

**DEVELOPED THERMO-HYDRAULIC MODEL OF
DIRECT STEAM GENERATION WITH RELIABLE TWO-
PHASE FLOW CONSIDERATION**

BY

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[Dedication]

To my mother and my father, for the early alphabet that they taught me

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TABLE OF CONTENTS

TABLE OF CONTENTS.....	vi
LIST OF TABLES	ix
LIST OF FIGURES	x
ABSTRACT.....	xvii
ملخص الرسالة	xix
CHAPTER 1 INTRODUCTION	1
1.1 Concentrating Solar Power	1
1.2 Concentrating Solar Power System Types.....	2
1.2.1 Parabolic Trough Solar Concentrator	3
1.2.2 Solar Power Towers.....	3
1.2.3 Solar Dish.....	4
1.3 Solar Energy applications	5
1.3.1 Air Heating.....	5
1.3.2 Desalination Processes.....	5
1.3.3 Industrial Processes.....	6
1.3.4 Refrigeration Systems	6
1.3.5 Power Plants.....	7
1.4 Gas-liquid mixer on two-phase pump.....	8
1.5 Objectives	10
CHAPTER 2 LITERATURE REVIEW.....	11
2.1 Solar Radiation.....	11
2.2 Direct steam generation two phase flow modeling.....	12

2.2.1 Prediction of Flow Patterns.....	14
2.3 Experimental data for DSG.....	17
2.4 The gas-liquid mixture on multiphase pump for homogenization.....	20
2.4.1 Flow homogenizer	20
2.4.2 Existing designs of flow homogenizers	21
CHAPTER 3 MATHEMATICAL MODELING & METHODOLOGY	24
3.1 Thermo hydraulic modeling.....	24
3.2 Flow pattern maps.....	28
3.2.1 Kattan Flow Pattern Map [64]	28
3.2.2 Zurcher et al. [65] new flow pattern transition criteria	30
3.2 Two-phase heat transfer coefficient correlations	32
3.3.1 Dimensionless number.....	32
3.3.2 Two-phase heat transfer coefficient correlations	34
3.4 Pressure drop of two-phase flow.....	42
3.4.1 Basic Definitions and terminologies.....	42
3.4.2 Two-phase flow pressure drop correlations	44
3.5 Proposed design of a flow homogenizer stage.....	56
3.5.1 Procedure of experiments and measurements.....	61
CHAPTER 4 RESULTS AND DISCUSSION.....	63
4.1 Flow pattern map	63
4.2 Heat transfer coefficient correlations.....	65
4.2.1 Validation with R134a experimental data.....	66
4.2.2 Validation with water experimental data	69
4.3 Wall and fluid temperatures validation.....	69
4.4 Void fraction validation	71

4.5	Pressure drop validation.....	72
4.6	Parameters optimization.....	72
4.7	Experimental validation of homogenizer 1 (2-stage pump) performance	74
4.7.1	Single-phase flow experiment of homogenizer #1	74
4.7.2	Two-phase flow experiment of homogenizer #1	76
4.8	Experimental Validation of 4 stage Pump Performance.....	91
4.8.1	Single Phase Flow experiments	91
4.8.2	Two-Phase Flow experiments:.....	93
4.9	Evaluation of Homogenizer #1 With Respect To Degree Of Perforation:	109
4.10	Observed flow patterns in the experiments.....	110
4.11	Uncertainty analysis.....	113
CHAPTER 5 CONCLUSION & FUTURE WORK.....		116
5.1	Conclusion	116
5.1.1	Thermo hydraulic modeling concluded remarks	116
5.1.2	Homogenizer designing, testing and validation concluded remarks	118
5.2	Future work.....	119

LIST OF TABLES

Table 2.1: Properties of previous studies	15
Table 2.2: Solar collector parameters [55].....	17
Table 2.3: Cases of DSS Plataforma Solar de Almeria data [57]	17
Table 2.4: Case1 DSS Plataforma Solar de Almeria data [57]	17
Table 2.5: Case 2 DSS Plataforma Solar de Almeria data [57]	18
Table 2.6: Case 3 DSS Plataforma Solar de Almeria data [57]	18
Table 2.7: Case 4 DSS Plataforma Solar de Almeria data [57]	18
Table 2.8: Experimental data [71]	19
Table 2.9: Schott PTR-70-DSG receivers' parameters [72]	19
Table 2.10: Experimental data of Feldhoff et al. [72].....	19
Table 3.1: Variables m , n , C_g and Cl	46
Table 3.2: Values of the constant C for Lockhart-Martenelli curves	46
Table 3.3: $c, d, t, a_1, b_1, a_2, b_2$ parameters	54
Table 3.4: The parameters $c, d, t, a_1, b_1, a_2, b_2$ for different flow patterns.....	55
Table 3.5: Experimental test matrix and pumps used	58
Table 3.6: Pumps input power & RPM [103].....	61
Table 4.1: Flow pattern map comparison	64
Table 4.2: Steam generated for different cases	73
Table 4.3: Observed flow pattern at inlet and outlet of the homogenizer at GVF=0.2 ..	111
Table 4.4: Observed flow pattern at inlet and outlet of the homogenizer at GVF=0.6 ..	112
Table 4.5: Estimated uncertainties for measured and calculated values.....	115

LIST OF FIGURES

Figure 1.1: Concentrating solar power system	2
Figure 1.2: Parabolic trough solar collector [8]	3
Figure 1.3: Solar power tower [9]	4
Figure 1.4: Solar dish [10]	4
Figure 1.5: Gas lock problem formation sequence in centrifugal pump [104]	8
Figure 2.1: Direct normal irradiation [52]	11
Figure 2.2: Flow homogenizer behavior [74]	21
Figure 2.3: Advanced handling device Schlumberger[75]	22
Figure 2.4: Axial flow in Poseidon pump Schlumberger [75]	22
Figure 2.5: XGC gas compressor[76]	22
Figure 2.6: Fluid homogenized system for gas segregated liquid hydrocarbon wells [79]	23
Figure 3.1: Different flow regimes along the pipe [78]	24
Figure 3.2: Fluid and wall temperature profiles when the phase change occurs	27
Figure 3.3: Homogenizer #1 details [103]	60
Figure 3.4: Pump Hydraulic performance curve [103]	60
Figure 3.5: Pumps numbering	61
Figure 3.6: Single-phase flow experimental testing facility	62
Figure 3.7: Two-phase flow experimental testing facility	62
Figure 4.1: Wojtan et al flow pattern map	64
Figure 4.2: HTC vs vapor quality	66
Figure 4.3: HTC error vs vapor quality	66
Figure 4.4: HTC vs vapor quality	66
Figure 4.5: HTC error vs vapor quality	66

Figure 4.6: HTC vs vapor quality	66
Figure 4.7: HTC error vs vapor quality	66
Figure 4.8: HTC vs vapor quality	67
Figure 4.9: HTC error vs vapor quality	67
Figure 4.10: HTC vs vapor quality	67
Figure 4.11: HTC error vs vapor quality	67
Figure 4.12: HTC vs vapor quality	67
Figure 4.13: HTC error vs vapor quality	67
Figure 4.14: Heat transfer coefficient vs quality	69
Figure 4.15: Heat transfer coefficient vs quality	69
Figure 4.16: Wall and fluid temperature distribution along pipe.....	69
Figure 4.17: Void fraction variation with the vapor quality	71
Figure 4.18: Pressure drop variation with mass quality.....	72
Figure 4.19: Outlet mass quality for different pipe diameters	72
Figure 4.20: H-Q curve for perforated and unperforated impellers.....	74
Figure 4.21: H-Q curve compared to the homogenizer	75
Figure 4.22: Head difference & head difference ratio between perforated and unperforated impellers	76
Figure 4.23: Inlet & outlet pressure variation with GVF for unperforated and perforated 2-stage pump at constant liquid flow rate.....	79
Figure 4.24: H-Q curve of 2-stage pump with 10 L/M.....	79
Figure 4.25: Variation of the inlet and outlet pressure with GVF of the unperforated and perforated two stage pump at constant range of total mixture flowrate	81
Figure 4.26: Variation of the inlet and outlet pressure with GVF for the unperforated case and the first stage perforated-second stage unperforated of the two-stage pump at a constant liquid flow rate.....	83

Figure 4.27: variation of the inlet and outlet pressure with GVF for the unperforated case and the first stage perforated-second stage unperforated of the two-stage pump at constant total mixture flowrate	85
Figure 4.28: Variation of the pressure rise with GVF for different scenarios at the lowest flow rate 2 GPM	86
Figure 4.29: Variation of the pressure rise with GVF for different scenarios at flow rate 4 GPM	86
Figure 4.30: Variation of the pressure rise with GVF for different scenarios at flow rate 6 GPM	87
Figure 4.31: Variation of the pressure rise with GVF for different scenarios at flow rate 8 GPM	87
Figure 4.32: Variation of the pressure rise with GVF for different scenarios at flow rate 10 GPM	88
Figure 4.33: Variation of the pressure rise with GVF for different scenarios at flow rate 12 GPM	88
Figure 4.34: Dimensionless pressure rise versus GVF for homogenizer 1 (perforated first stage and unperforated second stage impeller) flow regimes are at the discharge, pressure difference across the homogenizer	89
Figure 4.35: Dimensionless pressure rise versus GVF (zoom in scale) (perforated first stage and unperforated second stage impeller) pressure difference across the homogenizer	90
Figure 4.36: Four stages Pump and perforated impeller stage details	92
Figure 4.37: H-Q curves for the 4-stage pump with and without perforated stage	93
Figure 4.38: Suction and discharge pressure distribution with GVF for unperforated and first stage perforated pump at low system pressure	96
Figure 4.39: Suction and discharge pressure variation with void fraction at low and high system pressure (unperforated impeller).....	97
Figure 4.40: Comparison between unperforated pump and pump with perforated first stage at high pressure.....	98
Figure 4.41: Pressure rise and head variations across the pump with void fraction at low and high system pressure	100
Figure 4.42: Homogenizer#1 details.....	101

Figure 4.43: Homogenizer#2 details.....	102
Figure 4.44: Suction and discharge pressure variation with GVF of 4-stage pump downstream of homogenizer#1.....	102
Figure 4.45: Pressure difference across the 4-stage pump downstream of homogenizer#1.....	103
Figure 4.46: Pressure difference across the 4-stage pump downstream of homogenizer#2.....	104
Figure 4.47: Comparison of pressure difference across the 4-stage pump downstream of homogenizer#1 and #2	105
Figure 4.48: Comparison of the suction and discharge pressure of the 4-stage pump downstream of homogenizer#1 and Homogenizer #2	106
Figure 4.49: Pressure difference across the 4-stage pump with and without homogenizers	107
Figure 4.50: Normalized pressure across the 4-stage pump downstream of homogenizer#1 versus GVF. The flow pattern is in the suction of the 4-stage pump and discharge of the homogenizer (all pressure data)	108
Figure 4.51: Normalized pressure across the 4-stage pump downstream of homogenizer#2 versus GVF. The flow pattern is in the suction of the 4-stage pump and discharge of the homogenizer (all pressure data)	108
Figure 4.52: Performance comparison at same conditions between perforated second stage and two stages perforated and solid impeller.	109
Figure 4.53: Digital turbine flowmeter	113
Figure 4.54: Omega pressure gauge.....	113
Figure 4.55: Air pressure gauge.....	114
Figure 4.56: Air flowmeter	114

LIST OF ABBREVIATIONS

DSG	Direct steam generation
DISS	Direct steam generation solar power plant
FP	Flow pattern
GVF	Gas void fraction
SEGS	Solar energy generating systems
CSP	Concentrating solar power
ESP	Electric submersible pumps
HTC	Heat transfer coefficient
PD	Pressure drop
HQ	Head vs flowrate

NOMENCLATURES

h	Heat transfer coefficient ($W/m^2.k$)
X_{tt}	Martenelli parameter (-)
ρ	Density (kg/m^3)
σ	Surface tension (N/m)
k	Thermal conductivity ($W/m.k$)
C_p	Heat capacity ($J/kg.k$)
μ	Liquid dynamic viscosity ($Pa.s$)
T	Temperature ($^{\circ}C$)
P	Pressure (kpa)
v	Specific volume (m^3/kg)
γ	Liquid holdup (-)
D	Absorber inner diameter (m)
G	Mass flux ($kg/m^2.s$)
i	Enthalpy (J/kg)
ΔP	Pressure drop (kpa)
x	Vapor quality (-)
$h_b S$	Bubble formation ($W/m^2.k$)
$h_L F$	Convection ($W/m^2.k$)
C_o	Convection number (-)
$S \& F$	Correction and enhancement factors (-)
ϕ_{fo}^2	Two phase multiplier
ϕ	Friction multiplier
C	Constant depends on if the flow is turbulent or laminar.
Ω	Correction factor
B_o	Boiling number $(-)= \frac{q}{h_{fg}\rho V}$
Pr	Prantdl number $(-)= \frac{C_p \cdot \mu}{k}$
Fr_H	Fraud number $= \frac{G^2}{gD\rho_h^2}$
E	$= (1-x)^2 + \frac{x^2 \rho_L f_{GO}}{\rho_G f_{LO}}$ in eq31
F	$= x^{0.78} (1-x)^{0.224}$ in eq 31
H	$= \left(\frac{\rho_L}{\rho_G}\right)^{0.91} \left(\frac{\mu_G}{\mu_L}\right)^{0.19} \left(1 - \frac{\mu_G}{\mu_L}\right)^{0.7}$ in eq 31
We	Weber number $= \frac{G^2 D}{\sigma \rho_h}$ in eq 31
ρ_h	homogenous flow density $= \left(\frac{x}{\rho_G} + \frac{1-x}{\rho_L}\right)^{-1}$
E, F and H are parameters depend on fluid properties	

SUBSCRIPTS

<i>tp</i>	Two-phase flow
<i>fo</i>	Single-phase flow
<i>g</i>	Gas
<i>l</i>	Liquid
<i>b</i>	Nucleate boiling
<i>tpn</i>	Nucleate boiling is dominant
<i>bc</i>	Bulk convective
<i>w</i>	Wall
<i>sat</i>	Saturation
<i>h</i>	Homogenous
<i>Bf</i>	Bankoff

ABSTRACT

Full Name : [Khalid Abdelbasit Khalid]

Thesis Title : [Developed Thermo-Hydraulic Model Of Direct Steam Generation
With Reliable Two-Phase Flow Consideration]

Major Field : [Mechanical Engineering]

Date of Degree : [April-2021]

[This research aims to develop a multi-phase flow model for direct steam generation (DSG) systems. The fluid used in the DSG system is water; required energy is obtained from solar energy. The water inside the absorber passes through different regions due to the phase change process as it goes from liquid phase to gas phase. In between, a two-phase flow region exists. In the two-phase flow region, the flow passes through different flow patterns before reaching the gas phase region, for example (bubble, stratified wavy, slug, annular, and mist flow). Another important aspect of this work is to optimize the influential parameters that maximize the steam generated by carefully considering the multi-phase flow characteristics, which have not been considered entirely in the open literature. This research identifies the most sensitive parameters affecting the generated steam by performing a detailed parametric study. Controlling the two-phase flow regime is achieved by changing the geometric pipe characteristics and flow conditions. We explored the essential parameters such as the mass quality, void fraction, and the two-phase pressure drop along with the absorber. Moreover, an experimental databank for direct steam generation systems from open literature was collected to validate the developed model. A user-friendly work-station is developed using MATLAB interface.

Pumping two-phase flow is a challenge that needs more investigation. This research also aims to design, test, and validate an intermediate flow homogenizer stage to enhance the

process of pumping two-phase flow if a need for pumping the two-phase flow using a commonly used centrifugal pump. The homogenizer showed that it improves the two-phase flow pumping but could not solve the problem completely. Several suggestions have been made to improve pumping the two-phase flow. |

ملخص الرسالة

الاسم الكامل: خالد عبدالباسط خالد عبدالباسط

عنوان الرسالة: نموذج هيدروليكي حراري مطور للتوليد المباشر للبخار مع اعتبارات للتدفق ثنائي الطور

التخصص: الهندسة الميكانيكية

تاريخ الدرجة العلمية: أبريل - 2021

يهدف هذا البحث إلى تطوير تدفق ثنائي الطور لنظام توليد مباشر للبخار (DSG). السائل المستخدم في نظام DSG هو الماء. الطاقة المطلوبة تم توفيرها من الطاقة الشمسية. يمر الماء داخل الانبوب عبر عدة أشكال بسبب عملية تغيير الطور حيث ينتقل من الطور السائل إلى الطور الغازي. فيما بينهما، توجد مناطق مختلفة ثنائية الطور. في منطقة التدفق ثنائي الطور، يمر التدفق من خلال أنماط تدفق مختلفة قبل أن يصل إلى منطقة الطور الغازي على سبيل المثال (الفقاعة ، والتدفق المموج الطبقي ، والتدفق الحلقي والضبابي). جانب آخر مهم من هذا العمل هو معرفة أكثر المعاملات المؤثرة في عملية توليد البخار والتي تزيد من البخار المتولد عن طريق النظر بعناية في خصائص التدفق متعدد الأطوار، والتي لم تؤخذ في الاعتبار في الدراسات السابقة. في هذا البحث، سوف نحدد أكثر العوامل حساسية التي تؤثر على البخار المتولد عن طريق إجراء دراسة مفصلة. يمكن التحكم في نظام التدفق على مرحلتين من خلال تغيير الخصائص الهندسية للأنبوب وظروف التدفق. سيتم استكشاف جميع المعاملات المهمة على سبيل المثال الضغط المفقود ونسبه الغاز الى السائل و طول الانبوب وغيرها. علاوة على ذلك، تم جمع بنك بيانات تجريبي لأنظمة التوليد المباشر للبخار من الدراسات السابقة للتحقق من صحة النموذج المطور. تم تطوير برنامج رياضي باستخدام واجهة (MATLAB). يعد ضخ التدفق ثنائي الطور تحديًا يحتاج إلى مزيد من التحقيق. يهدف هذا البحث أيضًا إلى تصميم واختبار والتحقق من مرحلة ريش متوسطة (مجانس) لتحسين عملية ضخ التدفق ثنائي الطور. أظهر المجانس أنه يحسن ضخ التدفق ولكنه لا يستطيع حل المشكلة تمامًا. تم تقديم العديد من الاقتراحات لتحسين ضخ التدفق على مرحلتين.

CHAPTER 1

INTRODUCTION

Several countries have turned to renewable energy because of an energy shortage. Energy is required for different applications in our life, for example, thermal water desalination, air heating, refrigeration, power generation, and power industries [1]. Solar energy is the greatest source on the planet for thermal power generation to provide light for houses and industrial applications. As the research is going in this field, solar energy appears to be the most hopeful energy source these days, especially for countries with a huge amount of solar radiation.

1.1 Concentrating Solar Power

The primary technology utilized to generate electric power using solar energy is the parabolic trough solar collector[2]. It generates steam at high temperatures without significant degradation in collector efficiency[3]. Solar energy generating system (SEGS) is a source of energy known as a low-cost source. It utilizes several collectors to accumulate the sun's radiation to gain high-temperature results in driving steam turbines to produce electricity. Solar radiation is utilized by different means of concentrating solar power systems to deliver high temperatures. For instance, parabolic trough solar collector provides steam to the steam turbine at a temperature of 477°C and rejects heat to the atmosphere at 27°C (60% ideal thermal efficiency/Carnot efficiency). Then a conventional

cycle, for instance, a gas turbine and the steam engine, operates by solar heat. Additional heat can be stored in a medium with a phase change to generate power in the nighttime.

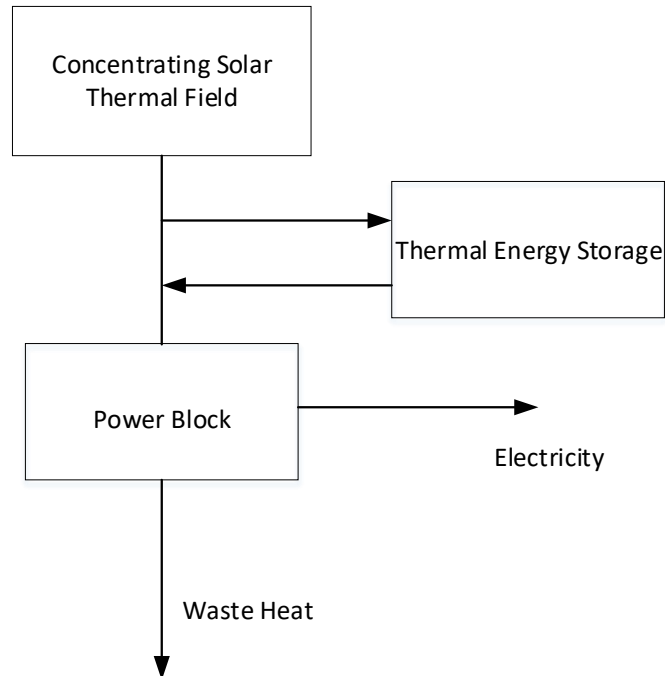


Figure 1.1: Concentrating solar power system

The wide-ranging classes classified by size are large (greater than 10 MW), medium (less than 10 MW), and small (less than 100 KW)[4].

Figure 1.1 presents a concentrating solar power system.

1.2 Concentrating Solar Power System Types

The concentrating solar power systems need sufficiently more than 5.2 kW/m²/day direct normal irradiance. In California, history shows impressive cost reduction; the parabolic trough solar plants worked effectively for over three decades. Concentrating solar power systems have a high potential and low-cost energy.

Types of concentrating solar power systems

1.2.1 Parabolic Trough Solar Concentrator

A parabolic trough solar collector shown in figure 1.2 is a solar thermal concentrator with a parabolic reflector with a single axis tracker that makes the sunlight reflect on the receiver (working fluid chamber). The sun radiation concentrated on the absorber tube after reflecting by the mirror to heat the fluid inside the tube to high temperature. The heated fluid that comes out from the collector is used to generate electricity or to drive machinery. This solar collector is known as a parabolic trough, and it is practically efficient in areas where direct solar irradiation is very high. A better design with coatings on the receiver tube results in significant plant efficiency enhancement [5], [6]. The plant effectiveness has increased by using of internally finned tube parabolic trough collector [7].

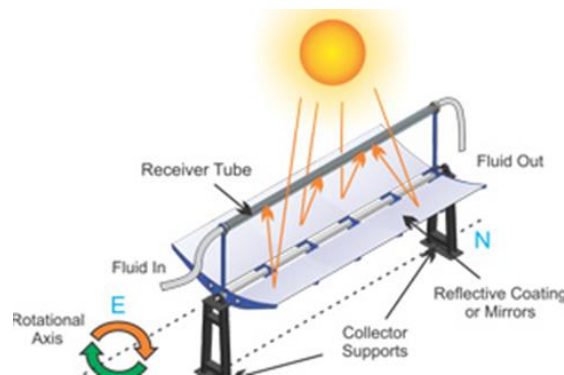


Figure 1.2: Parabolic trough solar collector [8]

1.2.2 Solar Power Towers

A solar tower arrangement shown in figure 1.3 contains a big field of flat mirrors used to track the sun called heliostats to reflect and focus light onto a receiver placed on the top of a tower. This light can be concentrated 1,500 times. The water is utilized as a heat-transfer fluid in some towers. Another design uses molten salt because of its energy storage

abilities—electricity produced during cloudy weather or at night by using a thermal storage system.

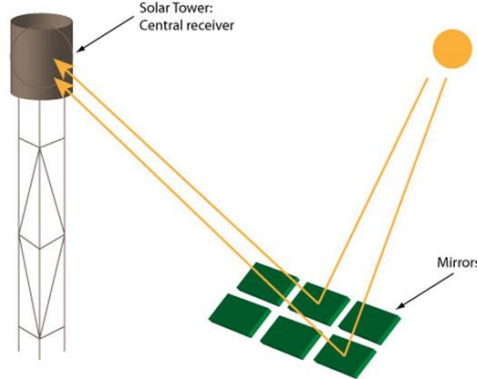


Figure 1.3: Solar power tower [9]

1.2.3 Solar Dish

The solar dish system presented in figure 1.4 uses a mirror parabolic dish consist of many smaller flat mirrors. This mirrors direct and concentrates sunlight onto a thermal receiver; the receiver absorbs heat and transfers it to electricity through a power conversion unit. The most public sort of heat engine used is the Stirling engine. In this system, a fluid is used to transfer heat from the receiver to a piston that produces mechanical power transferred to electricity through a generator. The electricity produced in this system in the range of 3 to 25 KW. The main components in this system are the concentrator ad the power conversion unit.

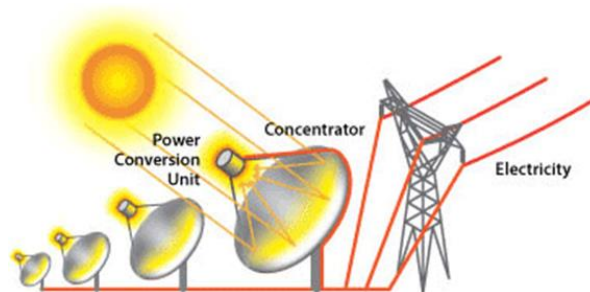


Figure 1.4: Solar dish [10]

1.3 Solar Energy applications

Solar energy consumption has enlarged significantly in local and manufacturing processes. Different applications are

1.3.1 Air Heating

Air heating system with solar energy used in agriculture, buildings, marine, foodstuffs, and textile for drying purposes. This energy has no effect on the atmosphere, but it depends on time, so a thermal storage system is necessary. Active, hybrid and passive systems are the types of air heating systems with the storage system and without. For those with storage systems, two methods were used, namely phase change and rock bed. Tyagia et al. [11] display that the storage of the latent heat is more accurate than the storage of the sensible heat. Simulation technique has been used to determine the optimum physical properties of the phase change material used in the energy storage [12], [13]. The Storage System stores energy by phase change. During the night, it recovers the energy through the change from liquid to solid. Commonly, fatty acids, hydrated salts, paraffin, and non-paraffins are used as thermal energy storage materials [14]–[17].

1.3.2 Desalination Processes

The desalination process is a method to convert salty water to sweet water. The desalination process is either thermal or reverse osmosis. Sharaf et al. [18] have completed an analysis of cogeneration of multi-effect distillation system with a cycle of solar organic. The outcomes showed that the gain ratio is higher and less solar field area is required. A standalone desalination plant works with renewable (solar and wind) energy studied by Mohamed & Papadakis [19]. The output capacity of the plant per day is $250m^3$. Karellas

et al. [20] studied a hybrid system of thermal organic Rankine cycle with reverse osmosis system to produce both water and electric power. Application of solar parabolic trough in desalination process makes it effective in producing water.

1.3.3 Industrial Processes

Solar energy is used in industrial applications instead of massive electricity consumption. Industrial processes need heating. Around 120 solar thermal systems operating for heat process according to international energy agency are reported around the world with production around 88 MW in a total area of 125000 m^2 [21]. The maximum temperature needed for all industrial processes is 260°C , and the parabolic trough solar concentrator can deliver a temperature of about 400°C [22]. Several developing countries have been using solar energy. Parabolic trough solar collectors can provide suitable energy for industrial applications.

1.3.4 Refrigeration Systems

Refrigeration and AC systems also utilize solar energy. An experimental study on solar collectors exposed that using solar energy for heating and electrical application reduces the auxiliary energy needed [23], [24]. To maintain steady thermal efficiency, a level of temperature about 60 to 80°C needed [25]. An increase of about 20% in the output electric power is shown as a result of using reflectors and coatings for the compound parabolic collector with mono crystal silicone cells[26], [27]. A group of researchers simulated an advanced adsorption system, and the results showed that the formation of decomposition outcomes decreases solar icemaker performance [28], [29]. The adsorption refrigeration performance coefficient increases because of optimization improvement [30], [31], [32]. There are several working fluids used in adsorption cooling system [33], [34], [35], [36],

[37]. Different studies performed on thermal collectors and photovoltaic theoretically, experimentally, and numerically on the refrigeration systems driven by solar energy [38], [39], [40], and [41].

