applications in industry use with different configurations of inflow obstacles geometry (Cullen, 2009). The inflow obstacles, consisting of metallic helical surface elements, corrugated sheets, parallel bars, small-diameter passages, or tabs sticking out from the wall, force the flow towards the pipe walls and back to the center; thus the mixing efficiency is improved (Figure 2.8) (Etchells III and Meyer, 2004; Cullen, 2009). Given enough helical baffles elements along the pipe, the radial inhomogeneity in temperature, velocity, and material composition improves gradually (Stamixco-mex, 2014). In static mixers the friction factor is many times larger than open duct flow, and this results in high pressure drop (depending on the number and the geometry of inflow elements); therefore extra energy is required to get the additional mixing effect (Etchells III and Meyer, 2004; Cullen, 2009).

Mixing with static mixers is independent of Reynolds numbers for a very low Reynolds number, Re<10. At higher Reynolds numbers (i.e., Re = 100), the mixed flow is not global chaotic—with islands, unmixed regions formed. Nevertheless, for Re> 1000, the mixed flow is mainly chaotic again, with some small islands. Therefore, mixing at Reynolds numbers higher than 100 but lower than 1000 may not be as effective as lower Reynolds numbers. In addition, more energy is consumed at higher Reynolds numbers because the pressure drop in static mixer is proportional to Reynolds number (Etchells III and Meyer, 2004; Cullen, 2009).
Figure 2.8. Different types of static mixer design options. From left: vortex mixer (type KVM), corrugated plate (type SMV), wall-mounted vanes (type SMF), cross-bar (type SMX), helical-twist (type KHT), cross-bar (type SMXL) (Etchells III and Meyer, 2004).

Twisted tubes and bend pipes

Another, passive chaotic advection mixer, which is used to equalize temperature and velocity profile in laminar flow, uses helically coiled tubes and bend pipes (90 and U-turn bend pipes). In twisted pipes and bend open flow tubes, the centrifugal force caused by the curvature of the tube generates *Dean roll cells* and creates a secondary flow, which modifies and reorients the main axial flow (Figure 2.9) (Khakhar et al., 1987; Changy et al., 2000; Etchells III and Meyer, 2004). For helically coiled tubes and bend pipes, the pressure drop is very similar to that of an open pipe regular flow, but efficiency mixing is limited in high Reynolds number values—close to the turbulent flow regime (Etchells III and Meyer, 2004).
Figure 2.9. Centrifugal forces create Dean roll-cells, causing periodic reorientation and mixing to main axial flow (Changy et al., 2000).

**Active chaotic advection mixers**

The first inline chaotic active continuous flow mixers were introduced in the late 1980s and early 1990s (Kusch and Otiino, 1992). One of the first chaotic advection mixers, proposed for continuous flow mixing, was the partitioned pipe mixer (PPM) (Khakhar et al., 1987). PPM is a simple construction mixer that consists of rectangular plates fixed orthogonally to each other inside the cross section of the pipe. The plates diced the pipe flow into a semi-circular duct flow, working like a small-scale static mixer. Semi-circular duct flow combined with an external rotation of the pipe produces a periodic reorientation of the flow, thus creating a spatially periodic velocity field, which leads to chaotic mixing (Figure 2.10) (Khakhar et al., 1987; Kusch and Otiino, 1992).
Another continuous flow mixer is the eccentric helical annular mixer (EHAM), which was introduced in 1989 from Ottino (1989). The system corresponds to the flow between eccentric cylinders. (Ottino, 1989; Kusch and Otiino, 1992; Mota et al., 2008). The two eccentric cylinders are rotated with different rotation frequency and with the reverse or the same cyclic direction to create a time periodic fluid motion, which leads to chaotic mixing on the annular flow between the two cylinders (Figure 2.11) (Ottino, 1989; Kusch and Otiino, 1992; Mota et al., 2008).
Kusch and Ottino (1992) used PPM and EHMM to compare experimentally and numerically the efficiency of mixing between spatial periodic and time periodic chaotic flows. Numerical studies using Pioncaré sections showed that the mixing on PPM depends on one dimensionless parameter of mixing strength $\beta$, given from the following equation:

$$\beta = \frac{4u_p L}{3\gamma \langle u_p \rangle R}$$  \hspace{1cm} (2.11)
Where $U_R$ is the rotation speed, $<U_z>$ is the average axial velocity of the main flow, $L$ is the length of the pipe mixer, and $R$ is the radius of the pipe mixer. $\gamma$ is given from the following equation (Ottino, 1989; Kusch and Ottino, 1992):

$$\gamma = \left(\frac{11}{3}\right)^{\frac{1}{2}} - 1 \quad (2.12)$$

According to Kusch and Ottino (1992) the dimensionless parameter $\beta$ can be considered, which gives the residence time of the fluid in the mixer, $L/(<U_z>)$, and the cross-sectional shear rate, $U_R/R$. From their numerical results, they found that mixing efficiency depended on the parameter of mixing strength $\beta$. From their numerical models, they concluded that, for both mixers, increasing $\beta$ so it is greater than the value of 30 unmixed regions decreased (KAM-tubes). For high $\beta$ values, $\beta>40$, no unmixed regions were observed.

In EHAM they observed that efficiency of mixing depended on the cross sectional velocity, which occurred from the frequency of rotation. Kusch and Ottino (1992) mentioned that the choice of the right frequency was very important because frequency influenced the angular displacement area of the cylinders, which determined the size of the chaotic cross section area. Changing the angular velocity of cylinders over a period did allow for the control of frequency of rotation. In their numerical calculation, they found that chaotic mixing can be achieved with a counter rotation of two cylinders and with a speed of 1080° per period for the outer cylinder and with the half-speed, 540°, for the inside cylindrical rod.

For their experiments, they used diluted glycerin with water, a Newtonian behavior fluid with a final viscosity of 1 Pa. Additionally, under the same conditions of flow rate at
200 mL/min, they studied the effects of these two mixers using a mixing strength number of $\beta=10$ for PPM and the counter rotation of two cylinders for EHAM with speeds at $1080^\circ$ per period for the outer cylinder and $540^\circ$ for the inside cylindrical rod. To test the two devices, they introduced the flow of fluorescent dyes, and they captured the mixing results with a UV light and a camera (Figure 2.12; Figure 2.13). They observed basic differences between spatially periodic and time periodic ducts. The PPM, a steady spatially periodic system, showed remarkably stable KAM islands under a variety of experimental conditions coexisting with chaotic advection. For EHAM, a time-periodic system, they observed that the KAM islands, which were initially created, were destroyed due to periodic injection of this unmixed area to the chaotic region.

Figure 2.12. Schematic experimental equipment for Kusch and Ottino (1992) experiments.
Figure 2.13. Experimental apparatus of EHAM and PPM. The inner cylinder for EHAM was a stainless steel rod with a diameter of 2.93 cm and length of 135 cm; for PPM the inside cross sectional obstacles were made of thin Pyrex plates (0.32 cm x 5.08 cm x 8.26 cm) (Kusch and Ottino, 1992).

*Periodic perturbation*

Active mixers have developed for micro-fluid continuous flow mixing in the laminar flow regime. This kind of micro mixer is based on the creation of time periodic or spatial periodic perturbation to create stretching and folding in the flow. Periodic perturbation can be
achieved using ellectrokinematical forces and magnetica field and pressure driven
perturbation (Lee et al., 2001; Bau et al., 2001; Niu and Lee, 2003; Nguyen and Wu, 2005).

Lee et al. (2001) introduced a mechanical chaotic mixer, where pressure perturbations
were applied from a pair of pressure reservoirs, which were connected in the channel T-pipe
transversally (Figure 2.14). The unsteady pressure drop, which is generated in the channel of
the pipe, creates a 2-D periodic flow at the channel, which leads to chaotic mixing (Lee et al.,
2001; Niu and Lee, 2003). Better mixing was obtained by increasing the number of side
channels and controlling the amplitude and frequency of perturbation. Numerical studies,
with the use of Poinceré section and Lyapunov exponents, showed that periodic pressure
perturbation will break up the KAM-tori close to the wall, and, by increasing the amplitude
and the frequency of perturbation, the stretching rate was enhanced, which resulted in the
increase of chaotic regions and reduction of the regular region.

The pressure driven source of perturbation can also be achieved by an electric or
magnetic field (Lee et al., 2001; Bau et al., 2001; Nguyen and Wu, 2005). Switching an AC
electric field, which contains charged or polarized particles, on and off periodically in the
flow can create dielectrophoresis (movement of neutral charged particles) for the electrical
field or Lorentz forces for the magnetic field. These forces can act like pressure driven
perturbation, creating a transversal time periodic flow, which results in chaotic particle
motion (Figure 2.15) (Lee et al., 2001; Bau et al., 2001; Nguyen and Wu, 2005).
Figure 2.14. (a) Schematic demonstration of experimental mixing with micro-mixer working with time periodic pressure perturbation, (b) Pressure driven sinusoidal perturbation mixer with spatial periodic mixing from different channels (Niu and Lee, 2003).

Figure 2.15. Schematic showing of experimental mixing with electrokinematical forces perturbation created from electrical field (Lee et al., 2001).
Rotated Arc Mixer

One of the most studied chaotic advection mixers is the Rotated Arc Mixer (RAM). RAM is a patented rotating device (Figure 2.16) that consists of two co-axial cylinders, one wrapped closely around the other (Metcalfe et al., 2004, Metcalfe et al., 2006; Metcalfe and Lester, 2009). A motor rotates the outer cylinder while the inner cylinder stays stationary. The gap between the two cylinders is 0.5–1% of the inner cylinder diameter, or 0.15–0.8 mm (Metcalfe et al., 2003, Metcalfe et al., 2006; Metcalfe and Lester, 2009). The inside cylinder has strategically, periodically located windows (apparatus), which expose the mainly axial flow to the drag of the rotation of the outer cylinder. In that way a transversal spatial periodic second flow is generated, which introduces a periodic reorientation of the main axial flow and generates the mechanism of chaotic advection (Figure 2.17) (Metcalfe et al., 2003, Metcalfe et al., 2006; Metcalfe and Lester, 2009). As in the case for a static mixer, RAM works better for low Reynolds number flows, Re< 100, and that is because secondary flow requires thick fluid to penetrate more in the cross section region of RAM. Otherwise, as the Reynolds number increases the transversal flow is limited to a thin layer close to the rotating boundaries (Metcalfe et al., 2006).
Figure 2.16. Rotated Arc Mixer (RAM) (Metcalfe et al., 2006).

Figure 2.17. Chaotic mixing of viscous glycerine-water mixture, with RAM (Metcalfe and Lester, 2009).
The mechanism of chaotic advection from RAM is based on periodic reorientation of the main axial flow. There are three parameters, two geometric (Figure 2.18) and one kinematic, that affect the mechanism of periodic reorientation and mixing efficiency produced by RAM (Metcalf et al., 2006). The two geometric parameters are the windows opening length, $\Delta$, and the angular offset, $\Theta$, between the two windows. The starting position of each window depends on $\Theta$ and is given by the angular position of the previous window plus $\Theta$, with $\Theta$ being either positive or negative. The kinematic parameter, as in the case of Kusch and Ottino (1992), is the linear stretching degree, $\beta$, created by transversal secondary flow (Metcalf et al., 2006).

Figure 2.18. Cross-sections surface view and geometric parameters of inside cylinder of RAM. Where $R$ is the inner radius of the inner stationary cylinder, $H$ is the window’s axial length, $\Delta$ is the angle of the window opening, and $\Theta$ is the angle by which each window is offset from its downstream neighbour (Metcalf et al., 2006).
Window length $A$ is the parameter that has been studied less on mixing. During experiments with RAM, a window opening of $A = \pi/4$ was usually selected, while $A$ lies to a range of values between $-360^\circ$ to $0^\circ$ (Metcalfe et al., 2006; Singh et al., 2008). Offset periodic angle $\Theta$ is an important designed parameter, which influences the mixing efficiency. The studies showed that a $\Theta$ of an opposite direction from the previous offset ($-\Theta$) will give a counter rotation protocol, which is the case in global chaotic motion, due to the breakdown of flow symmetries (Metcalfe et al., 2006; Speetjens et al., 2006). A usual value of $\Theta$, which was used in studies with RAM, was $\Theta = -\pi/4$, where $\Theta$ can take values from $-180^\circ$ to $180^\circ$ (Metcalfe et al., 2006; Singh et al., 2008). Shear thinning fluids require additional windows placed at different values of $\Theta$ than the basic apparatuses of windows used to produce global mixing for shear thickening and Newtonian fluids. Numerical results showed that an additional offset of $\Theta = \pi/2$ every 3 to 6 windows can result to global mixing for a shear thinning fluid with flow behavior index, $n$, equal to 0.29.

The degree of stretching, $\beta$, is adjustable during the operation of RAM and is given by the ratio of angular over the average axial velocity of the main flow (Equation 2.13). As in the case of Kusch and Ottino (1992), the mixing enhancement depends on $\beta$, with better mixing observed for higher values of $\beta$ (Metclafè, et al., 2006).

$$\beta = \frac{\Omega R}{\bar{U}}$$

(2.13)

Where $\Omega$ is the velocity of rotation, $R$ is the radius of the inner cylinder of RAM, and $\bar{U}$ is the average axial velocity of the flow (Metclafè, et al., 2006).
According to Metcalfe and Lester (2009), a heat exchanger (RAMex) can be built around the RAM. It is like a tube-and-shell heat exchanger, where the usual tube enclosed by the shell is replaced by the RAM (Metcalfe and Lester, 2009). Heat transfer on yoghurt, with the use of RAMex has been numerically studied by Metcalfe and Lester (2009). The numerical results were very promising for the enhancement of heat transfer in viscous food products, with less energy required compared to the energy required from static mixers. In their numerical experiments, they found that the heat transfer rate of RAMex was increased against the heat transfer rate of static mixers by six times and three times for Newtonian and non-Newtonian viscous product, respectively.

*Baffled tube mixer*

Another category of active chaotic advection mixers is the oscillatory flow in a tube with smoothly periodic baffled structures inside (Figure 2.19) (Ni and Gough, 1997; Zheng and Mackley, 2008). A typical system of an oscillatory baffled tube consists of a tube with baffled structures and the oscillating unit, which provides reciprocal pulsation (Zheng and Mackle, 2008). Mackley and Ni (1991) observed that just the oscillation of fluid did not create chaotic mixing. However, chaotic flow occurs with a combination of oscillatory flow in a baffled structure tube (Figure 2.20). The baffled structures, which are periodically placed inside the tube, combined with oscillatory flow can create chaotic advection through to a periodically reversing flow in the tube (Ni and Gough, 1997; Ni et al., 2003).
Figure 2.19. Configuration of two different oscillatory flow mixers (a) ring-shaped baffled tube and (b) meso-tube (Zheng and Mackle, 2008).
Figure 2.20. (a) and (b) Oscillatory flow, (c) regular no chaotic flow for oscillatory flow in a smooth wall column, (d) and (e) chaotic flow in an oscillatory flow in a baffled column (Ni et al., 2003).

*Pulsated Synthetic jet*

Pulsated synthetic jet formation is an active chaotic advection mixer, which can give well mixed viscous materials with low Reynolds number laminar flows. The secondary transversal synthetic jets create chaotic mixing with periodic out-phase folding and stretching, which increases the interfacial area and, thus, the radial mixing of main axial laminar flow (Xia and Zhong, 2012; Xia and Zhong, 2013). Synthetic jets are produced from
the periodic ejection and injection of fluid, from a piston (shaker). Synthetic jets are introduced transversally to the main flow through orifices located periodically and offset (usually with a difference of 180°) to create a pair of synthetic jets, resulting in mixing enhancement (Figure 2.21). The efficiency of mixing from synthetic jets depends on the penetration depth of synthetic jets, which is a function of the magnitudes of the jet velocity and the velocity of the flow parallel to the surface (Milanovic and Zaman, 2005). Xia and Zhong (2013) observed that at higher amplitude and frequency of pulsation better mixing results through the increase of the stretching (penetration depth of synthetic jets) and folding of the main flow.

Figure 2.21. Schematic demonstration of synthetic jet mixer (Xia and Zhong, 2013).
Parameters affecting chaotic mixing

Wall boundary conditions; No-slip wall and thermal boundary conditions

Chaotic mixing is the mixing produced by chaotic advection (mixing through exponential stretching of fluid elements) with the added mixing from diffusion (Aref et. al., 2014). The fluid elements characterized by some scalar quantities of concentration, temperature, and pressure, as the get stretched gradients of concentration, temperature, and pressure increase exponentially, results in the enhancement of diffusion. Typically the phenomenon of chaotic mixing decays exponentially, as uniformity of these scalar fields appears. Walls play an important role in mixing because no-slip wall conditions reduce the exponential stretching of chaotic advection. The no-slip boundary conditions formed at the wall lead all the points on a stationary wall to act as parabolic fixed-points from a dynamical system point of view. That results in the creation of closed streamlines near the wall, which reduce the advection from the wall zones and reduce the mixing efficiency for the whole mixed area. (Gouillart et al., 2007; Gouillart et al., 2008; Thiffeault et al., 2011; Aref et al., 2014). Moving walls have been experimentally proven to recover the dynamical system behavior close to the wall and recover the exponential stretching of fluid particles for chaotic mixing (El Omari and Le Guer, 2010 a; El Omari and Le Guer, 2010 b; Thiffeault et al., 2011). Studies with no-slip wall condition and thermal boundary conditions proved that through the use of an alternative rotation of wall boundaries, the development of closed streamlines near the wall can be avoided (El Omari and Le Guer, 2010 a; El Omari and Le Guer, 2010 b; Thiffeault et al., 2011).
Effects of fluid rheology

The rheological behavior of the fluid, Newtonian or non-Newtonian, has a distinct effect on chaotic advection application. Shear-thickening fluids are characterized by the most efficient mixing with application of chaotic advection, compared to Newtonian and shear-thinning fluids (El Omari and Le Guer, 2010 b; Metcalfe et al., 2006; Aref et al., 2014). Higher viscosity of shear thickening fluids close to the wall leads to higher driving force from moving walls (e.g., rotating walls (El Omari and Le Guer, 2010 b; Metcalfe et al., 2006), which generates larger vortices and increased mixing. On the other hand, shear-thinning fluids represent the most difficult cases for mixing, compared to Newtonian and shear-thickening fluids. Because the shear originates at the region close to the moving wall, the viscosity in this region decreases, and, thus, the transversal flow generates mixing to penetrate less deeply into the flow (Metcalfe et al., 2006; Aref et al., 2014). A result of this is the large, unmixed islands that appeared in the Metcalfe et al. (2006) Pionceré section with RAM (Figure 2.22).
Figure 2.2. Pionceré section results for RAM, with a shear-thinning fluid with flow behavior index, n, of 0.29. On the left side, the Pionceré section with the usual aperture used with RAM, with a big unmixed island formation. On the right Pionceré section with an extra offset window in RAM to get full mixing (Metcalf et al., 2006).

**Blending of fluids with high viscosity ratio**

Mixing of flow characterized by a high viscosity ratio is limited for regularly designed static mixers (Etchells III and Meyer, 2004). Elongational flow static mixers (Figure 2.23) are required. Due to the high viscosity ratio, the resistance at the interface between flow streams is very strong. In this kind of situation, the phenomenon, called the *miscible interface*, arises and blocks the mixing (Etchells III and Meyer, 2004). This phenomenon can arise due to significant differences in molecular weights, crystallinity, and from significant viscosity difference inside the flow. To overcome the resistance at the
interface, the mixer must be designed to operate at uniform shear stress (Etchells III and Meyer, 2004). According to Grace (1982), the application of the elongational shear has demonstrated more successful in fixing this problem as it makes the resistant layers of different viscosity thinner (Rauwendaal, 2014).

Figure 2.23. (a) Picture of Elongational Pin Mixer (EPM), (b) elongation flow created in EPM (Rauwendaal, 2014).
2.1.5 Chaotic advection and heat transfer

Turbulence flow is a very important factor in heat transfer application. Turbulence flow enhances the mixing close to the boundaries of the flow and results in the enhancement of the convection heat transfer coefficient due to increased temperature gradient close to heat transfer boundaries. In laminar flow, which is the characteristic flow regime for viscous food products, the temperature gradient at heat transfer boundaries is less than turbulence, and that makes heat transfer very slow. Furthermore, poor mixing of laminar flow leads to a wide temperature distribution inside the flow. Since the early 1990s, many numerical and experimental studies have proposed chaotic advection as a method for enhancing heat transfer and equalizing the temperature inside the fluid. These studies can be separated in two basic classes: the studies that used rotating cylindrical pipes for heat transfer and thermal mixing, and those related to studies of heat exchangers designed with twisted or bent pipes (Lefevre et al., 2003).

In the first category, most research is based on numerical studies. Most studies of the first class are focused on heat exchangers with two cylindrical rotating elements (Ghosh et al., 1992; Ganesan et al., 1997; Saatdjian and Leprevost, 1998; Lefevre et al., 2003; Rodrigo et al., 2006; Mota et al., 2007, Mota et al., 2008; Saatdjian, et al., 2011). The studies observed that heat transfer was enhanced due to cross-stream or transversal heat transfer created from chaotic advection. Parameters, such as eccentricity and protocol of rotation (e.g., frequency of rotation; counter or co-counter rotation of two cylinders), were affected the most with the heat transfer enhancement. The numerical results showed that the highest heat transfer enhancement rate was observed when the two cylinders where non-concentric,
with a periodic alternative rotation (Ghosh et al., 1992; Ganesan et al., 1997; Saatdjian and Leprevost, 1998; Lefevre et al., 2003; Rodrigo et al., 2006; Mota et al., 2007, Mota et al., 2008;). For the frequency of rotation, there was a specific frequency in which the maximum heat transfer was observed (Saatdjian and Leprevost, 1998; Lefevre et al., 2003; Rodrigo et al., 2006; Mota et al., 2007, Mota et al., 2008;). The same conclusion, a specific rotation frequency with the maximum heat transfer, was observed also for rotating cylinders with an elliptical cross section (Mota et al., 2007). A different geometry with rotating elements using two cylindrical rods inside a cylinder, named a two rod-mixer, was used for studies of thermal mixing and heat transfer (Figure 2.24) (El Omari and Le Guer, 2010 a, El Omari and Le Guer, 2010 b; Le Guer and El Omari, 2012). For the two rod-mixer, the best heat transfer enhancement and thermal mixing were observed with a discontinuous alternative periodic rotation protocol. Finally, numerical research studied a heat exchanger based on RAM, as a part of tube-and-shell heat exchanger (RAMex) in which the usual tube enclosed by the shell is replaced by the RAM (Metcalfe and Lester, 2009). According to Metcalfe and Lester (2009), numerical results were very promising for the enhancement of heat transfer in viscous food products, such as yoghurt.

More experimental studies have been done for the second category, where the overall heat transfer enhancement in laminar flow regime was examined for different values of Reynolds number using chaotic advection (Morkani et al., 1997; Lefèvre et al., 2003). Using the concept of chaotic advection, shell-tube heat exchangers, chaotic heat exchangers, were designed and tested, using as the shell part of the heat exchangers multiple twisted and bend pipes with cylindrical and cubic cross sections (Figure 2.25).
Due to the change in the curvature of pipes, secondary Dean-toll flows are generated; combined with periodic change of geometry of flow, chaotic flow is produced, which leads on heat transfer enhancement (Acharya et al., 1992; Peerhossaini et al., 1993; Acharya et al., 2001; Changy et al., 2000; Lefevre et al., 2003; Kumar and Nigam, 2007; Kumar et al., 2007). Numerical studies based on the different modulation multiple twisted and 90° bend pipes (Figure 2.26) were done to better design these heat exchangers. The research was
focused on the effect of different periodic switching angles between the bend pipes. Pionceré section studies showed that for switching angles less than 45°, switching angles between the bend pipes, large unmixed islands formed—in contrast with higher than 45° switching angles (between 45°-180° same results for Pionceré section), where the unmixed area was significant decreased (Yamagishi et al., 2007). For various Dean numbers (Dn=50-400) studies, a switching angle of 135° showed the best results, with a small unmixed area for Pionceré section and with the highest mean Lyapunov exponents (Yamagishi et al., 2007). Experiments were performed to compare the efficiency of heat transfer, using a chaotic heat exchanger consisting of 90° bend pipes with a 90° switching angle against a helical heat exchanger and a regular tube in tube heat exchanger, for a large range of Reynolds numbers (from 30 to 30,000) for Newtonian fluids (Figure 2.26; Figure 2.27). The results showed an improvement of 15-20% on heat transfer for the chaotic heat exchanger; for low Reynolds number, Re<70, flows (Changy et al., 2000) above a Re>1000, no difference between helical and chaotic heat exchanger configurations was observed. The greatest enhancement on the overall heat transfer coefficient for Changy et al. (2000) chaotic configuration heat exchanger was observed for a Reynolds number around 250. Chaotic configuration heat exchangers showed both numerically and experimentally the creation of strong chaotic flow— independently for the geometry of the cross section, circular or cubic (Peerhossaini et al., 1993; Changy et al., 2000; Zheng et al., 2013). The most important designed parameter of chaotic advection heat exchangers is the periodic switch angle between twisted or bend pipes, which affects the efficiency of mixing and heat transfer (Yamagishi et al., 2007).
Figure 2.25. Multiple 90° pipe bends with different periodic switching angle; (a) 45°, (b) 180°, (c) 90°, (d) 0° (Yamagishi et al., 2007).
Figure 2.26. (a) Chaotic configuration heat exchanger, with 33 $90^\circ$ bend pipes and a $90^\circ$ periodic switch angle between the bends (Changy et al., 2000); (b) Chaotic configuration heat exchanger, with four $90^\circ$ bend pipes with a cubic cross section and a $90^\circ$ periodic switch angle between the bends (Peerhossaini et al., 1993).
2.1.6 Chaotic flow produced by pipe vibration on continuous flows

Recently, an increased number of studies have shown interest in the influence of pipe vibration on flow behavior, heat transfer, and mixing of viscous materials through the production of chaotic fluid motion. Numerical and experimental results have observed that the effects of vibration motion are generally influenced by the type of vibration, longitudinal, transversal, or rotational (Figure 2.28), the rheological properties of fluid (Newtonian and non-Newtonian) and vibration parameters, amplitude of vibration (A), and the frequency (f) of vibration.
2.1.6.1 Flow enhancement of viscous fluids with pipe vibration

Vibration has been investigated as a method for flow enhancement of viscous Newtonian and non-Newtonian isothermal fluids. Experimental and numerical (simulation with Computational Fluid Dynamics (CFD)) results of superposed longitudinal and rotational vibration showed that the time-average flow rate behavior depends on flow behavior. More specifically, the flow rate of Newtonian fluids (n=1) was unaffected, the flow rate of shear thinning fluids (n<1) had a considerable enhancement, and flow retardation was observed for
shear thickening fluids (n>1) (Deshpande and Barigou, 2001; Eesa and Barigou, 2008). Similar results for the flow enhancement of Newtonian and non-Newtonian fluids (shear-thinning fluids) were observed by the Shin et al. (2003) experiments with a transversal acoustic vibration in a capillary tube on Newtonian fluids and human blood. Wall vibration introduces an extra applied shear, which leads in reduction of apparent viscosity (Deshpande and Barigou, 2001; Eesa and Barigou, 2008). In addition, this study applied shear leads on enhancement of average flow rate for shear-thinning (pseudoplastic) fluids and an increase in flow resistance for shear thickening (dilatant) fluids (Deshpande and Barigou, 2001; Eesa and Barigou, 2008). Moreover, Shin et al. (2003) concluded that transverse acoustic vibration increased the flow rate, creating a particle-free layer near the wall (Figure 2.29). According to Shin et al. (2003), particles, having higher inertia than fluid, were forced to move into the center layer, resulting in a particle-free layer close to the wall. The enhancement rate of a shear thinning fluid increased within the range of studied vibration frequency (0-40 Hz) and amplitude (0-4 mm); however, a different combination of frequency and amplitude gave the same results of the peak acceleration ($A\omega^2$), resulting in similar flow enhancement (Deshpande and Barigou, 2001). For rotational vibration, Eesa and Barigou (2008) observed no enhancement of flow after a frequency of 300 Hz. Furthermore, the enhancement effects became less significant for larger tube diameters, higher external pressure (Deshpande and Barigou, 2001), and higher flow rates (Shin et al., 2003). Eesa and Barigou (2008) concluded that the enhancement obtained from longitudinal vibration was greater than rotational, with the lowest being the transversal vibration. Mechanical vibration at the sonic frequencies scale had a limited effect on low and moderate viscous fluids, but high viscous fluids ultrasonic
vibration was very successful at enhancing flow (Piau and Piau, 2002; Eesa and Barigou, 2008). For all viscous materials, flow remained in the laminar regime, with the Reynolds number less than 100, for vibrated flows at sonic range of frequency and less than 400 in the ultrasonic region (Eesa and Barigou, 2008).

Figure 2.29. Experimental set up of acoustic transverse vibration in a capillary tube for studies on flow enhancement on Newtonian fluids (water, silicone, oil) and human blood (Shin et al., 2003).

2.1.6.2 Vibration effects on radial mixing; temperature uniformity

Eesa and Barigou (2011) composed a CFD study on the effects of a transversal sinusoidal vibration at the pipe wall on a non-isothermal fluid. Eesa and Barigou (2011) observed that pipe wall vibration induced a vigorous swirling fluid motion with strong
vorticity field and complex spiraling fluid streamlines and trajectories. This chaotic fluid motion had a substantial impact on the radial temperature distribution of fluid and on the mean heat transfer coefficient. By increasing amplitude (1.0, 1.5 and 2.0 mm) and frequency (0-75 Hz), radial temperature distribution presented a better profile. However, in contrast to flow enhancement, which is a function of both vibration frequency and amplitude, radial mixing and temperature distribution depended more on amplitude of vibration (Eesa and Barigou, 2010). A uniformity of 90% was observed when the ratio (A/D) of vibration amplitude (A) with the pipe diameter (D) was A/D=0.1 and the frequency was at 75 Hz (Eesa and Barigou, 2010). Less viscous products showed a better uniformity of the temperature profile after vibration; more viscous products required more energetic vibration, with higher amplitude and frequency (Eesa and Barigou, 2010). Eesa and Barigou (2011) concluded that the strong chaotic flow induced by vibration reduced the thermal entrance length by making the temperature profile develop very rapidly and also reduced fouling by promoting cleaning close to the pipe wall.

2.2 Acoustics

An active method that can produce chaotic fluid motion or micro-turbulence fluid motion close to the boundaries of the flow is acoustics. In this part of this review, an introduction to acoustics dynamics, produced in a fluid medium from acoustic wave, is provided, and the way these dynamics influence mixing and heat transfer is discussed.

2.2.1 Acoustic wave

Acoustic wave is a mechanical sine wave, which is propagated through an elastic medium (fluid or solid) by the vibration of molecules. Acoustic wave can be described as the
passage of pressure fluctuation (compressions and rarefactions), which creates the oscillation of the elastic medium (Raichel, 2006). An acoustic wave is characterized by the frequency, the wavelength, and the amplitude. The quantification of sound is determined by the acoustic power and the acoustic intensity (Raichel, 2006).

**Wavelength** ($\lambda$) is the distance between two areas of maximal compression or rarefaction (Raichel, 2006).

**Frequency** ($f$) is the number of wavelengths that pass per unit time. It is measured as cycles (or wavelengths) per second (Hz). Depending on the frequency, acoustic waves are categorized by infrasonic (0-20 Hz), sonic (20 Hz-20 kHz), and ultrasonic (20 kHz-1 MHz) (Raichel, 2006).

**Amplitude** ($A$) is the height above the baseline and represents maximal compression. It is expressed in decibels, which is a logarithmic scale (Raichel, 2006). Amplitude is a measure of energy. The more energy carried an acoustic wave, the higher its amplitude (NDT, 2013).

**Acoustic power** ($P$) is the amount of acoustic energy generated per unit of time (W) (Raichel, 2006; NDT, 2013).

**Acoustic intensity** is the power density or concentration of power within an area (W/m$^2$) (Raichel, 2006; NDT, 2013).

### 2.2.2 Acoustic vibration

**Acoustic streaming**

When sound waves pass through a fluid, they are attenuated and lose energy. The loss of energy from the sound waves is absorbed and dispersed in the medium. At low frequencies
of sound, there are linear acoustic waves. When a medium is acted upon by linear acoustic waves, the medium does not move but, rather, oscillates. This oscillation transmits acoustic energy (Uchida et al., 1995). When high frequency sound waves propagate, nonlinear acoustic waves appear. Nonlinear sound waves are attenuated, because of the viscosity and the inertia of the medium, resulting in a pressure gradient along the wave propagation direction. This gradient of pressure creates the shelf for the medium to move. This phenomenon is called acoustic streaming (Uchida et al., 1995; Ro and Loh, 2011).

### 2.2.3 Acoustic Cavitation

Acoustic waves, which have a frequency higher than the upper limit of human hearing range (16 kHz - 20 kHz), are called ultrasound (20 kHz - 1 MHz). Ultrasound is responsible for generating many physical effects, such as shock waves, microjets, turbulence, and shear forces (Ashokkumsr, 2011). When an ultrasound wave propagates through a fluid, it may cause two phenomenon: acoustic streaming and acoustic cavitation (Legay et al., 2011). As it has been described from many authors, acoustic cavitation is the major phenomenon that appears from the propagation of ultrasound into a fluid (Neppiras, 1980; Cai et al., 2010; Legay et al., 2011). Cavitation is the formation, growth, and collapse of microbubbles (the bubble size is about $10^{-4} \text{m}$) within a liquid (Figure 2.30) (Neppiras, 1980; Lauterborn and Ohl, 1997; Leighton 1998). A sinusoidal varying pressure, which is imposed from the sound wave onto the already existing cavitations of the liquid, is responsible for the formation, growth, and collapse of the microbubble. Analytically, when the existing vapor microbubbles, known as weak spots in the liquid or solid particles, are exposed to negative pressure cycle, they start oscillating, and they are gradually pulled apart from the liquid.
(Leong et al., 2011). The instantaneous pressure decreases (sufficiently below the vapor pressure) to such an extent that the microbubbles cannot remain stable. Therefore, the microbubbles increase their volume to a new equilibrium value. The number of the microbubbles that are produced depend at the proportion of the density of the weak spots of the liquid (Leong et al., 2011). The bubble size grows because of two factors: First, for a single microbubble, the increase of the bubble size is a result of oscillating sound waves. With the oscillation of the bubble, the bubble's size increases. A bubble grows until it reaches a critical size, called resonance size. Resonance size is critical for the microbubbles. There are two possibilities after a microbubble reaches the critical resonance size: the bubble collapses (transient cavitation), or the bubble continues oscillation, approximately near the resonance size (stable cavitation). The second reason for the bubble growth is the coalescence of bubbles. When two bubbles meet each other, they combine to form a large bubble (Ashokkumr, 2011; Leong et al., 2011). It is important to explain that the generation of bubbles after 100 kHz is decreased and the cavitation effect above the 1 MHz becomes weaker (Leong et al., 2011).
2.2.3.1 Resonance frequency for cavitation

The collapse of cavitation bubbles is related to the critical resonance size. The resonance size of the bubbles depends on the applied acoustic frequency (Legay et al., 2011). When the acoustic frequency is close to a system's natural frequency of vibration, the system has the tendency to absorb more energy. The tendency of a system to oscillate at maximum amplitude is called resonance (Kinsler et al., 1982). The frequency of resonance occurs at very large amplitudes of oscillation and causes the collapse of the bubble. The critical bubble size $R_c$ can be approximately calculated using:

$$R_c = \sqrt{\frac{3\gamma p_{\infty}}{\rho \omega^2}}$$

(2.14)
Where $\gamma$ is the specific heat ratio of the gas within the bubble, $p_\infty$ is the ambient liquid pressure, $\rho$ is the liquid density, and $\omega$ is the angular frequency of ultrasound (Legay et al., 2011).

Eq. 2.14 shows that for small ultrasound frequency (16-100 kHz), large bubbles are formed, which are responsible for the formation of strong microjets, shear forces, shock waves, and turbulence. Above the 100 kHz, the bubble size is much smaller and the bubble's collapse generates high temperatures within the bubble (2000 K - 5000 K) and highly reactive radicals (Ashokkumar, 2011; Legay et al., 2011).

2.2.3.2 Jet formation and shock wave from acoustic cavitation

A characteristic of cavitation bubbles is their exceedingly fast collapse, which can result in the creation of liquid jets, shock waves, and even light (sonoluminescence). The energy from a microbubble collapse is converted into kinetic energy in the form of a liquid jet and shock wave (Brujan et al., 2005). As Lauterborn and Claus-Dieter Ohl (1997) illustrated, a flat, solid surface nearby causes the bubble to involute from the top (surface below the bubble) and to develop a high-speed liquid jet towards this solid surface. When the jet hits the opposite bubble wall from the inside, it pushes the bubble wall ahead, causing a funnel shaped protrusion with the jet inside. When a bubble collapses, a shock wave is created; this shock wave reflects off the wall and then reflects off the inside surface of the bubble. Thus, a high-speed liquid jet (jet shock wave) develops, with gas and vapor, towards this solid surface (Figure 2.31). The jet, which is been created, has decays instantly into many other microbubbles. This results in the continuation of the acoustic cavitation phenomenon, where the new microbubbles act as new creation points.
The combination of many liquid jets and jet shock waves is responsible for the creation of one outgoing shock wave (cavitation bubble cloud) (Lauterborn and Claus-Dieter Ohl, 1997; Legay et al., 2011).

Shock waves have some very interesting characteristics in relation to the maximum radius of the bubble and the distance of shock waves from the solid surface. Increasing the maximum radius of the bubble increases shock wave amplitude and duration. Additionally, Shaw and Spelt (2010) observed that the decay of shock wave formation occurs far from the solid surface.
2.2.4 Acoustic wave and boundary layer

2.2.4.1 Acoustic streaming and turbulence flow

Rayleigh was the first to study the phenomenon of acoustics near the solid boundary surface. In the literature, the acoustic streaming, which developed into boundary layers, is called Rayleigh streaming. As Riley (1998) mentioned to his research, Reynolds stresses are responsible for the creation of Rayleigh streaming. Acoustic streaming generates turbulence due to the wave dissipation and the Reynolds stress (Figure 2.32). Where the wave dissipation is created, so too is the decrease of acoustic energy created (Nyborg, 1958; Monnier et al., 1999). The turbulence flow is developed from the acoustic streaming and can enhance the convection heat transfer (Figure 2.33) (Legay et al., 2011).

Figure 2.32. Picture of acoustic streaming generated from a probe horn (Monnier et al., 1999).
2.2.4.2 Acoustic cavitation and turbulence flow

When a bubble collapses, microturbulence flow appears due to the formation of a liquid jet and shock wave. The turbulence intensity near the wall may break down the viscous boundary layer (Figure 2.34). The increase of turbulence in the near-wall boundary layers leads to the significant enhancement of convection heat transfer (Lee and Choi 2002; Nomura et al., 2002; Cai et al., 2008).
2.2.5 Heat transfer enhancement by acoustic streaming and acoustic cavitation

The bubble collapsing is an instant physical reaction, which takes place in a matter of microseconds (depending on bubble size and frequency). Because of that, the magnitude of particle displacement velocity during bubble implosion can be estimated at about 100 m/s (Legay et al., 2011). According to Legay et al. (2011), the velocities of microturbulence from acoustic cavitation are two or three orders of magnitude greater than the velocities of microturbulence caused from acoustic streaming. Therefore, acoustic cavitation can be considering the major factor for the enhancement of heat transfer by ultrasound.

From the literature, we can also conclude that acoustic cavitation plays a major role in heat transfer enhancement by ultrasound. Cai et al. (2010), in their research, compared the experimental results of heat transfer enhancement by acoustic streaming with the
experimental results of heat transfer enhancement by acoustic cavitation. The results indicate that the enhancement ratio (comparing heat transfer coefficients with and without cavitation) of heat transfer with mechanical vibration from acoustic, without the phenomenon of cavitation, was 51% (Lemlich and Hwu, 1961) and 40% (Bergles and Newell, 1965) against a 300% enhancement ratio with acoustic cavitation in Nomura et al. (2002).

2.2.5.1 Heat transfer enhancement by ultrasound vibration

Ultrasound vibration of surfaces and ultrasound vibration of fluids are new methods for enhancement of heat transfer (Bergles, 2011). When a surface or a fluid is vibrated by ultrasound, high-intensity acoustic waves appear (Loh et al., 2002). The propagation of high-intensity acoustic streaming and cavitation have a great impact on the thermal boundary layer and, thus, on enhancement of heat transfer (Nomura et al., 2002; Legay et al., 2011).

One of the first interesting studies with ultrasound vibration was that of Bergles and Newell (1965). In their study, Bergles and Newell investigated the influence of high intensity ultrasound vibration on heat transfer to heated water. They found an increase of 40% in heat transfer coefficients, and they concluded that the influence of ultrasound on heat transfer depends on the conditions of flow and the extent of cavitation in the annular gap. Another interesting experiment was that of Wong and Chon (1969). They obtained the influence of the critical sound pressure on convection heat transfer, and they found a remarkable increase on heat transfer coefficients in the natural convection when the pressure was above the critical sound pressure. However, for approximately 30 years until the early 2000's, there are few published papers on the ultrasound effect on heat transfer. However, in the last ten years, research has begun to explain the enhancement of natural convection by ultrasound and the
parameters that effect this (Legay et al., 2011). The next paragraphs describe, in chronological order, the most important experiments on heat transfer enhancement by ultrasound in liquids.

Nomura et al. (2002) observed that the acoustic jet can increase the heat transfer coefficients for both tap and indigassed water. They illustrated the turbulence intensity from microjets, and they concluded that the high intensity of turbulence was close to the heated surface. Additionally, Nomura et al. (2002) found an approximate enhancement of 300% in heat transfer coefficients when the distance between the horn and the heated surface was at 15 mm. They also observed that the heat transfer coefficient increased at higher ultrasonic power; however, the heat transfer coefficient of water remained stable at an ultrasonic power of 15W.

Zhou et al. (2004) investigated the effects of sound source intensity, vibrator location, sound distance, fluid temperature, and thermo physical properties of working fluid on convective heat transfer from a horizontal tube. They observed a significant enhancement of heat transfer when the sound source worked at a high intensity and when the vibrator was located at the center, 20 mm above the tube. For a constant heat flux, they recorded an enhancement of heat transfer at low intensity of 0.5 A. Zhou et al. (2004) worked on three different fluids: acetone, ethanol, and water. They observed that the heat transfer depended on the thermo physical properties of the working fluid. They concluded that the amount of cavitation bubbles increased when the Prandtl number, the saturation vapor pressure, and the temperature of working fluid increased.
Nomura et al. (2005) studied the effect of ultrasonic heat transfer enhancement with an obstacle in front of the heating surface. They observed that the ultrasound wave was easily propagated through the plate, and they concluded that the enhancement of heat transfer depends on the material and the thickness of the plate. They also observed the highest increase in heat transfer coefficients when the plate was made of acrylic and metallic aluminum and when the thickness was less than or equal to 5 mm.

Jeong and Kwon (2006) studied the effect of ultrasound (40 kHz) on critical heat flux, under natural convection conditions, of a subcooling (5, 20, 40 ºC), pre-heated, distilled water bath. They observed that the critical heat flux increased due to the fluid mixing from acoustic streaming and acoustic cavitation. Furthermore, they found that the highest rate of critical heat flux augmentation occurred when the heated surface position was facing downward toward the ultrasound vibrator, with an enhancement ratio of 13% against 4% when at vertical position.

Baffigi and Bartoli (2009) investigated the heat transfer enhancement from ultrasound (40 kHz) waves in subcooling boiling condition for three different generator ultrasonic output powers: 300, 400, and 500 W. They observed that acoustic cavitation was responsible for the enhancement of heat transfer, with a maximum enhancement of 45% when the ultrasound power was at 500W and the subcooling degree at 35 ºC.

Cai et al. (2010) studied the influence of acoustic cavitation on convective heat transfer coefficients in a horizontal circular tube. Furthermore, they experientially investigated the influence of operating distance, cavitation intensity, and heat flux on heat transfer. They operated the ultrasonic transducer in resonance frequency (18 KHz) to get the
strongest vibration and the highest intensity of cavitation. High cavitation intensity caused an increase in the enhancement ratio of heat transfer. For low heat flux, heat transfer enhancement had a better augmentation. Concerning the location of the acoustic source, Cai et al. (2010) observed that the higher enhancement ratio of heat transfer occurred when the location of that ultrasound transducer was over the midpoint (central location) and at a distance of 15 mm from the horizontal tube.

2.2.5.2 Heat transfer enhancement by ultrasound in heat exchangers

Over the past ten years, many researchers have tried to investigate the extension of hydrodynamic phenomena from ultrasound to heat exchangers. The first results were very promising; the use of ultrasound appears to be an innovative method for the enhancement of heat transfer in heat exchangers. Kurbanow and Melkumov (2003) found an enhancement of 20% on heat transfer in a refrigeration heat exchanger system, and they observed the contribution of cavitation microjets on disruption of the thermal boundary layer. Furthermore, Monnot et al. (2007) studied the effect of ultrasound on cooling between water inside a vessel and water flowing in a coil. Monnot et al. (2007) investigated the effect of high frequency ultrasound, up to 1600 KHz, and they observed an augmentation in overall heat transfer coefficient up to 2.04 times, at 800 KHz.

Ultrasound effect on Tube-Tube Heat exchanger (Vibrating Heat exchangers)

Several new studies have investigated the performance of heat transfer in tube-tube heat exchangers, due to ultrasound vibration. This new type of heat exchanger, assisted by acoustic vibration, can be found in the literature as Sono-Exchanger (Legay et al., 2012c). Two types of tube-tube heat exchanger were studied:
i) Shell-and-tube, where the inner pipe had a U shape (Gondrexon et al., 2010; Yao et al., 2010; Legay et al., 2012 a).

ii) Tube-Tube, with two concentric straight pipes (Legay et al., 2012 a; Legay et al., 2012 b)

All the studies worked at high intensity ultrasound conditions with water as working fluid, where cold water was flowing in the annular pipe and hot water in the inner pipe.

*Ultrasound effect on shell-and-tube heat exchanger*

The results of heat transfer improvement are very promising for the shell-and-tube heat exchanger. However, the enhancement ratio of heat transfer depends on different parameters.

Flow rate is a very important factor in ultrasonic heat transfer enhancement. Lower flow rate for both annular flow and inside flow increases the augmentation influence of ultrasound (Gondrexon et al., 2010, Yao et al., 2010), with the augmentation rate becoming more sensitive in annular side flow (Gondrexon et al., 2010).

The hot water (inner tube) temperature affected the enhancement ratio. In general, increasing temperature decreases the viscosity but increases the vapor pressure, which is important for the formation of bubbles in cavitation. Yao et al. (2010) found an ideal temperature of liquid could exist where higher intensity cavitation and thus higher heat transfer enhancement can be achieved. For, Yao et al. (2010) that was at 55 °C. Yao et al. (2010) studied the influence of ultrasound power on heat transfer. They observed that the heat transfer enhancement ratio increased with the increase of ultrasound power, with a 13% and 17% enhancement ratio for 60 W and 100 W respectively. For ultrasound power lower
than 40 W, the enhancement of heat transfer can be considered negligible, only 3% (Yao et al., 2010).

The heat transfer coefficients in the Yao et al. (2010) experiment was approximately up to 1130 W/(m$^2$K), instead of 1700 W/(m$^2$K) as in Gondrexon et al. (2010), with both at 100 W generator output power. This difference could be explained by the different set up of the two heat exchangers (i.e., diameter of inner shell tube, diameter of outer pipe, material of pipes, etc.) (Figure 2.35; Figure 2.36). The enhancement of heat transfer presented a greater ratio when the ultrasound vibrator was located at the center of the heat exchanger and the diameters of tubes were smaller (Gondrexon et al., 2010).

Figure 2.35. Shell-and-tube structure of Yao et al. (2010).
Ultrasound effect on tube-tube heat exchanger

The first experiment with tube-and-tube heat exchanger presented a slightly lower enhancement ratio of overall heat transfer than that of shell-and-tube heat exchanger (Legay et al., 2012 a). Legay et al. (2012 b) changed the setup of their first experiment (Legay et al., 2012 a) by decreasing slightly the internal diameter of inner tube to ensure a better stability of the generator that supplies ultrasonic power to the system (Figure 2.37). Legay et al. (2012 b) observed a similar increase in the heat transfer coefficient to the shell-and-tube heat exchanger (Gondrexon et al., 2010), at 257%. In addition, the ultrasound effect in heat transfer showed a higher enhancement in lower flow of cold water (outer tube).
Figure 2.3. Tube in tube heat exchanger, (a) Legay et al. (2012 a), (b) Legay et al. (2012 b), all the dimensions in mm.