1.3.5 Power Plants

The parabolic trough solar collector is used to generate electric power in different places around the world, and that is because of the higher limit of temperature and higher concentration ratio. For power generation, high steam pressure is needed with a large amount, 18 bar obtained for a feasibility study of parabolic trough collector [42]. A study was performed to increase the annual power generation of solar parabolic trough [43]. Another study was performed on the solar collector to increase the output power performance [44]. The fluid used in the plant is a synthetic oil with a parabolic trough collector to generate a temperature of 400° C [45], then the direct steam generation is presented to enhance the solar power plant [2] to supply the electricity continuously, thermal energy storage needed. It can provide electricity for 6 hours after sunset.

The power generated using water as a working fluid to generate steam, but these days researchers are studying different types of working fluid for better performance of power plant [46], [47]. The total capacity of solar power plants around the world is around 2828.8 MW, while the capacity of the solar power plants under construction is 2463.9 MW [48]. More steam is required to increase the oil production; for example, Kuwait is planning to increase the oil production to 270,000 barrels per day, so it requires 900,000 barrels of water to generate steam every day, as Geiger [49] indicates. This amount of steam is provided normally by natural gas, which is almost 19 GW of thermal energy. However, DSG could reduce the consumption of natural gas between 25 to 100%. In Oman, two solar

enhanced oil recovery plants have already been commissioned, one is a seven MW pilot, and the other is currently under construction, which is more than 1 GW plant.

1.4 Gas-liquid mixer on two-phase pump

In many applications, the pumped fluids' gas content (gas and liquid) can exceed the gas moving ability of centrifugal pumps. Due to inertial effects, the gas separates and form large gas pockets leading to the gas-locking problem explained in figure 1.5. The balancing holes balance the axial thrust but do not contribute much toward flow homogenization that is used in multistage vertical pumps.

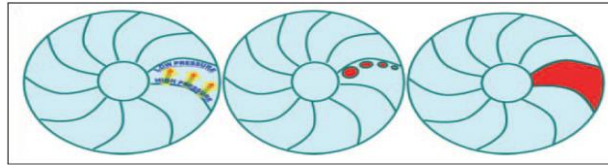


Figure 1.5: Gas lock problem formation sequence in centrifugal pump [104]

Electric submersible pumps are commonly used to recover liquids from subterranean wellbores and generally comprise an electric motor that operates a multi-stage centrifugal pump. Centrifugal pumps are designed to handle all liquid fluids; in some applications, we require to pump gas with liquid. The gas cause trouble for centrifugal pumps (gas locked). The conventional centrifugal pumps stop moving fluid when the gas content exceeds a certain volume percentage depending on the gas pressure and the pump size and type. Centrifugal pump performance (head and capacity) decreases when the gas percentage inside the mixture increases due to liquid-gas phase separation. Impeller configuration is also part of the problem. Centrifugal pumps formed from several stages; each stage includes a stationary diffuser and a rotating impeller [50]; the pump starts to produce lower outputs when the gas-liquid volume ratio of Gas Void Fraction, GVF, increases above a certain value (10-15%)

Bagci et al. [51] discussed the challenges of obtaining a complete potential of the electric submersible pump to pump fluid in the presence of free gas at the intake of the pump. Their results confirmed that ESP (which is basically a centrifugal pump) could successfully manage fluid with gas fraction reaches 75 % at pump intake by using appropriate gas handling equipment and special method. They suggested a management plan for gas handling procedure as follows:

- For the GVF of 20% or less: the standard ESP can handle this range of free gas.
- For the $20\% < \text{GVF} < 45\%$: homogenizer must be installed upstream of the pump. These are mainly axial flow machines or mixed flow pumps to mix the gas and liquid, and some cases compress the gas into liquid before they get into the ESP.
- For $45 < \text{GVF} < 95$: the gas handling unit must be used in addition to a gas separator. The gas separator directs the gas to move through the annulus; then, it can collect before the wellhead.

1.5 Objectives

In this research, we focused first on searching all the multi-phase flow models used in thermos-hydraulic modeling literature used in the process of direct steam generations (DSG), the main tasks in this work are

- 1) To conduct extensive literature review
- 2) To predict direct steam generation process using friendly prediction tool
 - a. Single and Multi-phase flow models
 - b. Heat transfer models
 - All convection Heat transfer coefficient correlations for multi-phase flow as a function of steam quality and flow pattern
 - c. Fluid flow correlations
 - Pressure drop correlations for multi-phase flow
 - Flow pattern map for water vapor two-phase flow as a function of steam quality
 - d. Experimental data bank and validation
- 3) To optimize the influential parameters to maximize the steam generation
- 4) To design, test, and validate a flow homogenizer stage to improve two-phase flow pumping |

CHAPTER 2

LITERATURE REVIEW

2.1 Solar Radiation

The capability of any CSP system in any position depends on the solar radiation characteristics at that place. Irradiance is the overall specific power radiated from the sun per unit area that arrives the absorber measured by W/m^2 . Solar radiation involves direct & diffuse beams, and the last one is the scattered part. The summation of these two components gives global radiation; the variation of these components during the day depends upon environmental and meteorological factors such as cloud cover, humidity, and the relative earth to sun geometry. The direct normal incident (direct beam radiation) shown in figure 2.1 is important to determine the existing solar energy, which always less than the global radiation by the diffuse component value.

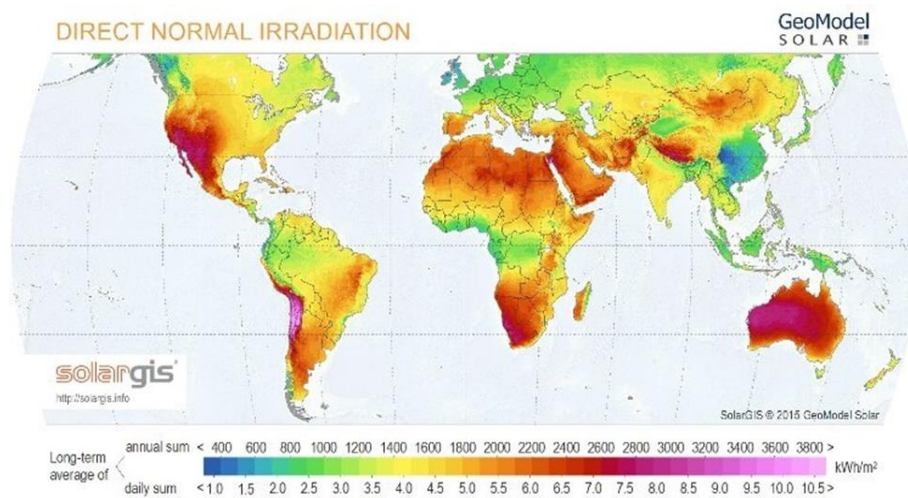


Figure 2.1: Direct normal irradiation [52]

Saudi Arabia has an enormous potential to use solar energy; the direct normal irradiance in Saudi Arabia is significant. The average solar radiation received in the Arabian Peninsula for 8.89 hours the average time is $2.2 \text{ GWh}/\text{m}^2$ while the average horizontal solar radiation is $5591 \text{ Wh}/\text{m}^2$ [52]. Saudi Arabia is progressively investing in solar energy; by 2032, Saudi Arabia planned to install solar power plants with a capacity of 41 GW where a 16 GW should be PV system [53].

2.2 Direct steam generation two phase flow modeling

The existing parabolic trough plants use thermal oil as a parameter to transfer heat with a temperature of 400°C . Using the DSG in a parabolic trough concentrator could achieve better performance with a temperature of 550°C . This type of technology does not exist commercially because of some complexity. In 2014, Alguacil et al. [54] performed a study on DSG in PTCs; they built a plant of 8MWht , and they tested the plant for one year as an outcome of this study, an advanced control system assures plant steadiness even in transient situations. In addition, the mechanical behavior of the receiver tube was designed and analyzed to achieve a temperature of 450°C . Two main problems related to the receiver tube: the possibility of stratified flow inside the tubes results in insufficient inner cooling for the solar collector, which can deform the tubes. This problem can be avoided by ensuring that the tube's minimum flow rate is sufficient to provide enough inner wall cooling and radial temperature distribution to prevent tube deformation [55].

During evaporation along the absorber tube, different flow patterns exist: stratified smooth/wavy, slug, bubbly, dispersed bubble, and annular flow categorized by quality distribution. The complexity of thermos hydraulic modeling appears because of two-phase

flow characteristics, such as flow transitions, instabilities, and interactions between phases [56]. It is necessary to know the flow patterns, to calculate temperature and pressure distributions across the receiver. Undesired flow patterns along the absorber tube may appear intermittent, and stratified flow regimes are not recommended inside the solar absorber tube because intermittent flow will cause instability of pressure, and stratified flow will cause different thermal stresses in the pipe which may cause a system failure. The thermo- hydraulic behavior of direct steam generation needs more research to ensure good controllability in the process [57]. Lobon et al. [57] present theoretical research of two-phase pressure drop in horizontal pipes throughout evaporation. The model ensures a smooth transition between patterns at the transition boundary. The powerful design of DSG require the maximum temperature difference across the absorber to be in a specific range because the increase of temperature difference due to dry-out and flow stratification may lead to failure (absorber glass breakage). Temperature difference of 3° C may occur at annular flow while it may reach 50° C in the case of stratified flow so that, annular flow is the most appropriate flow regime for save operation of DSG system.

Numerous researches performed to develop a transition model for two-phase flow, for example[58], [59], [60] Taitel and Dukler, in 1976, Barnea et al. in 1985 and Hanratty in 1986 Hanratty 1987. The summary of these models reported by Taitel in 1990 [61]. A hydrodynamic model has been established by Odeh et al. [62] based on [61] and connected to the thermal model to assess the flow pattern and pressure drop for horizontal and inclined tubes in a DSG collector. The model used to compare DSG collectors with oil-based collectors. The results showed that the horizontal tube has a higher possibility of stratified and intermittent flow than the inclined tube at high radiation. In contrast, low radiation

results in an increase of intermittent flow in the case of an inclined collector and stratified flow in the case of a horizontal collector. Pressure drop is higher for the inclined collector by 19% because of pressure drop in the connecting pipes between the inclined collectors. The absorber tube diameter affects the flow pattern and pressure drop along the absorber tube; larger tube diameter results in stratified and intermittent flow, which is unfavorable in DSG. On the other hand, a small diameter results in higher collector cost and pressure drop since it needs very accurate concentrators. Sun et al. [63] examined the operation strategies of the direct steam generation loop; they used Taitel and Dukler [58] flow pattern map to perform the analysis.

Once through, recirculation and injection are the operation modes used for direct steam generation systems mentioned in the literature [6]. Each one has its own benefits and drawbacks. However, one through mode is the best performance, lower cost, and the most straightforward arrangement.

2.2.1 Prediction of Flow Patterns

Several maps were established to predict the flow pattern inside the horizontal tubes, for example, Taitel and Duckler [58]. In 1998, Kattan et al. [64] developed an enhanced two-phase flow pattern map for evaporation in horizontal tubes. The map covered an inclusive range of fluid qualities and mass velocities for five refrigerants. The flow pattern map is effective for adiabatic and diabatic evaporating flows, and it includes the estimation of dry-out considering the flow parameters and heat flux. After few years (4 years) in 2002 Zurcher et al. [65] developed the flow pattern map of Kattan et al. [64]. The modifications allow gaining an accurate prediction for the fluid flow pattern. In 2004, Wojtan et al. [66] and [67] introduced transition curves of annular to dry-out and dry-out to mist, producing

a new flow pattern map and the heat transfer coefficient as a function of the flow pattern and vapor quality. The new transition curves are used to calculate the beginning and ending of dry-out in horizontal tubes. This map was established from the refrigerants database. However, the analytical estimation of the transition between the patterns depends on physical mechanisms. The dimensionless form gives a general flow pattern map that allows predicting the flow patterns at other conditions, not only that determined experimentally. Table 2.1 shows the previous studies properties.

Table 2.1: Properties of previous studies

Study	El_Safi 2015 [68] case 1	Odeh et al 2000 [62]
Pipe diameter (m)	Plata forma Solar de Almeria Outer/inner (0.07/0.05)	Outer/inner (0.114/0.097), (0.07/0.054) and (0.048/0.038)
Dimensions of the collector(m)	•Single collector (50/25), total (500) and aperture width (5.76)	Collector area (547), tube walls (0.005-0.008) Compared to the oil-based collector (0.002) and the length (600)
Flow pattern consideration	•Wojtan et al. [78] flow pattern map and Quiben and Thome pressure drop model. •After 35% of the two-phase region intermittent for 15% annular for 47%	•stratified, annular, dispersed bubble and intermittent •intermittent for large absorber tube diameter dispersed bubble flow occurs only at very high superficial liquid velocities 15% intermittent and 85% annular in the two-phase region
Mass flow rate(kg/s)	0.47±0.05	1.25
DN Irradiance	822±10	beam radiation of 200 to 1000
Pipe inclination	0	0-10
Operating conditions	P=3.42±0.06(MPa) and T=205 C	P _{in} =100bar,T _{in} =210°C,T _{exit} =450° C
Study	Lobon et al. [57] 2014 case 1	Eduardo Zarza et al. [69] 2001
Pipe diameter (m)	Outer/inner (0.035/0.025)	Outer/inner (0.07/0.05)
Dimensions of the collector(m)	13 parabolic trough connected in series with 700m total length., parabola width (5.76), and focal length of (1.71)	11 collectors, Total length (550), aperture width (5.76), and area of (3000)

Flow pattern consideration	Local homogenous boiling model is used	The first nine collectors are the evaporation section, and the last two are the superheated section (experimental work)
Mass flow rate(kg/s)	0.45-0.73	0.37-0.90
DN Irradiance	822±10	600-850
Pipe inclination	0	0, 2 and 4
Operating conditions	P=3.42±0.06(MPa) and T=205 C	P _{in} =40(bar),T _{in} =210C P _{out} =30(bar),T _{out} =300C
Study	Odeh et al. [2] 1998	Eck and Steinmann [70] 2001
Pipe diameter (m)	Inner diameter 0.054 and 0.038	Plata forma Solar de Almeria Inner diameter (0.054)
Dimensions of collector(m)	Total 600m long 104m joining pipe length for inclined absorber	Total is 500 long
Flow pattern consideration	•Inclined absorber is the lower probability for stratified and intermittent flow at high radiation •In the inclined absorber, the intermittent flow increases at lower radiation, while in the horizontal collector, the stratified flow increases at low radiation •Stratified, dispersed bubble, intermittent and annular flow are considered in this study •The flow tends to be intermittent for large diameter at low radiation	•Annular flow in the absorber is indicated because of temperature gradient less than 20K (0m-425m) •Maximum temperature gradient of 40K due to lower heat transfer coefficient in steam flow region (425m-500m)
Mass flow rate(kg/s)	1.25	0.53
DN Irradiance	1000	-
Pipe inclination	0 and 10	Inclination lead to better volume to hight ratio
Operating conditions	Saturated steam at 100 bar	P=100bar and T= 400C

It is important to study the thermal stresses in the absorber, especially when there is water inside; when the flow is stratified, the tube may fail under bending

2.3 Experimental data for DSG

Zarza [55] reported direct steam generation power plant during the DISS project, at the Plataforma Solar de Almería. The study carried out, for the zones of water preheating, evaporation and steam superheating in DISS collectors for horizontal absorbers.

DISS facility sited at Plataforma Solar de Almeria data, the solar collector parameters showed in table 2.2.

Different cases reported for this project in tables 2.3, 2.4, 2.5, 2.6 and 2.7, the experimental data of Hachicha et al. [72] showed in table 2.8 while tables 2.9 and 2.10 showed the receiver parameters and experimental data of Feldhoff et al. [73] respectively.

Table 2.2: Solar collector parameters [55]

length of a single collector	50/25m
aperture width	5.76m
outer diameter	70mm
inner diameter	50mm

Table 2.3: Cases of DSS Plataforma Solar de Almeria data [57]

Cases	Pressure (Mpa)	Temperature(C)	Mass flow (Kg/s)	Direct Normal Irradiation(W/m ²)
Case 1	3.42 ± 0.06	205 ± 2	0.47 ± 0.05	822 ± 10
Case 2	3.38 ± 0.06	196± 2	0.47 ± 0.05	807 ± 10
Case 3	6.25 ± 0.06	239 ± 2	0.55± 0.05	971 ± 10
Case 4	6.23 ± 0.06	235 ± 2	0.50 ± 0.05	850 ± 10

Table 2.4: Case1 DSS Plataforma Solar de Almeria data [57]

case 1	Length (m)	Temperature(°C)	Length (m)	Pressure(bar)
	0.0	205.4	0.0	34.174
	51.3	241.4	50.7	34.087
	102.7	239.6	101.3	34.000
	154.0	238.8	152.0	33.913
	205.3	240.5	204.4	33.478
	254.8	239.6	255.1	33.304

	306.2	237.9	305.8	33.000
	357.5	237.9	356.4	32.609
	408.8	237.9	408.9	32.217
	460.2	237.0	459.5	31.696
	509.7	246.7	510.2	31.391

Table 2.5: Case 2 DSS Plataforma Solar de Almeria data [57]

Case2	Length (m)	Temperature(°C)	Length (m)	Pressure(bar)
	0.0	196.1	0.0	33.707
	51.3	239.8	50.7	33.578
	102.7	239.8	101.3	33.491
	154.0	239.8	152.0	33.405
	205.3	238.9	204.4	33.103
	254.8	239.8	255.1	32.888
	306.2	238.0	305.8	32.586
	357.5	237.1	356.4	32.198
	410.7	238.0	407.1	31.810
	458.3	236.3	457.7	31.293
	509.7	263.0	510.2	31.207

Table 2.6: Case 3 DSS Plataforma Solar de Almeria data [57]

Case 3	Length (m)	Temperature(°C)	Length (m)	Pressure(bar)
	0.0	239.2	0.0	62.496
	51.5	278.4	50.7	62.310
	101.2	278.4	101.3	62.204
	152.7	277.0	152.0	62.097
	204.2	277.0	204.4	61.991
	255.7	279.7	255.1	61.912
	305.4	275.7	305.8	61.805
	356.9	277.0	356.4	61.593
	408.4	277.0	407.1	61.301
	458.0	277.0	459.5	61.088
	509.5	286.5	510.2	60.982

Table 2.7: Case 4 DSS Plataforma Solar de Almeria data [57]

Case 4	Length (m)	Temperature(°C)	Length (m)	Pressure(bar)
	0.0	236.2	0.0	62.305
	50.9	276.6	51.4	62.086
	101.8	276.6	102.7	62.007
	152.6	276.6	152.7	62.007

	203.5	276.6	204.1	61.907
	254.4	277.6	254.1	61.808
	305.3	275.5	306.8	61.709
	356.1	276.6	356.8	61.609
	408.4	277.6	408.1	61.411
	457.9	291.0	458.1	61.212
	508.8	315.9	509.5	61.113

- Hachicha et al. 2018 [71]

Table 2.8: Experimental data [71]

Length (m)	T(°C)	Length (m)	Pressure (Mpa)
0.0	205.0	0.0	3.418
50.4	241.2	52.0	3.407
100.9	240.5	104.0	3.403
153.3	240.5	156.0	3.392
203.7	240.5	208.0	3.350
256.1	240.5	260.0	3.331
306.6	238.2	310.1	3.300
357.0	238.2	362.1	3.262
407.5	238.2	414.1	3.221
459.9	237.0	468.0	3.171
510.3	247.3	520.0	3.141

- Feldhoff et al. 2016 [72]

Table 2.9: Schott PTR-70-DSG receivers' parameters [72]

Length	150 m
aperture width	5.78 m
outer diameter	70 mm
inner diameter	58.8 mm

Table 2.10: Experimental data of Feldhoff et al. [72]

DNI 300 w/m ²	Length (m)	T(°C)	DNI = 600 w/m ²	Length (m)	T(°C)	DNI = 900 w/m ²	Length (m)	T(°C)
	0.0	123.5		0	123.494		0.0	123.5
	194.1	216.9		211.765	222.892		229.4	234.9
	420.0	216.9		420	225.904		420.0	234.9
	631.8	216.9		631.765	225.904		631.8	234.9
	843.5	216.9		843.529	222.892		843.5	231.9
	991.8	219.9		1002.35	228.916		988.2	228.9
	1055.3	256.0		1214.12	340.361		1055.3	250.0
	1214.1	352.4		1284.71	391.566		1214.1	334.3

	1351.8	448.8		1348.24	436.747		1351.8	430.7
	1351.8	403.6		1351.76	403.614		1351.8	388.6
	1500.0	503.0		1496.47	500		1500.0	497.0

2.4 The gas-liquid mixture on multiphase pump for homogenization

Deterioration of pumping head starts to be appreciable for gas void fractions (GVF)

above 6%. It was also found that rotodynamic multiphase pumps cannot handle gas void fractions above 20% due to gas locking problems; however, mixed-type pumps can handle fluids with a gas void fraction that may reach 30% or slightly higher.

2.4.1 Flow homogenizer

In the case of strong incoming slug flow with a succession of liquid slugs and gas pockets at a period of less than one or two minutes, the sudden changes in mixture density or composition produce associated variations of drive power and rapid and repetitive torque changes or pulsing. This behavior was eliminated simply with a flow homogenizer Figure 2.2 left diagram [73].

The homogenizer is a small vessel with a static mixer that is installed upstream of the pump. Its main functions are to dampen the energy of liquid slugs and to smooth out GVF fluctuations. It can also act as a sand trap. The flow homogenizer can also be useful at the start-up of a well to avoid the pump's dry running (Figure 2.2 right diagram) [73]. Obviously, when the flow characteristics only vary slightly from the average values, there is no need to install a homogenizer [73].

Some pump suppliers have stages designed to better handle-free gas. These stages are installed at the pump intake and build pressure in the presence of free gas. After the gas is sufficiently compressed, additional standard pump stages are used to generate the desired

pump discharge pressure [74]. Examples of these special “gas handling” stages are the Schlumberger Advanced Gas Handler [75] (AGH)TM and PoseidonTM, the Centrilift Multi Vane Pump (MVP)TM, and Wood Group XGSTM system [76]. A few of these stages in a pump, in a gassy environment, even pumping under a packer, may be able to build pressure in the pump such that gas is not a problem for the remainder of conventional stages in the pump assembly[74].

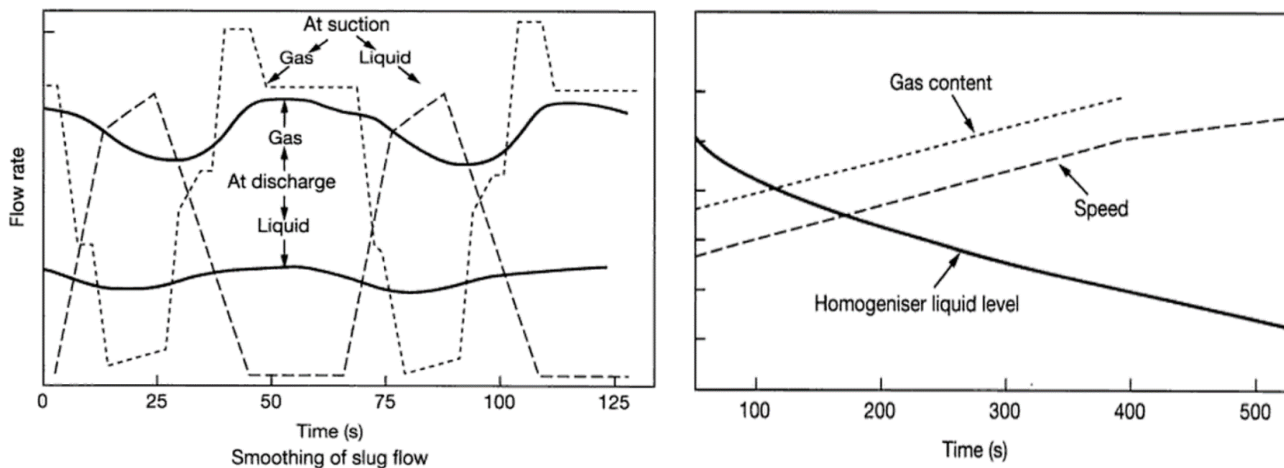


Figure 2.2: Flow homogenizer behavior [74]

2.4.2 Existing designs of flow homogenizers

- Advanced gas handling stage

The AGH stage presented in figure 2.3 has a recirculation path to keep bubbles from accumulating and to keep some pressure building in the pump. It enables the pump to handle gas without gas locking. This is achieved by homogenizing the gas-liquid mixture, which reduces gas bubble sizes by putting the gas bubble back in the liquid solution [75]. It is used in applications where GVF is 45% (wells), it reduces vapor bubbles sizes, homogenizes the gas-liquid mixture to behave like single-phase fluid before it enters the pump [75].

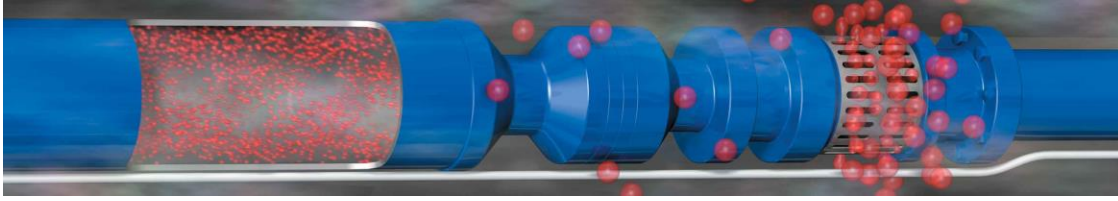


Figure 2.3: Advanced handling device Schlumberger[75]

- Poseidon

Poseidon showed in figure 2.4 is a helicoaxial multiphase pump that handles up to 75% of gas. It has a special stage design that allows the axial flow of the mixture to reduce gas locking

Poseidon pump pushes the gas/liquid mixture through the production pump stages and increases the mixture pressure to reduce the gas volume. Also, it breaks large gas bubbles and homogenizes the fluid for a good mixture between gas and liquid.

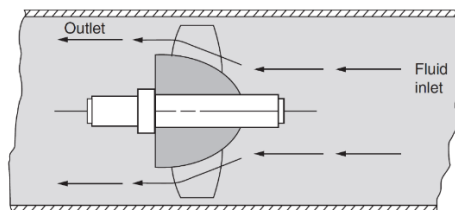


Figure 2.4: Axial flow in Poseidon pump Schlumberger [75]

- XGC system

The XGC system presented in figure 2.5 has a compression chamber downstream from the tandem gas separators. Free gas is compressed back into the solution by the compression chamber. It also breaks the bigger gas bubbles into smaller bubbles at the same time. This provides an increasingly homogenized solution, which a submersible pump stage can handle without gas locking [76]

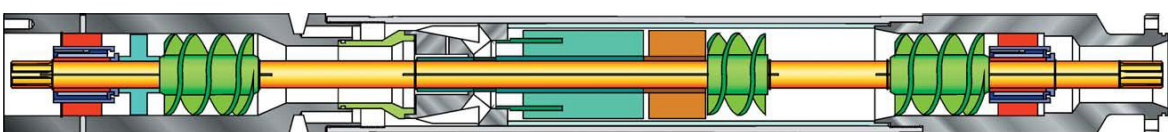


Figure 2.5: XGC gas compressor[76]

- Fluid homogenizer system for gas-segregated liquid hydrocarbon wells and method of homogenizing liquids produced by such wells Patent no. US9353614 [77]

This device explained in figure 2.6 includes a fluid homogenizer system for gas-segregated liquid hydrocarbon wells. It aims to improve the flow characteristics in gas slugging wells. The slugged gas is first separated and then re-injected into the liquid column in a controlled method at a downstream location to improve the production fluid's homogeneity and flow characteristics. The system comprises an annulus section and a nozzle [77]

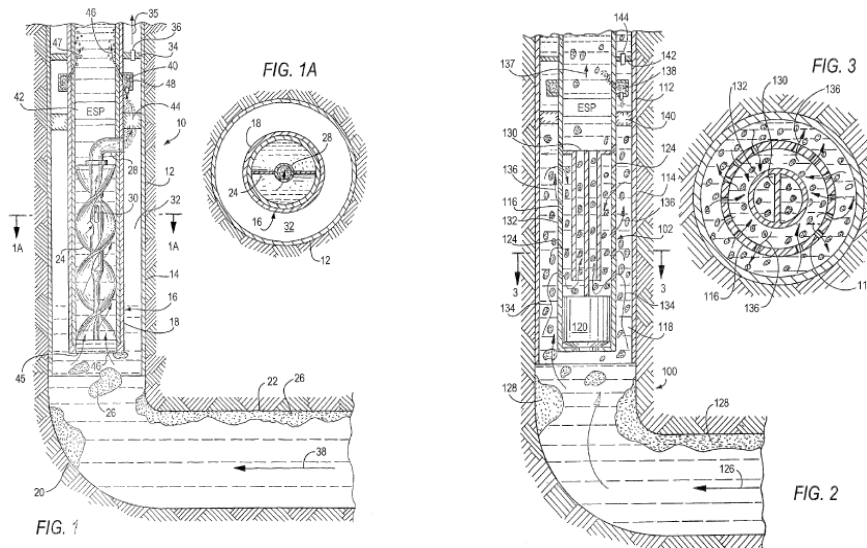


Figure 2.6: Fluid homogenized system for gas segregated liquid hydrocarbon wells [79]

CHAPTER 3

MATHEMATICAL MODELING & METHODOLOGY

3.1 Thermo hydraulic modeling

The heat absorbed by the collector's receiver in the form of solar radiation comes from the sun to the absorber tube. Then, the heat is absorbed by water inside the tube. A phase change occurs from the single-phase liquid (water) to the single gas phase (steam), passing through the two-phase region (water-steam), and superheated steam with specific temperature will come out from the tube. The hydraulic model of direct steam generation deals with all regions of flow inside the tube (single-phase liquid (water) region, (preheating region), two-phase flow (water-steam) region, and single-phase superheated steam region). In the two-phase flow region, there are different flow regimes. Firstly the flow pattern is identified depending on the mass flux, vapor quality, heat flux, and the distance of that region from the beginning of the pipe. After that, the heat transfer coefficient and pressure drop correlations corresponding to the flow pattern applied to the model. The preheated and superheated regions are considered as a single-phase regions.