For the general behavior of the tube-tube heat exchanger, Legay et al. (2012 b) concluded:

- The parallel-flow or counter-flow did not affect any notable difference in the overall heat transfer enhancement.
- The highest heat flow was observed in counter-flow because of the highest temperature difference on tube-tube heat exchanger.
- The influence of ultrasound on heat transfer decreased when the transition/turbulent flow was \( \geq 3 \) L/min.
- Acoustic streaming effect was insignificant in heat transfer because the fluids was already flowing.

- Ultrasound vibration and cavitation contributed to heat transfer by disrupting the thermal boundary layer of annular cold water flow (Figure 2.38).

- The thermal boundary layer of the internal pipe and internal pipe side (hot water) (Figure 2.38) estimated that the energy balance of the system was not affected by the ultrasound wave or the vibration. Possible explanations for this behavior include the material and the thickness of the inner tube (0.5 mm, stainless steel).

Figure 2.38. Illustration of ultrasound effects: heat transfer enhancement inside the heat exchanger (Legay et al., 2012 b).
2.2.6 Ultrasound in viscous products

In the literature, only a few studies were found on heat transfer of viscous products assisted by ultrasound. However, there is a number of studies involving micromixing of viscous products or microfluids (with a very small Reynolds number) with ultrasound application. The research showed that acoustic cavitation could significantly enhance the micromixing (Wang et al., 2012). The tendency of bubble formation due to cavitation is increased at a low Reynolds number (Lee and Choi 2002). However, the transition of microturbulence, which is formed from a bubble collapsing, is more difficult in viscous products than in water (Monnier et al., 1999). Nevertheless, for low ultrasound frequencies (20 KHz), studies showed that high viscosity products mixing with ultrasound could be as efficient as mechanical mixers with less required energy (Monnier et al., 1999).

Experiments on convective heat transfer between metal aluminium particles and fluid of different concentration/viscosity, showed a significant enhancement of heat transfer coefficients (Lima and Sastry, 1990). Lima and Sastry (1990) used a rectangular ultrasound batch type tank at 1000 W ultrasound output power, and they observed the enhancement of heat transfer between particles and liquid due to sonic agitation. They showed that for lower viscosity fluids, the heat transfer enhancement was higher than for high viscous fluids. Lima and Sastry (1990) investigated the influence of viscosity by using a different concentration of Carboxy methycellulose. They observed higher overall heat transfer coefficients, up to 2100 W/m² °C for lower concentrations, against 700 W/m² °C for high concentration solutions. Lima and Sastry (1990) also noticed that the position of particle in the batch influenced the heat transfer. For lower viscosity fluid, the heat transfer enhancement was greatest at the
center than in the corners of the vessel. As the fluid viscosity increased, the differences in overall heat transfer coefficients between center and corner was reduced.

2.2.7 Ultrasound in food processing

Ultrasound technology is not a new application in the food industry (Mason et al., 1996). The food industry takes advantage of chemical reactions and physical forces, which are generated from ultrasonic waves, to improve food processing for many food products (Chandrapala et al., 2012). Ultrasonic waves are used for producing food emulsions, tenderizing meet products, filterating food ingredients, inspecting food packages, in drying, in homogenization, in freezing, in dairy processing, and in several other applications (Mason et al., 1996, Chandrapala et al., 2012).

Experiments on improving the rheological properties of food products (corn, potato, and waxy starch) by controlling the viscosity with ultrasound and heat have shown very promising results. The modification of viscosity through the use of ultrasonic technology has been found to be possible. The ultrasound technology has made the process simple and rapid, with no chemicals or additives and with minimal changes in physicochemical structure of the food system (Iida et al. 2008; Chandrapala et al., 2012). The modification of viscosity depends on the ultrasonic frequency, the temperature, and the time of treatment (Zou et al., 2009). Zou et al. (2009) worked at 211 kHz for different temperatures. At low temperature (25 ºC), they did not observe any difference between the viscosity of sonicated and non-sonicated samples. For higher temperatures (up to 65 ºC) close to the onset temperature of gelatization, they observed a significant reduction in viscosity of waxy rice starch after 60 min of treatment (Zou et al., 2009). Similar results were also obtained by Iida et al. (2008).
They observed a reduction on viscosity of starch (potato, tapioca, sweet potato, and corn starch) solutions, after gelatinization, when up two orders of magnitude after 30 min of treatment. They also studied the influence of different ultrasonic frequencies (183, 143, 99 and 44 kHz) and found a greater decrease in viscosity at lower frequencies. The decrease in viscosity of starch is based on the fact that the starch granule can be degraded by cavitation forces (Jambrack et al., 2010). The particle size measurements of the heated and sonicated granules were smaller to that of the heated, non-sonicated starch granule (Iida et al. 2008; Zou et al., 2009). Moreover, scanning electron microscopy tests showed no effect on the starch granule surface, and the size exclusion chromatography did not show any reduction in the size of the starch molecules from ultrasound treatment (Zou et al., 2009).

Ultrasound technology has a great influence on the freezing rate of food products (Zheng and Sun, 2006, Delgado et al., 2009). The enhancement of freezing rate by ultrasound can improve product quality. Sun and Li (2002) found an improvement in the structure of potato tissue using ultrasound at 15.85 W for 2 min. Higher ultrasound power input and time of treatment has negative results on the quality of potato tissue due to the formation of larger crystals (Sun and Li, 2002).

2.2.7.1 Non-thermal and thermal treatment with ultrasound

The physical (strong shear forces) and chemical (free radians) effects of acoustic cavitation, with or without heat and pressure, have been used for inactivation of microbial (bacteria) and enzymes (e.g. Pectinmethylesterase (PME) and polygalacturonase (PG)) (Cullen et al., 2012).
High ultrasound frequencies, usually up to 500 kHz, are used for the inactivation of pathogen and spoilage bacteria. The formation of free radicals, hydrogen peroxide, shear forces, and the gradient of pressure drop can affect the damage of cell of bacteria (Cullen et al., 2012). High ultrasound frequencies can be used as a non-thermal processing agent, which is capable of achieving the 5-log for food-borne pathogens requirement of the FDA (Cullen et al., 2012).

The inactivation of enzymes with ultrasound needs prolonged exposure periods (Cullen et al., 2012). However, the combination of mild heat, pressure, and ultrasound (at low frequencies 20-100 kHz), *manothermosonication* and *thermosonication*, can achieve higher rates of inactivation for enzymes than thermal processing at similar temperatures (Burgos et al., 1999; Wu et al. (2008); Cullen et al., 2012).

### 2.2.7.2 Quality parameters of food products after ultrasound

Ultrasound treatment (depending on frequency, power, amplitude of ultrasound and process time), could possibly influence quality parameters of liquid foods, like color, anthocyanins, and ascorbic acid. However, these changes are considered too small and, thus, non-problematic (Mason et al., 2005; Culles et al., 2012). Furthermore, research found that quality parameters, like cloud stability (quality parameter for color, flavor, and mouthfeel) in fruit juice, have improved after manothermosonication (Knorr et al., 2004; Culles et al., 2012).

*Color changes after manothermosonication*

Gomes-Lopez et al. (2010) observed some instrumental changes in color of orange juice containing calcium after ultrasound treatment at 20 kHz, for 6 min at ultrasound
amplitude of 89.25 μm. Therefore, these changes were observed by sensory evaluation based on panel-tasted, color aroma, and flavor (Gomes-Lopez et al. 2010; Culles et al., 2012). Experiments with ultrasound (20 kHz, 1500W) on freshly squeezed orange juice samples presented a significant effect on the browning index and color parameters (with no significant changes in pH and °Brix) (Tiwary et al., 2008). Tiwary et al. (2008) explained these results as being caused by cavitation. Acoustic cavitation could effect color degradation of fresh orange juice, such as accelerating chemical reactions, increasing diffusion rates, dispersing aggregates, or breaking down susceptible particles, such as enzymes and microorganisms (Tiwary et al., 2008). Physical conditions produced from acoustic cavitation accelerated the carotenoid isomerization, leading to color degradation of fresh orange juice (Tiwary et al., 2008).

Ultrasound effects on viscosity of tomato paste and juice

The viscosity of tomato juice and paste is related to pectin molecules and pectin particle size, with large particle size determining high viscosity (Vecret et al., 2002; Wu et al., 2008). Reducing the particles size to one point decreases the viscosity. However, after this point, the viscosity starts increasing again due to the greater interaction between the particles (Vecret et al., 2002). The rheological characteristics of tomato juice and paste improved after manothermosoniction (20 kHz, 2 kg pressure, 117 μm amplitude and 70 °C) (Vecret et al., 2002) and thermosonication (24 kHz, 400 W, 25, 50 and 75 μm amplitude, 60, 65 and 70 °C) (Wu et al., 2008). Acoustic cavitation could effect particle size reduction, which leads to high viscosity (1.6 times higher than thermal treated samples), and consistency of tomato paste and juice (Vecret et al., 2002). Wu et al. (2008) surmised that the
increase of viscosity was due to a decrease in particle size from acoustic cavitation and shear stress produced from ultrasound. The decrease of particle size resulted in stronger inter-particle interactions, which lead to an increase in viscosity. Wu et al. (2008) examined the particle size distribution using *lazer diffraction* and *photo-micrograph* methods. They concluded that the particle decrease depends on ultrasound amplitude and treatment time, where greater amplitude and treatment time lead to greater reduction of particle size. Moreover, Vecret et al. (2002) hypothesized that acoustic cavitation could effect the molecule breakage of pectin, meaning the reduction of molecule weight of pectin. This assumption of Vecret et al. (2002), based on previous experiments on controlled degradation of polymer solutions, was assisted with ultrasound, where they observed the reduction of molecule weight (Price, 1990).

*Thermosonication effects on quality parameters of milk*

The milk, after thermosonication treatment had a slightly higher lightness (Bermudez-Aguirre et al., 2009). Bermudez-Aguirre et al. (2009) also detected a decrease of 0.26 % in protein content and a slightly increase in fat content due to the size reduction of fat globules of milk.

2.2.8 Acoustic reactors

In the literature, three principle types of ultrasound units are described for experimentally enhanced heat transfer: *horn* (Nomura et al., 2002; Cai et al., 2010), *cup-horn* (Monnot et al., 2007), and *resonating tube (sonotrode) (or Sonitube)* (Gondrexon et al., 2010; Legay et al., 2012 a; Legay et al., 2012 b). Comparisons between these three types of
ultrasound reactors showed that the resonant tube has many advantages over the other reactors (Romdhane et al., 1995; Faid et al., 1998 b). The advantages of the resonant tube are:

1. It provides three times higher cavitation intensity than the other reactors at the same acoustic intensity (Faid et al., 1998 b);

2. The ultrasonic field of the tube seems to be more homogeneous (Romdhane et al., 1994; Faid et al., 1998 b); and

3. The resonant tube can be operated continuously (Romdhane et al., 1995).

**Horn**

The horn ultrasound reactor consists of a generator (G), a transducer (C), which converts the electric energy into a vibrating mechanical energy, and an applicator (S) (horn), which allows the transmission of the wave into the medium (Figure 2.39) (Faid et al., 1998 b). At low input power, standing waves are being developed but at higher input power (above 25W), and the standing waves lose their pattern. Additionally, close to the emitting surface under the horn, high intensity cavitation and significant acoustic streaming occur. However, the intensity of cavitation has an important variation at the maxima (surface under the Horn) and the minima (Faid et. al., 1998 b).
Figure 2.39. Image of Horn, Ultrasound reactor at 20 kHz (Faid et al., 1998 b).

*Cup-Horn*

Cup-Horn is an ultrasound reactor, designed by SODEVA (SODEVA TDS, Savoie Hexapôle rue Charles Montreuil 73420 Méry, France; www.sodeva.com). The ultrasound reactor is located under the emitted surface (Figure 2.40), and it involves a generator and a transducer (Faid et al., 1998 b). Standing waves are observed over the whole range of input power, but the intensity of cavitation is much higher near the emitter surface than the free surface (Faid et al., 1998 b).
Resonating tube

The experiments, with a resonating tube as ultrasound reactor, used a commercialized metal (stainless steel, titanium, aluminium) resonating tube called the Sonitube (Vibert and Arnaud Perrier, 2012), which is an ultrasound reactor designed and patented by SODEVA.

The Sonitube can operate up to 2000 W of ultrasound power and over a range of frequencies 5-100 kHz. It consists of a generator (G), a transducer (C), a booster (B), which can adjust the displacement amplitude between the transducer and the tube, a modular unit (M), made from a metal collar and a tubular metal body, and the two pipe resonators or sonotrodes (R). The ultrasound emitter is mounted radially from the modular unit and the resonators are placed axially to the modular unit (Figure 2.41). The main idea of the resonating tube is that the frequency of transducer is equal to the vibrating frequency of
Collar (Figure 2.42) and to the longitudinal vibration of metal body. The length of the modular unit and the inner and the outer diameter of the collar determine the axial and the radial vibration, respectively. The reactor uses the radial vibration to cause cavitation during continuous liquid flow (Vaxelaire, 1995; Faid et al., 1998 b).

The length and the radius of the modular unit and resonator are determined by the half-wavelength of ultrasound frequency (Romdhane et al., 1995; Vaxelaire, 1995; Faid et al., 1998 b). The length of the Sonitube can be extended by an integer multiple of the half-wavelength of ultrasound frequency, and more than one multireactor can be used in the inline combination (Figure 2.42.) (Vaxelaire, 1995).

Figure 2.41. Image of stainless steel resonating tube at 20 kHz (with inner diameter at 42 mm), where C: transducer, B: booster, M: modular unit (Faid et al., 1998 b).
The wavelength is given from the equation:

$$\lambda = \frac{C}{f}$$  \hspace{1cm} (2.15)

Where $\lambda$ is the wavelength (cm), $f$ is the working frequency, and $C$ is the speed of sound (cm/s) in the metal of Sonitube (Faid et al., 1998 a).

The inner and outer radius (Figure 2.42) of the collar of Sonitube are given from the equation:

$$f = \frac{1.08 C}{(d_i + d_e)\pi} \cdot \frac{1}{2}$$  \hspace{1cm} (2.16)

Where $f$ is the frequency of vibration, $C$ is the speed of sound (cm/s) in the metal, which is made the collar of modular unit, $d_i$ is the inner diameter of collar, and $d_e$ is the outside diameter of the collar (Figure 2.42) (Vaxelaire, 1995).

Figure 2.42. Image of the collar of Modular unit of Sonitube (Vaxelaire, 1995).
Accurate design of resonance dimensions and frequency of Sonitube

To determine and design the accurate resonance dimensions and frequency of the Sonitube, Powerultrasonic (2013) suggests the use of a computer program called Sonoanalayzer (www.sonoanalyzer.com). Sonoanalayzer is a computer program that is used for the accurate design of different shapes of sonotrode through use of the Finite Elements Method.

Characteristics of acoustic cavitation by resonating tube

The cavitation intensity has a radial homogeneity in the resonating tube but does not have axial homogeneity. The intensity of cavitation decreases towards the extremities of the tube (higher in the center). An interesting experiment from Faid et al. (1998 a) investigated the ultrasound power by studying the thermal effects (the temperature difference) of a thermoelectriical probe along the axial length inside and outside a resonant tube (Figure 2.43; Figure 2.44). The temperature difference, ΔT is defined as Teq- To, where Teq is the equilibrium temperature and To is the initial temperature of the medium. Faid et al. (1998 a) observed higher cavitation power inside the Sonitube. Moreover, they found very good radial homogeneity of cavitation but a decrease of cavitation towards the axis.
Figure 2.43. Experimental set up for investigation of ultrasound power of Soini tube by studying the thermal effects ($\Delta T$) (Faid et al., 1998 a).

Figure 2.44. Thermal effect of ultrasound power by Soni tube, at the center $Y=0$mm and at the radius of Resonator $Y=13$mm. Where $T_{eq}$ equilibrium temperature, $T_0$ is the initial temperature of the medium, and $X$ is the axial length (Faid et al., 1998 a).
Tube-Tube Heat transfer exchanger using Resonating tube (Sonitube)

Sonitube has been used in many experiments as part of the vibrating tube-tube heat exchangers (Gondrexon et al., 2010; Legay et al., 2012a; Legay et al., 2012b). Legay et al., (2012c) used a titanium sonitube (sonotrode) and built around it the tube-tube heat exchanger. More specifically, they set up an inner stainless steel pipe, which was centered within the Sonitube reactor (Figure 2.45) (Legay et al., 2012a). The shape of tube-tube heat exchanger allowed it to resonate and create strong cavitation in the annular flowing liquid (Legay et al., 2012b). Furthermore, structural damages of heat exchanger from ultrasound vibration has been a concern, although no damage has been reported from the previous experiments (Legay et al., 2012a).

Figure 2.45. Structure of Heat exchanger with outer tube and reactor a titanium Sonitube at 35 kHz (Legay et al., 2012 b).
2.2.8.1 Heat transfer enhancement using a sonitube-tube heat exchanger

The experiments showed a significant enhancement in heat transfer when using a resonating tube (sonitube)-tube heat exchanger (Gondrexon et al., 2010; Legay et al., 2012 a; Legay et al., 2012 b). The ultrasonic vibration of the heat exchanger creates acoustic cavitation and powerful micro-streaming, which agitate the fluid near the boundaries, disturb the dynamic boundary layers, and increase heat and mass transfer (Legay et al., 2012 c).

Research showed that the enhancement of heat transfer did not depend on ultrasound amplitude vibration. Gondrexon et al. (2010), using a 100 W output power generator, studied the influence of ultrasound amplitude on heat transfer. They tested different ultrasound amplitudes, from 100% of the initial amplitude of the generator down to 50% decreased. Gondrexon et al. (2010) observed similar enhancement in all the different amplitudes. According to Legay et al. (2012 b), a decrease on ultrasound amplitude (using a booster after transducer) was necessary for the best heat exchanger energetic efficiency possible.

The flow rate of annular flow side had an important effect on heat transfer (Gondrexon et al., 2010; Legay et al., 2012 b). Gondrexon et al. (2010) observed a range of enhancement in the overall heat transfer coefficient from 123 up to 257%, depending on the flow rate, where the increase was higher for lower flow rates (0.15-0.5 L/min). Legay et al. (2012 b) observed that the ultrasound effect in heat transfer was negligible when the flow rate was $\geq$3 L/min. The inside flow also influenced the heat transfer, but this influence was less important than the rate of outside flow (Gondrexon et al., 2010; Legay et al., 2012 b).
2.3 Conclusions

Chaotic advection is the mechanism of mixing in laminar flow regimes. Chaotic advection is related with a chaotic trajectories motion, which is generated with passive mixers, like a static mixer, by changing the geometry of the flow and through active mixers, such as RAM, by creating a secondary transversal time or spatial periodic flow. Chaotic fluid motion forces the fluid elements to an exponential rate of stretching (which affects mixing efficiency) and folding of fluid elements, increasing the interfacial fluid surface and improving mixing.

Another active method for generating chaotic advection is through periodic transversal acoustical or mechanical vibration at low frequencies (sonic frequency range) of vibration. Acoustic or mechanical vibration creates a transversal periodic fluid motion, known as acoustic streaming or steady streaming, which generates chaotic dynamics and improves mixing and transportation of fluid elements. Acoustic and mechanical vibration at ultrasonic frequency range generates acoustic cavitations. Acoustic cavitation is the phenomenon of formation and collapse of microbubbles. At low ultrasound frequencies (20-100 kHz), the collapse of microbubbles creates strong shear forces and shock waves. The strong microturbulence, which has been created, is responsible for significant mixing enhancement at the regions close to the boundaries of the flow.

Mechanisms of chaotic advection and acoustic cavitation have been studied and used as mechanisms of thermal mixing (improvement of temperature uniformity) and heat transfer enhancement during the processing of highly viscous and sensitive materials.
This study is focused on the application of acoustical and mechanical low frequency vibration as an active mechanism for generating chaotic flows during continuous flow cooling of viscous and multiphase food products. Chaotic advection is used to achieve thermal mixing during cooling, expecting to improve the total cooling process and maximize the final food product.
### NOMENCLATURE

**Latin letters**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C$</td>
<td>speed of sound (cm/s) in the metal</td>
</tr>
<tr>
<td>$c_p$</td>
<td>specific heat of fluid (J/kg·°C)</td>
</tr>
<tr>
<td>$D$</td>
<td>diffusivity of either heat (thermal diffusivity) or concentration (diffusivity)</td>
</tr>
<tr>
<td>$d$</td>
<td>diameter of the pipe (m)</td>
</tr>
<tr>
<td>$d_i$</td>
<td>inner diameter of Collar Modular unit (cm)</td>
</tr>
<tr>
<td>$d_o$</td>
<td>outside diameter of the Collar Modular unit (cm)</td>
</tr>
<tr>
<td>$f$</td>
<td>working frequency</td>
</tr>
<tr>
<td>$\bar{u}$</td>
<td>average fluid velocity (m/s)</td>
</tr>
<tr>
<td>$k$</td>
<td>thermal conductivity of fluid (°C·m/W)</td>
</tr>
<tr>
<td>$L$</td>
<td>length of the pipe (m)</td>
</tr>
<tr>
<td>$Pe$</td>
<td>Péclet number, defined by Eq. (2.2) (dimensionless)</td>
</tr>
<tr>
<td>$p_\infty$</td>
<td>ambient liquid pressure</td>
</tr>
<tr>
<td>$R$</td>
<td>radius of the pipe (m)</td>
</tr>
<tr>
<td>$R_c$</td>
<td>critical bubble size (m)</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number, defined by Eq. (2.1) (dimensionless)</td>
</tr>
<tr>
<td>$T_{eq}$</td>
<td>equilibrium temperature of the medium (°C)</td>
</tr>
<tr>
<td>$T_o$</td>
<td>initial temperature of the medium (°C)</td>
</tr>
<tr>
<td>$U_R$</td>
<td>rotation speed</td>
</tr>
<tr>
<td>$&lt;U_z&gt;$</td>
<td>average axial velocity of the main flow</td>
</tr>
</tbody>
</table>
\( V \) velocity field

\( \mathcal{X} \) initial position of fluid particle

**Greek letters**

\( \beta \) mixing strength or stretching strength

\( \gamma \) specific heat ratio of the gas within the bubble

\( \Delta \) window length of RAM

\( \Delta T \) temperature difference (\( ^{\circ}\text{C} \))

\( \Theta \) offset periodic angle

\( \lambda \) wavelength

\( \mu \) viscosity of fluid (Pa/s)

\( \rho \) fluid density is the density of fluid

\( \Psi(x, y) \) scalar stream function

\( \omega \) angular frequency of ultrasound

\( \Omega \) velocity of rotation

**Subscripts**

\( e \) referring to the outside area of the pipe

\( z \) referring to Cartesian coordinate system

\( i \) referring to the inside area of the pipe

\( R \) referring to pipe radius
Symbols

—  volume average value
< > volume average value

Abbreviations

2-D  two-dimensional
3-D  three-dimensional
EHAM  Eccentric Helical Annular Mixer
EPM  Elongational Pin Mixer
KAM  Kolmogorov-Arnold-Moser theorem
PPM  Partitioned Pipe Mixer
RAM  Rotated Arc Mixer
REFERENCES


Ashokkumar M., 2011. The characterization of acoustic cavitation bubbles–An overview, Ultrasonics Sonochemistry 18, p. 864–872.


PowerUltrasonics, 2013. SonoAnalyzer software,


Rauwendaal, 2014. Elongational Pin Mixer (EPM),


CHAPTER 3: Mathematical solution and its computer implementation for temperature predictions during continuous flow cooling of non-Newtonian products

Abstract

Accurate calculation of temperature distribution and bulk product temperature of highly viscous foods (i.e., applesauce) during continuous flow cooling was obtained using a mathematical solution of the Graetz problem, for non-Newtonian flows, using the technique of the Sturm-Liouville integral transforms and its computer implementation in a code written on Maple 16. Numerical simulation studies of the cooling process of applesauce, using the mathematical model, showed a significant improvement of 61.4% in total cooling time with the application of thermal mixing during applesauce cooling.

Key words: Graetz-Nusslet problem, non-Newtonian fluid, computer modeling, temperature distribution, bulk temperature, temperature uniformity, thermal mixing

3.1 Introduction

During the cooling cycle of continuous thermal processing of highly viscous food products, like fruit and vegetable purees, laminar flow, non-Newtonian flow behavior (characteristic, for the most, of processed foods (Tokisangyo, 2014)), and low conductivity lead to a wide radial temperature distribution within the product and, consequently, an unequal cooling treatment and degradation of final food quality. Thermal mixing, that is,
homogenization of temperature profile at the exit of the cooling section, could enhance the cooling process and reduce cooling time.

The temperature distribution for viscous, Newtonian flow behavior, fluids with a fully developed velocity field, has been mathematical solved and analyzed in the Graetz-Nusslet or Graetz problem (Johnston, 1994). The solution of the Graetz-Nusslet gives the temperature distribution of an incompressible Newtonian fluid with constant fluid properties, flowing in laminar regime, under a fully developed velocity field and by ignoring heat generation via viscous dissipation. Only a few studies have addressed the Graetz problem for non-Newtonian flows, characteristic for viscous and multiphase food products. For non-Newtonian flows, the Graetz problem has been solved via analytical and numerical techniques. Analytical techniques solved the Graetz problem for non-Newtonian flows by converting the initial problem to a Sturm-Liouville problem (Johnston, 1994; Rice and Do, 1995). Numerical solutions were focused on finite difference (McKillop, 1964; Parikh and Mahalingam, 1988) and orthogonal collocation methods to solve the generalized Graetz problem.

The present study focused on the development of a mathematical solution and its computer implementation of the generalized Graetz problem for non-Newtonian flows, with the purpose of estimating the radial temperature distribution and the bulk temperature of highly viscous foods, such as applesauce and sweet potato puree, at the exit of the cooling section of a bulk sterilization system. The developed computer model will be used to establish the benefits of thermal mixing and temperature profile homogenization during
continuous flow cooling of viscous products by calculating and comparing the results of total required cooling time with and without application of thermal mixing.

### 3.2 Mathematical solution

The analytical technique of Sturm-Liouville integral transforms, as described in Johnson (1994) and Rice and Do (1995), was used to estimate the temperature at the exit of the cooling section during the processing of viscous food materials. Starting with the energy equation for constant physical properties, we have:

\[
\rho c_p \frac{DT}{Dt} = (k \nabla^2 T) + \mu \Phi_n \tag{3.1}
\]

Where \( \rho \) (kg/m\(^3\)) is the density of the food product, \( c_p \) (J/kg·°C) is specific heat of the food product, \( k \) (°C·m/W) is the thermal conductivity of the food product, \( \mu \) (Pa/s) is the viscosity of the food product, \( T \) is temperature of the product, and \( \Phi_n \) is the viscous dissipation function. The expression \( c_p \frac{DT}{Dt} \) on the left side is a material derivative of enthalpy. In cylindrical coordinates, Eq. (3.1) can be written as:

\[
\rho c_p \left[ \frac{\partial T}{\partial t} + u_r \frac{\partial T}{\partial r} + u_0 \frac{\partial T}{\partial \theta} + u_z \frac{\partial T}{\partial z} \right] = k \left[ \frac{1}{r \partial r} \left( r \frac{\partial T}{\partial r} \right) + \frac{1}{r} \frac{\partial^2 T}{\partial \theta^2} + \frac{\partial^2 T}{\partial z^2} \right] + \mu \Phi_n \tag{3.2}
\]

with

\[
0 \leq r \leq R, \\
0 \leq x \leq L, \\
0 \leq \theta \leq 2\pi, \\
t > 0
\]
The left hand side of Eq. 3.2 is the rate of increase of enthalpy per unit volume, while the right hand side is the rate of energy input by heat conduction per unit volume and the rate of conversion of mechanical to thermal energy per unit volume (viscous energy dissipation).

Where \( R \) (m) is the radius of the pipe, \( L \) (m) is the total length of cooling section.

The following assumptions of the generalized non-Newtonian Graetz problem (constant fluid properties, fully developed laminar flow, and neglect of viscous dissipation) and with the Dirichlet boundary condition (constant wall temperature) were made:

**Assumptions**

Fully developed velocity profile, for a power law fluid:

\[
\begin{align*}
\frac{\partial}{\partial \phi} u &= 0 \\
\frac{\partial}{\partial r} u &= 0 \\
\frac{\partial}{\partial x} u &= u(r) = u_m \left[ \frac{3 \cdot n + 1}{n + 1} \left( 1 - \left( \frac{r}{R} \right)^n \right) \right]^\frac{n+1}{n}
\end{align*}
\]

Where \( u_m \) is the mean velocity of the flow (m/s), \( n \) is the flow behavior index of the fluid.

Assuming axisymmetric temperature field, that is,

\[
\frac{\partial T}{\partial \theta} = 0
\]

and negligible viscous dissipation

\[
\mu \Phi_n = 0
\]

the energy equation is simplified to:
\[ \rho c_p \left[ u_m \left( \frac{3 \cdot n + 1}{n + 1} \right) \right] \left[ 1 - \left( \frac{r}{R} \right)^{n+1} \right] \frac{\partial T}{\partial x} = k \left[ \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial T}{\partial r} \right) + \frac{\partial^2 T}{\partial x^2} \right] \] (3.9)

with

\[ 0 \leq r \leq R, \]
\[ 0 \leq x \leq L \]

According to Johnston (1994), for the calculation of bulk temperature \( T_{\text{bulk}} \), the term of axial conduction of Eq. 3.9 can safely be ignored for \( \text{Pe} > 100 \), converting Eq. 3.9 to Eq. 3.10:

\[ \rho c_p u(r) \frac{\partial T}{\partial x} = k \left[ \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial T}{\partial r} \right) + \frac{\partial^2 T}{\partial x^2} \right] \] (3.10)

### 3.2.1. Graetz problem for non-Newtonian fluids with Dirichlet boundary conditions

Considering Eq. 3.10 and applying the boundary conditions of Graetz problem for Non-Newtonian fluids with Dirichlet boundary conditions, the following problem is built (Figure 3.1):
Boundary conditions

Considering Eq. 3.10 and applying the boundary conditions of Graetz problem for Non-Newtonian fluids with Dirichlet boundary conditions, the following problem is built:

\[
p\rho c_p u(r) \frac{\partial T}{\partial x} = k \left[ \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial T}{\partial r} \right) + \frac{\partial^2 T}{\partial x^2} \right] \quad (3.10)
\]

with

\[
0 \leq r \leq R, \\
0 \leq x \leq L
\]

Uniform initial temperature \( (T_0) \) at the entrance of cooler

\[
T(r,0) = T_0 \quad (3.11)
\]
Constant wall temperature \( (T_w) \)

\[
T(R, x) = T_w
\]  
(3.12)

Radial symmetric

\[
\frac{\partial T(0, x)}{\partial r} = 0
\]  
(3.13)

*Non-Dimensionalization*

For easier analysis of the problem, the following dimensionless forms are employed:

For the constant wall temperature, define

\[
\theta = \frac{T_w - T(r, x)}{T_w - T_0}
\]  
(3.14)

The following dimensionless distance variables:

\[
y = \frac{r}{R}
\]  
(3.15)

with

\[
0 \leq y \leq 1
\]

\[
z = \frac{x}{R Pe}
\]  
(3.16)

where \( Pe \) is the Peclet number. The Péclet number quantifies the ratio between advection and diffusion, where a \( Pe \to 0 \) means the diffusion is rapid in a very slow flow, while when \( Pe \to \infty \) means a very slow diffusion in a rapid high flow. The equation for \( Pe \), for a pipe flow heat transfer problem, is given from the following:

\[
Pe = \frac{R Pe \rho u_m}{\kappa}
\]  
(3.17)
The fully developed non-dimensional velocity profile is given by:

\[ u(y) = \frac{u(r)}{u_\infty} = \left[ \frac{3 \cdot n + 1}{n + 1} \right] \left[ 1 - \left( \frac{r}{R} \right)^{n+1} \right] = \left[ \frac{3 \cdot n + 1}{n + 1} \right] \left[ 1 - \left( \frac{y}{n} \right)^{\frac{n+1}{n}} \right] \]  

(3.18)

**Non-dimensional governing equation and boundary conditions**

Hence, the non-dimensional governing equation and boundary conditions are:

\[ u(y) \frac{\partial \theta}{\partial z} = \frac{1}{y} \frac{\partial}{\partial y} \left( y \frac{\partial \theta}{\partial y} \right) \]  

(3.19)

with

\[ 0 \leq y \leq 1 \]  

(3.20)

and

\[ 0 \leq z \leq \frac{L}{R \cdot Pe} \]  

(3.21)

Uniform initial temperature

\[ \theta(y,0) = 1 \]  

(3.22)

Constant wall temperature

\[ \theta(1,z) = 0 \]  

(3.23)

Radial symmetric

\[ \frac{\partial \theta(0,z)}{\partial y} = 0 \]  

(3.24)
3.2.2 Solution of Graetz problem for non-Newtonian fluids

The solution of Eq. 3.19 is based on the Sturm-Liouville integral transform (Johnson, 1994). The following associated eigenproblem (Equation 3.25) contains similar boundary conditions to Eq. 3.19:

\[
\frac{1}{y} \frac{\partial}{\partial y} \left( y \frac{dK_i(y)}{dy} \right) + \xi_i^2 K_i(y) = 0, i = 1,2,3,\ldots\infty
\]  

(3.25)

with

\[ 0 \leq y \leq 1 \]

(3.26)

\[ K_i(1) = 0 \]

(3.27)

where \( K_n \) defines the kernel transformation. The solution of Eq. 3.25 is given by:

\[ K_i(y) = J_0(\xi_i y), i = 1,2,3,\ldots\infty \]

(3.28)

where \( J_0 \) is the Bessel functions of order zero of the first kind, \( \xi_i \) is the eigenvalue, the \( i^{th} \) positive zero of \( J_0 \).

Multiplying Eq. 3.19 with \( y \cdot K_i \) results in:

\[ y K_i \left( \mu(y) \frac{\partial \theta}{\partial z} \right) = \frac{1}{y} \frac{\partial}{\partial y} \left( y \frac{\partial \theta}{\partial y} \right) y K_i, i = 1,2,3,\ldots\infty \]

(3.29)

And integrating Eq. 3.29 with respect of \( y \) from 0 to 1:
\[
\int_0^1 yK_i \left( u(y) \frac{\partial \theta}{\partial z} \right) \, dy = \int_0^1 \left( \frac{1}{y} \frac{\partial}{\partial y} \left( y \frac{\partial \theta}{\partial y} \right) \right) yK_i \, dy, \ i = 1, 2, 3, \ldots \infty \quad (3.30)
\]

Therefore, for right side of Eq. 3.30 we have:

\[
\int_0^1 \frac{\partial}{\partial y} \left( y \frac{\partial \theta}{\partial y} \right) yK_i \, dy = \int_0^1 \frac{\partial}{\partial y} \left( y \frac{\partial \theta}{\partial y} \right) yK_i \, dy + \int_0^1 \frac{\partial}{\partial y} \left( y \frac{\partial K_i}{\partial y} \right) \theta \, dy - \int_0^1 \frac{\partial}{\partial y} \left( y \frac{\partial K_i}{\partial y} \right) \theta \, dy \quad (3.31)
\]

Recall from Chorlton (1981):

\[
\int_0^1 \frac{\partial}{\partial y} \left( y \frac{\partial \theta}{\partial y} \right) yK_i \, dy - \int_0^1 \frac{\partial}{\partial y} \left( y \frac{\partial K_i}{\partial y} \right) \theta \, dy = \left[ \left( y \left( \frac{\partial \theta}{\partial y} \right) - \frac{\partial K_i}{\partial y} \right) \right] \bigg|_0^1, \ i = 1, 2, 3, \ldots \infty \quad (3.32)
\]

From the boundary conditions of Eq. 3.19 and 3.23, Eq. 3.32 is equal to 0.

Recall from Eq. 3.25:

\[
\frac{\partial}{\partial y} \left( y \frac{dK_i(y)}{dy} \right) = -\xi^2_i \, K_i(y) \quad (3.25)
\]

Therefore, for right side of Eq. 3.30 we have:

\[
\int_0^1 \frac{\partial}{\partial y} \left( y \frac{\partial \theta}{\partial y} \right) K_i \, dy = \int_0^1 \frac{\partial}{\partial y} \left( y \frac{\partial K_i}{\partial y} \right) \theta(y,z) \, dy = \int_0^1 -\xi^2_i K_i(y) \theta(y,z) \, dy = -\frac{\xi^2_i}{\xi^2_i} \int_0^1 K_i(y) \theta(y,z) \, dy \quad (3.33)
\]

**Integral transformation**

Defining the integral transformation:

\[
< \theta, K_i > = \int_0^1 K_i(y) \theta(y,z) \, dy \quad (3.34)
\]
Through the inverse transformation of Eq. 3.34, we calculate the dimensionless temperature, \( \theta \), as (Rice and Do, 1995):

\[
\theta (y, z) = \sum_{i=1}^{\infty} \frac{K_i}{< K_i, K_i >} < \theta, K_i >
\]

(3.35)

With \( < K_i, K_i > \) as:

\[
< K_i, K_i > = \int_0^1 K_i^2 i \, dy
\]

(3.36)

Recall Eq. 3.19:

\[
u(y) \frac{\partial \theta}{\partial z} = \frac{1}{y} \frac{\partial}{\partial y} \left( y \frac{\partial \theta}{\partial y} \right)
\]

(3.19)

Applying the inverse form of \( \theta \) for the left side of Eq. 3.19:

\[
u(y) \sum_{i=1}^{\infty} < K_i, K_i > \frac{\partial < \theta, K_i >}{\partial z} = \frac{1}{y} \frac{\partial}{\partial y} \left( y \frac{\partial \theta}{\partial y} \right)
\]

(3.37)

Homogeneous boundary conditions allow us to make this substitution (Johnson, 1994).

Next is the application of integral transformation on Eq. 3.37 for the eigenfunction \( K_j \).

Recall Eq. 3.25 for the eigenfunction \( K_j \):

\[
\frac{\partial}{\partial y} \left( y \frac{dK_j(y)}{dy} \right) = -\xi_j^2 K_j(y) y
\]

(3.25)

Multiplying Eq. 3.37 by Eq. 3.25 and integrating with respect of \( y \) from 0 to 1 we get:
\[
\int_0^1 u(y) \sum_{i=1}^\infty \frac{K_i}{<K_i,K_i>} \frac{\partial <\theta,K_i>}{\partial z} y K_j \, dy = -\xi_j^2 <\theta,K_j> \Rightarrow 
\]

(3.38)

\[
\int_0^\infty \sum_{i=1}^\infty u(y) \cdot y \cdot K_j \cdot K_i \frac{\partial <\theta,K_i>}{\partial z} \, dy = -\xi_j^2 <\theta,K_j> \Rightarrow 
\]

(3.39)

\[
\sum_{i=1}^\infty <u(y) \cdot K_i,K_j> \frac{\partial <\theta,K_i>}{\partial z} \, dy = -\xi_j^2 <\theta,K_j> 
\]

(3.40)

where \(<u(y) K_i,K_j>\) is given from:

\[
<u(y)K_i,K_j> = \int_0^1 u(y)K_i \, K_j \, dy 
\]

(3.41)

Introducing the Kornecker Delta \(\delta_{i,j}\), recall the Property of Kornecker Delta (Wolfarm, 2013):

\[
K_j = \sum_{i=1}^\infty K_i \delta_{i,j} 
\]

(3.42)

Then the right side of Eq. 3.40 can be written as:

\[
\sum_{i=1}^\infty <u(y)K_i,K_j> \frac{\partial <\theta,K_i>}{\partial z} \, dy = -\sum_{i=1}^\infty \xi_j^2 \delta_{i,j} <\theta,K_i> 
\]

(3.43)

**Vectors and Matrices**

We define the following vectors and matrices:

**Vector \(\Theta\):**

\[
\Theta = \langle <\theta,K_i> \rangle, \, i = 1,2,3,\ldots,\infty 
\]

(3.44)

**Matrices \(A, D\):**

\[
A = \begin{bmatrix} <u(y) \cdot K_i,K_j> \end{bmatrix}, \, i = 1,2,3,\ldots,\infty, \, j = 1,2,3,\ldots\infty 
\]

(3.45)
Eq. 3.44 becomes:

\[ \sum_{i=1}^{\infty} A \frac{\partial \Theta}{\partial z} = -\sum_{i=1}^{\infty} D \Theta \]  \hspace{1cm} (3.47)

Eq. 3.47 is in the form of a system of linear first ordinary differential equations in the form:

\[ A \frac{\partial \Theta}{\partial z} = -D \Theta \]  \hspace{1cm} (3.48)

The solution of Eq. 3.48 is:

\[ \Theta = \Theta_o \cdot e^{-\frac{zD}{A}} \]  \hspace{1cm} (3.49)

\[ \Theta_o \] is a constant, which is calculated from the initial conditions by applying the integral transformation of Eq. 3.27 to Eq. 3.23 (initial boundary condition of Eq. 3.25) (Johnson, 1994). Hence, the vector \[ \Theta_o \] is given from:

\[ \Theta_o = \left\{ \int_0^1 y \cdot K_i dy \right\} = \{<1, K_i>\}, i = 1, 2, 3, \ldots \infty \]  \hspace{1cm} (3.50)

### 3.2.3 Temperature distribution solution

Recall Eq. 3.35 for the dimensionless temperature, \[ \theta \]:

\[ \theta (y, z) = \sum_{i=1}^{\infty} \frac{K_i}{<K_i, K_i>} <\theta, K_i> \]  \hspace{1cm} (3.35)

From Eq. 3.49 we calculate \[ <\theta, K_i> \], \[ K_i \] is given from Eq. 3.28 and \[ <K_i, K_i> \] from Eq. 3.36. Knowing the non-dimensional temperature distribution \[ \theta \] we can calculate from Eq. 3.14, the temperature distribution \[ T(r, x) \] comes from:
\[ T(r, x) = T_w - (T_w - T_0) \theta \]  

(3.51)

### 3.2.4 Bulk temperature at the end of cooling unit

Knowing the radial temperature distribution \( T(r, L) \) at the end of cooler \((x=L)\), we can calculate the bulk temperature. The bulk temperature, \( T_{\text{bulk}} \), is calculated from the following equation (Whitaker, 1983):

\[
T_{\text{Bulk}} = \frac{\int_0^R T(r, L)r \, u(r) \, dr}{\int_0^R r \, u(r) \, dr} 
\]

(3.52)

### 3.3 Compute modeling on Maple of temperature distribution solution and bulk temperature at the exit of the cooling unit

Using the solution of Graetz problem for non-Newtonian fluids presented above, a computer code was written on Maple 16 software in order to estimate the temperature distribution and the bulk product temperature at the exit of the cooling section during cooling in a shell and tube heat exchanger of a highly viscous food product initially at uniform temperature.

Here, then, is the introduction of the constant fluid properties of thermal conductivity, specific heat, density, and the flow behavior index into the program; given the parameters of the cooling unit, such as the radius and the length of the tube of the cooling unit, the average volumetric flow rate, the initial (uniform) temperature of the product, and the constant wall temperature (Dirichlet boundary conditions) (Figure 3.2), the program calculates and plots the radial temperature distribution and the value of bulk temperature at the exit of the cooling section (Figure 3.3).
Figure 3.2. Picture of input values of constant fluid properties thermal conductivity \(k\), density of the food \(\rho\), specific heat \(c_p\), and flow behavior index \(n\); parameters of the cooling unit, such as the radius \(R\), and the length \(L\) of the tube of the cooling section; average volumetric flow rate \(V\) and the mean velocity \(U_m\), the initial uniform temperature value \(T_i\) and the constant wall temperature \(T_w\), and Peclet number \(Pe\) calculation.
Figure 3.3. Radial temperature distribution curve from computer model and bulk temperature, $T_{bulk}$, calculation using the final temperature distribution function, $T_{func}$, estimated from the mathematical solution of Graetz problem and the computer model.

3.3.1 Comparison of computer model results with values from literature

The bulk temperature results from the computer model were compared with bulk temperature values from the literature (Johnson, 1994) for different flow behavior index values ($n=1$; $n=0.5$) and for different values of non-dimension distance variable $z$ (recall
equation 3.12). The results for all the examined cases were in absolute agreement with literature values (Table 3.1).

Table 3.1. Comparison of non-dimensional bulk temperature $\theta_b$ values from this work with these from literature; for different values of flow behavior index, $n$, and different values non-dimension distance variable $z$.

<table>
<thead>
<tr>
<th></th>
<th>$\theta_b$, This work</th>
<th>$\theta_b$, Johnson (1994)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$n=1$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$z$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.002</td>
<td>0.96175</td>
<td>0.96175</td>
</tr>
<tr>
<td>0.020</td>
<td>0.83622</td>
<td>0.83622</td>
</tr>
<tr>
<td>0.100</td>
<td>0.57890</td>
<td>0.57879</td>
</tr>
<tr>
<td>$n=0.5$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$z$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.040</td>
<td>0.73616</td>
<td>0.73616</td>
</tr>
<tr>
<td>0.100</td>
<td>0.55683</td>
<td>0.55683</td>
</tr>
</tbody>
</table>

3.3.2 Comparison of computer model results with experimental results

Simulated temperature distribution data from the computer model was compared with experimental time-temperature data collected from experiments on thermal mixing of applesauce during the cooling cycle. The temperature of applesauce was measured at the exit of a shell and tube heat exchanger, which was used as cooler (The experiment on thermal
mixing of applesauce is described on Chapter 5). Input data from literature were used for thermal conductivity (0.5846 °C·m/W), density (1031 k·g/m$^3$), and specific heat (3730 J/kg·°C) of applesauce (Lozano, 2006; Toledo, 2007; Aqua-calc, 2013). A flow behavior index equal to 0.14 was used based on experimental shear rate vs. shear stress data. Experimental data were obtained for a volumetric flow rate (3.75 L/min = 0.0000625 m$^3$/s), a shell and tube heat exchanger with $L=1.708$ m and $R=0.0174$ m; the average values of experimental temperature data for the inlet at the cooling section temperature and the constant wall temperature, temperatures of 106 °C and 82 °C, were used for the inlet and the wall temperature, respectively.

The fitted plot for the computer model temperature distribution curve at the exit of the cooling section and the experimental temperatures at the exit of cooling section, at 3 different cross sectional points of the pipe—at the center ($r=0$ m, $T=106.7$ °C), at the wall ($r=0.017$ m, $T=83.7$ °C) and at an intermediate point ($r=0.006$ m, $T=105.0$ °C)—are shown in Figure 3.4. As it is presented in Figure 3.4, the simulated and experimental data were in excellent agreement.
Figure 3.4. Comparison of the radial temperature distribution curve at the exit of the cooling section, predicted from computer model, with experimental temperature measurements at three different cross sectional points.

3.3.3 Numerical simulations of thermal mixing effects on enhancement of continuous flow cooling

Using the temperature distribution computer model, a numerical investigation was performed in order to estimate how thermal mixing and improved radial temperature uniformity could enhance cooling of viscous and multiphase food products. It was assumed that cooling was achieved by passing the product through a series of independent cooling
units (shell and tube heat exchanger). For that reason, the time and the number of cooling units needed to get the product from a high uniform initial temperature of 106 °C down to room temperature (25 °C) were compared for two different cases:

i) With application of thermal mixing, the temperature profile at the exit of each cooling unit homogenized, so that the product was entering into the next cooling section with a uniform temperature profile.

ii) Without application of thermal mixing at the exit of the cooling section, the product was entering into the next cooling section with the same temperature profile as the one that was achieved at the exit of the previous cooling section.

For the solution of both two simulations, the following assumptions were made: the wall temperature reaches the value of 25 °C rapidly, the flow rate was constant at 5.6 L/min (an approximately value of 1.5 gal/min), and the parameters of each cooling section were $L=1.708 \text{ m}$ and $R=0.0174 \text{ m}$. For both simulations, the fluid properties of applesauce (thermal conductivity, specific heat, density, and flow behavior index) were the same as the values from the literature, as in section 3.3.2 (Lozano, 2006; Toledo, 2007; Aqua-calc, 2013).