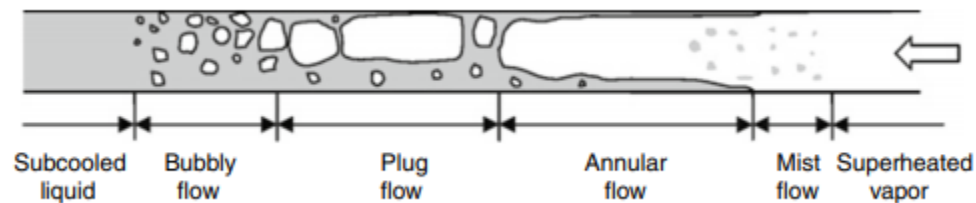


Figure 3.1: Different flow regimes along the pipe [78]

The methodological procedure of how to find the total length of the pipe by knowing the heat flux, mass flux, internal pipe diameter, inlet conditions, and final superheated steam temperature as follows:

- The single-phase liquid region

From 0 (inlet condition) to 1 (saturated liquid)

$$Q = \frac{\dot{m}}{A_{cross}} C_p (T_1 - T_0) \text{ or } q = \dot{m} C_p (T_1 - T_0) = \dot{m} (h_1 - h_0) \text{ and } G = \frac{\dot{m}}{A_{cross}} \quad (3.1)$$

Where C_p (heat capacity) is a function of T (temperature)

Q : heat flux

$\dot{m}$: mass flow rate

A_{cross} : cross sectional area

$$Q \pi D_i L_{0 \leftrightarrow 1} = \dot{m} (h_1 - h_0) \quad (3.2)$$

$$L_{0 \leftrightarrow 1} = \frac{\dot{m} (h_f - h_i)}{Q \pi D_i} = \frac{G A_{cross} (h_f - h_i)}{Q \pi D_i} = \frac{G D_i (h_f - h_i)}{4Q} \quad (3.3)$$

where $h_1 = h_f$ and $h_0 = h_i$

- The two-phase water-steam region

From 1 (saturated liquid) to 2 (saturated gas)

$$Q \pi D_i L_{tp} = \dot{m} h_{fg} \quad (3.4)$$

h_{fg} is the latent heat of vaporization

$$L_{tp} = \frac{GD_i h_{fg}}{4Q} \quad (3.5)$$

- The single-phase gas region

By knowing the final superheated steam temperature

$$Q\pi D_i L_{2\leftrightarrow 3} = \dot{m}(h_3 - h_2) \quad (3.6)$$

$$h_2 = h_g$$

$$L_{2\leftrightarrow 3} = \frac{GD_i(h_3 - h_2)}{4Q} \quad (3.7)$$

The total length

$$L_T = L_{0\leftrightarrow 1} + L_{1\leftrightarrow 2} + L_{2\leftrightarrow 3} \quad (3.8)$$

The quality is a function of pressure, mass flux, heat flux, and pipe distance (input variables), then from the flow pattern map, the flow pattern is predicted. After that, the heat transfer coefficient and the pressure drop correlations were selected, corresponding to that flow pattern predicted.

The methodological procedure of how to find the thermodynamic quality and void fraction (α) as a function of z :

- For the two-phase region ($0 < X < 1$)

The thermodynamic quality (X) and void fraction (α) of the vapor-liquid mixture at any pipe distance z is given by:

$$X(z) = \frac{(h(z) - h_f)}{h_{fg}} \quad (3.9)$$

$$\pi D \phi = \dot{m} \frac{dh}{dz}, \quad \dot{m} = \frac{\pi D^2 G}{4}, \quad G = \text{mass flux} = \rho v_{average}$$

$$X(z) = \frac{4Q(z - Z_{sc})}{DGh_{fg}} \quad (3.10)$$

$Q = \text{heat flux}, \quad Z_{sc}$

$= \text{tube length needed to heat the liquid up to saturation enthalpy}(h_f)$

$$\alpha(z) = \frac{X(z)}{X(z) + \left(\frac{(1 - X(z))\rho_g}{\rho_l} s \right)}, \quad s = \text{slip ratio} = \frac{v_g}{v_l} \quad (3.11)$$

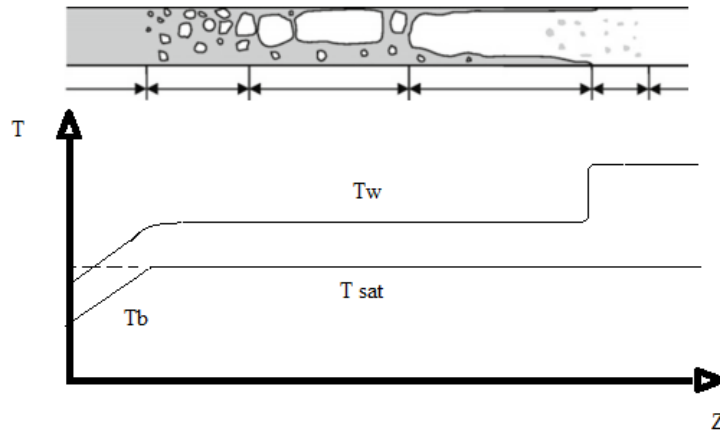


Figure 3.2: Fluid and wall temperature profiles when the phase change occurs

Figure 3.1: Different flow regimes along the pipe shows the process of changing phase from liquid to gas when a constant heat flux is considered around the pipe. In the beginning, the flow was a single (liquid) phase flow; then the flow starts to boil, forming bubbly flow

because of the heat flux considered; then, as the pipe length increases from left to right, the quality and void fraction increases at which the flow became intermittent. After an intermittent flow pattern, the flow goes into the annular flow region when the dry out of the liquid film occurs before it reaches the single-phase gas region.

Figure 3.2 presents the fluid and wall temperature distribution when the phase changes from liquid phase to liquid-gas phase then to gas phase only at the end.

3.2 Flow pattern maps

Right understanding of the physical structure of flow patterns is helpful in the general understanding of the mechanism of mass, momentum and energy transfer. Several maps are established to predict the FP inside the horizontal tubes, in the next part it is explained how flow pattern maps are generated.

3.2.1 Kattan Flow Pattern Map [64]

The equations govern the transition from pattern to other reported as follows

Stratified to stratified wavy transition

It occurs when the shear work and the pressure on a wave overcome the viscous dissipation, using sheltering coefficient $s=0.005$ then; Steiner finally obtains this equation.

$$G_{Strat-Steiner} = \left[\frac{226.3^2 \widehat{A}_L \widehat{A}_v^2 \rho_v (\rho_l - \rho_v) \mu_l g \cos v}{\pi^3 x^2 (1-x)} \right]^{\frac{1}{3}} \quad (3.12)$$

G Mass flux ($kg.m^{-2}.s^{-1}$)

ρ_g Gas density ($kg.m^{-3}$)

g Gravity ($m.s^{-2}$)

$\widehat{A}_l = A_l/D^2$ And $\widehat{A}_v = A_v/D^2$ non-dimensional cross-sectional areas, which are functions of the void fraction. However, for horizontal flow, the angle v is zero.

Stratified wavy to Annular/Intermittent transition

Steiner included the effect of surface tension produced by capillary waves

$$G_{Wavy-Stainer} = \left[\frac{16\widehat{A}_v^3 g D (\rho_l \rho_v)}{\pi^3 x^2 (2\widehat{h}_l - 1)^2} \left[\frac{\pi^2}{25\widehat{h}_l^2} \left(\frac{Fr}{We} \right)_l + \left(\frac{1}{\cos v} \right) \right] \right]^{\frac{1}{2}} \quad (3.13)$$

Liquid height in dimensionless form $\widehat{h}_l = h_l/D$, cross-sectional area in dimensionless form $\widehat{A}_v = A_v/D^2$

$$\left(\frac{Fr}{We} \right)_l = \frac{\sigma}{g D^2 \rho_L} \quad (3.14)$$

The percentage of inertia to surface tension force is the Weber number (We), while the Froude number is the ratio of inertia to gravity forces. Kattan proposed two changes to the above boundary. In the beginning, they presented the heat flux effect on the onset of dry-out by changing the ratio of (Froude to Weber number) as in the following equations:

$$(1 - x)^{G1} \left[\left(\frac{Fr}{We} \right)_l \right]^{G2} \quad (3.15)$$

$$G1 = -48.24 \left(\frac{\dot{q}}{\dot{q}_{crit}} \right) \quad (3.16)$$

$$G2 = 9.65 \left(\frac{\dot{q}}{\dot{q}_{crit}} \right) + 1.03 \quad (3.17)$$

$\dot{q}$ is the heat flux and $\dot{q}_{crit}$ is the critical heat flux for nucleate boiling, which can be calculated as follows

$$\dot{q}_{crit} = 0.131 \sqrt{\rho_- v} \Delta h_{evap} [g(\rho_l - \rho_v) \sigma]^{\frac{1}{4}} \quad (3.18)$$

The effect of heat flux results in changing stratified wavy to annular transition boundary to lower vapor qualities with increasing heat flux

Steiner and Kattan et al. proposed the transition of intermittent to annular flow at a value of Martinelli parameter of 0.34 for liquid and vapor turbulent flow and value of 0.51 for laminar, turbulent liquid, and vapor flow:

$$X_{tt} = \left(\frac{\mu_l}{\mu_v}\right)^{0.125} \left(\frac{1-x}{x}\right)^{0.875} \left(\frac{\rho_v}{\rho_l}\right)^{0.5} = 0.34 \quad (3.19)$$

Annular to mist transition

$$G_{MF-Steiner} = \left[\frac{7680 A_V^2 g D \rho_L \rho_V}{x^2 \pi^2 \xi_{ph}} \left(\frac{Fr}{We}_L \right) \right]^{0.5} \quad (3.20)$$

ξ_{ph} is the rough pipes friction factor given as follows

$$\xi_{ph} = \left[1.138 + 2 \log \left(\frac{\pi}{1.5 A_l} \right) \right]^{-2} \quad (3.21)$$

3.2.2 Zurcher et al. [65] new flow pattern transition criteria

The void fraction was used for several transition criteria to calculate the cross-sectional surface areas and dimensionless velocities. These parameters depend on the stratified configuration. The void fraction is the liquid height prediction. During the flow pattern transition, the void fraction is not continuous.

The void fraction is different for different flow patterns, and it is a function of mass velocity.

Critical heat flux

Evaluated using Kutateladze method

$$\dot{q}_{crit} = 0.131 \sqrt{\rho_v} \Delta h_{evap} [g(\rho_l - \rho_v) \sigma]^{\frac{1}{4}} \quad (3.22)$$

Stratified to stratified wavy transition

This boundary calculated using the void fraction model proposed by Taitel and Duckler as a function of void fraction and dimensional parameter

$$X_{tt}^2 \left[\frac{(S_l)^{1.2}}{(1-\alpha)^3} \right] - \left[\frac{(S_v+S_l)^{0.2}}{\alpha^2} \left(\frac{(S_v+S_l)}{\alpha} + \frac{S_l}{(1-\alpha)} \right) \right] = 0 \quad (3.23)$$

$$G_{Strat} = \left[800 \cdot \frac{\alpha^2(1-\alpha)}{x^2(1-x)} \cdot \rho_v g (\rho_l - \rho_v) \mu_l \cdot \cos v \right]^{\frac{1}{3}} \quad (3.24)$$

v=0 for horizontal flow

New stratified wavy to annular transition

The Rouhani and Steiner void fraction model used in the annular flow

$$\alpha_{Rouhani} = \frac{x}{\rho_v} \left[\left(1 + 0.12(1-x) \right) \left[\frac{x}{\rho_v} + \frac{1-x}{\rho_l} \right] + \frac{1.18(1-x)[g\sigma(\rho_l-\rho_v)]^{\frac{1}{4}}}{G\rho_l^{\frac{1}{2}}} \right]^{-1} \quad (3.25)$$

$$G_{Wavy,New} = \left[\frac{gD\rho_l\rho_v\alpha^3\pi}{2x^2\sqrt{2(1-\cos\theta_{wet})}} \left(1 + \frac{\pi^2}{25h_l^2}(1-x)^{G1} \left(\frac{Fr}{We} \right)_L^{G2} + \frac{h_l\lambda G^2 x^2}{Br\rho_v x^{\frac{5}{2}}} \right) \right]^{\frac{1}{2}} \quad (3.26)$$

$$G_1 = -24.12 \left(\frac{\dot{q}}{\dot{q}_{crit}} \right) \quad (3.27)$$

$$G_2 = 4.825 \left(\frac{\dot{q}}{\dot{q}_{crit}} \right) + 1 \quad (3.28)$$

$$G_{Wavy,New} = \frac{650\mu_l}{(1-x)D} \quad (3.29)$$

New intermittent to annular transition

$$X_{tt} = \left(\frac{\mu_l}{\mu_v} \right)^{0.125} \left(\frac{1-x}{x} \right)^{0.875} \left(\frac{\rho_v}{\rho_l} \right)^{0.5} = 0.34 \quad (3.30)$$

If $x < \frac{x_1}{A}$ Then the flow is intermittent.

New annular to mist transition

Using the initial value of $G_{MF-initial} = 1000$, the void fraction of Rouhani calculated as per eq (2.14).

$$G_{MF} = \frac{\alpha}{x} \left[1.138 + 2 \log \left(\frac{8}{3(1-\alpha)} \right) \right] * \left[480 g D \rho_L \rho_V \left(\frac{Fr}{We} \right)_L \right]^{0.5} \quad (3.31)$$

Flow Pattern Arrangement

Stratified	$G < G_{Strat}$
Stratified- wavy	$G_{Strat} < G < G_{wavy}$
Intermittent	$G_{wavy} < G < G_{MF}$ and $x < x_{A/I}$
Annular	$G_{wavy} < G < G_{MF}$ and $x > x_{A/I}$
Stratified-wavy, Dry	$G_{Strat} < G < G_{Wavy}$ and $G < G_{min.Wavy}$ and $x > 0.9$
Onset of dry-out	$G_{Strat} < G < G_{Wavy}$ and $G \geq G_{min.Wavy}$

3.2 Two-phase heat transfer coefficient correlations

Firstly, common dimensionless numbers are presented in the next part then the two-phase heat transfer coefficient correlations are reported

3.3.1 Dimensionless number

Convection number

$$Co = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_g}{\rho_L} \right)^{0.5} \quad (3.32)$$

x Vapor quality (-)

ρ_l Liquid density ($kg.m^{-3}$)

ρ_g Gas density ($kg.m^{-3}$)

Froude Number

$$Ft = \left(\frac{x^3 G^2}{\rho_g^2 g d (1-x)} \right)^{0.5} \quad (3.33)$$

x Vapor quality (-)

G Mass flux ($kg.m^{-2}.s^{-1}$)

ρ_g Gas density ($kg.m^{-3}$)

d Absorber inner diameter (m)

g Gravity ($m.s^{-2}$)

Boiling Number

$$K_f = \frac{q}{Gh_{fg}} \quad (3.34)$$

q Heat flow (W)

G Mass flux ($kg.m^{-2}.s^{-1}$)

h_{fg} Enthalpy of vaporization ($kJ.kg^{-1}$)

Nusselt Number

$$N = \frac{\left(\frac{Q}{A}\right)d}{K(T_w - T_m)} = \frac{hd}{k} \quad (3.35)$$

h Heat transfer coefficient ($Wm^{-2}K^{-1}$)

d Pipe diameter (m)

k Thermal conductivity ($Wm^{-1}K^{-1}$)

Phase change Number

$$Ph = \frac{c_p(T_s - T_{w,i})}{h_{fg}} \quad (3.36)$$

T_s Saturation temperature ($^{\circ}C$)

$T_{w,i}$ Inner wall temperature ($^{\circ}C$)

C_p Heat capacity ($kJ.kg^{-1}.K^{-1}$)

h_{fg} Enthalpy of vaporization ($kJ.kg^{-1}$)

Equivalent liquid Reynolds number

$$Re_{eq} = \frac{G_{eq}d}{\mu_L} \quad (3.37)$$

$$G_{eq} = G \left((1 - x) + x \left(\frac{\rho_l}{\rho_g} \right)^{0.5} \right) \quad (3.38)$$

x Vapor quality (-)

G Mass flux ($kg.m^{-2}.s^{-1}$)

ρ_g Gas density ($kg.m^{-3}$)

ρ_l Liquid density ($kg.m^{-3}$)

d Absorber inner diameter (m)

After identifying the FP, the fluid and wall temperatures are calculated using different 2-phase HTC correlations. Different correlations of 2-phase HTC are presented.

3.3.2 Two-phase heat transfer coefficient correlations

- DENGLER AND ADDOMS (1956) [79] equation for heat transfer coefficient.

$$h_{tp} = 3.5 h_{fo} \left(\frac{1}{X_{tt}} \right)^{0.5} \quad (3.39)$$

$$X_{tt} = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_g}{\rho_f} \right)^{0.5} \left(\frac{\mu_f}{\mu_g} \right)^{0.1} \quad (3.40)$$

h_{tp} Heat transfer coefficient of two-phase flow ($W.m^{-2}.K^{-1}$)

h_{fo} Single-phase heat transfer coefficient ($W.m^{-2}.K^{-1}$)

X_{tt} Martinelli parameter (-)

ρ_g Gas (vapor) density ($kg.m^{-3}$)

- Froude number must be calculated to evaluate the two-phase heat transfer coefficient,

$$Fr = \frac{G^2}{\rho_l^2 g D_{ab,i}} \quad (3.41)$$

Fr Froude number (-)

G Mass flux ($kg.m^{-2}.s^{-1}$)

ρ_l Liquid density ($kg.m^{-3}$)

g gravity ($m.s^{-2}$)

$D_{ab,i}$ Absorber inner diameter (m)

If $Fr < 0.04$, the flow is stratified, and the HTC is presented by Shah [80]

$$h_{tp} = 3.9 h_{fo} Fr^{0.24} \left(\frac{x}{1-x} \right)^{0.64} \left(\frac{\rho_l}{\rho_g} \right)^{0.4} \quad (3.42)$$

h_{fo} Single-phase HTC ($W.m^{-2}.K^{-1}$)

Fr Froude number (-)

x Vapor quality (-)

ρ_g Gas (vapor) density ($kg.m^{-3}$)

ρ_l Liquid density ($kg.m^{-3}$)

h_{fo} It can be calculated using the Dittus-Boelter equation

$$h_{fo} = 0.023 \left(\frac{k_1}{D_{ab,i}} \right) \left(\frac{G(1-x)D_{abi}}{\mu_L} \right)^{0.8} (Pr_L)^{0.4} \quad (3.43)$$

k_1 Thermal conductivity for saturated water ($W.m^{-1}.K^{-1}$)

$D_{ab,i}$ Absorber inner diameter (m)

G Mass flux ($kg.m^{-2}.s^{-1}$)

x Vapor quality (-)

μ_l Liquid dynamic viscosity ($Pa.s$)

Pr_l Prandtl number (-)

If $Fr > 0.04$, the flow is annular, and the HTC is determined by Chan correlation

$$h_{tp} = h_b S + h_l F \quad (3.44)$$

$h_b S$ Bubble formation ($W.m^{-2}.K^{-1}$)

$h_l F$ Convection ($W.m^{-2}.K^{-1}$)

h_b Nucleate boiling HTC ($W.m^{-2}.K^{-1}$)

h_L Saturated water HTC ($W.m^{-2}.k^{-1}$)

S & F Are correction and enhancement factors (-)

h_b It can be calculated using the empirical relation of Stephan 1992 [81]

$$h_b = 3800 \left(\frac{q}{20000} \right)^n Fp \quad (3.45)$$

q Heat flux (w/m^2)

$$n = 0.9 - 0.3P_n^{0.15} \quad (3.46)$$

$$F_p = 2.55P_n^{0.27} \left(9 + \frac{1}{1-P_n^2} \right) P_n^2 \quad (3.47)$$

$$P_n = \frac{P}{P_{cr}} \quad (3.48)$$

P Is working pressure (bar)

P_{cr} Is critical pressure for water (221 bar)

$$S = \frac{1}{1+(1.15*E-6*F^2*Re^{1.17})} \quad (3.49)$$

$$F = 1 + (2.4 * E4 * Bo^{1.16}) + 1.37X_{tt}^{-0.86} \quad (3.50)$$

$$Re = \frac{G(1-x)D_{abi}}{\mu_L} \quad (3.51)$$

Re Reynolds number (-)

$D_{ab,i}$ Absorber inner diameter (m)

G Mass flux ($kg.m^{-2}.s^{-1}$)

x Vapor quality (-)

μ_l Liquid dynamic viscosity ($Pa.s$)

- HEAT TRANSFER COEFFICIENT BY KANDLIKAR 1990 [82]

$$h_{tp} = hf_o * ((1.1360 * C_o^{-0.9} * (25 * Fr_l)^{0.3}) + (667.2 * B_o^{0.7})) \quad (3.52)$$

$$h_{tpn} = hf_o * ((0.6683 * C_o^{-0.2} * (25 * Fr_l)^{0.3}) + (1058 * B_o^{0.7})) \quad (3.53)$$

h_{tp} Two-phase HTC when convective boiling is dominant ($W.m^{-2}.k^{-1}$)

h_{tpn} Two-phase HTC when nucleate boiling is dominant ($W.m^{-2}.k^{-1}$)

B_o Boiling number (-)

C_o convection number (-)

Fr_l Froude number (-)

- CHEN [83]

$$h = h_{nb} + h_{bc} \quad (3.54)$$

h_{nb} Nucleate boiling local heat transfer coefficient contribution ($Wm^{-2}k^{-1}$)

h_{bc} Bulk convective local heat transfer coefficient contribution ($Wm^{-2}k^{-1}$)

$$h_{nb} = 0.00122 \left(\frac{k_l^{0.79} C_{pl}^{0.45} \rho_l^{0.49}}{\sigma^{0.5} \mu_l^{0.29} i_{fg}^{0.24} \rho_{fg}^{0.24}} \right) [T_w - T_{sat}(P_l)]^{0.24} [P_{sat}(T_w) - P_l]^{0.75} S \quad (3.55)$$

k_l Liquid thermal conductivity ($W.m^{-1}.K^{-1}$)

C_{pl} Liquid heat capacity ($J.kg^{-1}.k^{-1}$)

ρ_l Liquid density ($kg.m^{-3}$)

σ Surface tension ($N.m^{-1}$)

μ_l Liquid dynamic viscosity ($Pa.s$)

i_{fg} Enthalpy of vaporization (J/kg)

ρ_{fg} Density ($kg.m^{-3}$)

T_w Wall temperature ($^{\circ}C$)

T_{sat} Saturation temperature ($^{\circ}C$)

P_{sat} Saturation pressure (kpa)

P_l Liquid pressure (kpa)

$$h_{bc} = 0.023 \left(\frac{k_1}{D} \right) \left(\frac{G(1-x)D}{\mu_L} F(X_{tt})^{1.25} \right)^{0.8} (Pr_l)^{0.4} \quad (3.56)$$

k_1 Thermal conductivity for saturated water ($W.m^{-1}.K^{-1}$)

D Absorber inner diameter (m)

G Mass flux ($kg.m^{-2}.s^{-1}$)

x Vapor quality (-)

μ_l Liquid dynamic viscosity ($Pa.s$)

Pr_l Prandtl number (-)

- BENNET-CHEN [84]

$$h = h_{nb} + h_{bc} \quad (3.57)$$

h_{nb} Nucleate boiling local heat transfer coefficient contribution ($W.m^{-2}.K^{-1}$)

h_{bc} Bulk convective local heat transfer coefficient contribution ($W.m^{-2}.K^{-1}$)

$$h_{nb} = 0.00122 \left(\frac{k_l^{0.79} c_{pl}^{0.45} \rho_l^{0.49}}{\sigma^{0.5} \mu_l^{0.29} h_{fg}^{0.24} \rho_{fg}^{0.24}} \right) [T_w - T_{sat}(P_l)]^{0.24} [P_{sat}(T_w) - P_l]^{0.75} S \quad (3.58)$$

k_l Liquid thermal conductivity ($W.m^{-1}.K^{-1}$)

C_{pl} Liquid heat capacity ($J.kg^{-1}.K^{-1}$)

ρ_l Liquid density ($kg.m^{-3}$)

σ Surface tension ($N.m^{-1}$)

μ_l liquid dynamic viscosity ($Pa.s$)

i_{fg} Enthalpy of vaporization (J/kg)

ρ_{fg} Density ($kg.m^{-3}$)

T_w Wall temperature ($^{\circ}C$)

T_{sat} Saturation temperature ($^{\circ}C$)

P_{sat} Saturation pressure (kpa)

P_l Liquid pressure (kpa)

$$h_{bc} = h_1 F(X_{tt}) (Pr_L)^{0.296}, h_1 = 0.023 \left(\frac{k_1}{D} \right) (Re_l)^{0.8} Pr_l^{0.4} \quad (3.59)$$

k_1 Thermal conductivity for saturated water ($W.m^{-1}.K^{-1}$)

D Absorber inner diameter (m)

G Mass flux ($kg.m^{-2}.s^{-1}$)

x Vapor quality (-)

μ_l Liquid dynamic viscosity ($Pa.s$)

Pr_l Prandtl number (-)

- GUNGOR AND WINTERTON [85]

$$h_1 = 0.033 \left(\frac{k_1}{D} \right) \left(\frac{G(1-x)D}{\mu_L} \right)^{0.8} (Pr_L)^{0.4} \quad (3.60)$$

$$h_{pool} = 55M^{-0.5}q^{0.67}pr^{0.12}[-\log_{10} Pr]^{-0.55} \quad (3.61)$$

k_1 Thermal conductivity for saturated water ($W.m^{-1}.K^{-1}$)

D Absorber inner diameter (m)

G Mass flux ($kg.m^{-2}.s^{-1}$)

x Vapor quality (-)

μ_l Liquid dynamic viscosity ($Pa.s$)

Pr_l Prandtl number (-)

M Molecular weight

q Heat flow (W)

$$E = 1 + 24000Bo^{1.16} + 1.37(X_{tt})^{-0.86}, S = (1 + 1.15 * E - 6E^2 Re_l^{1.17})^{-1} \quad (3.62)$$

Bo Boiling number (-)

X_{tt} Martinelli parameter (-)

Re Reynolds number (-)