Numerical simulation for applesauce showed that for the first case (with temperature profile equalization after each cooling section), 930 s were needed to decrease the temperature of the product from 106 °C to 25 °C. For the second case (without temperature equalization after each cooling section), 2408 s were needed. It is easy to understand that improved temperature distribution during cooling results in more rapid cooling process, with an improvement of roughly 61.4%.
3.4 Conclusions

In this study, an accurate mathematical model was developed and used to estimate the temperature distribution and the bulk product temperature during continuous flow cooling of highly viscous food materials. Numerical simulation of the cooling process of applesauce showed a significant improvement of 61.4% of total cooling time with the application of thermal mixing during applesauce cooling.
NOMENCLATURE

*Latin letters*

**A** matrix defined by Eq. (3.45)  

**c**\(_p\) specific heat of fluid (J/kg·°C)  

**D** matrix defined by Eq. (3.46)  

**J**\(_0\) Bessel functions of order zero of the first kind  

**k** thermal conductivity of fluid (°C·m/W)  

**Kn** kernel transformation  

**L** total length of cooling section (m)  

**n** flow behavior index (dimensionless)  

**Pe** Péclet number (dimensionless)  

**R** radius of the pipe of cooling section (m)  

**r** radial distance variable (m)  

**T** temperature function  

**T(r,x)** temperature distribution function  

**Tfunct** temperature distribution function  

**T**\(_0\) initial uniform temperature at the inlet of cooling section (°C)  

**Ti** initial uniform temperature at the inlet of cooling section (°C)  

**T**\(_w\) constant wall temperature (°C)  

**u**\(_m\) average fluid velocity (m/s)  

**V** volumetric flow rate (L/s)
x \quad \text{axial distance variable (m)}

y \quad \text{radial distance variables (dimensionless)}

z \quad \text{axial distance variables (dimensionless)}

\textit{Greek letters}

\begin{align*}
\delta_{ij} & \quad \text{Kronecker Delta} \\
\mu & \quad \text{viscosity of fluid (Pa/s)} \\
\rho & \quad \text{fluid density is the density of fluid} \\
\Theta & \quad \text{vector defined by Eq. (3.49)} \\
\Theta_o & \quad \text{vector defined by Eq. (3.50)} \\
\theta & \quad \text{dimensionless temperature function} \\
\theta & \quad \text{angular position (radians)} \\
\zeta_i & \quad \text{eigenvalue of the } i^{\text{th}} \text{ positive zero of } J_0 \\
\Phi_n & \quad \text{viscous dissipation function}
\end{align*}

\textit{Subscripts}

i \quad \text{referring to the initial temperature at the inlet of cooling section} \\
o \quad \text{referring to the initial temperature at the inlet of cooling section} \\
w \quad \text{referring to the wall temperature}
REFERENCES


CHAPTER 4: Thermal mixing via acoustic vibration during continuous flow cooling of sweet potato puree

Abstract

For this study, a complete continuous flow thermal system, consisting of a continuous flow microwave (MW) system (formed from a series of home MW ovens), a cooler, and a mixing unit (consisting of a 180° bend pipe and an acoustic vibrator), was built. Acoustics were used to impose transversal sinusoidal vibration into a 180° bend pipe in order to generate chaotic advection and enhance thermal mixing during the continuous flow cooling process of sweet potato puree in a temperature range of 60 to 105 °C. Uniform radial temperature distribution, with a temperature difference at 0-3 °C between the product temperature at center and close to the pipe wall, was observed when applying continuous vibration at the maximum amplitude and at 20±4 Hz, a frequency close to the resonance frequency of vibration of the mixing system (acoustic vibrator, 180° bend pipe and product). Initial temperature difference between the product temperature at the center and the temperature close to the wall, before the application of vibration, was the most important factor in causing uniformity of temperature upon exiting the mixing unit. A critical value of 12-15 °C was found; above this value a rather wide temperature distribution (5-12 °C) remained, while for initial temperature difference equal to or less than this value, homogenization of radial temperature profile was observed at the exit of mixing unit.

Key words: Continuous flow thermal process, cooling, acoustics, mechanical vibration, thermal mixing, sweet potato puree
4.1 Introduction

The efficiency of thermal processing (heating, holding, and cooling) of highly viscous and multiphase food products with low thermal conductivity is a major challenge and problem in food processing and food engineering (Barigou et al., 1998; Metcalfe and Lester, 2009). This problem for the heating and holding portion of continuous thermal processing has been addressed through advanced continuous flow heating technologies, such as continuous flow microwave (MW) system, ohmic, and radiofrequency heating technologies (Coronel et al., 2005; Steed et al., 2008; Cullen et al., 2012). Advanced heating technologies provide a uniform rapid volumetric heating process, which is utilized by the processors to minimize the holding time and maximize the final food product quality (Coronel et al., 2005; Steed et al., 2008; Cullen et al., 2012).

Due to lack of advanced technologies, cooling depends on slower convection heat transfer methods. During the cooling cycle of thermal processes, laminar flow, non-Newtonian flow behavior (characteristic for most processed foods (Tokisangyo, 2014)), and low conductivity of highly viscous foods lead to a wide radial temperature distribution within the product, resulting in unequal cooling treatment and degradation of final food quality. The cooling process of highly viscous, low conductivity food products can be enhanced significantly by improving mixing (Metcalfe and Lester, 2009; Eesa and Barigou, 2010; Saatdjian, et al., 2011).

In laminar flow, two-dimensional (2-D) steady (time independent) flow, and in small scale eddies in turbulence, viscous forces are dominated and fluid particles flow within streamlines, resulting in limiting mixing (Franjione and Ottino, 1991). However, mixing in
the laminar regime can be enhanced through the mixing technique of chaotic advection. Chaotic advection exists when highly complicated trajectories are observed in the Lagrangian frame (time dependence trajectories). Chaotic advection is a method to achieve mixing in very viscous materials (i.e., as food purees, soups, etc.) where turbulence flow cannot be achieved (Cullen, 2009). Chaotic advection can be achieved through passive methods, by periodically changing the flow geometry, or through active methods, with the application of non-linear periodic forces into the fluid to create a 2-D time periodic (time dependent) flow or a three-dimensional (3-D) flow (Aref, 1984; Ottino, 1989; Aref, 2002; Cullen, 2009).

Chaotic advection applies simple non-turbulence flows (chaotic flow) to create rapid mixing from the periodic stretching and folding of fluid elements; this results in an exponential increase of interfacial area and pattern with high gradients (density, pressure, temperature, etc.) to facilitate chaotic mixing (a combination of chaotic advection and diffusion) (Ottino, 1989; Arref, 2002; Cullen, 2009).

For temperature homogenization, which was the goal of this study, non-diffusive mixing is required and rapid mixing can be achieved via the mixing mechanism of chaotic advection; this is in contrast to heat transfer problems in which diffusion is also involved in mixing (Lester et al., 2009). In mixing by chaotic advection, the degree of stretching is the factor that affects the degree of mixing the most (Ottino, 1989; Kusch and Ottino, 1992; Metcalfe et al., 2006; Singh et al., 2008). The degree of stretching is an important factor in achieving global chaotic flow and in breaking down flow symmetries inside the flow, which can work as barriers, during mixing with chaotic advection.

Chaotic advection has been used in heat transfer studies as a mechanism to increase heat transfer coefficients (Acharya et al., 1992; Acharya et al., 2001; Metcalfe and Lester,
2009) and thermal mixing (improving radial temperature uniformity) (Lester et al., 2009; Eesa and Barigou, 2010; El Omari and Le Guer, 2010 a, El Omari and Le Guer, 2010 b; Saatdjian et al., 2011; Le Guer and El Omari, 2012). In continuous flow thermal treatment of viscous and multiphase food products, such as fruit and vegetables purees, dairy products, and soups, the application of heat exchangers based on the concept of chaotic advection has been studied, with the most common examples being those of the scraped surface heat exchangers (Carlson, 1991) and RAMex, a shell and tube heat exchanger based on a chaotic advection mixing device, the Rotated Arc Mixer (RAM). RAM contains two co-centric cylinder: the inside cylinder is stationary with periodic placed windows (openings), and the outside rotating cylinder imposes spatial periodic transversal secondary flow to the initial axial flow (Lester et al., 2009; Metcalfe and Lester, 2009). On thermal mixing application, the most commonly studied and used inline mixers for improving radial temperature distribution are static mixers (Etchells III and Meyer, 2004). The complex geometry of static mixers works as obstacles or baffles, creating periodic cuts and folds of fluid elements, which result in radial mixing (Etchells III and Meyer, 2004).

In the food industry, most solid-liquid mixtures of highly viscous processed foods contain a high solid particles concentration, which can be damaged from the high shear imposed by the blades of a scraped surface heat exchanger or can be trapped and damaged at the windows of RAM. Furthermore, the large pressure drop created by the physical obstacles-baffles of static mixers may block the solid-liquid mixture flow and clog with solid particles. In this study, the goal was to design and build a new mixing unit that would combine active and passive techniques to generate chaotic advection in an open duct flow.
system in attempt to achieve a reduced shear and pressure drop and to retain the maximum integrity of food particles.

Acoustic and sinusoidal transversal mechanical vibration techniques are two methods that have been studied as non-linear methods to generate chaotic advection in open duct flows. Acoustic vibration at low acoustic frequencies (60 Hz) is a method used industrially for the batch mixing process of viscous materials (Resodyn, 2014). Furthermore, acoustic vibration at low ultrasonic frequencies (20 kHz and 35 kHz) has been studied as a method of mixing and heat transfer enhancement—more because of acoustic cavitation via ultrasound and less because of chaotic advection via transversal vibration (Gondrexon et al., 2010; Legay et al., 2011; Legay et al., 2012 a; Legay et al., 2012 b). Mechanical vibration at low acoustic frequencies (0-75 Hz) is a technique that has been shown by Computational Fluid Dynamics (CFD) and experimental studies to produce mixing and improved heat transfer of viscous, Newtonian, or non-Newtonian fluids (Khalaf and Sastry, 1996; Eesa and Barigou, 2010). According to validated CFD models, the degree of stretching and thermal mixing from the use of mechanical vibration imposed transversally on a pipe is influenced more by normalized amplitude (ratio of amplitude of vibration with radius of pipe) of vibration than by frequency of vibration (Eesa and Barigou, 2010), with the best thermal mixing observed for high normalized amplitude at a low frequency range of vibration (Eesa and Barigou, 2010). Other passive methods on chaotic advection that create less of a pressure drop than the most used, passive technique of static mixers are bend or curved pipes. In curved pipes, such as 180° bend pipes, centrifugal forces generate Dean flow, which works as a secondary transversal flow and can generate chaotic flow (Chagny et al., 2000). Heat transfer
enhancement has been achieved with the use of shell and tube heat exchangers with chaotic advection configuration, with the shell consisting from periodically connected bend pipes (Changy et al., 2000).

In this work, a new mixing unit was built and tested using acoustics to generate acoustic and mechanical vibration on a 180° bend pipe to achieve thermal mixing during cooling of sweet potato puree. Thermal homogenization of sweet potato puree during the cooling cycle was studied for different parameters of acoustic and vibration, such as frequency, amplitude, and protocols of vibration. The application of an efficient method of thermal mixing during the cooling process at the exit of each cooling section (cooling time or cooling unit) will improve the radial temperature distribution inside the food product and will therefore result in more equal thermal treatment of all food parts, maximizing the final food product quality and enhancing the total cooling process as a result of the entire continuous flow thermal processing.

4.2 Materials and methods

Problem definition and hypothesis

The development of advanced continuous flow heating technologies, like MW continuous flow systems, provided uniform rapid volumetric heating processes on viscous and multiphase food products that minimize the total time of the heating process and maximize the final food quality (Figure 4.1). On the other hand, during the conventional continuous cooling process, laminar flow and low thermal conductivity of highly viscous foods lead to a wide temperature distribution inside the product, resulting in unequal cooling treatment of food material, slow cooling process, and degradation of food quality (Figure
Applying efficient thermal mixing at the exit of each cooling section (cooling time or cooling unit) will improve the radial temperature distribution inside the food product, and that will result in a more equal thermal treatment of all food parts, enhancement of the total cooling process, and maximization of final food product.

Figure 4.1. Time temperature history of three radial cross-sectional points of the pipe (center, intermediate, wall) of sweet potato puree during heating (until 1700 s) with a continuous flow MW unit (pilot-scale 5 kW, 915 MHz system; Model number P09Y5KA02, Industrial Microwave Systems, Morrisville, NC) and cooling with shell in tube heat exchanger.
4.2.1 Tested Materials

For these series of experiments, sweet potato puree was used as the tested material. The samples of sweet potato puree were aseptically processed and stored in freezers (-20°C) at the pilot plan of the department of Food Science of North Carolina State University (NCSU). Sweet potato puree was chosen because it is a product that shows very good results during the application of advanced heating technologies, such as MW heating, and also because it has rheological characteristics of a highly viscous and multiphase food product with shear thinning behavior. Furthermore, it has similar apparent viscosity between the initial raw product and the final thermal treated sweet potato puree (Cornel et al., 2005), while also presenting an elastic solid behavior (Ahmed and Ramaswamy, 2006).

4.2.2 Experimental set up

To study thermal mixing during the cooling cycle of continuous flow thermal processing of highly viscous foods, a reticulating complete thermal treatment system was built with three basic sections: heating, cooling, and mixing. The thermal treatment system consisted of a food grade progressive cavity dosing pump Seepex MD-012 12 (Fluid Engineering Inc., Birmingham, AL), a continuous flow MW system used for the heating part, a shell in tube heat exchanger used as cooler, and a closed reticulating system with the mixing unit at the exit of cooler for equalizing the temperature profile (Figure 4.2). Approximately five gallons of sweet potato puree were adequate to fill the system and allow for recirculation of the product at a flow rate of 3.75 L/min for all the experiments.
Figure 4.2. a) Schematic diagram and b) picture of experimental apparatus of continuous flow thermal processing system.
Heating

MW technology was used to heat up the tested material. Continuous flow MW heating is an emerging technology in the food industry that has the potential to replace the conventional heating process for viscous and multiphase foods (Coronel et al., 2005; Kumar et al., 2008). In contrast to conventional heating, heating with MW technology provides rapid, uniform volumetric heating of the entire food product (Coronel et al., 2005). This advantage of MW heating allows the processors to eliminate the heating time and minimize the holding time during thermal processing, which results in the maximization the final food product quality by maintaining most of the color, flavor, texture, and nutrients of the initial raw product (Kumar et al., 2008; Cullen et al., 2012). For the purpose of this study, a new continuous flow MW system was designed and built. The MW system consisted of thirteen bench-top Panasonic Inverter MW ovens, each working at 2450 MHz and nine of them delivering a maximum power of 1.250 kW, three MW ovens with 1.300 kW, and one MW oven with high power input at 1.200 kW. The thirteen bench-top Panasonic Inverter MW ovens were split into four rows of MW series, with the first row consisting of four MW ovens and the rest three rows having three MW ovens each (Figure 4.3). In order to construct a continuous flow microwave system, each MW oven was drilled with 2" diameter holes located at the center of the bottom and top of the oven. A polypropylene pipe with an outside diameter of 1.5" and an inside diameter of 1.37" was used to connect the first row of the MW system, while Teflon pipe with the outside diameter of 1.5" and inside diameter of 1.37" was used to connect the MW ovens in the other rows (Figure 4.4). The MW system's rows were connected with a 1" inside diameter silicon pipe. To ensure continuous full pipe flow inside
the MW ovens, the MW system was designed in such way as to ensure upstream flow inside all MW ovens. In order to measure the temperature of the test materials, type-T thermocouples were placed at the inlet and outlet of each MW row. The continuous flow MW system designed and used in this study is a prototype MW heating system tested for the first time in this study. The use of this MW system provides all the advantages of MW technology on the processing of highly viscous and multiphase foods, delivering a high total maximum power input of 16.350 kW in the commercial used MW frequency of 2450 MHz, while minimizing the cost to build, use and maintain through the use of batch home MW ovens.

Figure 4.3. Figure of continuous flow MW system.
Figure 4.4. Pictures of two MW ovens from the first row of MW series continuously connected with polypropylene pipe.

Cooling

Sweet potato puree, after the MW system, was pumped through a shell in tube heat exchanger, which was used as the cooler unit. The heated food was pumped through the inside shell configuration, while water was used on the outside of heat exchanger as cooling fluid (with an initial temperature of approximately 16 °C). Type-T thermocouples were placed at the inlet and outlet of the cooler to measure the temperature of tested food.
Mixing

The mixing unit, which was located at the exit of the cooler, consisted of a 180° bend pipe mounted on the top of a low sonic frequency tactile audio transducer, *Buttkicker LFE* (The Guitammer Company, Westerville, OH). Buttkicker LFE is a cylindrical shape bass shaker standing at 5.375" tall and 5.375" wide and with a total weight of 11 pounds (4.98 kg). It has a minimum handling loudspeaker power (electrical power transferred from an audio amplifier to the transducer) of 400 W and a maximum of 1900 W. Buttkicker LFE contains a magnetically suspended piston, which responds in a low sonic frequency range of 5 Hz and 240 Hz and can produce powerful sinusoidal acoustic/mechanical vibration. To control and generate a sinusoidal harmonic transversal vibration, Buttkicker LFE was mounted into four metal extension springs, which were connected into an aluminum box with the following dimensions: 29"x20"x29" (length, width, height respectively) (Figure 4.5).

The tested material was pumped through an open duct 180° bend pipe, which was mounted at the top of the Buttkicker LFE, with the center of U-turn elbow positioned right above of the Buttkicker LFE's magnetron (Figure 4.5). The 180° bend pipe with an outside diameter of 1.5" and inside diameter of 1.37" was made of a stainless steel U-turn elbow, which was connected with two flexible tubular parts, each 21" in length (Figure 4.6). The two flexible tubular parts were located on the top of a metal base with half of the length of the flexible tubes mounted on the metal base with metal clamps; the other half of the flexible tubes, close to the U-turn elbow, vibrated freely. Rubber was placed between the metal base and the tubular part of the U-turn, where the metal clamps were located, to avoid any damage to the pipe. Rubber was also used as "steps": by increasing the number of rubbers used along
the length of the U-turn in the direction of the flow, at its junction of metal clamp, the 180°
bend pipe, and the metal base, upstream flow and full pipe flow inside the mixing unit was
ensured. To test and determine the effect of vibration on thermal mixing, it was necessary to
control and measure different parameters of vibration, such as frequency, amplitude, and
duration of vibration. In order to establish the efficiency of the thermal mixing process, the
temperature of tested food was measured and recorded during mixing. Time-temperature data
were measured every second using T-type thermocouples in multiple positions through the
system and recorded using a data acquisition system (model DAS-16, Keithley Metrabyte
Inc., Taunton, MA). A specifically designed type of T-type thermocouples, using Keyhole
Multi-point probes (Windridge Sensors LLC, Holly Springs, NC), were used to measure the
thermal profile of tested material at the inlet and the outlet of the 180° bend pipe of the
mixing unit. Keyhole Multi-point probes were made from ultra-thin, stainless steel probe tips
equipped with multipoint thermocouples, which provided information about the radial
temperature profile of the flow. In this study, the Keyhole Multi-point probes were used at
the inlet and outlet of mixing unit, measuring the temperature at three different radial points:
at the center of the pipe, close to the pipe wall, and at an intermediate point (Figure 4.6).

Frequency of acoustic vibration was controlled via a computer program, FreqGen
1.13 (Digital River Inc., Minnetonka, MN). FreqGen 1.13 is a computer frequency generator
program that turns the signal from the soundcard into the right waveform/voltage and allows
the user to control the frequency and time length of the vibration. The computer with the
FreqGen 1.13 was connected inline with an amplifier type BKA1000-N power amplifier (The
Guitammer Company, Westerville, OH), which was also connected inline with the Buttkicker
LFE. Adjusting the volume of the amplifier was the method used to control the amplitude of vibration. Frequency and amplitude of vibration during the mixing process were measured and recorded via an accelerometer, a vibration data logger device called the SlamStick™ (Mide Inc., Medford, MA). The SlamStick™ is a high-speed, ultra-portable, rechargeable USB data logger capable of measuring the acceleration of vibration of all three axes (x, y, z). The SlamStick™ was mounted at top of U-turn elbow to record the time-history of vibration. To process the results from SlamStick™, the computer program, Slamstick viewer (Mide Inc., Medford, MA), was used. Slamstick viewer uses the data from the SlamStick™ and plots the time-history of the acceleration of vibration as a function of time for each axis and provides a one-dimension Fast Fourier Transformation (1-D FFT) plot, which provides information about the frequency distribution over the entire recording. 1-D FFT plot also provides information for the resonance frequency, the frequency with the highest acceleration of vibration during the vibration process.
Figure 4.5. Picture a) of aluminum base of Buttkicker LFE and b) Buttkicker LFE placed at the mounted at the springer of the base with the U-turn elbow located at the top of Buttkicker LFE.
4.2.3 Experimental methods

This study focused on determining the influence of the parameters of vibration, amplitude, and frequency, and the protocols of vibration on thermal mixing during the continuous flow cooling of sweet potato puree. Frequency of vibration was tested first,
following by experiments on the effects of vibration amplitude; finally, tests on different protocols of vibration (continuous vs. discontinuous vibration) were done at the test-acquired best range of frequencies and the best amplitude of vibration.

Experimental studies on frequency were done in two steps: the first step before the beginning of thermal processing and the second step, along with studies on the effect on amplitude of vibration and different protocols of vibration, during the cooling process of sweet potato puree. For the experimental studies on the effect of vibration parameters during cooling, sweet potato puree was initially heated using the MW system. The product was continuously circulated until the product reached the temperature of 100-105 °C. During the heating cycle, the cooler was closed. When sweet potato puree reached the goal temperature, the MW system was turned off and the cooling and the mixing stage of experiments began for the testing of different parameters of vibration.

*Frequency of vibration*

The first step of each experiment was to determine the acoustic frequency that creates the highest magnitude of acceleration of vibration and the resonance frequency of the mixing system. The mixing system consisted from the Buttkicker LFE, the 180° bend pipe, the SlamStick™, and the flowing tested material. The experiments on resonance frequency were done before the beginning of thermal processing, with the sweet potato puree at room temperature. The resonance frequency was determined by changing the acoustic frequency at low sonic range of 10-100 Hz, at a fixed volume on the amplifier, and at 25% of the volume intensity of the BKA1000-N. FreqGen 1.13 was programmed to increase the acoustic frequency by 10 Hz; then, after 10 sec of vibration at each tested frequency, five seconds of
silence (frequency at 0 Hz) followed. SlamStick™ was used to record the data (acceleration and frequency) of vibration. The resonance frequency of the mixing system was determined as the frequency with the highest magnitude of acceleration on the 1-D FFT plot of the Slamstick viewer.

The second step in the frequency vibration studies occurred during the cooling process of sweet potato puree. During cooling of sweet potato puree, different frequencies around that which was determined from the first step resonance frequency were tested. Temperature uniformity at the three radial points of Keyhole Multi-point probes at the exit of the mixing unit were studied for vibrations of 120, 60 and 30 s duration at the maximum volume level on amplifier for the resonance frequency and frequencies in the range of ± 5 Hz from the resonance frequency.

Different frequencies and durations of vibration were tested through different protocols. Based on the results of resonance frequency tests, different frequencies close to the resonance frequency of 20 Hz were studied. The tested frequencies were 15 Hz, 16 Hz, 18 Hz, 24 Hz, 25 Hz and 28 Hz. Additional experiments were done on 30 Hz and 40 Hz. A 60 s duration of vibration was used for the experiments on frequency effects. The effects of different durations of vibration at 120, 60, and 30 s were studied for frequencies of 16 Hz, 20 Hz and 24 Hz. Additionally, during different protocols of vibration, continuous periods (periods with constant tested vibration) vs. discontinuous periods (periods of non-vibration between the tested vibration), were tested for the frequencies of 20 Hz, 24 Hz and 25 Hz. For all these series of experiments, the volume of sound was fixed at 100% of the volume of amplifier and the flow rate was at 3.75 L/min.
Amplitude of vibration

The effect of the amplitude of vibration on thermal mixing during the continuous flow cooling of sweet potato puree was investigated by adjusting the volume level on the BKA1000-N amplifier. Temperature homogenization at the exit of the mixing unit was studied on different vibration amplitudes for a vibration duration of 2 min at the fixed frequency of 20 Hz.

Different volumes of sound were tested for their effects on thermal mixing during the cooling of sweet potato puree in a temperature range of 70-100 °C. Three different volumes were experienced through the act of adjusting the level of the amplifier to 100%, 75%, and 50% of the volume. The frequency of vibration was fixed at 20 Hz, and the duration of vibration was at 120 s for each tested amplitude. A 30 s period of non-vibration was applied between the tests of each different amplitude, working from the highest volume of amplifier to the lowest.

Protocols of vibration

An unmixed region, also known as KAM (from the theorem of Kolmogorov-Arnold-Moser about the persistence of quasi-periodic motions in dynamical systems) “islands” or KAM-tori (Ottino, 1989; Metcalfe, 2010), formed around elliptic periodic points—fluid elements that return to their initial position after a number of periods (Ottino, 1989). In order to investigate which conditions of vibration give the most efficiency mixing, different protocols of vibration, continuous vibration vs. discontinuous, and continuous vibration on resonance frequency vs. a combination of different frequencies close to resonance frequency were examined. The equalization of the temperature radial profile was studied for the
different protocols of vibrations at the best amplitude and frequencies of vibration observed on the first experiments of this study.

4.3 Results and discussion

4.3.1 Time-temperature results during heating

Time-temperature results at the end of recirculation heating process showed that approximately 600 s were required to heat sweet potato puree from a temperature of 25 °C to 105 °C at the exit of MW system, at a flow rate of 3.75 L/min (Figure 4.7).

Figure 4.7. Time-temperature history outlet of the MW system.
4.3.2 Vibration effects on temperature distribution

Resonance frequency

The first set of experiments was completed to determine the frequency with the highest magnitude of acceleration of vibration. All the experiments with sweet potato puree began with resonance frequency determination prior to the start of the thermal process. The results from 1-D FFT plot showed that the resonance frequency for the mixing system was at 20 Hz (Figure 4.8).

![1-D FFT plot, showing acceleration (using as unit Gs) magnitude of vibration of different tested frequencies, at a range of 10-100 Hz.](image)

Figure 4.8. 1-D FFT plot, showing acceleration (using as unit Gs) magnitude of vibration of different tested frequencies, at a range of 10-100 Hz.
Amplitude of vibration

Time temperature history at the inlet and outlet of mixing unit during cooling and vibration experiments at three different points of the pipe (at the center, at the wall, and at an intermediate point) showed an improvement of temperature homogenization due to vibration; the best results were observed at 100% of the volume of amplifier (Figure 4.9). Comparing the temperature difference between the center of the product and at the wall ($\Delta T_{c-w}$), of roughly the same fluid element at the inlet and the outlet of the mixing unit (Figure 4.10), the best mixing was observed at 100% of the volume of amplifier; starting at a $\Delta T_{c-w}$ of 25-30 °C at the inlet of the 180° bend pipe, the $\Delta T_{c-w}$ reduced at 2-3 °C (Figure 4.10). Furthermore, during acoustic vibration at the highest volume, an improvement on the inlet $\Delta T_{c-w}$ was observed (Figure 4.10). For the other two tested amplifier volumes, an outlet $\Delta T_{c-w}$ of 5 °C was recorded for the volume of amplifier at 75%, with an improved inlet $\Delta T_{c-w}$ at 15-20 °C (Figures 4.9 and 4.10). An outlet $\Delta T_{c-w}$ from the 180° bend pipe of 12-15 °C was established for the volume of amplifier at 50%, with an inlet $\Delta T_{c-w}$ at approximately 25-30 °C. The improvement from the inlet and to the outlet $\Delta T_{c-w}$ for the case of 50% of amplifier volume was due to the thermal mixing at the region between the wall and the intermediate point; there was no thermal mixing in the center region and the intermediate point, and the temperature close to the center of the pipe was the same at the inlet and the outlet of the mixing unit (Figures 4.9 and 4.10).
Figure 4.9. Time-temperature history during vibration at different amplitudes at a) the inlet and b) the outlet of the mixing unit, at three different points of the pipe: at the center, at the wall and at an intermediate point. The dashed lines represent the beginning and the solid lines represent the end of vibration at each different volume (50, 75, and 100%) of amplifier. Note that the average time it took fluid particle to move from the inlet to the outlet of U-turn was 24 sec.
Figure 4.10. Comparison of temperatures of center ($T_c$) and wall ($T_w$) elements at the inlet and the outlet of the mixing unit. The dashed lines represent the beginning and the solid lines represent the end of vibration at each different volume (50, 75, and 100%) of amplifier.

From the data of 1-D FFT, the highest acceleration of vibration for each tested volume of sound was in the case of 100% volume of amplifier with a magnitude of acceleration at 6.75 Gs at 20 Hz, whereas the magnitude of acceleration was at 3.50 Gs and 2.31 Gs at 20 Hz for 70% and 50% volumes of amplifier, respectively (Figure 4.11). In 1-D FFT plots, it was observed that, except for the fundamental mode (programmed tested frequency), overtone modes of vibration also existed in frequencies other than the tested frequency. In the 1-D FFT plot for 75% volume of amplifier, overtones at 60 Hz and 80 Hz presented the same peak of acceleration with the tested frequency of 20 Hz.
From the equation of sinusoidal motion (Eq. 4.1), the peak-to-peak amplitude, D can be estimated based on the magnitude or peak acceleration of each different volume of sound (Spaceagecontrol, 2014):

\[ D = \frac{G A}{2\pi^2 F^2} \]  

(4.1)

Where \( D \) is the peak-to-peak amplitude of sinusoidal vibration (mm), \( G \) is a constant for metric system equal with 9.806645 m/s\(^2\), \( A \) is the peak acceleration (Gs), \( \pi \) is a constant equal to 3.141593, and \( F \) is the frequency of sinusoidal vibration (Hz).

The calculated peak-to-peak amplitudes of vibration were 8.38 mm, 4.35 mm, and 2.86 mm, for 100%, 70 %, and 50% volumes of amplifier respectively; this showed that better temperature homogenization was produced at an increased amplitude of vibration.
Figure 4.11. 1-D FFT plots, during vibration at 20 Hz, a) at 100% volume of amplifier, b) at 75% volume of amplifier and c) at 50% volume of amplifier.
**Frequency, duration, and protocols of vibration**

*Frequency of vibration*

Vibration for 60 s at 16 Hz, 20 Hz, 24 Hz, 30 Hz, and 40 Hz, and at a radial temperature range of 80-100 °C was used to examine the effect of frequency on the thermal mixing of sweet potato puree. Time temperature results at the inlet and the outlet of the mixing unit showed that thermal mixing of sweet potato puree, due to vibration, can be achieved at the frequencies of 16 Hz, 20 Hz, and 24 Hz (Figure 4.12). However, no difference on outlet $\Delta T_{c-w}$ resulted from vibration at 30 Hz and 40 Hz: from an inlet $\Delta T_{c-w}$ of 26-28 °C, the outlet $\Delta T_{c-w}$ remained at approximately 15 °C, the same with or without vibration of 30 Hz and 40 Hz, and the temperature at the center remained constant at 98-100 °C at the inlet and at the outlet of mixing unit (Figure 4.12). For the frequencies of 16 Hz, 20 Hz, and 24 Hz the best mixing was observed at 16 Hz, where an improvement of inlet and outlet $\Delta T_{c-w}$ was observed due to vibration. Starting vibration with an inlet $\Delta T_{c-w}$ of 23 °C, the inlet temperature was decreased to 18 °C. For this inlet $\Delta T_{c-w}$ range, very good outlet $\Delta T_{c-w}$ was recorded with a temperature range of 2-5 °C (Figure 4.13). For the frequencies of 20 Hz and 24 Hz, as well as 16 Hz, when an inlet $\Delta T_{c-w}$ above 25 °C was formed, improvement on thermal mixing was observed, but it was not as good as it was in the case where the inlet $\Delta T_{c-w}$ was below 20 °C (Figure 4.13). In that case, no improvement of inlet temperature distribution was noted, and the outlet $\Delta T_{c-w}$ was improved to 5-7 °C from an outlet $\Delta T_{c-w}$ of 16-17 °C without vibration to a $\Delta T_{c-w}$ of 10-12 °C for all these three frequencies. It is important to note that the outlet temperature at the center was reduced at least 3-4 °C due to the vibration at 16 Hz, 20 Hz and 24 Hz.
Figure 4.12. Time-temperature history at a) the inlet and b) the outlet of the mixing unit, at three different cross sectional points of the pipe (at the center, at the wall and at an intermediate point) during continuous vibration of 60 s at different frequencies of 16 Hz, 20 Hz, 24 Hz, 30 Hz, and 40 Hz.
Figure 4.13. Comparison of temperatures between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 60 s at different frequencies of 16 Hz, 20 Hz, 24 Hz, 30 Hz, and 40 Hz.

**Frequency and duration of vibration**

The next series of experiments aimed to determine the most efficient frequency on thermal mixing at the frequency range of 15-25 Hz with the application of 60 s continuous vibrations. The influence of duration and different protocols of vibration were examined through tests of continuous vs. discontinuous vibration and in different durations of vibration: 60 s at 16 Hz, 18 Hz, and 25 Hz, and 120 s at 20 Hz.
Time-temperature history and $\Delta T_{c-w}$ results at the inlet and the outlet of mixing unit showed that vibration at 100% of amplifier volume for 60 s at the region of 15-25 Hz provided very good thermal mixing (Figure 4.14 and 4.15). This experiment tested the effects of 15 Hz, 20 Hz, and 25 Hz in a duration of 60 s following a discontinuous protocol with a 60 s period of non-vibration between of each tested frequency. In these series of tests, the most uniform radial temperature distribution at the outlet of mixing unit was observed at the frequency of 25 Hz, with a $\Delta T_{c-w}$ of 0-2 °C (Figure 4.15). The value of $\Delta T_{c-w}$ at inlet of mixing is a critical factor for the degree of uniformity of the outlet radial temperature distribution. In the case of 15 Hz, the inlet $\Delta T_{c-w}$ started at 30 °C and kept rising during vibration due to the cooling process, while the outlet $\Delta T_{c-w}$ presented a reduction—starting at 22 °C and decreasing to a value of 17 °C at the end of 60 s at 15 Hz (Figure 4.15). An inlet $\Delta T_{c-w}$ of 30-32 °C had been established at the beginning of 60 s vibration at 20 Hz, and after the 60 s vibration, a considerable reduction on both inlet and outlet $\Delta T_{c-w}$ were recorded. At the end of vibration at 20 Hz, the inlet $\Delta T_{c-w}$ was reduced to 12 °C from an initial value of 30-32 °C, while the outlet started at 16 °C and reduced at 5 °C at the end of vibration (Figure 4.15). The inlet $\Delta T_{c-w}$ was at 20 °C at the end of the non-vibration period of 60 s, before the beginning of a 60 s vibration period at 25 Hz. During vibration at 25 Hz, a significant reduction to both inlet and outlet $\Delta T_{c-w}$ was observed with values at 5 °C and between 0-2 °C, respectively (Figure 4.15).

A continuous protocol with a total of 180 s of vibration was tested at the end of 60 s vibrations at 25 Hz; this was followed by 120 s vibrations at 20 Hz. The results at the first 150 s of vibration were significant: As mentioned above, after the first 60 s at 25 Hz the
outlet $\Delta T_{cw}$ was at -2 °C and remained at this region for at least 90 s of vibration at 20 Hz. At the last 30 s of vibration, a suddenly increase of inlet and outlet $\Delta T_{cw}$ was observed, with the final outlet $\Delta T_{cw}$ at the range of 6-8 °C (Figure 4.15). After 180 s of non-vibration (the effects being tested at a radial temperature range of 60-85 °C) no significant results were observed on thermal mixing from vibration; the inlet and outlet temperature of sweet potato puree at the center of pipe remained unchanged at 85 °C (Figure 4.14). It is important to mention that in the last 30 s of this test, Buttkicker LFE turned off because it was overheated.

Two different experiments tested the effects of 120 s vibrations, at the frequency range of 16-20 Hz, on thermal mixing during cooling of sweet potato puree in a radial temperature range of 70-105 °C. In the first experiment, 120 s of vibration was applied in three different frequencies (16 Hz, 18 Hz, and 20 Hz), and at different time-periods of the experiment; the second experiment vibration of 120 s at 20 Hz was applied in different time periods and temperature ranges of the cooling process of sweet potato puree.

The time-temperature history and $\Delta T_{cw}$ results at the inlet and the outlet of mixing unit showed that the most uniform radial temperature distribution at the outlet of mixing unit, was at the frequency of 20 Hz, with a $\Delta T_{cw}$ of 1-3 °C (Figure 4.16; Figure 4.18). Again, the most important factor in the uniformity of final exit temperature from the U-turn was the initial inlet $\Delta T_{cw}$ (Figure 4.17; Figure 4.19). Improvement on radial temperature distribution was observed for all the frequencies and for all the different temperature ranges for the case of 20 Hz and 16 Hz. Reduction of center product temperature was also recorded for all the cases, with the only exception being the results from 18 Hz, where the temperature at the center of the pipe for sweet potato remained unchanged during vibration (Figure 4.16).
Figure 4.14. Time-temperature history during vibration of 60 s at different frequencies, 15 Hz, 20 Hz, and 25 Hz, followed by 120 s of vibration at 20 Hz, a) the inlet and b) the outlet of the mixing unit, at three different cross sectional points of the pipe: at the center, at the wall and at an intermediate point.
In the set of tests with 120 s of vibration at 20 Hz, boiling of the sweet potato was observed before the beginning of the cooling process, and full pipe flow was not observed during the first 180 s of vibration tests in which the temperature of the sweet potato puree in the region between the intermediate point and the center of the pipe was between 96-102 °C. As the pipe was not full, due to the gas produced from boiling, the test did not show what was observed in previous tests but resulted in a very good mixing result with an outlet $\Delta T_{c-w}$ at 1-3 °C, down from the inlet $\Delta T_{c-w}$ at 26-28 °C (Figure 4.19).
Figure 4.16. Time-temperature history a) the inlet and b) the outlet of the mixing unit, at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during vibration of 120 s at different frequencies, 16 Hz, 18 Hz, and 20 Hz, and at different time-periods of cooling.
Figure 4.17. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during vibration of 120 s at different frequencies, 16 Hz, 18 Hz, and 20 Hz, and at different time-periods of cooling.
Figure 4.18. Time-temperature history a) the inlet and b) the outlet of the mixing unit, at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during vibration of 120 s at 20 Hz and at different time-periods of cooling.
Figure 4.19. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit, during vibration of 120 s at 20 Hz, at different time-periods of cooling.

Two different durations of vibration were tested at a radial temperature range 60-95 °C: 120 s at 20 Hz and 30 s at 20 Hz and 25 Hz. A vibration of 30 s was applied in a discontinuous protocol of vibration for a total time of 65 s and consisted of two 30 s periods of vibration at 20 Hz and 25 Hz with an intermediate period of 5 s of non-vibration.

The first application of 120 s of vibration at 20 Hz started at the time the cooler turned on, so the temperature of the tested product was increasing during the first test in this experiment (Figure 4.20). Therefore, independently if the vibration that was applied during
heating or during cooling, very good uniformity of radial temperature distribution occurred for all the cases of duration and protocols of vibration, with an outlet $\Delta T_{c-w}$ at 2-5 °C (Figure 4.21) when the inlet at mixing unit $\Delta T_{c-w}$ of sweet potato puree was below 13 °C (Figure 4.21). Non-uniform temperature was observed for all the types of vibration when the inlet $\Delta T_{c-w}$ was at 27-30 °C with continuous vibration of 120 s to present better uniformity of outlet radial temperature and with an outlet $\Delta T_{c-w}$ of 15 °C (Figure 4.21).

*Discontinuous protocol of vibration*

Different protocols of discontinuous vibration were tested with purpose of addressing the problem of large, unmixed areas (KAM-tori) and with the purpose of determining the most efficient and energy saving method of thermal mixing via vibration. For that reason, discontinuous protocols of vibration with a duration of approximately 90 s were applied at the frequencies of 20 Hz and 24 Hz in a radial temperature range of 70-90 °C for sweet potato puree.

Two different protocols of discontinuous vibration of 90 s and 91 s, respectively, were tested. The first protocol applied nine cycles of vibration of 7 s followed by 3 s of non-vibration at 20 Hz and 24 Hz, and the second seven cycles of 10 s of vibration at 20 Hz were followed by 3 s of non-vibration for each cycle.

Satisfactory results were shown with an outlet $\Delta T_{c-w}$ of 5-10 °C, due the low starting inlet $\Delta T_{c-w}$ of 15-20 °C; in the case of discontinuous vibration, thermal mixing was observed but had no constant efficient behavior. Finally, for the second protocol at 20 Hz, no improvement in the temperature of the product at the center of the pipe was observed between the inlet and the outlet temperature (Figure 4.22; Figure 4.23).
Figure 4.20. Time-temperature history a) the inlet and b) the outlet of the mixing unit, at three different cross sectional points of the pipe (at the center, at the wall and at an intermediate point) during vibration at 20 Hz of 60 s at the start of the experiment and 120 s at the end of the experiment, with an intermediate tested vibration protocol of 75 s combining 30 s of vibration at 20 Hz and 25 Hz, with 5 s of non-vibration period in between.
Figure 4.21. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit, during vibration at 20 Hz of 60 s at the start of the experiment and 120 s at the end of the experiment, with an intermediate tested vibration protocol of 75 s combining 30 s of vibration at 20 Hz and 25 Hz, with 5 s of non-vibration period in between.
Figure 4.22. Time-temperature history a) the inlet and b) the outlet of the mixing unit, at three different cross sectional points of the pipe (at the center, at the wall and at an intermediate point) during vibration of two different protocols of vibration of 90 s and 91 s, respectively. The first one with 7 s vibration followed by 3 s of non-vibration tested at two different frequencies at 20 Hz and 25 Hz, and the second with 10 s vibration at 20 Hz followed by 3 s of non-vibration.
Figure 4.23. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during vibration of two different protocols of vibration of 90 s and 91 s, respectively. The first one with 7 s vibration followed by 3 s of non-vibration tested at two different frequencies at 20 Hz and 25 Hz, and the second with 10 s vibration at 20 Hz followed by 3 s of non-vibration.

Continuous protocol of vibration

In the final experiment of this study, continuous vibration of 4 min at different frequencies, 16 Hz, 20 Hz, 24 Hz, and 28 Hz, with a duration of 60 s at each frequency during thermal processing of sweet potato puree with a radial temperature range of 85-100 °C was examined (Figure 4.24). Application of 60 s of vibration at 16 Hz started at the time the cooler was turned on, with the inlet $\Delta T_{c-w}$ at low temperature of 10 °C at the beginning of
mixing. The results showed very efficient thermal mixing through the whole mixing application, with a $\Delta T_{c-w}$ of 5-8 °C at the inlet and 0-3 °C at the outlet of the U-turn, respectively (Figure 4.25).
Figure 4.24. Time-temperature history during continuous vibration of 4 min at different frequencies, 16 Hz, 20 Hz, 24 Hz and 28 Hz with a duration of 60 s at each frequency a) the inlet and b) the outlet of the mixing unit, at three different cross sectional points of the pipe: at the center, at the wall, and at an intermediate point.
Figure 4.25. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 4 min, with 60s min vibration at frequencies of 16 Hz, 20 Hz, 24 Hz, and 28 Hz.

1-D FFT plots during vibration

In all the 1-D FFT plots for all the different series of experiments studying amplitude, frequency, and different protocols of vibration, it was observed that, except in the fundamental mode (programmed tested frequency), overtone modes also existed that probably helped to produce more uniform mixing by destroying the KAM-tori unmixed regions (Figure 4.26). Overtones were different for each tested frequency, with 45 Hz, 60
HZ, and 75 Hz to be the most characteristic overtones for 15 Hz, 20 Hz, and 25 Hz, respectively (Figure 4.26).

Additionally, based on the results of magnitude of acceleration at fundamental mode of vibration, the degree of thermal mixing in the intermediate and center point regions depended on the acceleration magnitude and, thus, the magnitude of the amplitude of vibration. Using as an example 1-D FFT plots from the experiment of 60 s of vibration at 15 Hz, 20 Hz, and 25 Hz, the magnitude of acceleration at 15 Hz, 20 Hz, and 25 Hz was recorded at 2.12 Gs, 4.20 Gs, and 3.20 Gs. The time-temperature history results reported that the temperature of sweet potato puree at the center of the pipe remained constant during vibration of 15 Hz; with vibrations at 20 Hz and 25 Hz, significant change of temperature at center of the pipe was observed (recall Figure 4.14).

Magnitude of acceleration and, thus, amplitude of vibration was the factor that affected the degree of mixing at the center region of the pipe. For all the experiments in which thermal mixing was extended at the region close to the center of the pipe, the recorded magnitude of acceleration was above 3.20 Gs at the fundamental mode of vibration.
Figure 4.26. 1-D FFT plot for vibrations tests of 60s at a) 15 Hz, b) 20 Hz, and c) 24 Hz.
4.4 Conclusions

Continuous flow thermal processing of sweet potato puree was studied in order to examine thermal mixing effects and the enhancement of continuous flow cooling. A continuous flow recirculation thermal process system was built to study the effects of sinusoidal mechanical vibration, generated from acoustics, on thermal mixing during the cooling process of sweet potato puree in a temperature range of 60-105 °C.

A newly designed continuous flow MW system at 2450 MHz, formed from a series of MW ovens, was built and used for the heating process of sweet potato puree. The MW system produced a very rapid heating process with an average time of 600 s required for the system to heat 5 gal of sweet potato puree up to 105 °C from an initial temperature of 25 °C.

Acoustics were used to impose transverse vibration on a 180° bend pipe and to generate chaotic fluid motion on sweet potato puree during the cooling process, thus achieving thermal mixing. A mixing system was built in which we were able to generate, record, and measure the effects of different acoustic frequencies (0-100 Hz), amplitudes, and protocols of vibration on the radial temperature distribution of sweet potato puree. Results showed that the most efficient mixing occurred with continuous vibration at the highest amplitude of vibration and with the resonance frequency of the system (acoustic vibrator, 180° bend pipe, and product) at 20 Hz. However, good thermal mixing was also observed in frequencies close to the resonance frequency at a range of ±4 Hz from the 20 Hz. The magnitude of the acceleration of vibration was the vibration parameter that most affected the degree of mixing at the fluid close at the center of the pipe: higher peak acceleration resulted in better mixing of sweet potato puree close to the central of the pipe.
Initial temperature difference between the temperature at the center of the pipe and the temperature close to the wall, $\Delta T_{c-w}$, before mixing was observed to be the most important factor that influenced the efficiency of radial temperature uniformity at the exit of the mixing unit. An initial temperature difference above 20-25 °C resulted in a wide radial temperature distribution at the exit of mixing unit, with an average recorded $\Delta T_{c-w}$ of 8-12 °C. However, with an initial temperature difference below 12-15 °C, results showed significant improvement in radial temperature profile with a $\Delta T_{c-w}$ of 0-3 °C at the exit of the mixing unit.

**Acknowledgments**

The author would like to acknowledge the Center for Advanced Processing and Packaging Studies (CAPPS) for their financial support.
NOMENCLATURE

Latin letters

\( A \)  
peak acceleration, Gs

\( D \)  
amplitude of vibration, mm

\( F \)  
frequency of vibration, Hz

\( G \)  
constant for metric system equal with 9.806645 m/s\(^2\)

Greek letters

\( \Delta T_{c-w} \)  
temperature difference between the temperatures at the center and the wall, °C

\( \pi \)  
constant equal with 3.141593

Subscripts

\( c \)  
center

\( w \)  
wall

Abbreviations

1-D FFT one-dimension Fast Fourier Transformation

2-D  
two-dimensional

3-D  
three-dimensional

CAPPS Center for Advanced Processing and Packaging Studies

NCSU  
North Carolina State University (NCSU)

MW  
microwave
RAM  Rotated Arc Mixer

s  second(s)

$T_c$  temperature at the center of the pipe

$T_w$  temperature at the wall of the pipe
REFERENCES


CHAPTER 5: Acoustic enhancement of continuous flow cooling of viscous and multiphase food products

Abstract

Acoustic vibration was used to enhance thermal mixing during continuous flow cooling of highly viscous, shear-thinning food products, such as banana puree, applesauce, sweet potato puree, and canned and aseptically processed cheese sauce, at the temperature range of 60-105 °C. Uniform temperature distribution, with a temperature difference of 0-4 °C between the product at the center of the pipe and the product close to the wall, was observed during the application of continuous vibration at the maximum level of amplifier volume and at 20 Hz for all the products except canned processed cheese sauce, which presented a temperature difference of 6-8 °C. Shear-thinning flow behavior, non-slip wall boundary conditions, and the magnitude of the initial temperature difference between the product temperature at center and close to the wall, before the application of vibration, effected the final uniformity of temperature for all the products. Initial temperature difference, and therefore the degree of viscosity ratio inside the product, was found to be the most important factor to influence the effectiveness of thermal mixing; with a critical value for all products to be found at 12-15 °C were above this value a rather wide temperature distribution (5-12°C) remained for all the tested products.