E Two-phase multiplier constant

- Modified Gungor and Winterton [86]

$$h_{tp} = h_{fo} \left\{ 1 + 3000Bo_o^{0.86} + 1.12 \left(\frac{x}{1-x} \right)^{0.75} \left(\frac{\rho_L}{\rho_g} \right)^{0.41} \right\} \quad (3.63)$$

$$h_{fo} = 0.023 \left(\frac{k_1}{D_{ab,i}} \right) \left(\frac{G(1-x)D_{abi}}{\mu_L} \right)^{0.8} (Pr_L)^{0.4} \quad (3.64)$$

k_1 Thermal conductivity for saturated water ($W.m^{-1}.K^{-1}$)

$D_{ab,i}$ Absorber inner diameter (m)

G Mass flux ($kg.m^{-2}.s^{-1}$)

x Vapor quality (-)

μ_L Liquid dynamic viscosity ($Pa.s$)

Pr_L Prandtl number (-)

ρ_l Liquid density ($kg.m^{-3}$)

ρ_g Gas density ($kg.m^{-3}$)

- LIU-WINTERTON

$$h_{tp} = ((F_{LW}h_{conv})^2 + (S_{LW}h_{nb})^2)^{0.5} \quad (3.65)$$

$$F_{LW} = \left\{ 1 + xPr \left(\frac{\rho_l}{\rho_g} - 1 \right) \right\}^{0.35} \quad (3.66)$$

$$S_{LW} = 1/(1 + 0.055F_{LW}^{0.1}Re_L^{0.16}) \quad (3.67)$$

x Vapor quality (-)

Pr_L Prandtl number (-)

ρ_l Liquid density ($kg.m^{-3}$)

ρ_g Gas density ($kg.m^{-3}$)

- WATTELET ET AL. [87]

$$h_{tp} = ((h_{nb})^{2.5} + (h_{bc})^{2.5})^{1/2.5} \quad (3.68)$$

h_{nb} Nucleate boiling local heat transfer coefficient contribution ($Wm^{-2}k^{-1}$)

h_{bc} Bulk convective local heat transfer coefficient contribution ($Wm^{-2}k^{-1}$)

$$h_{nb} = 55M^{-0.5}q^{0.67}pr^{0.12}[-\log_{10} Pr]^{-0.55} \quad (3.69)$$

Pr Prandtl number (-)

M Molecular weight

q Heat flow (W)

$$h_{bc} = F \cdot h_1 \cdot R \quad (3.70)$$

R Multiplier constant of two-phase

F Multiplier constant of two-phase

- Silver-Bell-Ghaly [88]

$$1/h_m = 1/h_c + Z_g/h_v \quad (3.71)$$

$$h_c = E_{new} h_{lo} \quad (3.72)$$

$$Z_g = x \cdot C p_v \cdot dT_{dew}/dt \quad (3.73)$$

$$E_{new} = 1 + 3000(Bo \cdot F_c)^{0.86} + 1.12 \left(\frac{x}{1-x} \right)^{0.85} \left(\frac{\rho_l}{\rho_v} \right)^{0.7} \quad (3.74)$$

Bo Boiling number (-)

ρ_l Liquid density ($kg \cdot m^{-3}$)

ρ_g Gas density ($kg \cdot m^{-3}$)

x Vapor quality (-)

F Two-phase multiplier constant

- GRANRYD [89]

$$h_m / h_{lo} = F_p / 1 + A = F_m \quad (3.75)$$

A Heat transfer area (m^2)

F_p Multiplier constant of two-phase

F_m Multiplier constant of two-phase

$$h_m / h_{lo} = F_p / 1 + A = F_m \quad (3.76)$$

$$A = \left(\frac{F_p}{C_{lg}} \right) x^2 \left[\frac{1-x}{x} \left(\frac{\mu_v}{\mu_l} \right) \right]^{0.8} \left(\frac{Pr_l}{Pr_v} \right)^{0.4} \left(\frac{k_l}{k_v} \right) \left(\frac{Cp_v}{Cp_w} \right), Cp_w = \left(\frac{\theta_i}{\theta_T} \right)_p \quad (3.77)$$

x Vapor quality (-)

Pr_l Liquid Prandtl number (-)

μ_l Liquid dynamic viscosity ($Pa.s$)

ρ_l Liquid density ($kg.m^{-3}$)

Pr_v Vapor Prandtl number (-)

μ_v Vapor dynamic viscosity ($Pa.s$)

ρ_v Vapor density ($kg.m^{-3}$)

3.4 Pressure drop of two-phase flow

Firstly, basic definitions are presented in the next part then the two-phase pressure drop correlations are reported

3.4.1 Basic Definitions and terminologies

- Volumetric flow rate $Q \left(\frac{m^3}{s} \right)$ is the mass flow rate $\dot{m} \left(\frac{kg}{s} \right)$ to density $\rho \left(\frac{kg}{m^3} \right)$ ratio

$$Q = \frac{\dot{m}}{\rho} \quad (3.77)$$

- $\dot{m}_{tp} \left(\frac{kg}{s} \right)$ defined as the summation of the liquid phase $\dot{m}_l \left(\frac{kg}{s} \right)$ and gas phase

$\dot{m}_g \left(\frac{kg}{s} \right)$ mass flow rates

$$\dot{m}_{tp} = \dot{m}_l + \dot{m}_g \quad (3.78)$$

- Total mass flux $G_{tp} (kg/m^2.s)$ is the summation of the liquid phase $G_l (kg/m^2.s)$ and the gas phase $G_g (kg/m^2.s)$ mass fluxes

$$G_{tp} = G_l + G_g \quad (3.79)$$

- Quality $x(-)$ is gas $\dot{m}_g \left(\frac{kg}{s}\right)$ to the total mass flow rates $\dot{m}_{tp} \left(\frac{kg}{s}\right)$ ratio

$$x = \frac{\dot{m}_g}{\dot{m}_{tp}} \quad (3.80)$$

- Slip ratio $S(-)$ is the gas phase $V_g(m/s)$ to the liquid phase average velocities $V_l(m/s)$ ratio

$$S = \frac{V_g}{V_l} \quad (3.81)$$

- Void fraction $\alpha(-)$ is the gas $A_g (m^2)$ to the total cross-sectional areas $A_t (m^2)$ ratio

$$\alpha = \frac{A_g}{A_l + A_g} \quad (3.82)$$

- Two-phase flow pressure drop (PD)

The total two-phase pressure drop $\frac{\Delta P}{\Delta L_{tp}} (kPa/m)$ is the sum of acceleration $\frac{\Delta P}{\Delta L_a} (kPa/m)$, frictional losses $\frac{\Delta P}{\Delta L_f} (kPa/m)$ and hydrostatic pressure drop $\frac{\Delta P}{\Delta L_h} (kPa/m)$

$$\frac{\Delta P}{\Delta L_{tp}} = \frac{\Delta P}{\Delta L_a} + \frac{\Delta P}{\Delta L_f} + \frac{\Delta P}{\Delta L_h} \quad (3.83)$$

Hydrostatic PD is the PD because of elevation variation. A frictional pressure drop occurs because of shear's irreversible work at the pipe walls. The frictional pressure drop is more complicated because it depends on different parameters, such as mass flux, pipe diameter, pipe inclination, the contact area between phases, and the fluid parameters. There are many correlations for frictional pressure drop in the open literature. The lack of understanding of two-phase flow has led researchers to develop different correlations with some restrictions of flow parameter ranges. The user of this correlation should be aware of this restriction.

3.4.2 Two-phase flow pressure drop correlations

Pressure drop correlations can be categorized depending on several criteria; for example, they can be classified based on flow pattern, pipe inclination, and development method. Correlation classification based on the pipe inclination can be horizontal, vertical, or any inclination pipe flow angle.

Classification of frictional PD correlation based on flow pattern is important because the frictional PD correlation depends on the flow pattern. Different flow models are found in the literature in which the correlations are developed, for instance, separation flow model, empirical models and homogenous flow model.

In the separated flow models, each phase flows separately from the other phase with different velocities, unlike the homogenous flow model in which the two fluids are assumed to have the equivalent velocity.

Normally the liquid two-phase friction multiplier is more preferred to be used than the gas two-phase friction multiplier, and the reason is the density of the liquid does not change much, while the density of the gas changes. In 1944, Martinelli et al. introduced the two-phase friction factor multiplier, and Martinelli and Nelson develop it in 1948. The authors proposed an empirical correlation based on experimental data.

- LOCKHART AND MARTINELLI [90]

In 1949, Lockhart and Martinelli introduced a correlation for two-phase pressure drop based on experimental data gathered from different studies. The authors assumed that the pressure drop (static) of the liquid and gas phases equal for all the flow patterns when there is no considerable radial static pressure difference. Also, the total volume of the pipe is equal to the sum of the volumes occupied by each phase, the mechanism of the liquid and

gas flow is categorized as laminar-laminar (viscous-viscous) (vv), laminar-turbulent (viscous-turbulent) (vt), turbulent laminar (turbulent-viscous) (tv) and turbulent-turbulent (tt) respectively. The authors introduced parameter X as a function of the mass fluxes, viscosities, densities, and pipe diameter. It relates the single-phase pressure drop for the liquid and gas as if each one is flowing in the pipe alone.

$$X^2 = \left[\frac{\frac{\Delta P}{\Delta L}_L}{\frac{\Delta P}{\Delta L}_g} \right] \quad (3.84)$$

$$X_{vv}^2 = \frac{\dot{m}_l \mu_l \rho_g}{\dot{m}_g \mu_g \rho_l} \quad (3.85)$$

$$X_{vt}^2 = Re_g^{-0.8} \frac{C_l \dot{m}_l \mu_l \rho_g}{C_g \dot{m}_g \mu_g \rho_l} \quad (3.86)$$

$$X_{tv}^2 = Re_l^{0.8} \frac{C_l \dot{m}_l \mu_l \rho_g}{C_g \dot{m}_g \mu_g \rho_l} \quad (3.87)$$

$$X_{tt}^{1.11} = \frac{\dot{m}_l \mu_l^{0.111} \rho_g^{0.555}}{\dot{m}_g \mu_g^{0.111} \rho_l^{0.555}} \quad (3.88)$$

$$f_l = \frac{C_l}{Re_l^n} \quad (3.89)$$

$$f_g = \frac{C_g}{Re_g^m} \quad (3.90)$$

Re_l, Re_g are the Reynolds number for liquid and gas when each one of them is running alone, f_l, f_g are the liquid and gas friction factors, C_l, C_g are the liquid and gas constants in the Blasius equation of friction factor. The variables m,n, C_g and C_l are presented in table 3.1.

Table 3.1: Variables m, n, Cg and Cl

Variables	Viscous-Viscous (v-v)	Viscous-Turbulent (v-t)	Turbulent-Viscous (t-v)	Turbulent-Turbulent (t-t)
m	1	0.2	1	0.2
n	1	1	0.2	0.2
C_g	0.046	0.046	16	16
C_l	0.046	16	0.046	16

The friction parameter ϕ is a function of Martenelli parameter X and it is used to relate single phase to two phase pressure drop, ϕ_l , ϕ_g are the friction multipliers when the flow is only liquid and only gas respectively.

$$\phi_l^2 = \left[\frac{\frac{\Delta P}{\Delta L}_{tp}}{\frac{\Delta P}{\Delta L}_l} \right] \quad (3.91)$$

$$\phi_g^2 = \left[\frac{\frac{\Delta P}{\Delta L}_{tp}}{\frac{\Delta P}{\Delta L}_g} \right] \quad (3.92)$$

$$\Delta p_l = \left[4f_L \left(\frac{L}{D} \right) G^2 (1-x)^2 \left(\frac{1}{2\rho L} \right) \right] \quad (3.93)$$

$$\Delta p_g = \left[4f_G \left(\frac{L}{D} \right) G^2 x^2 \left(\frac{1}{2\rho G} \right) \right] \quad (3.94)$$

$$\phi_L^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \quad (3.95)$$

$$\phi_G^2 = 1 + C + X^2 \quad (3.96)$$

The values of the constant C are presented in table 3.2

Table 3.2: Values of the constant C for Lockhart-Martenelli curves

Liquid	Gas	C
Turbulent	turbulent	20
Viscous	turbulent	12
Turbulent	Viscous	10
Viscous	Viscous	5

To compute the 2ph frictional PD utilizing Lockhart-Martinelli method, firstly, the frictional PD for any individual phase flowing calculated then, the Lockhart-Martenelli parameters ϕ_L, ϕ_g calculated. This correlation developed for horizontal pipe two-phase flow for the low-pressure system.

- BANKOFF [91]

Bankoff derived an expression for axial velocity variation and void fraction inside the tube.

The frictional PD is

$$\frac{dp}{dz_{fric}} = \frac{dp}{dz_{Lo}} \phi_{Bf}^{\frac{7}{4}} \quad (3.97)$$

Frictional PD for liquid only

$$\frac{dp}{dz_{Lo}} = \frac{f_{lo} 2G^2}{D\rho L} \quad (3.98)$$

The liquid friction factor

$$f_{Lo} = \frac{0.079}{Re_{Lo}^{0.25}}, \quad Re_{Lo} = \frac{GD}{\mu L} \quad (3.99)$$

Bankoff Multiplier is

$$\phi_{Bf} = \frac{1}{1-x} \left[1 - \gamma \left(1 - \frac{\rho_G}{\rho_L} \right) \right]^{\frac{3}{7}} \left[1 + x \left(\frac{\rho_L}{\rho_G} - 1 \right) \right] \quad (3.100)$$

$$\gamma = \frac{0.71 + 2.35 \left(\frac{\rho_G}{\rho_L} \right)}{1 + \left(\frac{1-x}{x} \right) \left(\frac{\rho_G}{\rho_L} \right)} \quad (3.101)$$

This technique is used for vapor qualities from zero to one.

- CICCHITTI ET AL.[92]

The author established a correlation for the systems of steam-water; the correlation works reasonably with other systems.

The frictional pressure drop is

$$\frac{dp}{dz_{fric}} = \frac{0.092}{D^{1.2}} G^{1.8} [\mu_L - x(\mu_L - \mu_G)]^{0.2} [v_L + x(v_G - v_L)] \quad (3.102)$$

v_L and v_G are the specific volumes of the liquid and gas, respectively

- THOM [93]

The author proposed a way to evaluate the PD through the 2ph forced circulation of water and steam boiling. From operating pressure and outlet vapor quality, the frictional, accelerational, and gravitational PD of 2ph flow is extracted from the curves. The pressure drop is given by

$$\Delta P_{total} = \Delta P_{static} + \Delta P_{mom} + \Delta P_{frict} \quad (3.103)$$

$$= G^2 \left(\frac{1}{\rho_L} \right) r_2 + 4f_{Lo} \left(\frac{L}{D} \right) G^2 \left(\frac{1}{2\rho_L} \right) r_3 + \rho_L g L r_4$$

r_2 , r_3 and r_4 are determined from the curves

- PIERRE[94]

Pressure drop and heat transfer measured by Pierre for refrigerants evaporating in horizontal tubes, he predicted the accelerational and frictional pressure drop. He derived the following equations from the energy equation

$$\Delta P_{total} = \Delta P_{mom} + \Delta P_{frict} \quad (3.104)$$

$$= \frac{\left[f_{BP} + \frac{(x_{out} - x_{in})D}{xL} \right] G^2 x v_G L}{D}$$

f_{BP} is the pure refrigerant 2ph friction factor valid when $\frac{Re}{K_f} > 1$

$$f_{BP} = 0.0185 \left(\frac{K_f}{Re_{Lo}} \right)^{0.25} \quad (3.105)$$

K_f is the Pierre boiling number

$$K_f = \frac{\Delta x h_{LG}}{Lg} \quad (3.106)$$

Re_{Lo} is the all liquid Reynolds number

$$Re_{Lo} = \frac{GD}{\mu L} \quad (3.107)$$

- BAROCZY[95]

The author presented a correlation to anticipate the two-phase frictional PD for single and two-component flow as a function of fluid properties. To compute the frictional PD in this method, equation no (3.97) was used. The frictional pressure drop then is

$$\left(\frac{dp}{dz}\right)_{fric} = \frac{2f_{fo}G^2v_f}{D}\phi_{fo(G=1365)}^2\Omega \quad (3.108)$$

v_f and v_g are the liquid and vapor specific volume and f_{fo} is the liquid friction factor

- CHAWLA[96]

Proposed correlation based on the gas pressure drop as it follows:

$$\left(\frac{dp}{dz}\right)_{fric} = \left(\frac{dp}{dz}\right)_{Go}\phi_{Chaw} \quad (3.109)$$

$$\left(\frac{dp}{dz}\right)_{Go} = \frac{f_{Go}2G^2}{D\rho G} \quad (3.110)$$

The gas friction factor

$$f_{Go} = \frac{0.079}{Re_{Go}^{0.25}}, \quad Re_{Go} = \frac{GD}{\mu_G} \quad (3.111)$$

Chawla two-phase multiplier

$$\phi_{Chaw} = x^{1.75} \left[1 + S \left(\frac{1-x}{x} \frac{\rho_G}{\rho_L} \right) \right]^{2.375} \quad (3.112)$$

S the slip ratio is:

$$S = \frac{u_G}{u_L} = \frac{1}{9.1 \left[\frac{1-x}{x} (Re_{Go} F_{rH})^{-0.167} \left(\frac{\rho_L}{\rho_G} \right)^{-0.9} \left(\frac{\mu_L}{\mu_G} \right)^{-0.5} \right]} \quad (3.113)$$

$$F_{rH} = \frac{G^2}{gD\rho_h^2} \quad (3.114)$$

The homogenous density ρ_h

$$\rho_h = \left(\frac{x}{\rho_G} + \frac{1-x}{\rho_L} \right)^{-1} \quad (3.115)$$

Valid for $0 < x < 1$. In annular flow, for high values of vapor quality works sensibly fine.

- CHISHOLM [97]

The 2ph frictional PD of Chisholm specified as follows

$$\Delta p_{fric} = \phi_{Ch}^2 \Delta p_{Lo} \quad (3.116)$$

The frictional PD for liquid and Gas phases are determined

$$\Delta p_{Lo} = 4f_{Lo} \left(\frac{L}{D} \right) G^2 \left(\frac{1}{2\rho_L} \right) \quad (3.117)$$

$$\Delta p_{Go} = 4f_{Go} \left(\frac{L}{D} \right) G^2 \left(\frac{1}{2\rho_G} \right) \quad (3.118)$$

Where turbulent friction factor is

$$f_{Lo} = \frac{0.079}{Re_{Lo}^{0.25}}, \quad Re_{Lo} = \frac{GD}{\mu L} \quad (3.119)$$

$$f_{Go} = \frac{0.079}{Re_{Go}^{0.25}}, \quad Re_{Go} = \frac{GD}{\mu_G} \quad (3.120)$$

The friction factor for laminar flow ($Re < 2000$) given as follows

$$f = \frac{16}{Re} \quad (3.121)$$

The ratio of gas frictional PD to the liquid frictional PD defined as the parameter Y

$$Y^2 = \frac{\Delta P_G}{\Delta P_L} \quad (3.122)$$

The Chisholm 2ph multiplier is

$$\phi_{Ch}^2 = 1 + (Y^2 - 1) \left[Bx^{\frac{2-n}{2}} (1-x)^{\frac{2-n}{2}} + x^{2-n} \right] \quad (3.123)$$

$n = 0.25$ from Blasius friction equation

For $0 < Y < 9.5$

$$B = \frac{55}{G^{0.5}} \text{ for } G \geq 1900 \frac{kg}{m^2 s}, \quad (3.124)$$

$$B = \frac{2400}{G} \text{ for } 500 < G < 1900 \frac{kg}{m^2 s},$$

$$B = 4.8 \text{ for } G \leq 500 \frac{kg}{m^2 s}$$

For $9.5 < Y < 28$

$$B = \frac{520}{YG^{0.5}} \text{ for } G \leq 600 \frac{kg}{m^2 s}, \quad (3.125)$$

$$B = \frac{21}{Y} \text{ for } G > 600 \frac{kg}{m^2 s},$$

For $Y > 28$

$$B = \frac{1500}{Y^2 G^{0.5}} \quad (3.126)$$

After calculating the value of B, then the 2ph multiplier can be determined, and the PD then calculated.

- FRIEDEL [98]

For vapor quality between $0 \leq x < 1$. It is one of the most accurate correlations to calculate pressure drop for two-phase flow.

$$\Delta P_{fric} = \Delta P_{LO} \phi_{fo}^2 \quad (3.127)$$

The liquid pressure drop ΔP_{LO}

$$\Delta p_{LO} = 4f_{Lo} \left(\frac{L}{D} \right) G^2 \left(\frac{1}{2\rho_L} \right) \quad (3.128)$$

The two-phase multiplier is

$$\phi_{fo}^2 = E + \frac{3.24FH}{Fr_H^{0.045} We_L^{0.035}} \quad (3.129)$$

Fr_H , E, F and H are as follows:

$$Fr_H = \frac{G^2}{gD\rho_h^2} \quad (3.130)$$

$$E = (1-x)^2 + \frac{x^2 \rho_L f_{GO}}{\rho_G f_{LO}} \quad (3.131)$$

$$F = x^{0.78} (1-x)^{0.224} \quad (3.132)$$

$$H = \left(\frac{\rho_L}{\rho_G} \right)^{0.91} \left(\frac{\mu_G}{\mu_L} \right)^{0.19} \left(1 - \frac{\mu_G}{\mu_L} \right)^{0.7} \quad (3.133)$$

We_L is weber number defined as:

$$We_L = \frac{G^2 D}{\sigma \rho_h} \quad (3.134)$$

ρ_h is the homogenous density

$$\rho_h = \left(\frac{x}{\rho_G} + \frac{1-x}{\rho_L} \right)^{-1} \quad (3.135)$$

The correlation is appropriate for vertical and horizontal flow; it is applicable for most working fluids and when the $\left(\frac{\mu_L}{\mu_G} \right) < 1000$.

- MULLER-STEINHAGEN AND HECK [99]

The frictional PD accordingly is

$$\left(\frac{dp}{dz} \right)_{fric} = F(1-x)^{\frac{1}{3}} + Bx^3 \quad (3.136)$$

F is a factor defined as

$$F = A + 2(B-A)x \quad (3.137)$$

A and B factors are the frictional PD for all liquid $\left(\frac{dp}{dz} \right)_{Lo}$ and all gas

$\left(\frac{dp}{dz} \right)_{Go}$ flow respectively:

$$\left(\frac{dp}{dz} \right)_{Lo} = \frac{f_{lo} 2G^2}{D\rho L} \quad (3.138)$$

$$\left(\frac{dp}{dz} \right)_{Go} = \frac{f_{Go} 2G^2}{D\rho G} \quad (3.139)$$

The liquid and gas friction factors

$$f_{Lo} = \frac{0.079}{Re_{Lo}^{0.25}}, \quad Re_{Lo} = \frac{GD}{\mu L} \quad (3.140)$$

$$f_{Go} = \frac{0.079}{Re_{Go}^{0.25}}, \quad Re_{Go} = \frac{GD}{\mu_G} \quad (3.141)$$

The present correlation has been checked, in addition to 14 correlations reported in the open literature against wide data of frictional PD. The correlation predicted the frictional pressure drop and the flow parameters influence with reasonable accuracy.

- PATANKAR ET AL. [100]

Predict the two-phase friction factor independent of flow pattern

$$f_m = F_2 + \frac{F_1 - F_2}{\left[1 + \left(\frac{Re}{t}\right)^c\right]^d} \quad (3.142)$$

F_1 and F_2 are the power laws defined as:

$$F_1 = a_1 Re^{b1}, \quad F_2 = a_2 Re^{b2} \quad (3.143)$$

The parameters $c, d, t, a1, b1, a2, b2$ shown in table 3.3

Table 3.3: c,d,t,a1,b1,a2,b2 parameters

$a1$	$b1$	$a2$	$b2$	c	d	t
13.98	-0.9501	0.0925	-0.2534	4.864	0.1972	293

The universal composite correlation friction factor is given by

$$f_m = 0.0925Re^{-0.2534} + \frac{13.98Re^{-0.9501} - 0.0925Re^{-0.2534}}{\left[1 + \left(\frac{Re}{293}\right)^{4.864}\right]^{0.1972}} \quad (3.144)$$

- D. D. JOSEPH[101]

The author fit the data for the 2060 experiment to power laws to obtain the composite correlation. Each experiment classified depend on the flow pattern, for instance, 1316 for slug flow, 40 dispersed bubble, 176 annular flow, and 528 stratified flow; for each flow pattern, the composite correlation has different values for the parameters as follows

Table 3.4 shows the values of the parameters for different flow patterns.

$$f_m = F_2 + \frac{F_1 - F_2}{\left[1 + \left(\frac{Re}{t}\right)^c\right]^d} \quad (3.145)$$

F_1 and F_2 are the power laws defined as:

$$F_1 = a_1 Re^{b_1}, \quad F_2 = a_2 Re^{b_2} \quad (3.146)$$

Table 3.4: The parameters $c, d, t, a_1, b_1, a_2, b_2$ for different flow patterns

Flow pattern	a_1	b_1	a_2	b_2	c	d	t
Slug flow	13.98	-0.9501	0.1067	-0.2629	3.577	0.2029	293
Dispersed bubble	13.98	-0.9501	0.1067	-0.2629	2.948	0.2236	304
Stratified flow	13.98	-0.9051	0.0445	-0.1874	9.275	0.0324	300
Annular flow	3.671	-0.6257	0.0270	-0.1225	2.191	0.2072	10000

The composite correlation for slug flow, dispersed bubble flow, stratified flow, and annular

flow respectively as follows

$$f_m = 0.1067e^{-0.2629} + \frac{13.98Re^{-0.9501} - 0.1067e^{-0.2629}}{\left[1 + \left(\frac{Re}{293}\right)^{3.577}\right]^{0.2029}} \quad (3.147)$$

$$f_m = 0.1067e^{-0.2629} + \frac{13.98Re^{-0.9501} - 0.1067e^{-0.2629}}{\left[1 + \left(\frac{Re}{304}\right)^{2.948}\right]^{0.2236}} \quad (3.148)$$

$$f_m = 0.0445Re^{-0.1874} \quad (3.149)$$

$$+ \frac{13.98Re^{-0.9501} - 0.0445Re^{-0.1874}}{\left[1 + \left(\frac{Re}{300}\right)^{9.275}\right]^{0.0324}}$$

$$f_m = 0.0270Re^{-0.1225} \quad (3.150)$$

$$+ \frac{3.671Re^{-0.6257} - 0.0270Re^{-0.1225}}{\left[1 + \left(\frac{Re}{10000}\right)^{2.191}\right]^{0.2072}}$$

- M.M. AWAD [102]

The acceleration part of PD represents the variation in the flow kinetic energy. Assuming uniform velocities of vapor and liquid in each phase, the acceleration part of PD found by

$$\Delta P_a = G^2 \left\{ \left[\frac{1 - x_o^2}{\rho_l(1 - \alpha_o)} + \frac{x_o^2}{\rho_g \alpha_o} \right] - \left[\frac{(1 - x_i)^2}{\rho_l(1 - \alpha_i)} + \frac{x_i^2}{\rho_g \alpha_i} \right] \right\} \quad (3.151)$$

ΔP_a is the accelerational pressure drop (*kpa*)

G is the mass flux (*kg/m²s*)

x_o is the outlet vapor quality (-)

x_i is the inlet vapor quality (-)

ρ_l is the liquid density (*kg/m³*)

ρ_g is the gas density (*kg/m³*)

α_i is the inlet void fraction (-)

α_o is the outlet void fraction (-)

3.5 Proposed design of a flow homogenizer stage

The main target of this part is to design and test an intermediate flow homogenizer stage. This stage's major function is to enhance the mixing of the two-phase fluids to improve the roto-dynamic pump's performance in handling two-phase mixtures with high GVF. The new homogenizer stage should have the same size as other pump stages but with a different impeller design and driven by the same shaft. However, in this part, a novel design of the impeller vane and novel concept of utilization, i.e., permitting installation at intermediate locations and each homogenizer stage will have the same size as the pump stage but with an impeller equipped with perforated vanes.

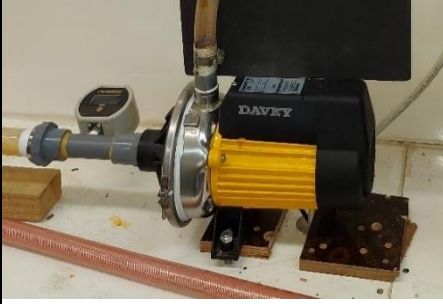
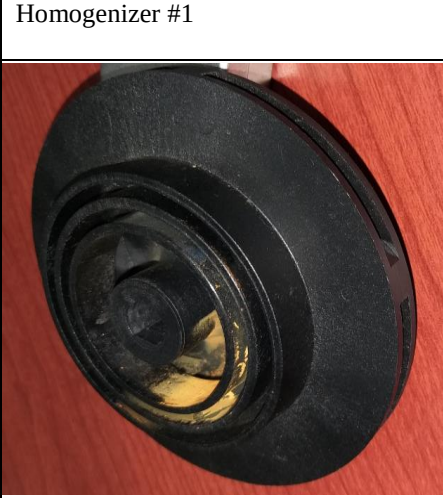
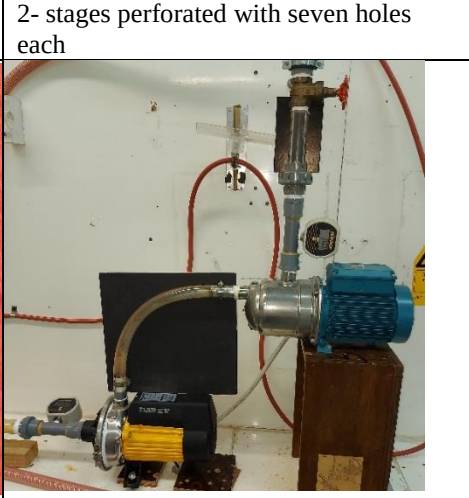
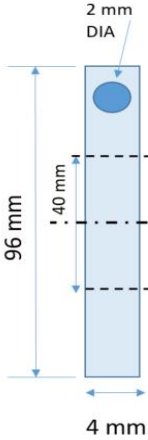
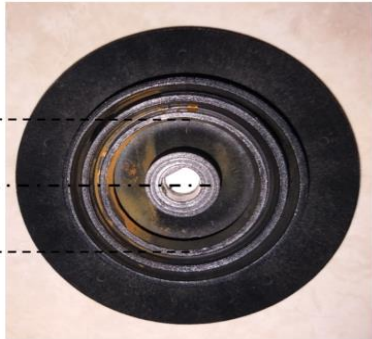
Experimental evaluation of the efficiency and the trade-off between head generation and homogenization investigated in the experimental work as a function of the GVF.

Figure 3.3 shows the setup used to generate the Head-Flow rate (H-Q) curve for the pump and the homogenizer for the single-phase case, and figure 3.4 shows the experimental setup to test the homogenizer and homogenizer followed by the main 4-stage pump under the two-phase flow conditions. The experimental facility is capable of measuring the flow rate of liquid (water) and flow rate of gas (air), and the pressure at the intake and discharge sides of the pump and homogenizer. The flow loop is equipped with two transparent sections in the pump's intake and discharge sides for visualizing the mixture homogeneity. Different tests were conducted to observe the level of homogenization through the transparent section downstream of the pump (compared to the condition in the transparent section upstream of the pump) and the change of pressure across the pump at a different gas void fraction as a function of the other parameters.

The experimental facility involves two stages radial impeller centrifugal pump (as a homogenizer) and a 4-stage, radial impeller centrifugal pump (as the main pump). The present and future experimental matrix and the pump arrangements used shown in table 3.5. A two-stage centrifugal pump was prepared as a homogenizer #1. Four sets of impellers were procured and prepared such that two of them are perforated, one with seven holes 2 mm DIA each and the other one is having four holes that have the same diameter of 2 mm. Later, three more holes were added to have similar perforated impellers with 7 holes 2 mm Dia. The details of homogenizer#1 (two stages pump) are shown in figure 3.3.

Table 3.5: Experimental test matrix and pumps used

Pump type	GVF	Impeller
Two-stage, radial impeller pump as a homogenizer #1	Single phase flow	Unperforated vane
		Single-stage perforated
		Two stages perforated
	Two-Phase flow	Unperforated
		One stage perforated
		Two stages perforated
Four stage radial impeller pump as the main pump	Single-phase flow	unperforated
		the first stage perforated only
		second stages perforated only
	Two-Phase flow	Unperforated
		first stage perforated
Two-stage pump (homogenizer#1) followed by the 4-stage main pump	Two-phase flow: variable gas and fixed inlet liquid flowrate	The two-stage pump has both stages are perforated with seven holes each, followed by the 4-stage unperforated pump
	Two-phase flow– variable gas and liquid flowrate with constant inlet pressure	The two-stage pump has both stages are perforated with seven holes each, followed by the 4-stage unperforated pump
Single-stage radial-pump (homogenizer# 2) followed by four stage pump	Two-phase experiments-constant inlet pressure	Impeller with extension inspired by the split vane pump

		
Homogenizer #1	2- stages perforated with seven holes each	Holes distributions
		
Impeller	Homogenizer connected to 4 stage pump	4 holes distribution
The arc length $L = 60 \text{ mm}$ Hole diameter is 2 mm in 4 mm width The void ratio is $2\text{mm times } 4 \text{ holes} / 60 = 13\%$ And for the 7 holes $2\text{mm times } 7 / 60 = 23\%$  		
Impeller details		

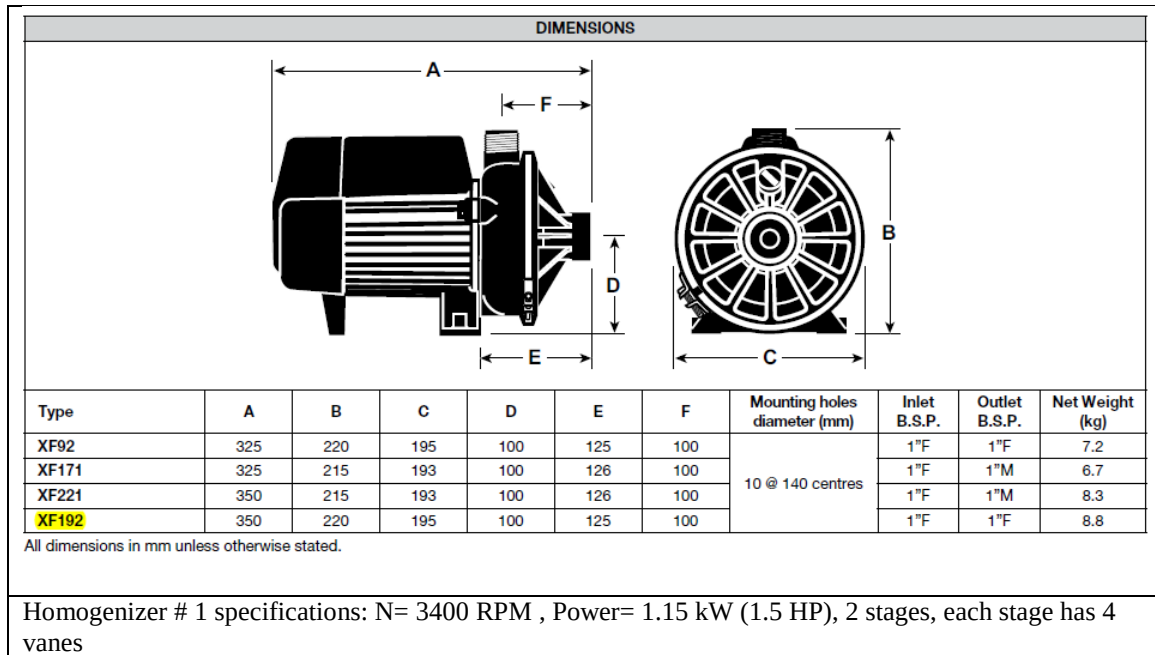


Figure 3.3: Homogenizer #1 details [103]

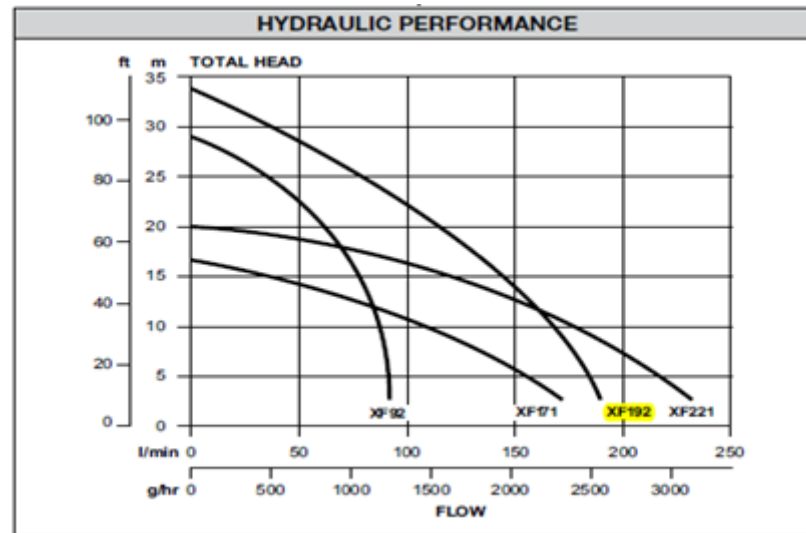


Figure 3.4: Pump Hydraulic performance curve [103]

- **Rated power and speed of all pumps used in the experimental part**

The brand name, input power, and the speed of all pumps used in this study shown in table 3.6. The pump numbers are shown in figure 3.5

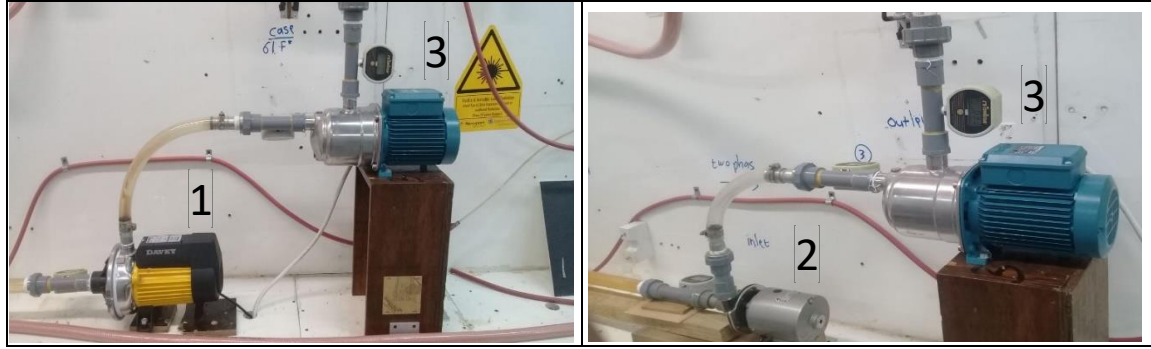


Figure 3.5: Pumps numbering

Table 3.6: Pumps input power & RPM [103]

#	brand	Power	R.P.M	Q		H(m)	Remarks
1	Davey XF 192	1.15 kW	3420	N.A		N.A	2 stage
2	Bersford PV22	1/8 HP	3400	N.A		N.A	1 stage
3	Calpeda MXPM 204-60/A	1 HP	3450	6 m ³ /hr (max)		16.5-58.5	4 stage

3.5.1 Procedure of experiments and measurements

A flexible test facility was designed and constructed at the fluid mechanics' laboratory at the Mechanical Engineering Department. Figure 3.6 & Figure 3.7 show the preliminary setup. The test rig can measure the flow rate of liquid (water) and flow rate of gas (air), and pressure at the pump's intake and discharge sides.

A two-stage centrifugal pump was prepared to conduct the first set of experiments. The first stage is perforated, as shown in figure 3.3. The pump specifications are shown in figure 3.3 & 3.4. In the next stage, the homogenizer is installed upstream of the 4-stage pump to study the influence of the number of stages downstream of the homogenizer on the level of flow homogeneity. 4 equal holes 1 mm each Impeller width, $b = 1/8$ inch, $D_2 = 3.75$ inch, $D_1 = 1.5$ inch.

CHAPTER 4

RESULTS AND DISCUSSION

In this chapter the results of the research are presented and discussed in details starting by the flow pattern maps used, heat transfer coefficient correlations, wall temperature, fluid temperature, void fraction, pressure drop correlations and finally the homogenizer designing, testing and validation.

4.1 Flow pattern map

Some of the flow patterns used in the adiabatic conditions when there is no heat transfer and others used in diabatic conditions when there is heat transfer to the working fluid are our direct steam generation cases. Wojtan et al. developed a new diabatic flow pattern map [66], [67], which is used in this study; the results have been compared with the unified model and Barneia showed in table 4.1.

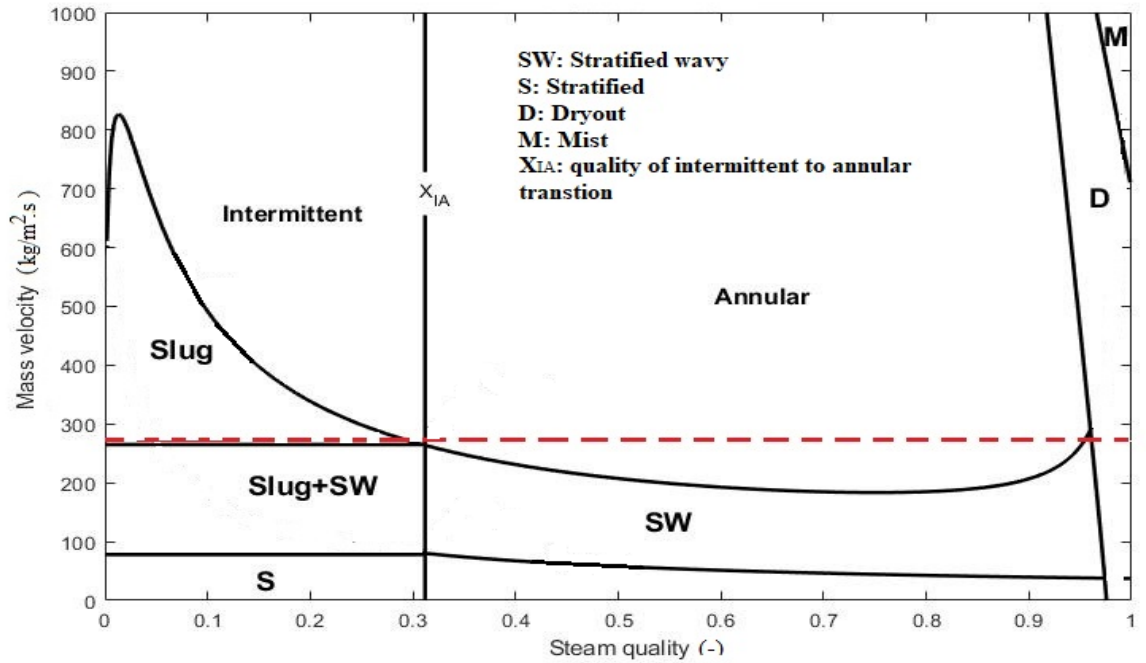


Figure 4.1: Wojtan et al flow pattern map

Table 4.1: Flow pattern map comparison

	Flow pattern		
x	unified model	Wojtan and Thome	Barneia
0	Single-phase liquid	Single-phase liquid	Single-phase liquid
0.03	Slug	Slug	Intermittent
0.06			
0.09			
0.12			
0.15			
0.18	Stratified	Slug	Annular
0.21			
0.24			
0.27			
0.3			
0.33			
0.36			
0.39			
0.42			
0.45			
0.48			
0.51	Annular	Annular	
0.54			

0.57					
0.6					
0.63					
0.66	Annular				
0.69					
0.72					
0.75					
0.78					
0.81					
0.84					
0.87					
0.9					
0.93					
0.96	Stratified wavy				
0.99	Dry out				
1	Single-phase vapor	Single-phase vapor	Single-phase vapor		

4.2 Heat transfer coefficient correlations

Different correlations of the two-phase heat transfer coefficient were reported previously: DENGLE AND ADDOMS correlation, KANDLIKAR correlation, CHEN correlation, BENNET-CHEN correlation GUNGOR AND WINTERTON correlation, Modified Gungor and Winterton correlation, LIU-WINTERTON correlation, WATTELET ET AL. Correlation, Silver-Bell-Ghaly correlation, and GRANRYD correlation. The previous correlations were compared and validated with experimental data for water and for refrigerants (R134a and R22).

4.2.1 Validation with R134a experimental data

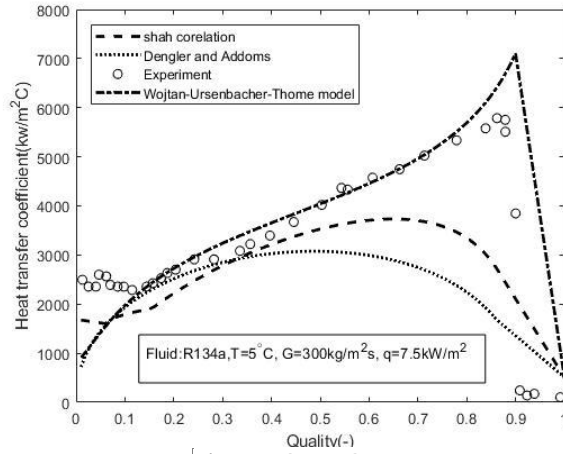


Figure 4.2: HTC vs vapor

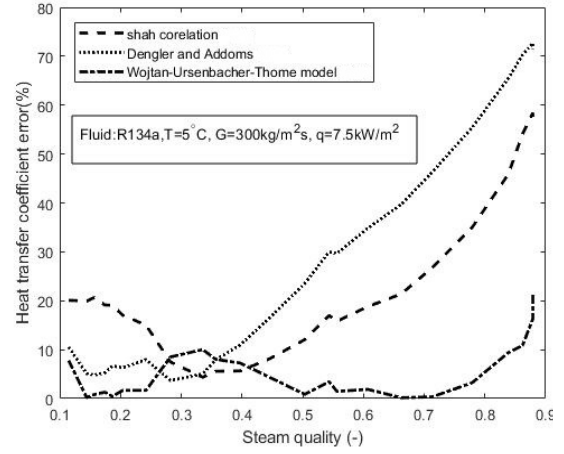


Figure 4.3: HTC error vs vapor quality

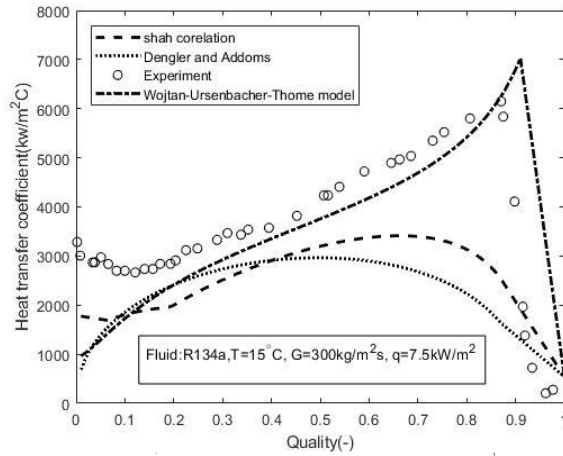


Figure 4.4: HTC vs vapor quality

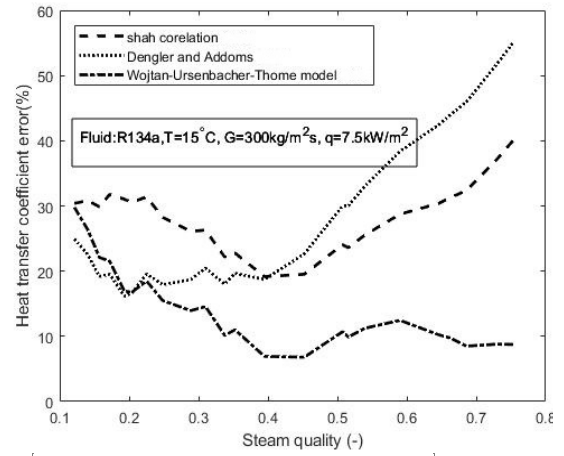


Figure 4.5: HTC error vs vapor quality

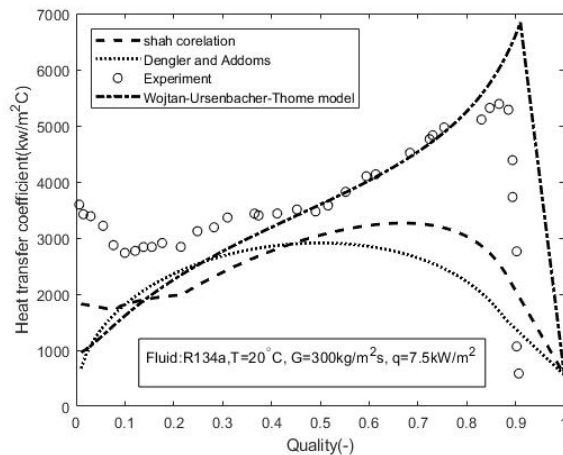


Figure 4.6: HTC vs vapor quality

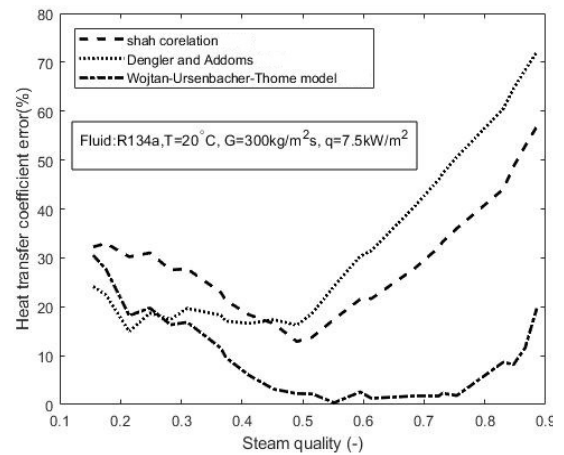


Figure 4.7: HTC error vs vapor quality

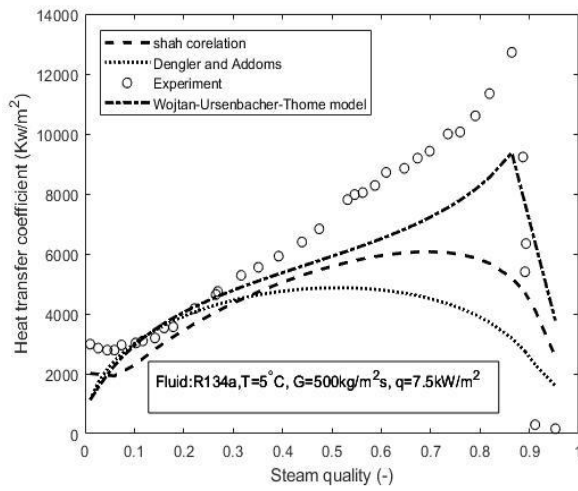


Figure 4.8: HTC vs vapor quality

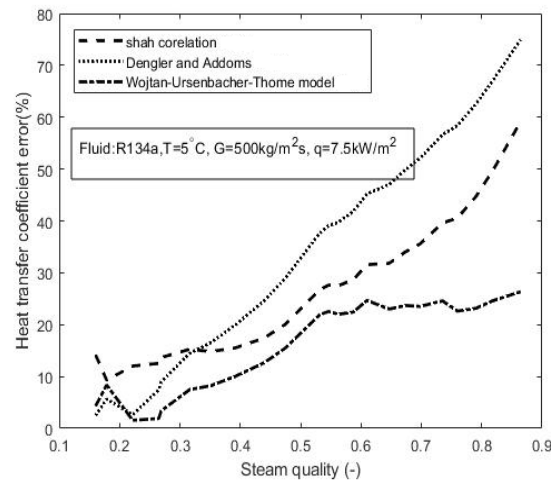


Figure 4.9: HTC error vs vapor quality

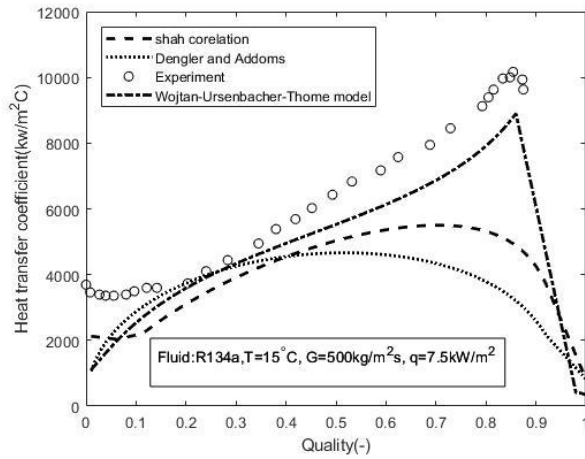


Figure 4.10: HTC vs vapor quality

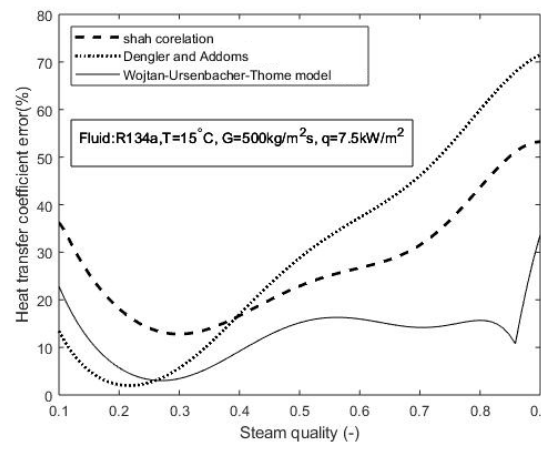


Figure 4.11: HTC error vs vapor quality

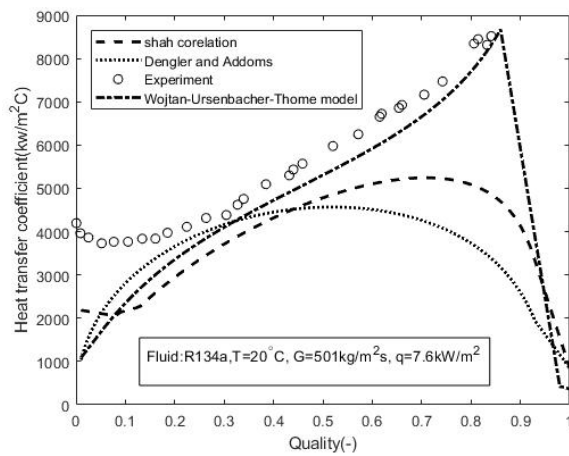


Figure 4.12: HTC vs vapor quality

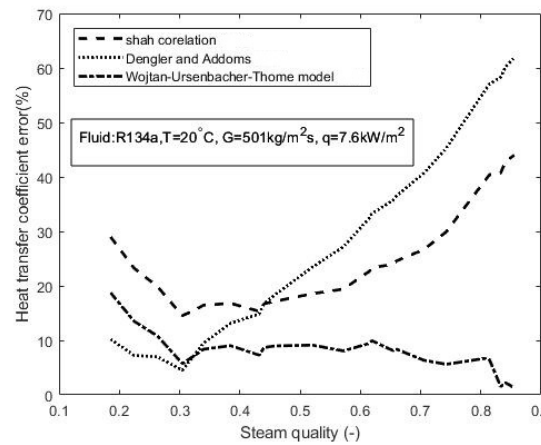


Figure 4.13: HTC error vs vapor quality

Figures 4.2-4.13 explain how the 2-phase HTC changes with quality at each point of the absorber for refrigerant R134a with different operating conditions. Temperature, mass velocity, and heat flux, respectively. Case 1 (5 °C, 300 kg/m² and 7.5 kW/m²), case 2 (15 °C, 300 kg/m² and 7.5 kW/m²), case 3 (20 °C, 300 kg/m² and 7.5 kW/m²), case 4 (5 °C, 500 kg/m² and 7.5 kW/m²), case 5 (15 °C, 500 kg/m² and 7.5 kW/m²), case 6 (20 °C, 501 kg/m² and 7.6 kW/m²). In all figures 4.2, 4.4, 4.6, 4.8, 4.10, and 4.12, when the water starts changing to steam and the quality increases, the 2-phase HTC increases in all correlations, and the experimental data also shows the same trend. However, when the mass quality reaches a certain value at which the FP changes from **A** to **D** flow regime, the 2-phase HTC decreases rapidly, which occurs because most of the fluid has been changed into vapor, resulting in a high temperature on the tube surface, which may cause tube failure. The Wojtan model has the best agreement with the data of the experiment. The percentage error of HTC is fluctuating when the mass quality increases for all models; however, the Wojtan et al. model agrees with the data of the experiment better than the Dengelor and Addoms and Shah [80] Correlations.