Key words: Continuous flow thermal process, cooling, acoustics, mechanical vibration, thermal mixing, sweet potato puree, banana puree, applesauce, cheese sauce
5.1 Introduction

In continuous flow thermal processing of viscous and multiphase food products, such as fruit and vegetable purees, soups, and typical dairy products, rapid heat transfer and uniform thermal treatment are vital in reducing time and cost of total process and in retaining the integrity of the final quality of the product (nutrients, color, texture, etc.). Advanced heating technologies, such as continuous flow microwave (MW) systems, ohmic, and radiofrequency heating technologies, provide uniform rapid volumetric heating, which minimizes the heating time and maximizes the final food product quality (Coronel et al., 2005; Steed et al., 2008; Cullen et al., 2012). For cooling processes, the absence of an advanced cooling technology in combination with the laminar flow characteristics and the low conductivity of viscous and multiphase food products leads to a wide temperature distribution within the product and results in unequal thermal treatment and reduction of final food quality.

Food production systems, which utilize advanced thermal methods, could benefit from operational enhancements of the cooling stage of the process, both for existing and future installations. The cooling processes of highly viscous, low conductivity foods can be enhanced significantly by improving radial mixing, which can be achieved by turbulent flow conditions or chaotic advection (Metcalfe and Lester, 2009; Eesa and Barigou, 2010; Saatdjian, et al., 2011). However, in highly viscous food products, due to the dominated viscous forces, turbulence flow is unfeasible or requires too much energy due to a high-pressure drop (Saatdjian, et al., 2011). Mixing in the laminar regime can be enhanced through the technique of chaotic advection. Chaotic advection is a mixing technique in the laminar
flow regime in which, through passive (by periodically changing the geometry of flow) and active (applying periodic non-linear forces) methods, chaotic fluid motion is generated. Chaotic advection generates chaotic trajectories and breaks the mixing barriers—the streamlines where the fluid elements flow in laminar flow (Aref, 1984; Ottino, 1989; Aref, 2002; Cullen, 2009). The characteristic technique of chaotic advection is the periodic stretching and folding of fluid elements, which results in an exponential increase of interfacial area and increased mixing (Aref, 1984; Ottino, 1989; Aref, 2002; Cullen, 2009). Chaotic advection is a method used in the thermal processing of highly viscous materials, for heat transfer enhancement (Acharya et al., 1992; Acharya et al., 2001; Metcalfe and Lester, 2009), and for thermal mixing—improvement of radial temperature uniformity (Lester et al., 2009; Eesa and Barigou, 2010; El Omari and Le Guer, 2010a and 2010b; Saatdjian et al., 2011; Le Guer and El Omari, 2012). Scraped surface heat exchangers and static mixers are the most commonly studied examples for heat transfer enhancement and thermal mixing, respectively (Carlson, 1991; Etchells III and Meyer, 2004). Another active thermal mixing technique proposed and studied is transverse vibration motion. Transverse vibration is a method that can produce chaotic fluid motion and radial mixing in a laminar steady flow (Eesa and Barigou, 2010). Experimental results of thermal mixing during the cooling process of sweet potato puree, presented in Chapter 4, showed that low amplitude, at low sonic frequency (20±4 Hz) continuous transverse vibration, can produce considerable thermal mixing.

In this study, the mixing mechanism proposed in Chapter 4, using acoustics to impose transverse vibration on a 180° bend pipe was used to examine and compare thermal mixing
of different viscous foods (e.g., sweet potato puree, banana puree, applesauce, and cheese sauce) during cooling.

5.2 Materials and methods

5.2.1 Tested Materials

For these series of experiments, sweet potato puree, applesauce, banana puree, and two types of cheese sauce were used as the tested material. The sweet potato puree samples were aseptically processed and stored in freezers at the pilot plan of the Department of Food Science of NCSU until used in experimentation. Aseptically processed applesauce and banana puree samples came from Aseptia/Wright Foods (Troy, NC) and were also stored in freezers at the pilot plan of the Department of Food Science of NCSU. Two different types of cheese sauce were used: one aseptically processed and the other in a container (can) came from Advanced Food Products (Clear Lake, WI). Both were stored at room temperature at the pilot plan of the Department of Food Science of NCSU.

5.2.2 Experimental set up

*Thermal mixing experiments during cooling*

To study thermal mixing and cooling enhancement of the different food products, the reticulating complete thermal treatment system, as described in Chapter 4, was used. The thermal treatment system consisted of a food grade progressive cavity dosing pump, Seepex MD-012 12 (Fluid Engineering Inc., Birmingham, AL), a continuous flow MW system working at 2450 MHz, formed from a series of home MW ovens used for heating, and a shell in tube heat exchanger used as cooler; at the exit of the cooler, the reticulating system was closed with the mixing unit, used to equalize the temperature profile. The mixing unit
consisted of a 180° bend pipe (formed from two flexible tubular parts, 21” in length, and a stainless steel U-turn elbow) mounted at the top of a low sonic frequency tactile audio transducer, *Buttkicker LFE* (The Guitammer Company, Westerville, OH). A computer frequency generator program, *FreqGen 1.13* (Digital River Inc., Minnetonka, MN), was used to control the frequency and length of vibration time. The amplitude of vibration was controlled via a power amplifier type BKA1000-N (The Guitammer Company, Westerville, OH), which was connected inline with the FreqGen 1.13 and the Buttkicker LFE. The parameters of vibration were measured and recorded using an accelerometer, a vibration data logger device called the *SlamStick™* (Mide Inc., Medford, MA), and the data were processing via the computer program *Slamstick viewer* (Mide Inc., Medford, MA).

T-type thermocouples were located in multiple positions throughout the MW system, cooler, and mixing unit. Special multipoint T-type thermocouples, *Keyhole Multi-point probes* (Windridge Sensors LLC, Holly Springs, NC), were used to measure the thermal profile of tested material at three different radial points at both the inlet and the outlet of the 180° bend pipe of the mixing unit: at the center of the pipe, close to the pipe wall, and at an intermediate point. Time-temperature data were recorded every second using a data acquisition system (model DAS-16, Keithley Metrabyte Inc., Taunton, MA).

### 5.2.3 Experimental methods

Thermal mixing of applesauce, banana puree, and the two different kinds of cheese sauce were examined using the method of transverse sinusoidal vibration generated from acoustics, as described in Chapter 4, as an active technique to generate chaotic advection.
All the tests, for all the different food products, started with experiments to determine the resonance frequency of the mixing system (consisting of the Buttkicker LFE, the 180° bend pipe, the SlamStick™, and the tested material). As described in Chapter 4, resonance frequency was determined by scanning acoustic frequencies in the low sonic range of 10-100 Hz and at increasing frequency steps of 10 Hz. 10 s of vibration was followed by 5 s of silence (0 Hz) at a fixed volume on the amplifier—25% of the maximum volume intensity of the amplifier. The SlamStick™ was used to record the data (acceleration and frequency) of vibration and the resonance frequency was determined as the frequency with the highest magnitude of acceleration on the one-dimensional Fast Fourier Transformation (1-D FFT) plot of the Slamstick viewer.

In addition to the resonance frequency measurements at the U-turn elbow of mixing unit, the same experimental method was used to determine the frequency with the highest magnitude of vibration in the flexible tube at the inlet of the mixing unit for the series of experiments with the aseptically processed cheese sauce (Figure 5.1). A second SlamStick™ was placed at the middle of the flexible tube, which was located at the inlet of the mixing unit, to measure and record the vibration data. The recorded results were processed with the 1-D FFT plot of the Slamstick viewer.
Figure 5.1. Picture of mixing system during resonance frequency tests, with two SlamStick™ devices: one at the top of U-turn elbow and the other at middle of the flexible tube close to the inlet of mixing unit.

Rheological tests

Rheological parameters, shear stress, shear rate, and apparent viscosity of all test materials were measured with a StressTech rheometer (Reologica Instruments AB, Lund, Sweden) using the serrated bob and cup measuring system. Rheological parameters were measured at different temperatures, in a range between 25 and 95 °C, using the same sample for all the different temperature measurements. The measurements started at 95 °C, and the
rest of the tests were done by decreasing the temperature by 10 °C at each subsequent measurement, with the last measurement at 25 °C.

5.3 Results

5.3.1 Banana puree

*Time-temperature results during heating*

Time-temperature results at the end of recirculation heating process showed that approximately 1360 s were required to heat banana puree from a temperature of 25 °C to 94 °C at the exit of MW system, at a flow rate of 3.75 L/min (Figure 5.2).

![Time-temperature history at the exit of the MW system for banana puree.](image)

Figure 5.2. Time-temperature history at the exit of the MW system for banana puree.
Thermal mixing during cooling of banana puree

Using the results from 1-D FFT plot, the resonance frequency of the mixing system with banana puree as the tested product was determined at 20 Hz for all the series of experiments with banana puree (Figure 5.3).

Figure 5.3. Effect of the acoustic frequency on the acceleration (using as unit Gs) magnitude of vibration of different tested frequencies, as determined from 1-D FFT plot, for banana puree.
Thermal mixing of banana puree during the cooling process in a temperature range of 60-94 °C was studied employing vibration for 120 s at 20 Hz and at 100% amplifier volume with a constant flow rate of 3.75 L/min, during all the experiments. Two different studies on banana puree thermal mixing where done in order to address the problem of non-slip wall conditions due to gel forming at the wall during the cooling of banana puree below 85 °C (approximately). In the first experiment, vibration started after the beginning of banana puree gelation so as to observe if vibration could break up the gel at the wall. In the second experiment, vibration started at the end of the heating cycle and was applied periodically during cooling.

Time temperature history at the inlet and outlet of the mixing unit during cooling experiments under vibration at three different points of the pipe (at the center, at the wall, and at an intermediate point) showed significant improvement in temperature uniformity, in the temperature range between 80 and 94 °C, with and without vibration (Figure 5.4; Figure 5.6). Comparing the temperature difference between the temperature at the center of the product and the temperature at the wall (ΔT_{c-w}) (Figure 5.5; Figure 5.7), a ΔT_{c-w} of 10-15 °C at the inlet of 180° bend pipe was reduced to 0-2 °C ΔT_{c-w} due to the effects of vibration (Figure 5.5). For the same inlet temperature range without vibration, a larger ΔT_{c-w} at the outlet, 3-4 °C, occurred due only to the radial mixing from the 180° bend pipe was recorded (Figure 5.5). Below 80 °C, in the temperature range of 75-80 °C, gel formation was observed, resulting in a gradual increase of the outlet ΔT_{c-w} to 3-4 °C, while the inlet ΔT_{c-w} was at 8-12 °C. Temperature variation was not clearly affected by vibration, with the outlet ΔT_{c-w} remaining at the same value with or without vibration, with an approximately improvement.
of 1-3 °C due to vibration (Figure 5.7). This is attributed to the non-slip wall conditions
created during gelation of banana puree, which had an effect on chaotic advection mixing by
reducing the effects of the secondary flow transverse to the main axial flow created by
vibration (Thiffeault et al., 2011).
Figure 5.4. Time-temperature history for banana puree at a) the inlet and b) the outlet of the mixing unit, at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 120 s at different 20 Hz. The dashed lines represent the beginning, and the solid lines represent the end of vibration. Note that the average time it took for the fluid particle to move from the inlet to the outlet of the U-turn was 24 sec.
Figure 5.5. Comparison of temperature differences between center ($T_c$) and wall ($T_w$) at the inlet and the outlet of the mixing unit during continuous vibration of 120 s at 20 Hz for banana puree.
Figure 5.6. Time-temperature history for banana puree at a) the inlet and b) the outlet of the mixing unit, at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 120 s at different frequencies of 20 Hz.
Figure 5.7. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 120 s at 20 Hz, for banana puree.

*Rheological tests*

The shear-dependent flow behavior for banana puree at different temperatures is depicted on Figure 5.8. The lines on Figure 5.8 are generated from the power law simulation (Eq. 5.1) when fitted to the experimental data.

$$\tau = K\gamma^n$$  \hspace{1cm} (5.1)

Where $\tau$ is the shear stress (Pa), $\gamma$ is the shear rate ($s^{-1}$), $K$ is the flow consistency index (Pa·s$^n$), and $n$ is the flow behavior index (dimensionless).
Figure 5.8. Shear stress vs. shear rate relationship of banana puree at different temperatures. Symbols refer to experimental data and lines refer to simulated values through Eq. 5.1.

The values for the flow consistency index ($K$) and the flow behavior index ($n$) at different temperatures are presented on Table 5.1.
Table 5.1. Flow consistency index \( (K) \) and flow behavior index \( (n) \) values for banana puree at different temperatures calculated through Eq. 5.1.

<table>
<thead>
<tr>
<th>( T ) (°C)</th>
<th>( K ) (Pa·s(^n))</th>
<th>( n )</th>
<th>( R^2 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>25</td>
<td>14.2256</td>
<td>0.3093</td>
<td>0.9972</td>
</tr>
<tr>
<td>35</td>
<td>12.7696</td>
<td>0.3043</td>
<td>0.9977</td>
</tr>
<tr>
<td>45</td>
<td>11.8055</td>
<td>0.2938</td>
<td>0.9959</td>
</tr>
<tr>
<td>55</td>
<td>9.9925</td>
<td>0.3035</td>
<td>0.9960</td>
</tr>
<tr>
<td>65</td>
<td>9.8932</td>
<td>0.2927</td>
<td>0.9921</td>
</tr>
<tr>
<td>75</td>
<td>9.2829</td>
<td>0.2895</td>
<td>0.9928</td>
</tr>
<tr>
<td>85</td>
<td>7.4107</td>
<td>0.3138</td>
<td>0.9876</td>
</tr>
<tr>
<td>95</td>
<td>12.8867</td>
<td>0.2350</td>
<td>0.9890</td>
</tr>
</tbody>
</table>

At all temperatures, a shear-thinning behavior of the banana puree samples, as indicated by the flow behavior index \( (n < 1) \) and shown on Figure 5.9 with the decrease of apparent viscosity with an increase of shear rate, is evident (in agreement with Figura and Teixeira (2007)). Excluding the data at 95 °C, an average flow behavior index, \( n \), equal to 0.2927\( \pm \)0.0092 was found. As reported in other studies (Metcalf et al., 2006), large unmixed regions were observed during mixing studies of shear thinning materials, which required an extra mixing process in comparison with Newtonian and shear thickening materials. In our case, imposed transversal vibration reduced the viscosity of product close to the vibrating wall, which resulting in less efficient chaotic advection results (reduction of stretching degree).
Figure 5.9. Effect of temperature on the apparent viscosities of banana puree at different shear rates.

As shown on Figure 5.9, the apparent viscosities of banana puree up to a temperature of 85 °C showed a significant decrease in temperature. The same is true for the K values (Table 5.1). The 95 °C data (Figure 5.8; Figure 5.9; Table 5.1) indicate a gelation process above the temperature of 85 °C. In addition, a big difference between the apparent viscosities at the temperatures of thermal mixing experiments, 60-94 °C, existed (Figure 5.9), indicating that a wide viscosity ratio existed within the banana puree during the thermal mixing application. Due to the high viscosity ratio, the resistance at the interface between flow streams is very strong. In such situations, a phenomenon called the *miscible interface* arises,
which blocks the mixing (Etchells III and Meyer, 2004). According to Grace (1982), application of an elongational shear has proved to be more successful in fixing this problem by making the resistant layers of different viscosities thinner (Rauwendaal, 2014).

5.3.2 Applesauce

Time-temperature results during heating

Time-temperature results at the end of recirculation heating process showed that approximately 1075 s were required to heat applesauce from a temperature of 25 °C to 100 °C at the exit of MW system at a flow rate of 3.33 L/min (Figure 5.10).

Figure 5.10. Time-temperature history at the exit of the MW system for applesauce.
Thermal mixing during cooling of banana puree

Using the results from 1-D FFT plot, the resonance frequency of the mixing system with applesauce as the tested product was determined at 60 Hz, at 50% of the level of the volume of amplifier and at 20 Hz at 25% of the level of the volume of amplifier for different series of experiments with applesauce (Figure 5.11).
Figure 5.11. Effect of the acoustic frequency on the acceleration magnitude of vibration of the mixing system with applesauce as the tested product, as determined by the 1-D FFT plot, at a) 50% and b) 25% of the level of the volume of amplifier.
Thermal mixing of applesauce during the cooling process in a temperature range of 60-95 °C was studied employing vibration for 120 s at 40 Hz and 60 Hz and 60 s at 20 Hz and 24 Hz, with the volume of amplifier fixed at the maximum level and the flow at 3.33 L/min.

Thermal mixing of applesauce was tested using vibration at 60 Hz (the resonance frequency measured during this experiment) and at 40 Hz (the second mode observed at the 1D-FFT plot). Vibration at 60 Hz started with a wide temperature distribution at the inlet of the mixing unit, with the initial inlet ΔTc-w at 35 °C and the initial outlet ΔTc-w at 25 °C (Figure 5.12; Figure 5.13). During vibration at 60 Hz, a considerable improvement of 6-8 °C on both inlet and outlet ΔTc-w values were shown, with final inlet at 26-28 °C and outlet at 18-20 °C (Figure 5.13). Vibration at 40 Hz did not show any improvement in thermal mixing of applesauce, with the outlet ΔTc-w remaining approximately the same value of 14 °C, before, during, and after vibration at 40 Hz (Figure 5.13).

The next series of experiments with applesauce were done with the resonance frequency of the mixing system recorded at 20 Hz. From time temperature history, results showed significant improvement in thermal mixing of applesauce with vibration at frequencies of 20 Hz and 24 Hz (Figure 5.14; Figure 5.15). Significant improvement in inlet and outlet ΔTc-w was recorded at 20 Hz. The inlet ΔTc-w was reduced from 18 °C to 6 °C at the end of vibration, while the outlet ΔTc-w was reduced from 14 °C to 6 °C. With vibration at 25 Hz, a uniform temperature profile was reported upon exiting the mixing unit, with an outlet ΔTc-w of 2-4 °C when the inlet ΔTc-w was at 9 °C (Figure 5.15).
Figure 5.12. Time-temperature history for applesauce at a) the inlet and b) the outlet of the mixing unit at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 120 s at 60 Hz and 40 Hz. Note that the average time it took a fluid particle to move from the inlet to the outlet of U-turn was 32 sec.
Figure 5.13. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 120 s at 60 and 40 Hz for applesauce.
Figure 5.14. Time-temperature history for applesauce at a) the inlet and b) the outlet of the mixing unit at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 120 s at 20 Hz and 24 Hz.
Figure 5.15. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 120 s at 20 Hz and 24 Hz, for applesauce.

Rheological tests

The shear-dependent flow behavior for applesauce at different temperatures is depicted on Figure 5.16. The lines on Figure 5.16 were generated from the power law simulation (Eq. 5.1) when fitted to the experimental data.
Figure 5.16. Shear stress vs. shear rate relationship of canned processed applesauce at different temperatures. Symbols refer to experimental data and lines refer to simulated values calculated through Eq. 5.1.

The values for the flow consistency index and the flow behavior index at different temperatures are presented on Table 5.2. At all temperatures, a shear-thinning behavior of the applesauce, as indicated by the flow behavior index \( n < 1 \), is shown with an average flow behavior index, \( n \), of 0.1368 ±0.0086.
Table 5.2. Flow consistency index and flow behavior index values for applesauce at different temperatures calculated through Eq. 5.1.

<table>
<thead>
<tr>
<th>$T$ (°C)</th>
<th>$K$ (Pa·s$^n$)</th>
<th>$n$</th>
<th>$R^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>25</td>
<td>51.7592</td>
<td>0.1544</td>
<td>0.9972</td>
</tr>
<tr>
<td>35</td>
<td>49.4612</td>
<td>0.1435</td>
<td>0.9977</td>
</tr>
<tr>
<td>45</td>
<td>47.7797</td>
<td>0.1378</td>
<td>0.9959</td>
</tr>
<tr>
<td>55</td>
<td>46.0007</td>
<td>0.1337</td>
<td>0.9960</td>
</tr>
<tr>
<td>65</td>
<td>44.6610</td>
<td>0.1298</td>
<td>0.9921</td>
</tr>
<tr>
<td>75</td>
<td>42.7568</td>
<td>0.1275</td>
<td>0.9928</td>
</tr>
<tr>
<td>85</td>
<td>38.3376</td>
<td>0.1331</td>
<td>0.9876</td>
</tr>
<tr>
<td>95</td>
<td>35.0429</td>
<td>0.1346</td>
<td>0.9890</td>
</tr>
</tbody>
</table>

As shown on Figure 5.17, the apparent viscosities of applesauce in the temperature range of 65-95 °C were influenced by the temperature, indicating that during the thermal mixing application, a wide viscosity ratio existed within the applesauce, which reduces mixing efficiency. While in the temperature range of 55-75 °C, no significant changes in apparent viscosities values were observed. Similar apparent viscosity values in the temperature range of 65-75 °C can explain the good thermal mixing results reported in that temperature range with vibration at 20 Hz and 24 Hz.
5.3.3 Cheese sauce (canned processed)

*Time-temperature results during heating*

Time-temperature results at the end of the recirculation heating process showed that approximately 2600 s were required to heat canned processed cheese sauce from a temperature of 25 °C to 112 °C at the exit of MW system at a flow rate of 3.50 L/min (Figure 5.18).
Figure 5.18. Time-temperature history at the exit of the MW system for canned processed cheese sauce.

*Thermal mixing during cooling of canned processed cheese sauce*

For all the series of experiments with canned processed cheese sauce resonance frequency of the mixing unit was determined at 20 Hz from 1-D FFT plot (Figure 5.19).
Figure 5.19. Effect of the acoustic frequency on the acceleration magnitude of vibration of different tested frequencies, as determined from 1-D FFT plot, for different series of experiments with canned processed cheese sauce.

Thermal mixing of canned processed cheese sauce during the cooling process in a temperature range of 60-112 °C was studied using vibration of 60 s at frequencies of 16 Hz, 20 Hz, and 25 Hz and 120 s at 20 Hz. During all the experiments on canned processed cheese sauce, the amplifier volume was fixed at the maximum level and the flow rate was constant at 3.50 L/min.

Time temperature history results showed that the best thermal mixing was at 20 Hz (Figure 5.20; Figure 5.21; Figure 5.23). An outlet $\Delta T_{c-w}$ of 6-8 °C was observed when the
inlet ΔT_{c-w} was between 10-15 °C, while an outlet ΔT_{c-w} of 10-12 °C was observed when the inlet ΔT_{c-w} was at 18-23 °C (Figure 5.21; Figure 5.23). For the other two tested frequencies, the best results were observed with a vibration of 25 Hz, where both inlet ΔT_{c-w} and outlet ΔT_{c-w} were improved during vibration with the inlet ΔT_{c-w} starting with a value of 20 °C and reduced to the value of 15 °C, while the outlet ΔT_{c-w} starting with a value of 15 °C was reduced to the value of 12 °C at the end of vibration (Figure 5.21). Vibration at 16 Hz had less of an effect on thermal mixing, with an increase of 3 °C on both values of inlet and outlet ΔT_{c-w} observed compared with the values of ΔT_{c-w} before the start of vibration at 16 Hz (Figure 5.21).
Figure 5.20. Time-temperature history for canned processed cheese sauce at a) the inlet and b) the outlet of the mixing unit at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 60 s at 16 Hz, 20 Hz, and 25 Hz, and 120 s at 20 Hz. Note that the average time it took the fluid particle to move from the inlet to the outlet of U-turn was 28 sec.
Figure 5.21. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 60 s at 16 Hz, 20 Hz, and 25 Hz and 120 s at 20 Hz for canned processed cheese sauce.
Figure 5.22. Time-temperature history for canned processed cheese sauce at a) the inlet and b) the outlet of the mixing unit at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 120 s at different 20 Hz.
Figure 5.23. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 120 s at 20 Hz for canned processed cheese sauce.

**Rheological tests**

The shear-dependent flow behavior of canned processed cheese sauce at different temperatures is depicted on Figure 5.24. The lines on Figure 5.24 are generated from the power law simulation (Eq. 5.1) when fitted to the experimental data.
Figure 5.24. Shear stress vs. shear rate relationship of canned processed cheese sauce at different temperatures. Symbols refer to experimental data and lines refer to simulated values calculated through Eq. 5.1.