4.2.2 Validation with water experimental data

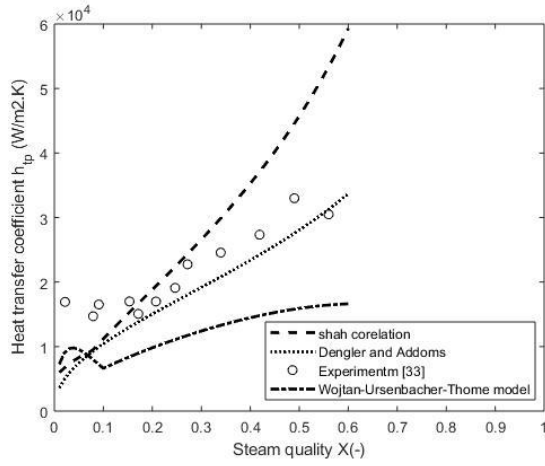


Figure 4.14: Heat transfer coefficient vs quality

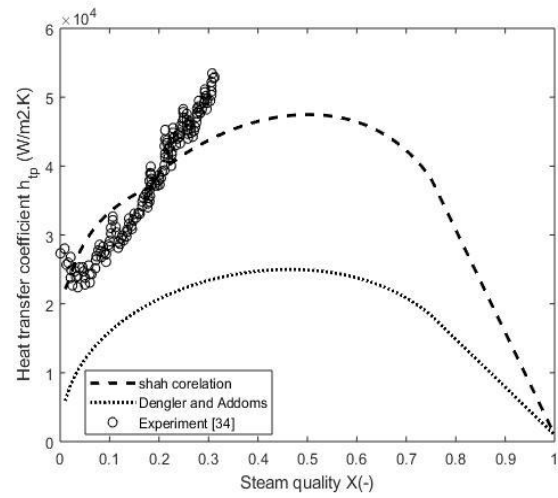


Figure 4.15: Heat transfer coefficient vs quality

Figures 4.14 and 4.15 clarify how the 2-phase HTC changes with the vapor mass quality at each point of the absorber for water. The results show that the Shah [80] correlation agrees more closely with the experimental data than Dengler and Addoms.

4.3 Wall and fluid temperatures validation

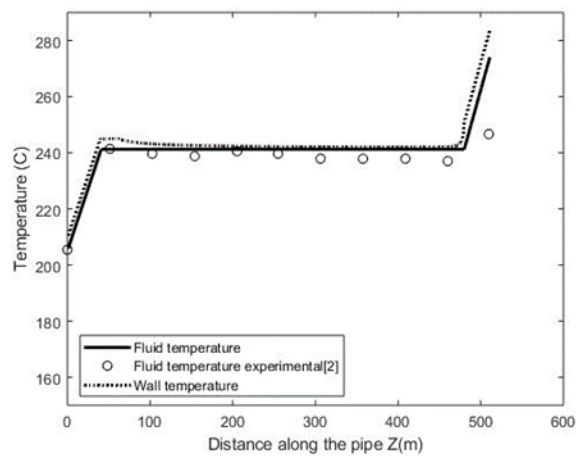


Figure 4.16: Wall and fluid temperature distribution along pipe

The absorber fluid is water, both water and wall temperatures going through three stages along the absorber tube. Firstly, when the quality is zero, the water and wall temperatures increase sharply until the quality starts to increase above zero. The wall and mixture temperatures are almost constant in this 2-phase region. After that, the quality increases above one, the wall and steam temperatures increase dramatically in this superheated area.

Considering the experiment of Plataforma Solar de Almeria[57] in which the fluid used is water, inlet pressure=34.2 bar, inlet temperature=205°C; pipe diameter=50 mm, L=510 m, $q=13257 \text{ w/m}^2\cdot\text{k}$, $\dot{m}=0.52 \text{ kg/s}$

Figure 4.16 explains how the fluid temperature changes along the pipe at specific inlet pressure and temperature of 34.2 bar and 205°C, respectively; increasing the inlet mass flow rate rises the distance along the pipe at which the superheating region starts. However, the liquid phase starts to change when $x > \text{zero}$ until it reaches one when the flow is entirely in a gaseous phase. The fluid (steam) then superheated to a specific outlet temperature depending on the inlet conditions. In the 2-phase area, the surface temperature depends on the HTC model.

4.4 Void fraction validation

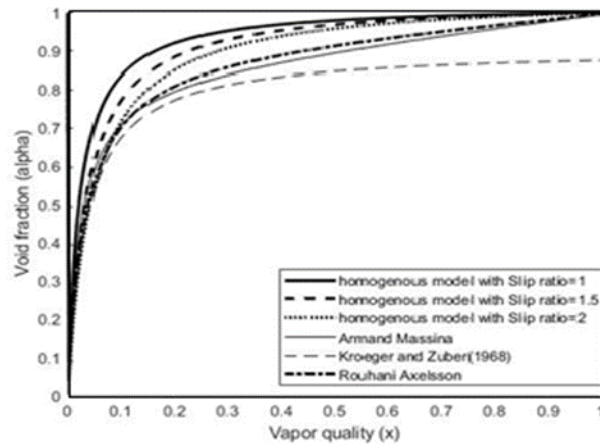


Figure 4.17: Void fraction variation with the vapor quality

The void fraction is a function of mass quality, which also increases from zero to one in the 2-phase region, as shown in figure 4.17.

Figure 4.17 shows how the void fraction changes across the pipe when the quality increases from zero to one. The void fraction is very important in calculating the boiling regime. The slip ratio affects the void fraction for homogenous models, while the Kroger and Zubair correlation does not give a good prediction for the void fraction at high qualities.

4.5 Pressure drop validation

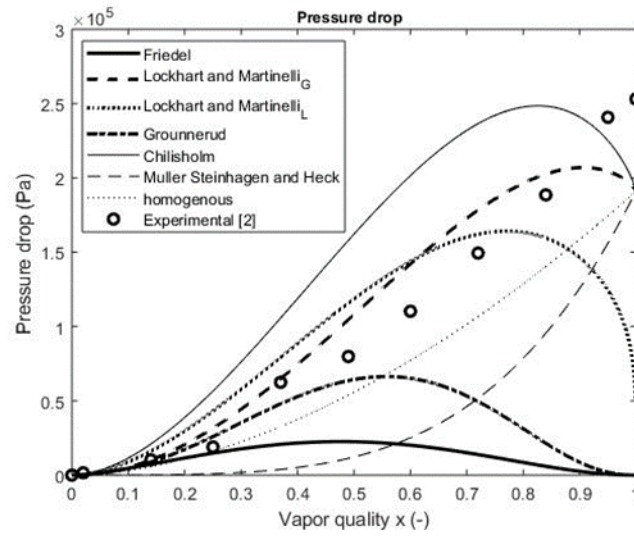


Figure 4.18: Pressure drop variation with mass quality

Figure 4.18 illustrates how the PD increases, while the quality and pipe length increase in the 2-phase region. Lockhart Martinelli and homogenous models are found to be in a good agreement with the experimental data.

4.6 Parameters optimization

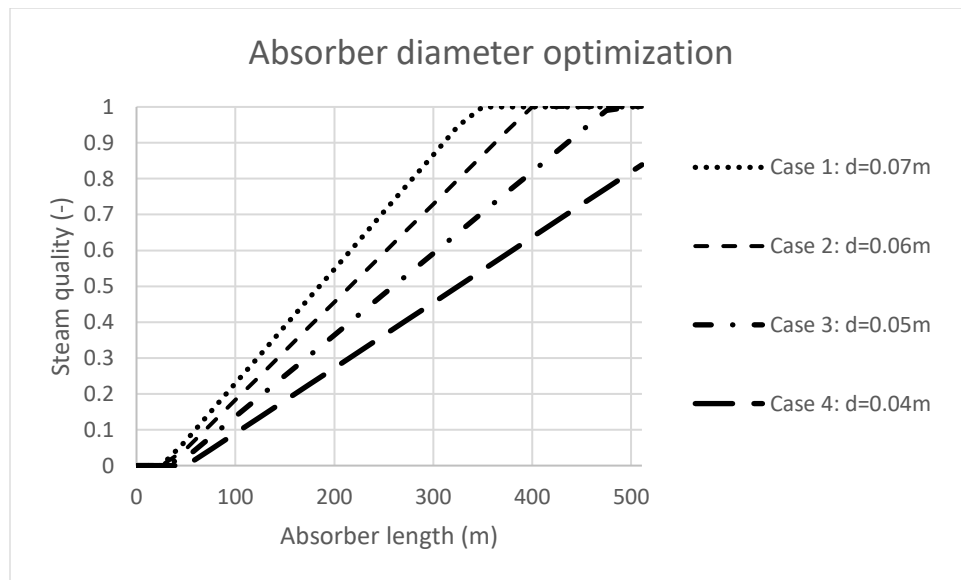


Figure 4.19: Outlet mass quality for different pipe diameters

Different parameters affect the direct steam generation process for example (pipe diameter, pipe length, heat flux, inlet pressure, inlet temperature and inlet flowrate), increasing the heat flux increases the steam generated while increasing the pipe length decreases the quantity of the steam generated. However, the most influential parameter is the absorber diameter. Figure 4.19 shows that increasing the pipe diameter increases the water quality until it reaches one (maximum steam generated). On the other hand, increasing the absorber diameter produces stratified flow pattern inside the absorber due to reduction in the mass velocity (unfavorable flow pattern leads to tube failure). The best case when we have annular flow pattern with maximum steam generated.

Table 4.2 shows the maximum steam generated for different scenarios

Table 4.2: Steam generated for different cases

Case	d (m)	Mass flowrate (kg/s)	Mass velocity (kg/m ² s)	Mass quality (-)	Heat flux (W/m ²)	Steam generated(kg/s)
1	0.07	0.52	135.10	1.00	13257	0.52
2	0.06	0.52	183.89	1.00	13257	0.52
3	0.05	0.52	264.80	1.00	13257	0.52
4	0.04	0.52	413.75	0.84	13257	0.44
5	0.03	0.52	735.55	0.61	13257	0.31
6	0.02	0.52	1655.00	0.37	13257	0.19
7	0.01	0.52	6619.99	0.14	13257	0.07

After the quality reaches one, increasing diameter is not recommended because the cost increases, the flow pattern transferred to stratified and no more steam will be generated.

The optimum case between the cases shown in figure 4.19 and table 4.2 is case 3 when the diameter is 0.05m because the outlet steam is the maximum (0.52 kg/s) and the flow pattern inside is annular.

4.7 Experimental validation of homogenizer 1 (2-stage pump) performance

4.7.1 Single-phase flow experiment of homogenizer #1

These experiments were conducted to validate the instrumentations H-Q curve of the homogenizer (perforated impeller) compared to the solid (unperforated) impeller. The test facility showed in figure 3.6 was used in the experiment. The flowrate and head varied by controlling the valve downstream. Digital turbine flowmeter (element number 3) (Fill-Rite TT10AN 1" 2-35 GPM (8-132 LPM) Digital In-line Turbine Meter, Aluminum) with the accuracy of 1%.used to measure the water flow rate. The suction and discharge pressure were measured using an Omega digital pressure gauge with an accuracy of 0.25% of full scale (element numbers 6 & 8).

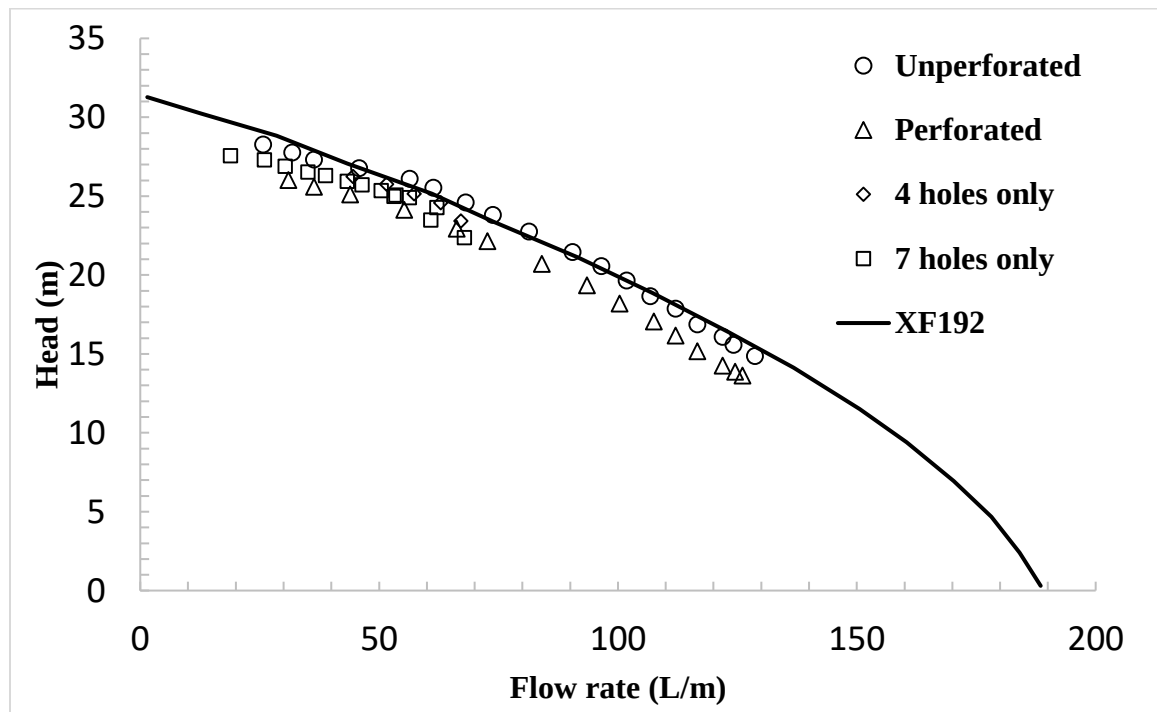


Figure 4.20: H-Q curve for perforated and unperforated impellers

Figure 4.20 shows the H-Q curve of the two-stage pump used in the present experiments obtained from the pump manual and the experimental data. The empty circle is the experimental data for the unperforated impeller, while the triangles are for the perforated impeller. It is worth to mention that the arrangement for the two stages perforated impeller was as follows: the seven holes in the outer stage (stage1) right after the intake and the four holes in the inner stage (stage 2).

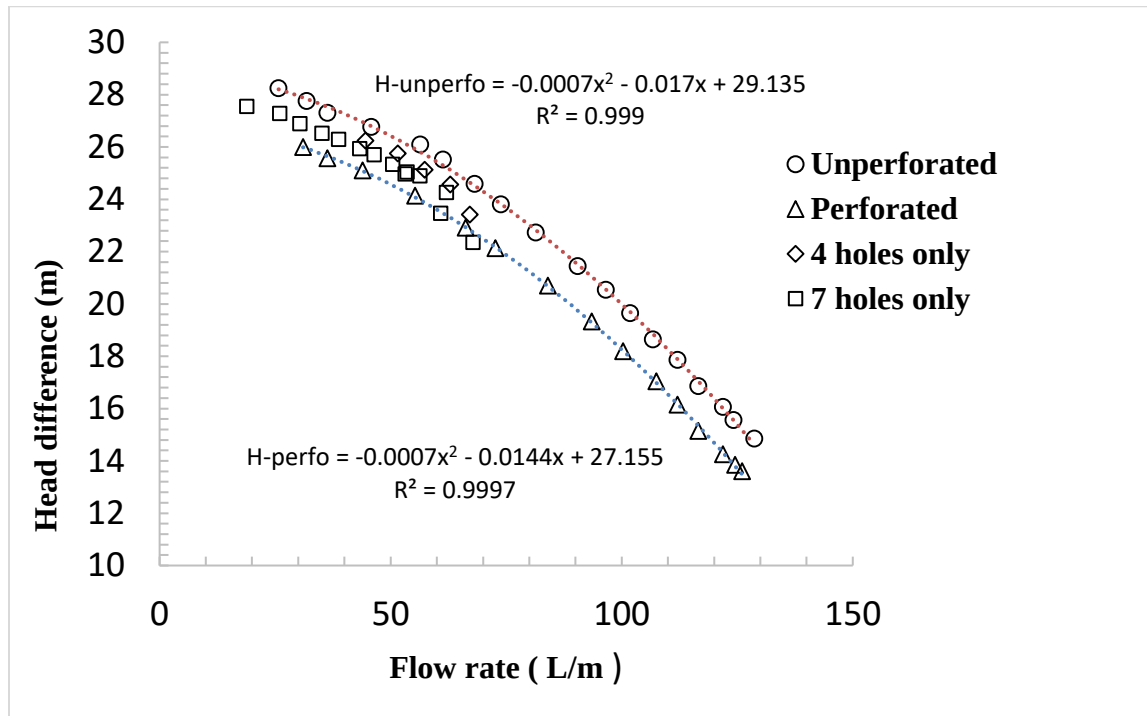


Figure 4.21: H-Q curve compared to the homogenizer

Figure 4.21 shows zoom in and the trend of the experimental data. Figure 4.22 shows the head difference and the head difference ratio to the head at the specific flow rate between the unperforated and perforated ($\Delta H = H_{\text{unperforated}} - H_{\text{perforated}}$ and $\Delta H/H = H_{\text{unperforated}} - H_{\text{perforated}} / H_{\text{unperforated}}$) as a function of the flow rate.

Figure 4.22 reveals that the head loss due to perforation is around 10% for most of the flowrate range. The maximum head lost due to perforation is 45% at the maximum flow rate (minimum head)

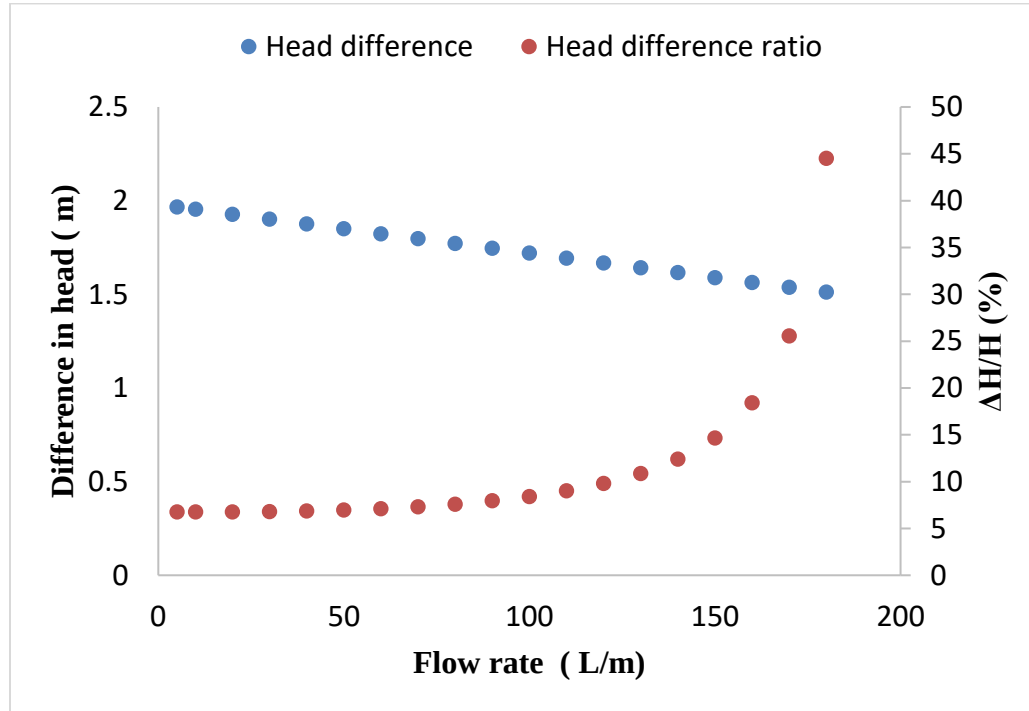


Figure 4.22: Head difference & head difference ratio between perforated and unperforated impellers

4.7.2 Two-phase flow experiment of homogenizer #1

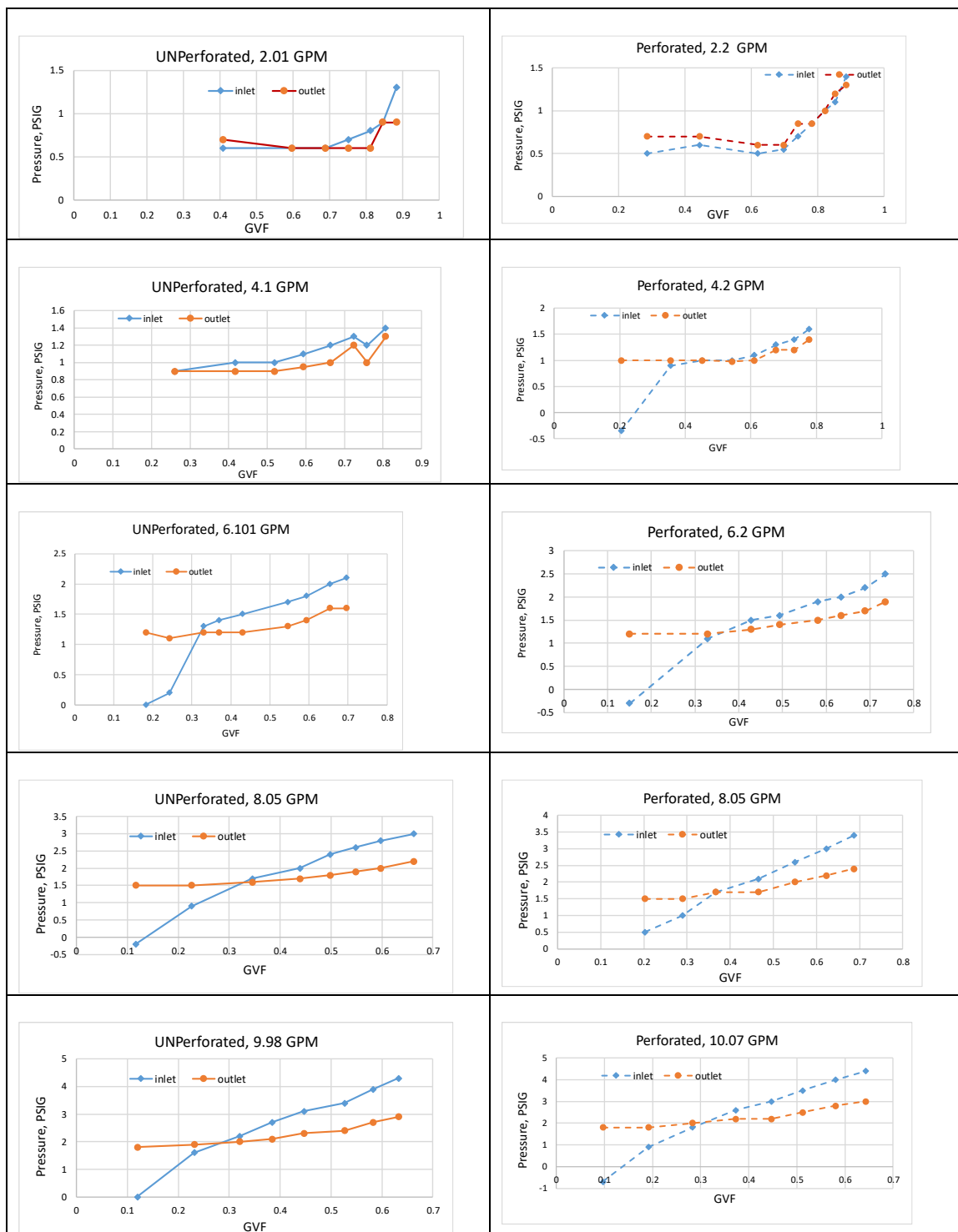
The two-phase flow experiments were conducted using the rig shown in figure 3.7, in which a booster pump provides the required water flow rate by adjusting the valve downstream of the booster pump and upstream of the homogenizer pump. The gas was injected around 100 cm upstream of the homogenizer. The pressure at the homogenizer's intake and discharge was recorded, also, the inlet liquid and gas flow rates. The flow pattern at the discharge was also reported. The flow pattern at the intake was mainly in the slug and stratified flow regimes.

a) Pump with two stages perforated or unperforated (Homogenizer #1 only)

Figure 4.23 shows a comparison between perforated and unperforated impellers of the two-stage pump acting as a homogenizer. The outlet and inlet pressure variation with the void fraction for the homogenizer are presented.

In the unperforated experiments, both stages were unperforated. In the perforated experiments, the impeller was arranged so that the outer stage next to the intake is the one perforated with 4 holes only, and the inner stage is with 7 holes. Figure 4.23 displays the inlet and outlet pressure distribution with the GVF for the perforated and unperforated 2-stage pump; at the lowest liquid flow rate, the perforated impeller performs better than the unperforated indicated by the higher discharge pressure over the suction pressure up to $\text{GVF} = 0.8$. However, when the liquid flow rate increases, the point at which the inlet and outlet pressure are equal moves to a smaller value of GVF. Considering the above analysis, one can argue that a constant total mixture flow rate should not be considered a constant liquid flow rate.

In the experiments shown in figure 4.23, the liquid flow rate was fixed, and the gas flow rate was increased until the pump suction pressure becomes greater than the discharge pressure. Assuming that the total mixture flowrate (gas+liquid) is a homogenous mixture, then the total flow might be at the end of the pump H-Q curve thus, the head might be very low at that flow rate compared to the ones at low total flowrate where the pump head is high. Thus, the comparison will be unfair. The experimental data were sorted, and a sector of about 10 L/M was considered to have approximately similar performance (same flow rate and head) as shown in figure 4.24 below and used for comparing the perforated with the unperforated cases as shown in figures 4.24 and 4.25.



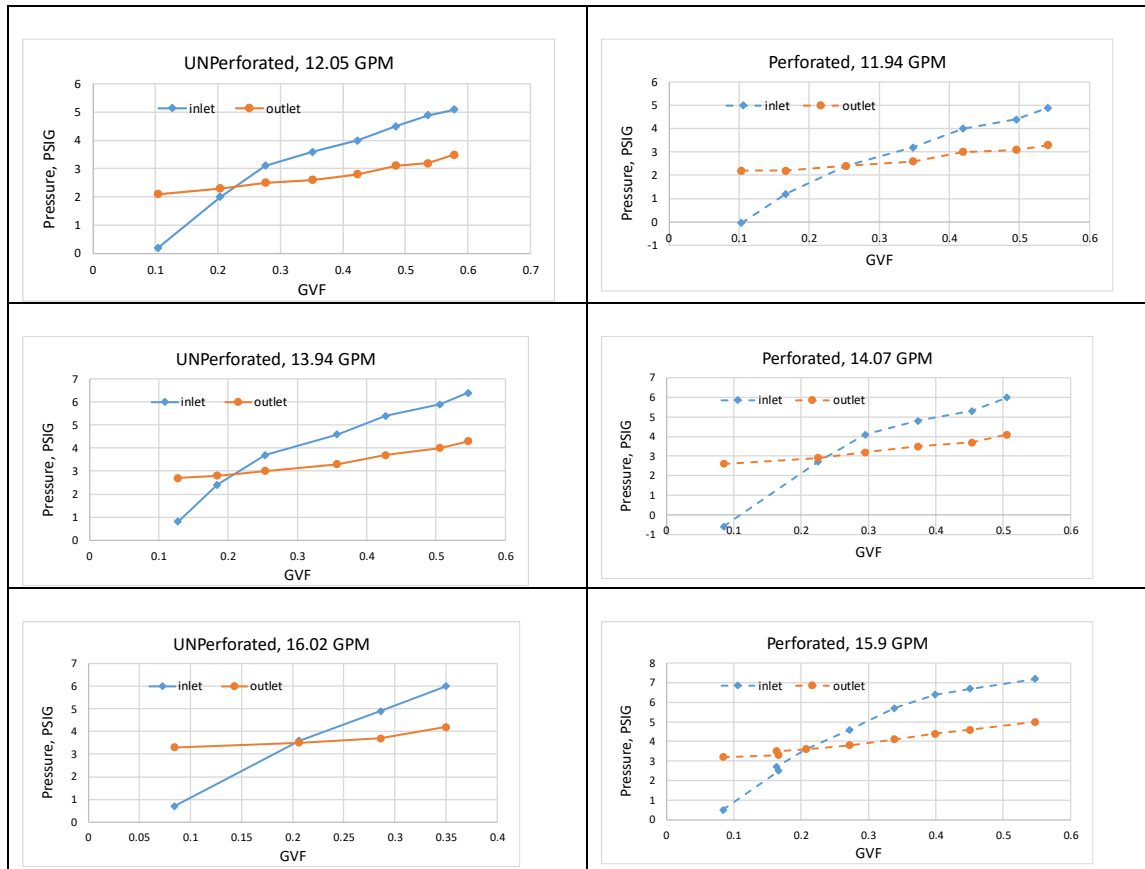


Figure 4.23: Inlet & outlet pressure variation with GVF for unperforated and perforated 2-stage pump at constant liquid flow rate

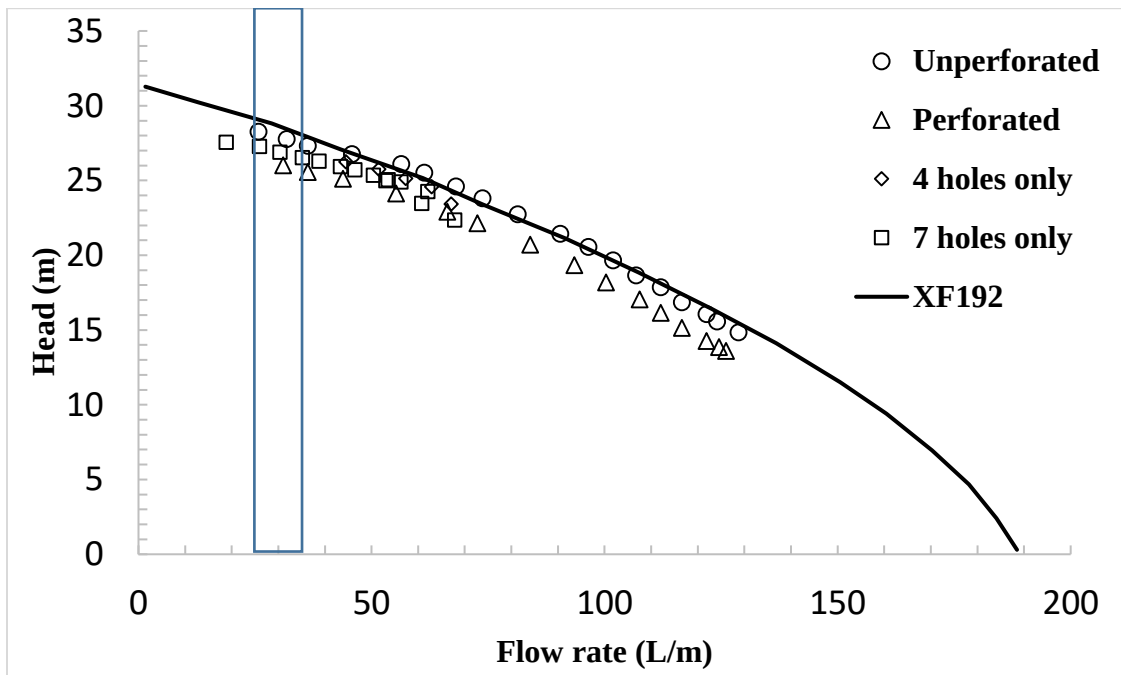
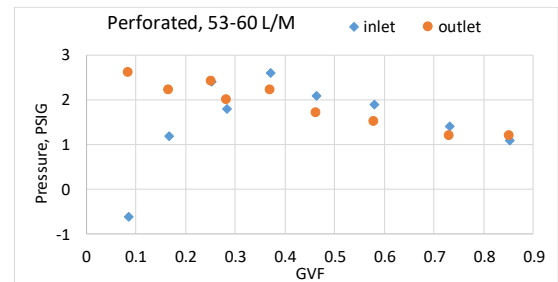
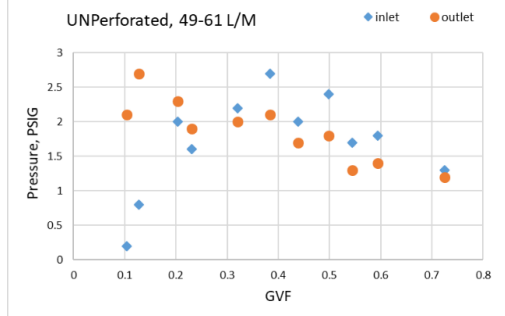
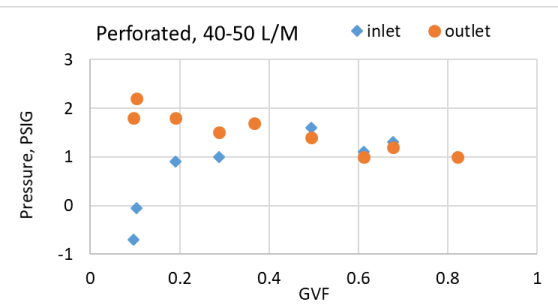
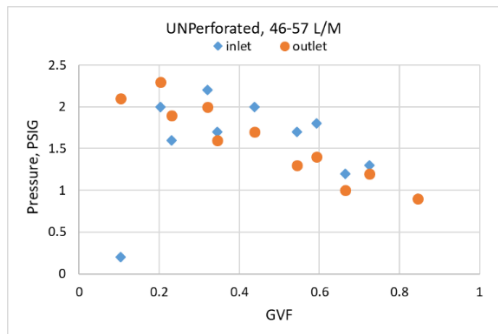
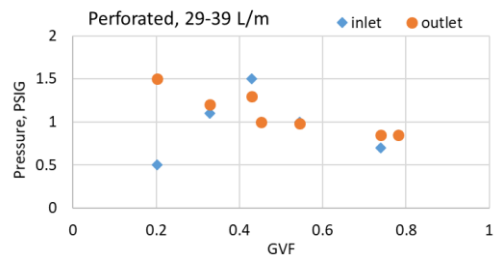
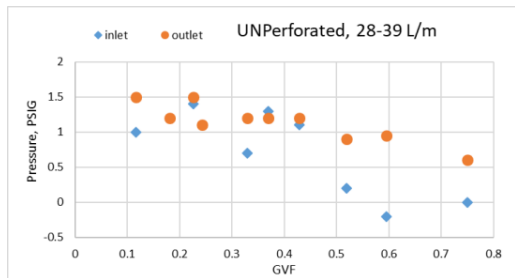
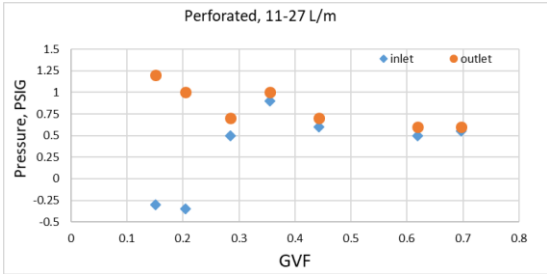
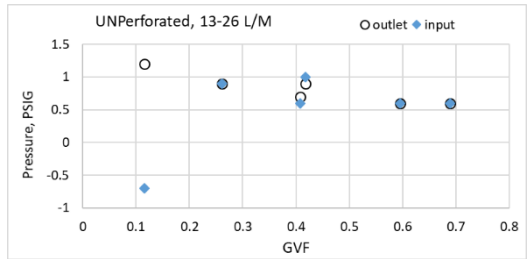


Figure 4.24: H-Q curve of 2-stage pump with 10 L/M



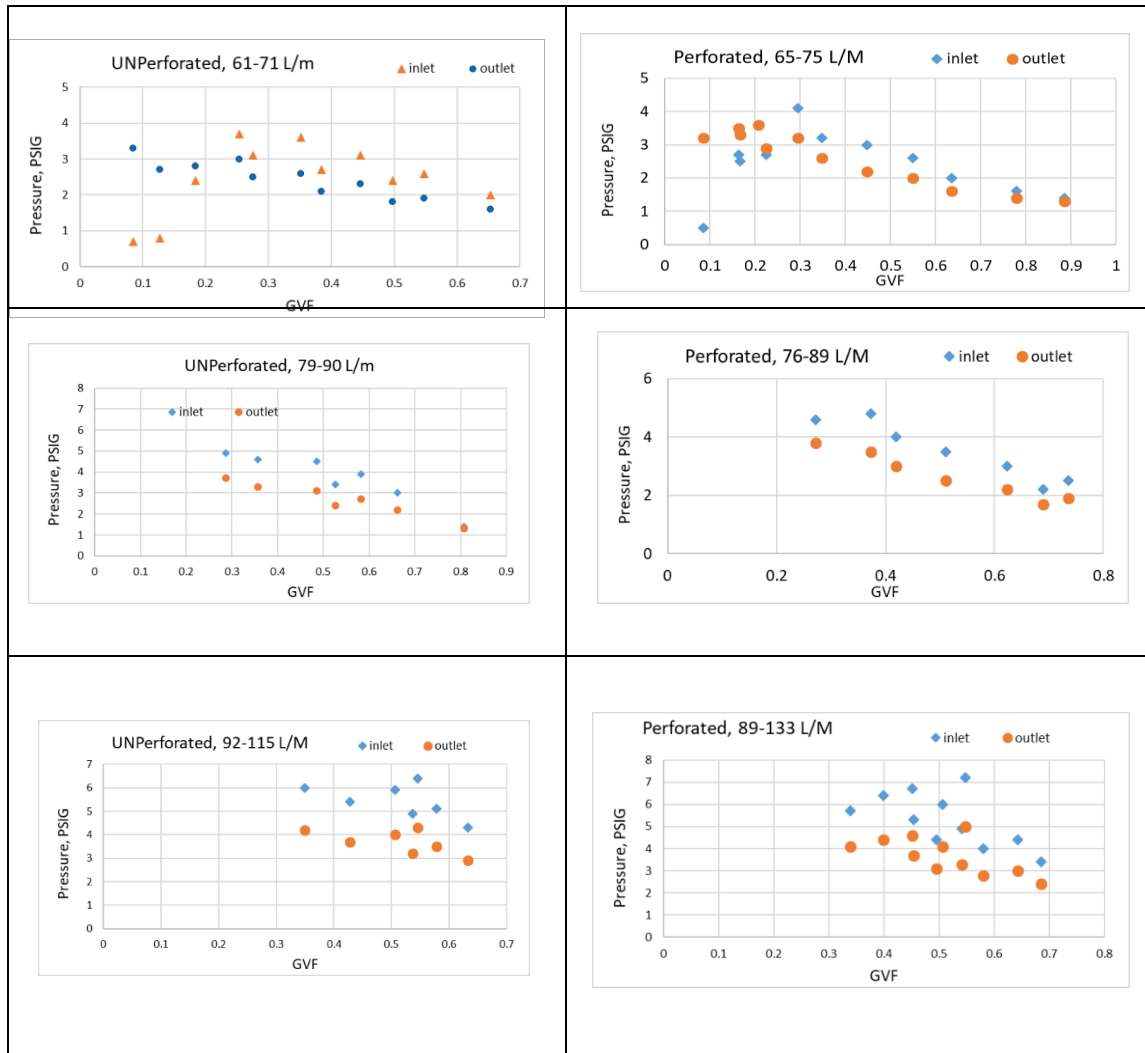
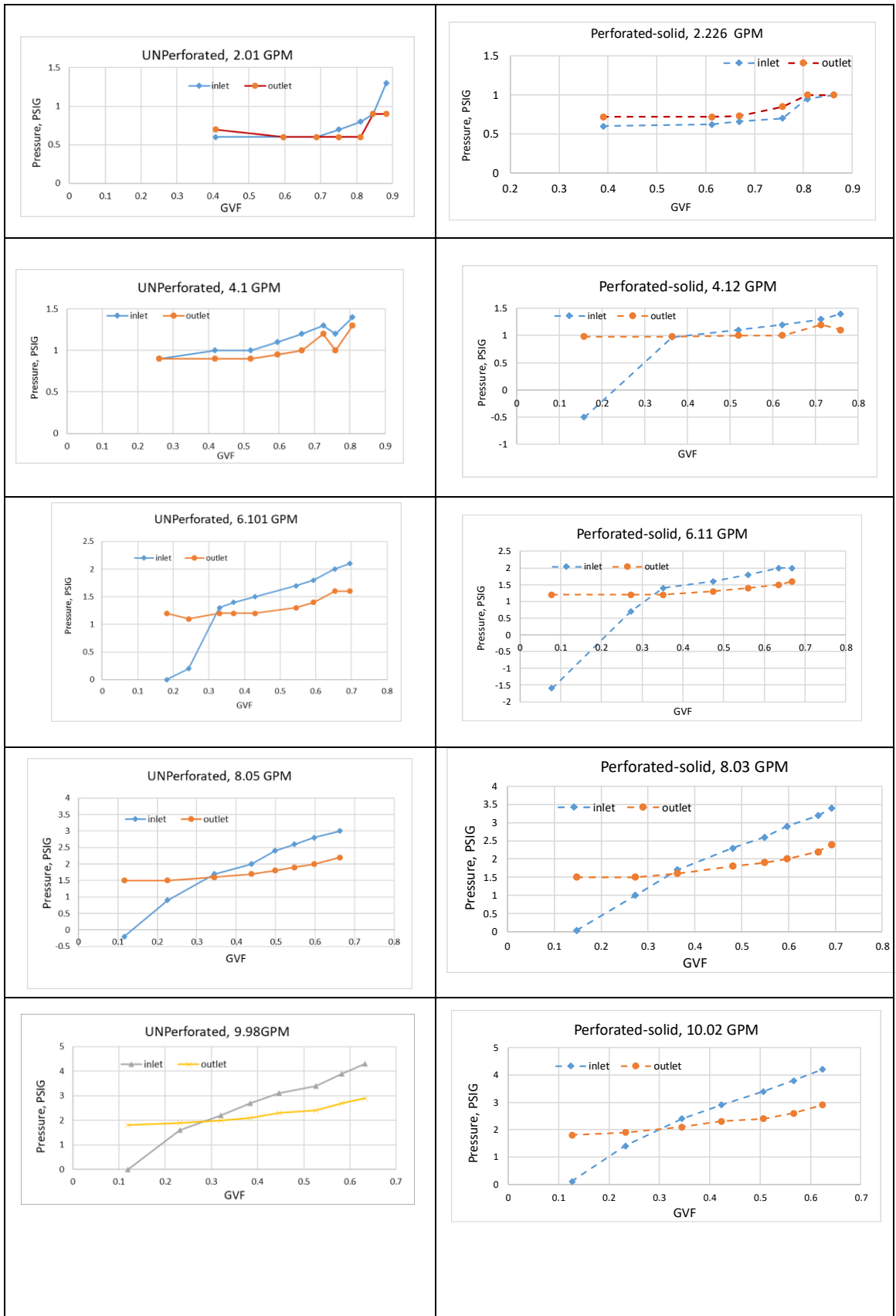


Figure 4.25: Variation of the inlet and outlet pressure with GVF of the unperforated and perforated two stage pump at constant range of total mixture flowrate

Figure 4.25 illustrates that the performance is better at low GVF and total flowrate.

b) Pump with the first stage perforated and the second unperforated

Figure 4.26 shows a comparison between an unperforated two-stage pump and the same pump with the first stage perforated. In these experiments and for the perforated case, the outer stage after the intake (the first) was the seven holes perforated stage and the next stage (the second) was the unperforated or solid impeller. It can be observed that at a low liquid flow rate and GVF the perforated pump performance is much better.



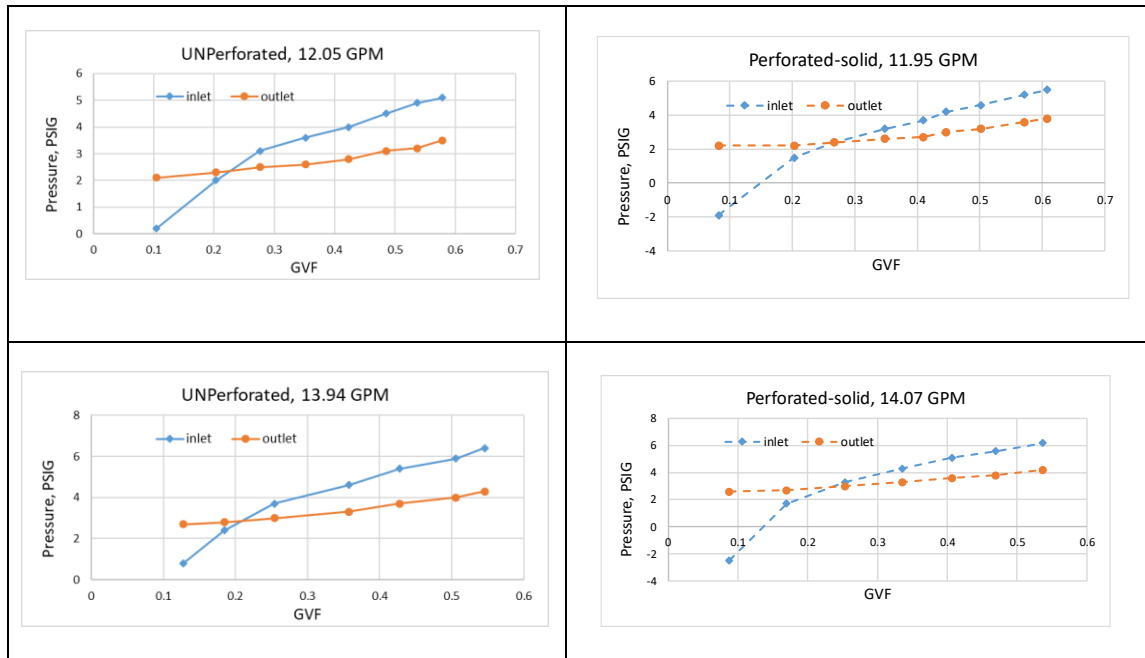
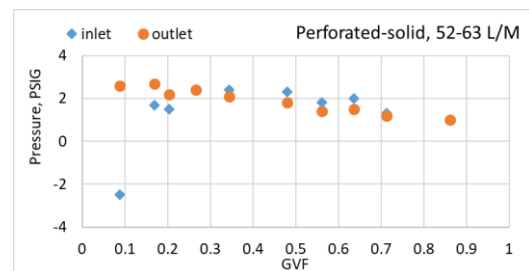
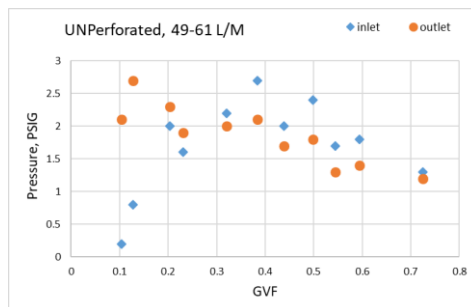
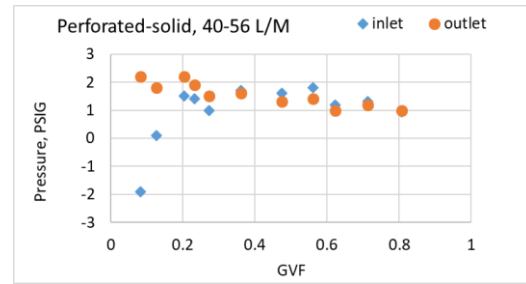
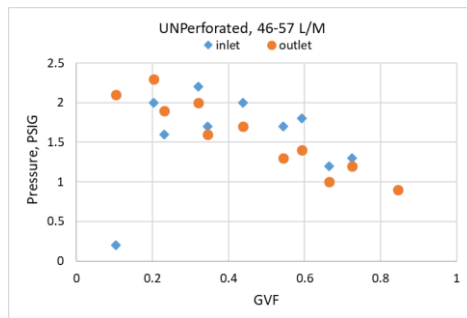
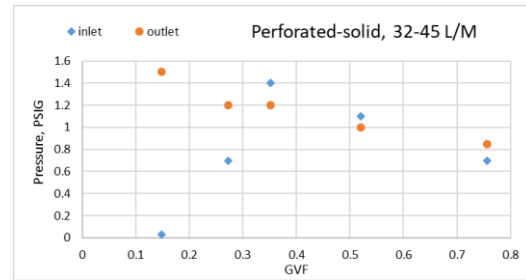
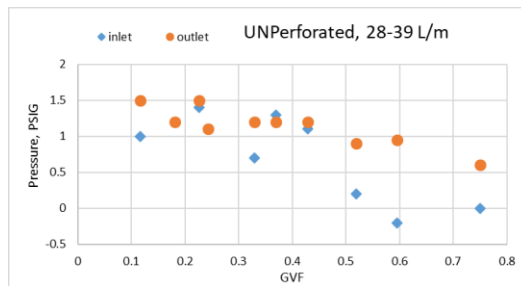
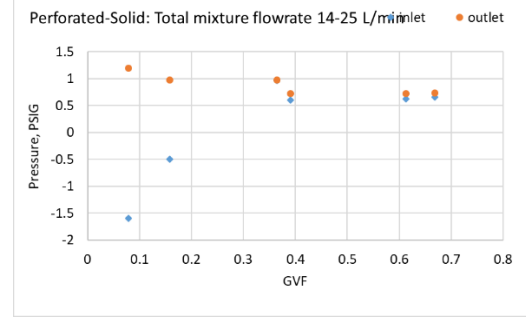
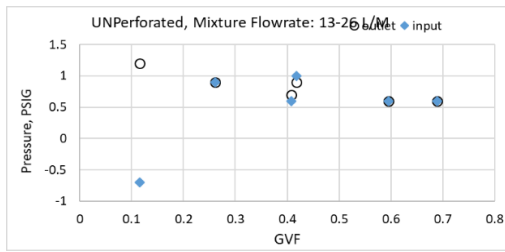


Figure 4.26: Variation of the inlet and outlet pressure with GVF for the unperforated case and the first stage perforated-second stage unperforated of the two-stage pump at a constant liquid flow rate

Figure 4.27 shows that the intake and discharge pressure is affected by the flow rate, GVF, and the situation of the stages (perforated-perforated, perforated-solid, or solid-solid). Thus, in the following figures, the pressure rise ($P_{\text{discharge}} - P_{\text{intake}}$) is presented versus the GVF. Figure 4.28 shows the best case at a liquid flow rate of 2 GPM. The pressure rise is positive up to $\text{GVF} = 0.8$ for the case of perforated impellers, while it is positive only up to $\text{GVF} = 0.6$ for the solid impeller.



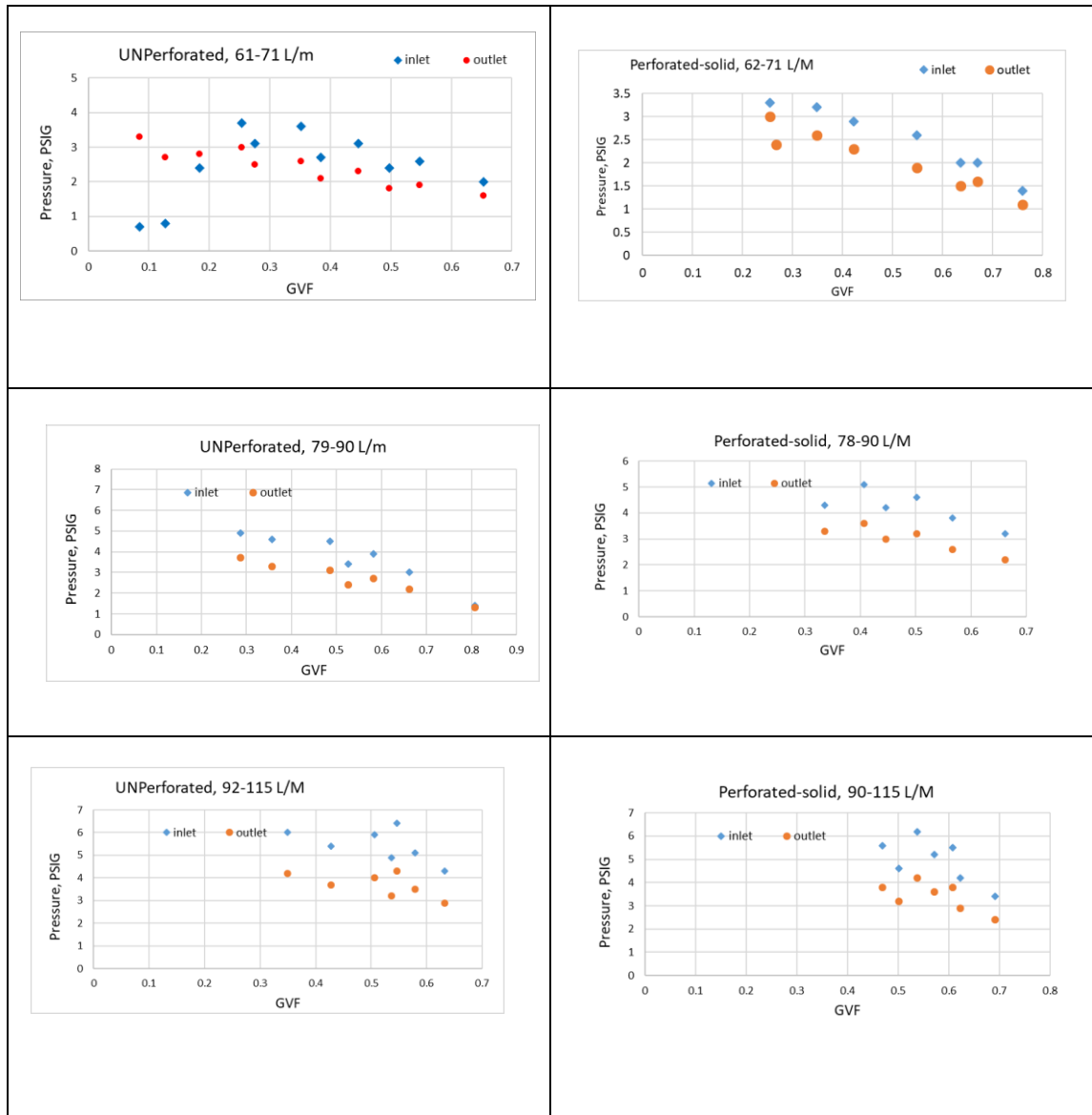


Figure 4.27: variation of the inlet and outlet pressure with GVF for the unperforated case and the first stage perforated-second stage unperforated of the two-stage pump at constant total mixture flowrate

It can be clearly seen that from Figure 4.27, at the constant total flow rate of the mixture, the performance of the perforated is better only at the lowest flowrate sector (14-25 L/m).

Figures 4.28-4.33 show the pressure rise across the homogenizer (pump 1) for the three different cases: the unperforated impeller, the first perforated, and the second stage unperforated, and both the first and the second stage are perforated. It can be seen that the perforated impeller performs better at low GVF and liquid flow rates.