The values for the flow consistency index and the flow behavior index at different temperatures are presented on Table 5.3. At all temperatures, a shear-thinning behavior of the canned processed cheese sauce, as indicated by the flow behavior index ($n < 1$), with average flow behavior index, $n$, of 0.352 ±0.0327 was observed. It is noticeable that the flow consistency index increased with temperature up to 45 °C and decreased thereafter.
Table 5.3. Flow consistency index and flow behavior index values for canned processed cheese sauce at different temperatures as calculated through Eq. 5.1.

<table>
<thead>
<tr>
<th>$T$ (°C)</th>
<th>$K$ (Pa·s$^n$)</th>
<th>$n$</th>
<th>$R^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>25</td>
<td>26.2795</td>
<td>0.3773</td>
<td>0.9972</td>
</tr>
<tr>
<td>35</td>
<td>45.8635</td>
<td>0.2711</td>
<td>0.9977</td>
</tr>
<tr>
<td>45</td>
<td>57.1640</td>
<td>0.2783</td>
<td>0.9959</td>
</tr>
<tr>
<td>55</td>
<td>48.7311</td>
<td>0.2999</td>
<td>0.9960</td>
</tr>
<tr>
<td>65</td>
<td>41.3700</td>
<td>0.3125</td>
<td>0.9921</td>
</tr>
<tr>
<td>75</td>
<td>35.2104</td>
<td>0.2966</td>
<td>0.9928</td>
</tr>
<tr>
<td>85</td>
<td>28.8616</td>
<td>0.2899</td>
<td>0.9876</td>
</tr>
<tr>
<td>95</td>
<td>22.8908</td>
<td>0.3087</td>
<td>0.9890</td>
</tr>
</tbody>
</table>

As shown on Figure 5.25, the apparent viscosities of canned processed cheese sauce presented a wide difference between the apparent viscosities at the temperatures of thermal mixing experiments, 60-95 °C, indicating that during the thermal mixing application a wide viscosity ratio existed within the canned processed cheese sauce, which reduced mixing efficiency.
Figure 5.25. Effect of temperature on the apparent viscosities of canned processed cheese sauce at different shear rates.

5.3.4 Cheese sauce (aseptically processed)

Time-temperature results during heating

Time-temperature results at the end of the recirculation heating process showed that approximately 1260 s were required to heat aseptically processed cheese sauce from a temperature of 25 °C to 98 °C at the exit of the MW system at a flow rate of 3.75 L/min (Figure 5.26).
Figure 5.26. Time-temperature history at the exit of the MW system for aseptically processed cheese sauce.

*Thermal mixing during cooling of aseptically processed cheese sauce*

Using the data from the measurements at the U-turn elbow of mixing unit, in the 1-D FFT plot, the resonance frequency of the mixing unit was determined at 20 Hz for all the experiments with aseptically processed cheese sauces (Figure 5.27). Additionally, the 1-D FFT plot showed that the frequency with the highest magnitude of vibration at the flexible tube at the inlet of the mixing unit was 60 Hz (Figure 5.28).
Figure 5.27. Effect of the acoustic frequency on the acceleration magnitude of vibration of different tested frequencies at the U-turn elbow, as determined from 1-D FFT plot, for different series of experiments with aseptically processed cheese sauce.
Thermal mixing of aseptically processed cheese sauce during the cooling process in a temperature range of 60-98 °C was studied employing vibration of 60 s at frequencies of 60 Hz and 20 Hz and 120 s and 240 s at 20 Hz. During all the experiments of aseptically processed cheese sauce, the amplifier volume was fixed at the maximum level and the flow rate was constant at 3.75 L/min.

Time temperature history results showed that the best thermal mixing was at 20 Hz (Figure 5.29; Figure 5.31; Figure 5.33). An outlet $\Delta T_{c,w}$ of 2-4 °C was observed when the
inlet $\Delta T_{c-w}$ was below 12-14 °C, while an outlet $\Delta T_{c-w}$ of 6-9 °C was observed when the inlet $\Delta T_{c-w}$ higher than 15 °C (Figure 5.30; Figure 5.32; Figure 5.34). Vibration at 60 Hz, the frequency with the highest acceleration magnitude of vibration at the flexible tube at the inlet of the mixing unit, was tested as a method to improve or control the inlet $\Delta T_{c-w}$. The results of vibration at 60 Hz showed no significant effects on outlet $\Delta T_{c-w}$ when the inlet $\Delta T_{c-w}$ was below 15 °C (Figure 5.32; Figure 5.34). While an improvement of 1-2 °C on both inlet and outlet $\Delta T_{c-w}$ in comparison to the results at 20 Hz was observed when the inlet $\Delta T_{c-w}$ was above 15 °C (Figure 5.32; Figure 5.34).
Figure 5.29. Time-temperature history for aseptically processed cheese sauce at a) the inlet and b) the outlet of the mixing unit at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 60 s at 60 Hz and 20 Hz and 120 s at 20 Hz. Note that the average time it took the fluid particle to move from the inlet to the outlet of U-turn was 24 sec.
Figure 5.30. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 60 s at 60 Hz and 20 Hz and 120 s at 20 Hz for aseptically processed cheese sauce.
Figure 5.31. Time-temperature history for aseptically processed cheese sauce at a) the inlet and b) the outlet of the mixing unit at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 60 s at 60 Hz and 20 Hz and 120 s at 20 Hz.
Figure 5.32. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 60 s at 60 Hz and 20 Hz for aseptically processed cheese sauce.
Figure 5.33. Time-temperature history for aseptically processed cheese sauce at a) the inlet and b) the outlet of the mixing unit at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 60 s at 60 Hz and 20 Hz and 120 s at 20 Hz.
Figure 5.34. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 60 s at 60 Hz and 20 Hz and 120 s at 20 Hz for aseptically processed cheese sauce.

Rheological tests

The shear-dependent flow behavior for aseptically processed cheese sauce at different temperatures is depicted on Figure 5.35. The lines on Figure 5.35 are generated from the power law simulation (Eq. 5.1) when fitted to the experimental data.
Figure 5.35. Shear stress vs. shear rate relationship of aseptically processed cheese sauce at different temperatures. Symbols refer to experimental data and lines refer to simulated values calculated through Eq. 5.1.

The values for the flow consistency index and the flow behavior index at different temperatures are presented on Table 5.4. At all temperatures, a shear-thinning behavior of the aseptically processed cheese sauce was observed, as indicated by the flow behavior index ($n < 1$) with an average flow behavior index, $n$, of 0.2731 ±0.0596. As in the case of canned processed cheese sauce, the flow consistency index increased to 45 °C, and thereafter decreased; likely responsible for that observation is the starch base used for the formation of both kinds of cheese sauce. According to the literature, temperature and starch base are the
two factors that affect the most the rheological behavior of cheese sauce (Guinee et al., 1994).

Table 5.4. Flow consistency index and flow behavior index values for aseptically processed cheese sauce at different temperatures as calculated through Eq. 5.1.

<table>
<thead>
<tr>
<th>$T$ (°C)</th>
<th>$K$ (Pa·s$^n$)</th>
<th>$n$</th>
<th>$R^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>25</td>
<td>11.5914</td>
<td>0.3814</td>
<td>0.9972</td>
</tr>
<tr>
<td>35</td>
<td>24.9827</td>
<td>0.2571</td>
<td>0.9977</td>
</tr>
<tr>
<td>45</td>
<td>37.2679</td>
<td>0.2105</td>
<td>0.9959</td>
</tr>
<tr>
<td>55</td>
<td>34.4939</td>
<td>0.2166</td>
<td>0.9960</td>
</tr>
<tr>
<td>65</td>
<td>31.4435</td>
<td>0.2328</td>
<td>0.9921</td>
</tr>
<tr>
<td>75</td>
<td>21.1200</td>
<td>0.2909</td>
<td>0.9928</td>
</tr>
<tr>
<td>85</td>
<td>17.8434</td>
<td>0.2926</td>
<td>0.9876</td>
</tr>
<tr>
<td>95</td>
<td>17.3159</td>
<td>0.3031</td>
<td>0.9890</td>
</tr>
</tbody>
</table>

The values of apparent viscosity of aseptically processed cheese sauces did not change where good thermal mixing was observed (Figure 5.36). While a wide difference between the apparent viscosities at the temperatures of 60-95 °C was presented, the temperature range in which the vibration effected thermal mixing was reduced.
Figure 5.36. Effect of temperature on the apparent viscosities of aseptically processed cheese sauce at different shear rates.

5.3.5 Sweet potato puree

*Thermal mixing during cooling of sweet potato puree*

Using the results from 1-D FFT plot, the resonance frequency of the mixing system with sweet potato puree as the tested product was determined at 20 Hz (Figure 5.37).
Figure 5.37. Effect of the acoustic frequency on the acceleration magnitude of vibration of the mixing system with sweet potato puree as the tested product, as determined from the 1-D FFT plot.

Thermal mixing of sweet potato puree during the cooling process in a temperature range of 68-98 °C was studied employing vibration for 120 s at 20 Hz with the amplifier volume fixed at the maximum level and the flow rate at 3.75 L/min.

In Chapter 4, the results of thermal mixing of sweet potato puree showed that the effectiveness of vibration on thermal mixing of sweet potato puree depended on the inlet ΔT_{c-w} value, with a critical value of 12-15 °C. In this study, two different cases with different inlet ΔT_{c-w} were tested to confirm the conclusions of Chapter 4. The first case referred to an
inlet $\Delta T_{c-w}$ above 12-15 °C (Figure 5.38; Figure 5.39), and the second case referred to an inlet $\Delta T_{c-w}$ below 12-15 °C (Figure 5.40; Figure 5.41). The results confirmed the conclusions of Chapter 4, with an outlet $\Delta T_{c-w}$ of about 10-14 °C for the first case of study and an outlet $\Delta T_{c-w}$ of 0-3 °C for the second case (Figure 5.39; Figure 5.41). Rheological tests on sweet potato puree were performed in attempt to explain this behavior.
Figure 5.38. Time-temperature history of sweet potato puree at a) the inlet and b) the outlet of the mixing unit at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 120 s at different 20 Hz. Note that the average time it took the fluid particle to move from the inlet to the outlet of U-turn was 24 sec.
Figure 5.39. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 120 s at 20 Hz for sweet potato puree.
Figure 5.40. Time-temperature history for sweet potato puree at a) the inlet and b) the outlet of the mixing unit at three different cross sectional points of the pipe (at the center, at the wall, and at an intermediate point) during continuous vibration of 120 s at different 20 Hz.
Figure 5.41. Comparison of temperature differences between $T_c$ and $T_w$ elements at the inlet and the outlet of the mixing unit during continuous vibration of 120 s at 20 Hz for sweet potato puree.

*Rheological tests*

The shear-dependent flow behavior of sweet potato puree at different temperatures is depicted on Figure 5.42. The lines on Figure 5.42 are generated from the power law simulation (Eq. 5.1) when fitted to the experimental data.
Figure 5.42. Shear stress vs. shear rate relationship of sweet potato puree at different temperatures. Symbols refer to experimental data and lines refer to simulated values calculated through Eq. 5.1.

The values for the flow consistency index and the flow behavior index at different temperatures are presented on Table 5.5. At all temperatures, a shear-thinning behavior of the sweet potato puree was observed, as indicated by the flow behavior index \( n < 1 \), with average flow behavior index, \( n \), of \( 0.1894 \pm 0.0150 \).
Table 5.5. Flow consistency index and flow behavior index values for sweet potato puree at different temperatures calculated through Eq. 5.1.

<table>
<thead>
<tr>
<th>$T$ (°C)</th>
<th>$K$ (Pa·s$^n$)</th>
<th>$n$</th>
<th>$R^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>25</td>
<td>77.2606</td>
<td>0.2128</td>
<td>0.9972</td>
</tr>
<tr>
<td>35</td>
<td>67.8757</td>
<td>0.2094</td>
<td>0.9977</td>
</tr>
<tr>
<td>45</td>
<td>61.3374</td>
<td>0.1807</td>
<td>0.9959</td>
</tr>
<tr>
<td>55</td>
<td>55.7371</td>
<td>0.1944</td>
<td>0.9960</td>
</tr>
<tr>
<td>65</td>
<td>51.0315</td>
<td>0.1812</td>
<td>0.9921</td>
</tr>
<tr>
<td>75</td>
<td>46.3909</td>
<td>0.1762</td>
<td>0.9928</td>
</tr>
<tr>
<td>85</td>
<td>44.1793</td>
<td>0.1723</td>
<td>0.9876</td>
</tr>
<tr>
<td>95</td>
<td>38.8071</td>
<td>0.1880</td>
<td>0.9890</td>
</tr>
</tbody>
</table>

As shown on Figure 5.43, the apparent viscosities of sweet potato puree in the temperature range of 25-95 °C were influenced by the temperature; this indicated that, during the thermal mixing application, a wide viscosity ratio existed within the sweet potato puree, which reduced mixing efficiency, as observed in the first case of study with sweet potato puree. In the temperature range of 75-85 °C no significant changes in apparent viscosities values were observed. Similar apparent viscosity values in the temperature range of 75-85 °C can explain the good thermal mixing results reported in the second case of study with sweet potato puree.
Figure 5.43. Effect of temperature on the apparent viscosities of sweet potato puree at different shear rates.

5.4 Conclusions

Acoustic vibration effectiveness on thermal mixing was studied during the cooling process of different highly viscous food products, such as banana puree, applesauce, cheese sauce, and canned and aseptically processed cheese sauce, and sweet potato puree. Good thermal mixing results were observed with vibration at 20 Hz, with the resonance frequency of the mixing system (acoustic vibrator, 180° bend pipe and product), and with at the maximum amplifier volume level for banana puree, applesauce aseptically processed cheese sauce and sweet potato puree. Uniform temperature was observed for these tested products,
with a temperature difference between the product at the center of the pipe and the product flowing close to the wall ($\Delta T_{c-w}$) of 0-4 °C upon exiting the mixing unit. Improvement in temperature uniformity, due to applied vibration, was observed for canned processed cheese sauce with the best $\Delta T_{c-w}$ at the exit of mixing unit at 6-8 °C.

Three different parameters were found to have the greatest effect on the degree of thermal mixing. With order of importance, these factors were the $\Delta T_{c-w}$ at the inlet of mixing unit, the shear thinning behavior, and the non-slip boundary conditions. Higher initial $\Delta T_{c-w}$ had, as a result, a high viscosity ratio between different product elements, resulting in a reduction of vibration effectiveness on thermal mixing. The critical value of inlet $\Delta T_{c-w}$, and, thus, the degree of viscosity ratio, was slightly different for each tested product and temperature range, with a critical range of 12-15 °C, above which the degree of temperature homogenization was reduced.

Shear-thinning behavior was observed for all tested food products. Vibration reduced the viscosity of each product close to the vibrating wall, resulting in less efficient chaotic advection results (reduction of stretching degree). Applesauce and sweet potato puree showed the most shear-thinning behavior, with a flow behavior index of 0.13 and 0.18, respectively. Results of thermal mixing of applesauce, especially during cooling at high temperature range of 80-100 °C, were effected mainly by the shear-thinning flow behavior. Thermal mixing was less effective, in comparison to sweet potato puree, due to the lower viscosity of applesauce.

Finally, a non-slip boundary condition were reported for all the tested materials, with banana puree presenting the greatest formation of non-slip boundary conditions due to
gelation phenomenon, occurred during the cooling process of these materials. Non-slip boundary conditions was a problem; transversal vibration was able to address it when vibration was started before the thick gel material was able to form on the wall.

Acknowledgments

The author would like to acknowledge the Center for Advanced Processing and Packaging Studies (CAPPS) for their financial support.
NOMENCLATURE

Latin letters

\( K \)  flow consistency index (Pa·s\(^n\))

\( n \)  flow behavior index (dimensionless)

Greek letters

\( \Delta T_{c-w} \)  temperature difference between the temperatures at the center and the wall (°C)

\( \tau \)  shear stress (Pa),

\( \gamma \)  shear rate (s\(^{-1}\))

Subscripts

\( c \)  center

\( w \)  wall

Abbreviations

1-D FFT  one-dimension Fast Fourier Transformation

CAPPS  Center for Advanced Processing and Packaging Studies

NCSU  North Carolina State University (NCSU)

MW  microwave

\( T_c \)  temperature at the center of the pipe

\( T_w \)  temperature at the wall of the pipe
REFERENCES


Rauwendaal, 2014. Elongational Pin Mixer (EPM),

CHAPTER 6: Conclusions

Thermal mixing effects and enhancement of continuous flow cooling were examined during continuous flow thermal processing of viscous and multiphase food products. A continuous flow recirculation thermal process system was built to study the effects of sinusoidal mechanical vibration generated from acoustics on thermal mixing during the cooling process of sweet potato, banana puree, applesauce and cheese sauce, in a temperature range of 60 to 105 °C.

A continuous flow MW system at 2450 MHz, formed from a series of MW ovens, was designed, built, and used for the heating process of the tested materials. The MW system produced a very rapid heating process with an average time of 1200 s required for the system to heat 5 gal of tested product from an initial temperature of 20 °C up to 105 °C. The best effects from the MW system were observed with sweet potato puree, requiring 600 s to reach 105 °C from an initial temperature of 25 °C, while the canned processed cheese sauce presented the slowest heating MW process, requiring 2600 s to heat to 115 °C from an initial temperature of 25 °C.

Thermal mixing as a method of enhancement of continuous flow cooling of highly viscous products was numerically studied. Temperature distribution and bulk product temperature during continuous flow cooling were estimated using a mathematical model, based on the solution of the Graetz problem and its computer implementation through a code written on Maple 16. Results from mathematical model showed a significant improvement of
61.4% on total cooling time with the application of thermal mixing during continuous flow cooling of applesauce.

A mixing unit was designed and used to generate thermal mixing during the cooling process of previously aseptically processed sweet potato, banana puree, applesauce, cheese sauce, and canned cheese sauce samples. Acoustics were used to impose transverse vibration on a 180° bend pipe and to generate chaotic fluid motion on tested products during the cooling process, thus achieving thermal mixing. A mixing system was built, through which we were able to generate, record, and measure the effects of different acoustic frequencies (0-100 Hz), amplitudes, and protocols of vibration on radial temperature distribution of different foods. The magnitude of acceleration of vibration was the vibration parameter that showed the greatest effect on the degree of mixing at the fluid close at the center of the pipe, with higher peak acceleration resulting in better mixing of product close to the center of the pipe.

Good thermal mixing results were observed for banana puree, applesauce, aseptically processed cheese sauce, and sweet potato puree with vibration close to the resonance frequency, 20 Hz, of the mixing system (acoustic vibrator, 180° bend pipe, and product) at the maximum amplifier volume. Uniform temperature was practically observed for these tested products, with a temperature difference between the product at the center of the pipe and the product flowing close to the wall of 0-4 °C upon exiting the mixing unit. A larger temperature difference of 6-8 °C was observed for canned processed cheese sauce.

The degree of thermal mixing was affected by rheological characteristics of foods, such as shear thinning flow behavior, non-slip wall boundary condition, and high viscosity
ratio inside the product. Shear-thinning behavior was observed for all the tested foods. Shear imposed from transversal vibration reduced the viscosity of product close to the vibrating wall, resulting in the reduction of the stretching degree of fluid elements and leading to less efficient chaotic advection and thermal mixing results. Applesauce and sweet potato puree showed the most shear-thinning behavior, with a flow behavior index of 0.13 and 0.18, respectively. Applesauce, as a less viscous product than sweet potato puree, was most affected by the shear thinning behavior, especially in the temperature range of 80-100 °C, where applesauce viscosity was reduced due to temperature effects. With the exception of applesauce in the temperature range of 80-100 °C, shear-thinning flow behavior was not reported as a major problem in thermal mixing results for the rest of the products. Additional treatment with higher input vibration power (i.e., with the addition of one extra Buttkicker LFE device) is expected to increase the degree of stretching and thermal mixing.

No-slip wall boundary conditions were reported for all the products, with banana puree presenting the most non-slip boundary conditions formed due to a gelation phenomenon, occurred during the cooling process of these materials, starting at the temperature range of 85-94 °C. No-slip boundary conditions effected chaotic advection mixing by reducing the effects of the secondary flow, transverse to the main axial flow, created by vibration. The problem of non-slip boundary conditions was addressed to some extent by starting the vibration before the thick gel material began to form on the wall.

The factor that had the greatest effect on the final temperature uniformity upon exiting the mixing unit was the high viscosity ratio between different product elements, which depended on the initial temperature distribution at the inlet of mixing unit. Wide
temperature distribution increased the viscosity ratio inside the product between the
temperature at the center of the pipe and close to the wall. The high viscosity ratio between
different product elements made the resistance at the interface between flow streams very
strong, reducing the mixing efficiency—a problem that can be addressed with the application
of an elongation shear.

A critical value of inlet temperature difference between the product at the center and
close to the wall at 12-15 °C was found; above this value, a rather wide temperature
distribution (5-12 °C) remained for all the tested products. The only case in which good
thermal mixing, with a temperature difference at 1-3 °C, was found was with sweet potato
puree, with an inlet temperature difference at 26-28 °C when vibration was applied without
full pipe flow conditions due to steam produced from boiling. In this study, to address this
problem during full pipe flow conditions, vibration at 60 Hz, the frequency that generated the
highest magnitude of acceleration at the inlet flexible pipe portion of the mixing unit, was
tested as a method to control and improve the temperature difference inside the product at the
inlet of the mixing unit. The result with vibration at 60 Hz for applesauce and aseptically
processed cheese sauce showed a slight improvement in both inlet and outlet temperature
difference of the mixing unit when starting with an inlet temperature difference above the
critical value. Furthermore, results of thermal mixing of aseptically processed cheese sauce
generated with a protocol of vibration combining vibration at 60 Hz, followed by vibration at
20 Hz, showed better results in comparison to vibration only at 20 Hz. With the vibration at
60 Hz, we were able to keep the temperature difference below the critical value at the inlet to
the mixing unit, and with application of vibration at resonance frequency, we were able to observe good thermal mixing results with an outlet temperature difference at 1-4 °C.

The results of this study showed that food production systems that utilize advanced thermal methods could benefit from operational enhancements of the cooling cycle by minimizing product temperature distribution at the entrance of each cooler section and improving radial mixing and thermal mixing. The method of transversal acoustic vibration showed to be a very efficient mechanism or homogenizing the temperature profile and enhancing the cooling process of food products. Rheological parameters of food materials, such as shear-thinning flow behavior, non-slip wall boundary conditions, and the degree of viscosity ratio inside the product, need to be fully addressed for industrial application of this method.
CHAPTER 7: Recommendations for future work

The results of this research showed that temperature homogenization, that is, thermal mixing during the cooling process, could be a very efficient technique for cooling enhancement, minimizing the total cooling time, and maximizing the final food product quality.

The new mixing unit, an open duct flow mixing system, designed for the purpose of this study was proposed as a system that could possibly minimize the degradation of food particles; this system is expected to reduce the shear and pressure drop that damages the particles with other thermal mixing devices already in use, such as a scraped surface heat exchanger and static mixers. Future work could be done to test and measure the particle size loss through the use of the proposed mixing unit to prove or disprove the above hypothesis.

Temperature homogenization of the tested products was caused by rheological parameters; these parameters influence the stretching degree of fluid elements. The factor that effected the greatest degree of thermal mixing was the high ratio of viscosity of different fluid product elements that depended on the wide temperature distribution inside the product. A method that was proposed and used for static mixers was the application of an elongation shear. A way to achieve elongation flow is to reduce the pipe diameter on the mixing unit section, in comparison to the diameter of the tube used for the rest of the continuous flow thermal process system, or to introduce periodic baffles into the mixing unit. However, application of these solutions would likely increase the shear and the pressure drop and would therefore damage the food particles.
As observed in the experiments with sweet potato puree, vibration on not-full pipe flow condition caused an almost perfect uniform temperature profile, with a very high initial temperature difference between the product elements. A possible use of this observation is to introduce sterilized air into the mixing unit during the cooling and the thermal mixing processes to achieve perfect temperature homogenization independent of the initial temperature distribution conditions.

Finally, as observed in the vibration parameter experiments on sweet potato puree, the degree of stretching and, thus, the degree of thermal mixing was related to the magnitude of vibration, which was controlled by adjusting the amplifier volume level and by identifying the resonance frequency of the mixing system. The results showed that the best mixing was achieved with the application of the maximum level of volume and the maximum energy input. The input energy, which was used to generate vibration to the whole mixing system, came from the Buttkicke LFE, the 180° bend pipe, and the tested product, with a very small amount of input energy to be used in product mixing. Future work could be done to design a mixing unit based on the technique of transversal vibration: to directly deliver the input energy on the product and maximize the vibration effects on thermal mixing.
REFERENCES