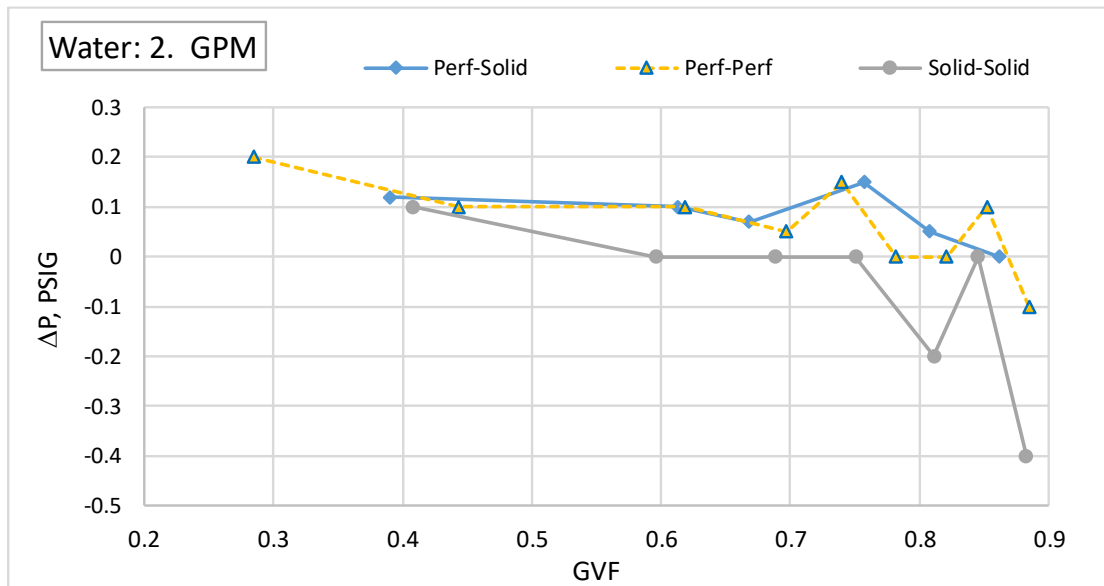


Figure 4.28: Variation of the pressure rise with GVF for different scenarios at the lowest flow rate 2 GPM

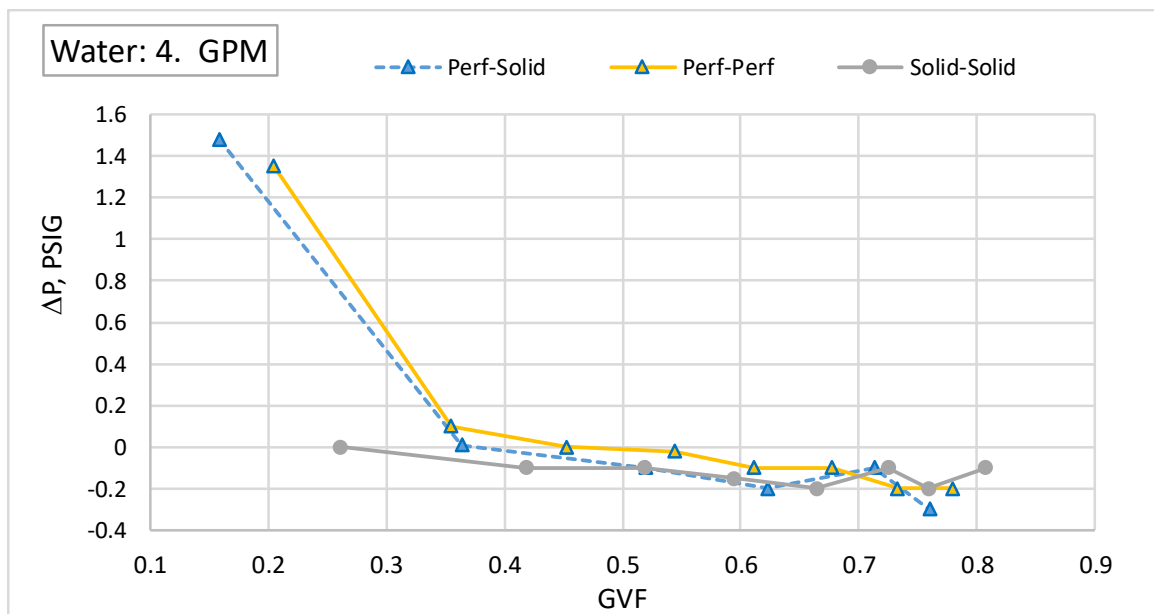


Figure 4.29: Variation of the pressure rise with GVF for different scenarios at flow rate 4 GPM

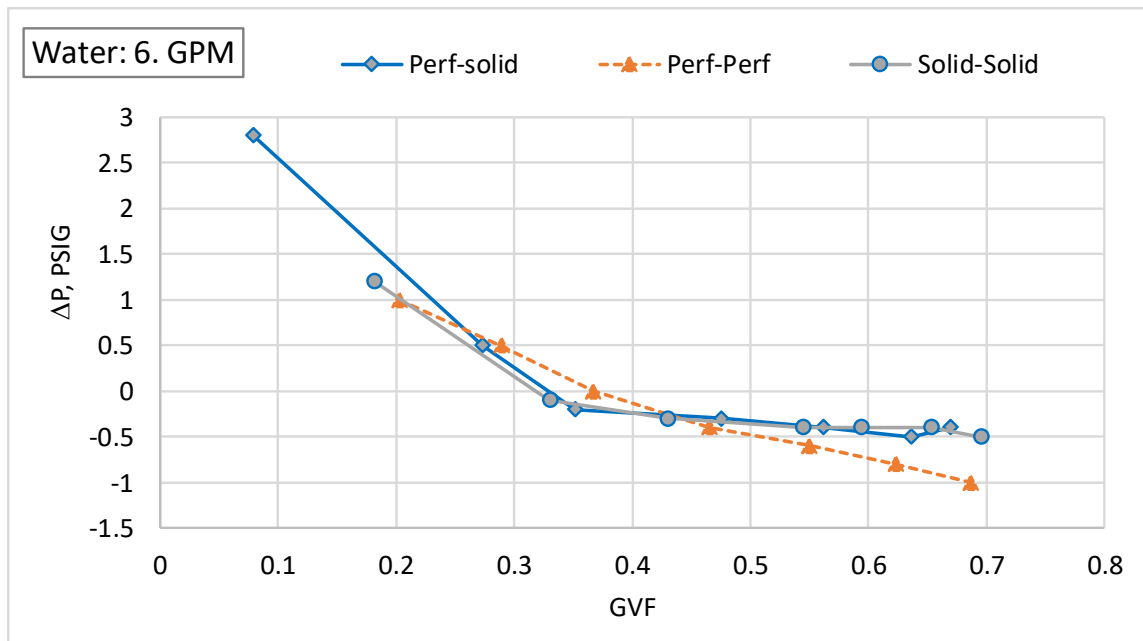


Figure 4.30: Variation of the pressure rise with GVF for different scenarios at flow rate 6 GPM

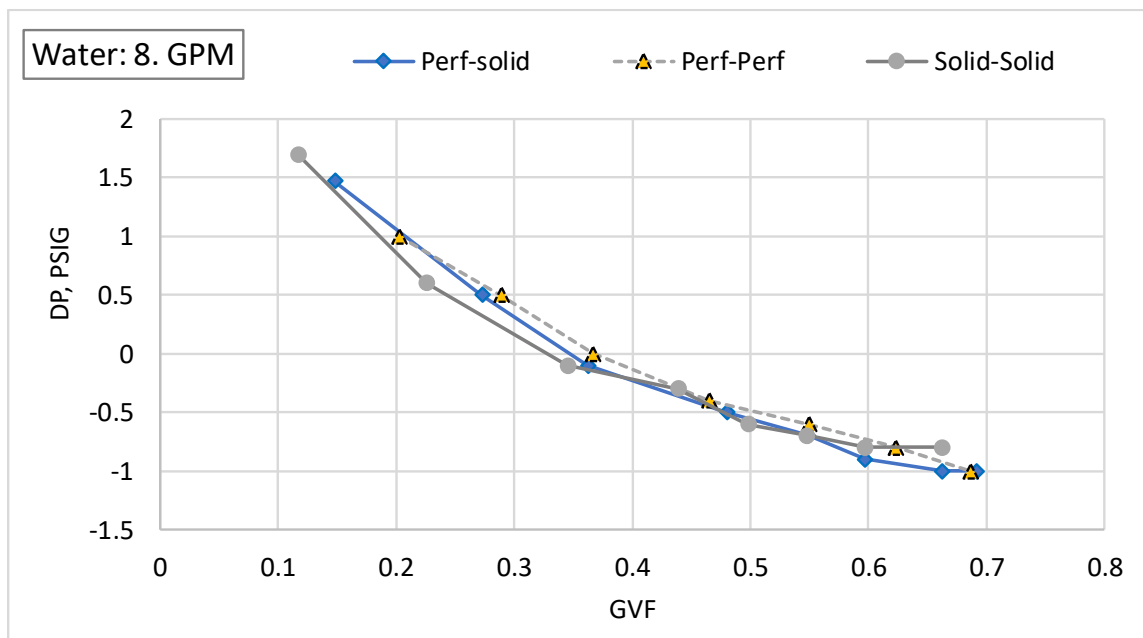


Figure 4.31: Variation of the pressure rise with GVF for different scenarios at flow rate 8 GPM

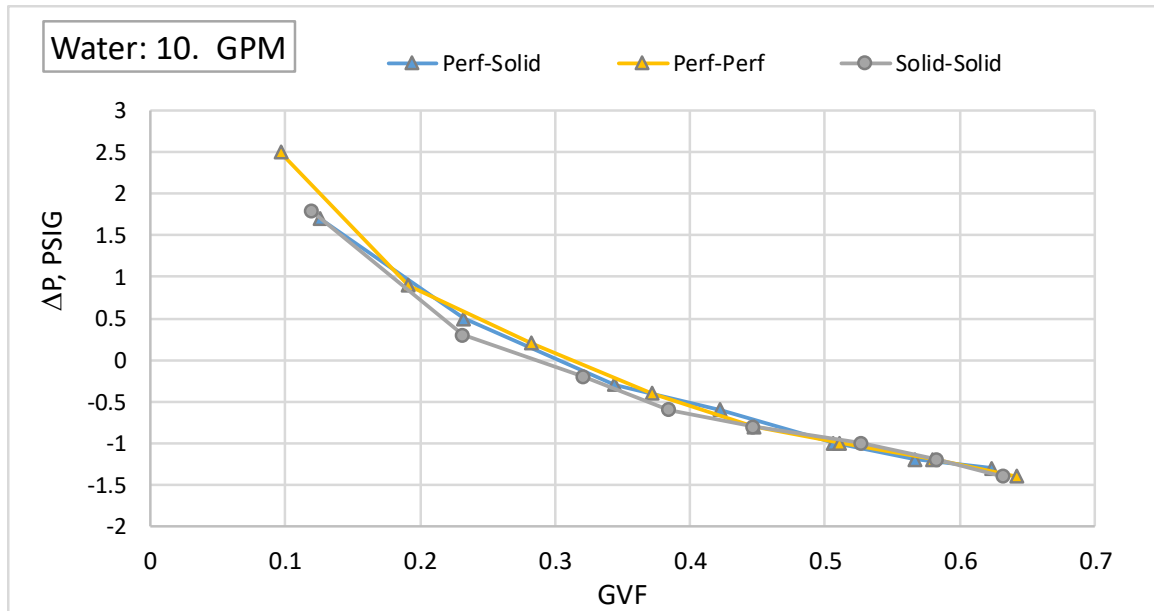


Figure 4.32: Variation of the pressure rise with GVF for different scenarios at flow rate 10 GPM

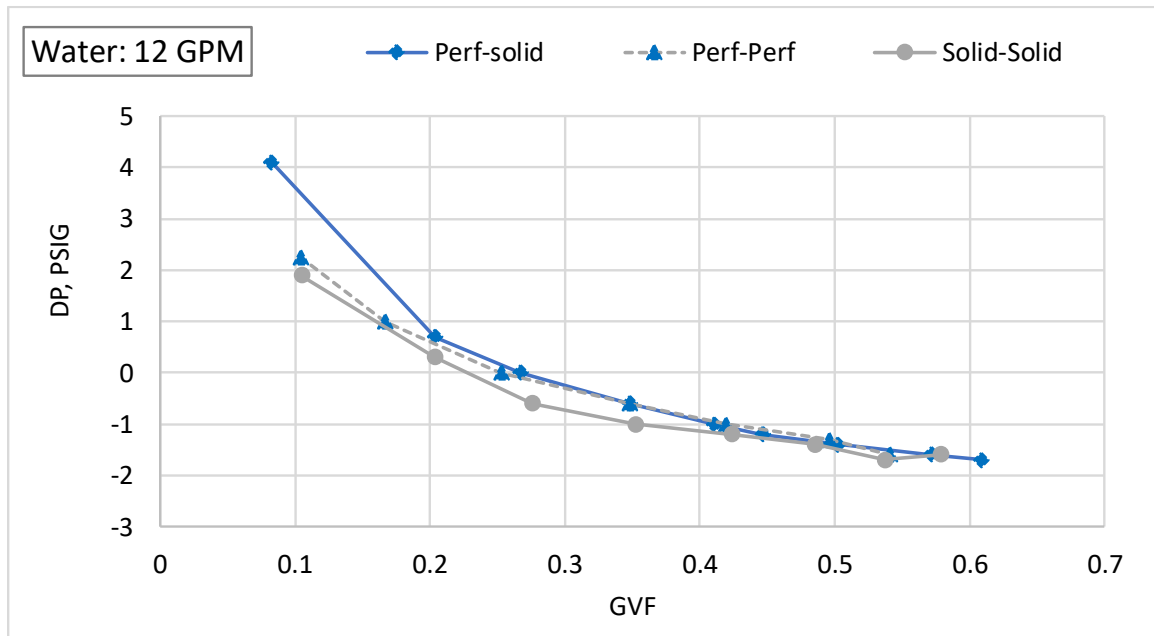


Figure 4.33: Variation of the pressure rise with GVF for different scenarios at flow rate 12 GPM

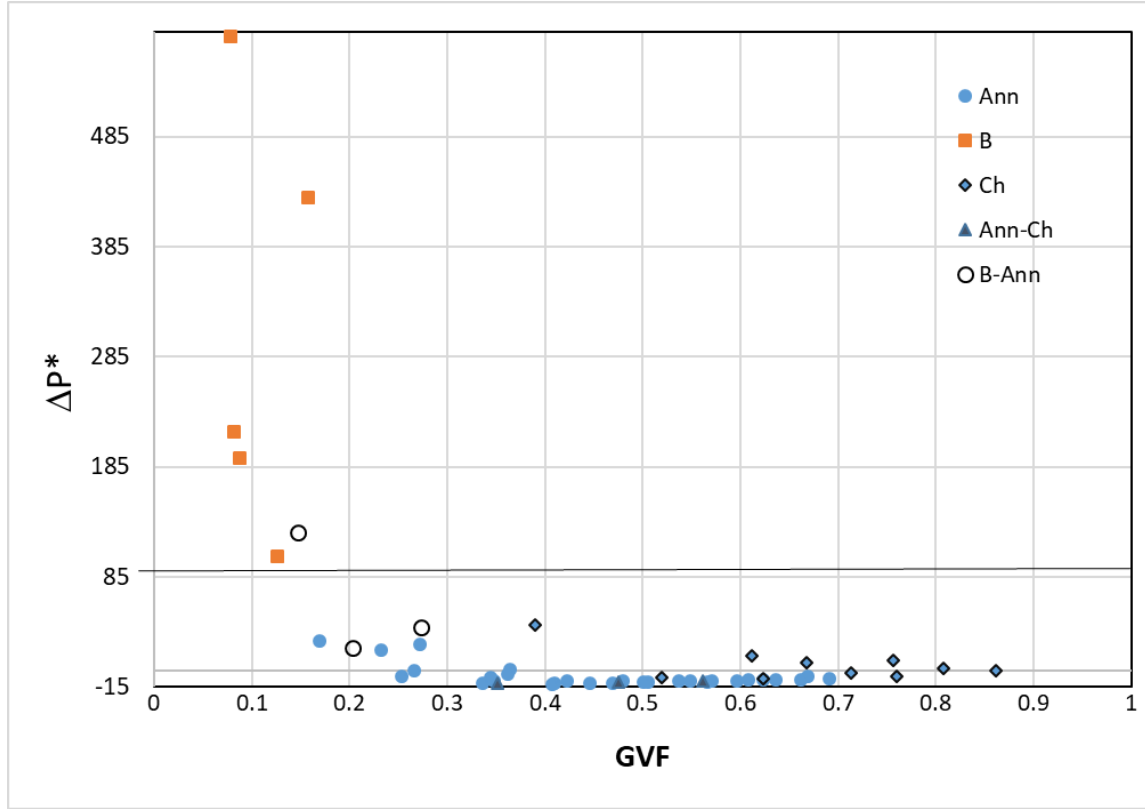


Figure 4.34: Dimensionless pressure rise versus GVF for homogenizer 1 (perforated first stage and unperforated second stage impeller) flow regimes are at the discharge, pressure difference across the homogenizer

An attempt to normalize all liquid flow rates in one figure is shown in figure 4.34. The pressure rise (or drop) is dimensionalized as

$$\Delta P^* = \frac{\Delta P}{\frac{1}{2} \rho_m V_m^2} \quad (4.1)$$

Where ρ_m is the inlet mixture density and V_m is the inlet mixture velocity ($V_{SL}+V_{SG}$).

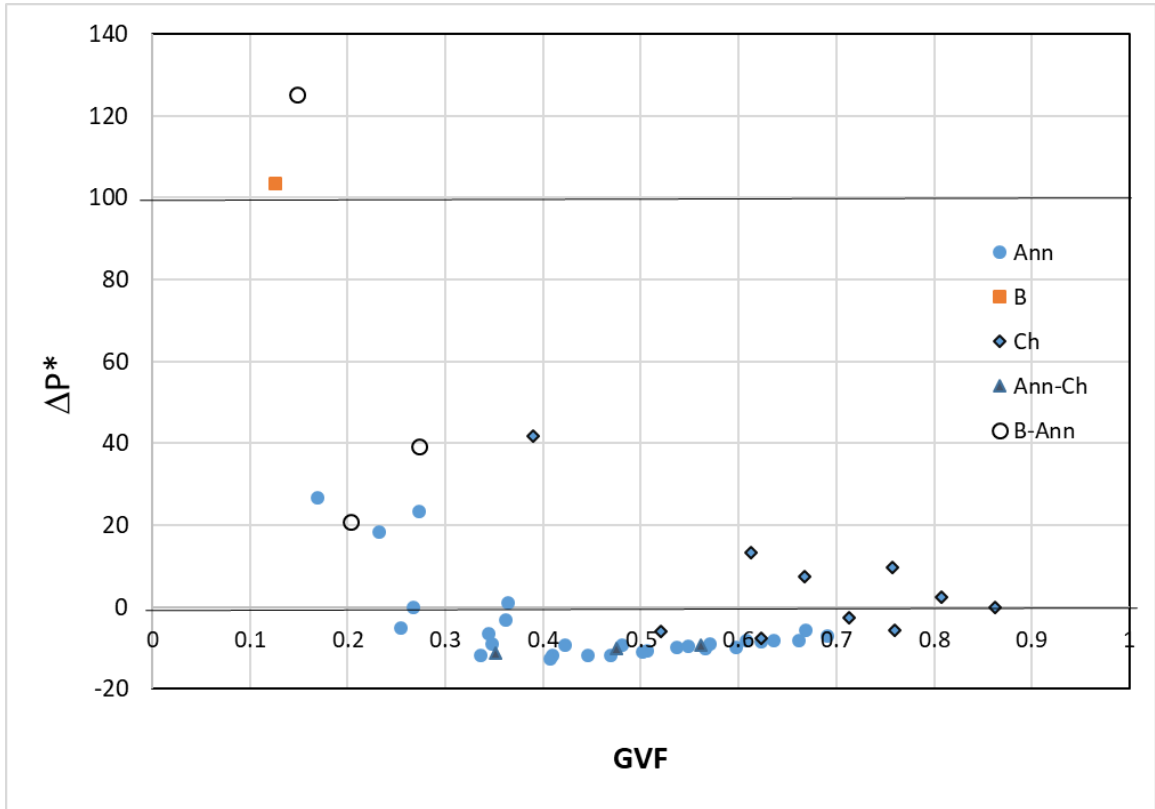


Figure 4.35: Dimensionless pressure rise versus GVF (zoom in scale) (perforated first stage and unperforated second stage impeller) pressure difference across the homogenizer

Figure 4.35 shows that for the dimensionless pressure above 100, the outlet flow pattern is bubbly while above zero, and below 100, the flow pattern is Churn or annular.

4.8 Experimental Validation of 4 stage Pump Performance

4.8.1 Single Phase Flow experiments

Single-phase flow experiments were conducted to validate the instrumentations and experimentally determine the main 4-stage pump's H-Q curve without a homogenizer. In addition, one more test with the first stage perforated is tested. The test facility shown in Figure 3.6 was used for these experiments with the same flowmeter, and pressure gauge as in the two-stage pump (homogenizer #1) flow loop explained previously. The impeller in this pump consists of 3 vanes (in the two-stage pump homogenizer, one impeller has four vanes). The first stage was perforated with nine holes of 1 mm diameter in staggered arrangement and eight holes of 2 mm diameter, as shown in figure 4.36. It is worth mentioning here that the pump suction port is not centered and located above the pump's axis, as shown in figure 4.36. Figure 4.37 shows the H-Q curve of the four-stage pump used in the present experiments obtained from the pump manual (empty squared legend) and the experimental data. The empty circle is the experimental data for the unperforated impeller, while the filled circles are for the pump, with the first stage being perforated. The H-Q results show that the perforation influence when running single-phase flow is negligible and the head developed is almost the same with and without perforation (within the uncertainty of the measurements). This can be attributed to the fact that the perforation is only in one stage out of four compared to the results in the previous section of the two-stage pump (homogenizer 1). Moreover, the test was repeated by installing the perforated stage in the second stage instead of the first stage and obtained the same results.

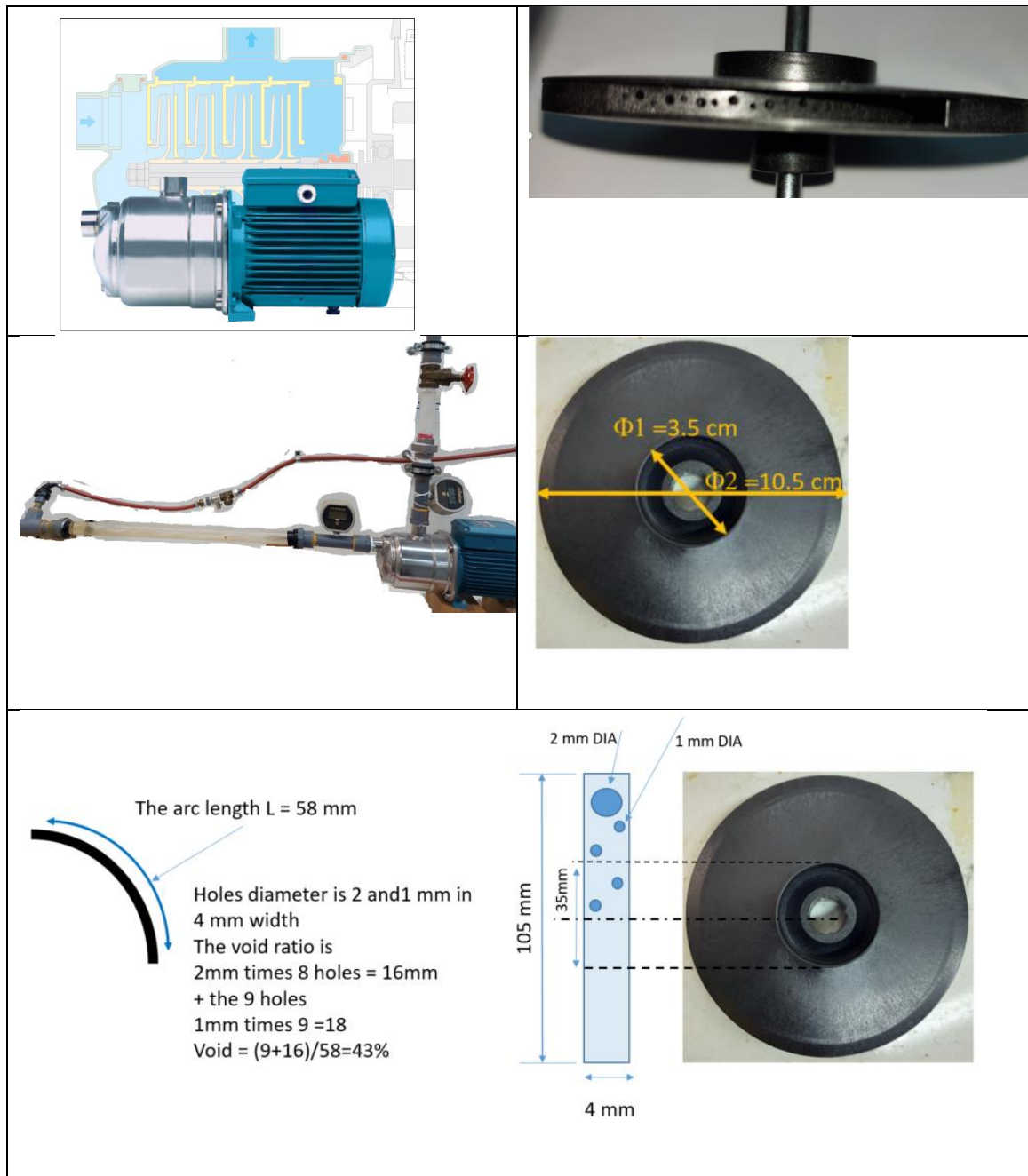


Figure 4.36: Four stages Pump and perforated impeller stage details

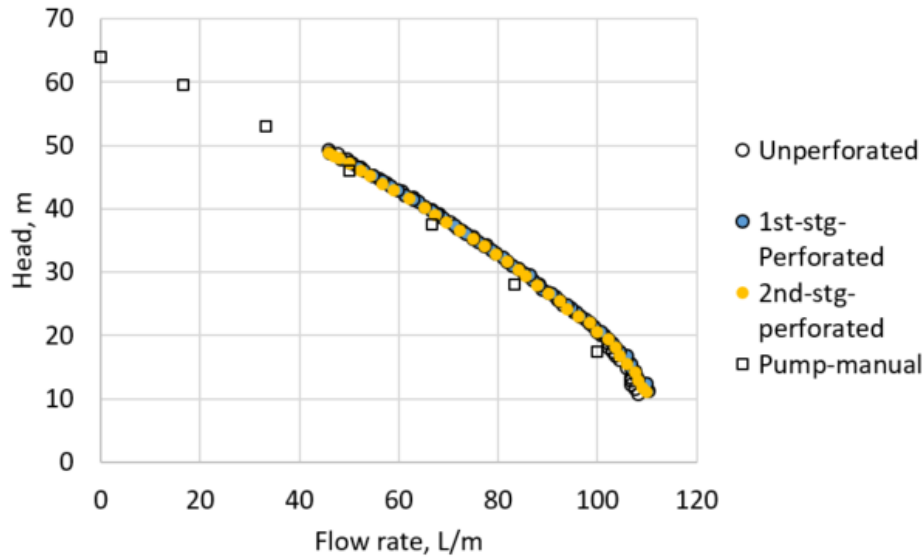


Figure 4.37: H-Q curves for the 4-stage pump with and without perforated stage

4.8.2 Two-Phase Flow experiments:

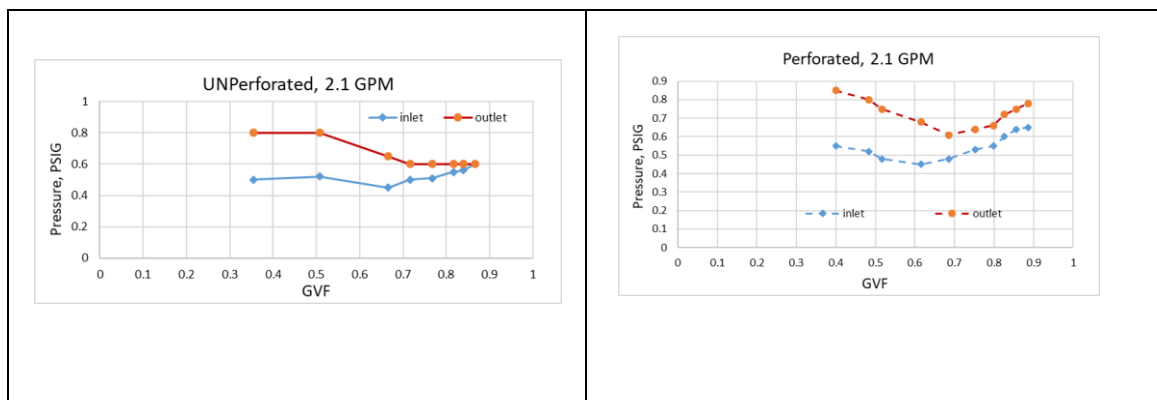
The two-phase flow experiments were conducted using the rig shown in figure 3.7, in which a booster pump provides the required water flow rate by adjusting the valve downstream of the booster pump. These experiments were conducted by fixing the inlet liquid flow rate and increasing the inlet gas flow rate up to the maximum GVF. The pressure at the pump's intake and discharge were recorded in addition to the inlet liquid and gas flow rates. The flow pattern at the discharge was also reported. The flow pattern at the intake was mainly slug, and some cases stratified.

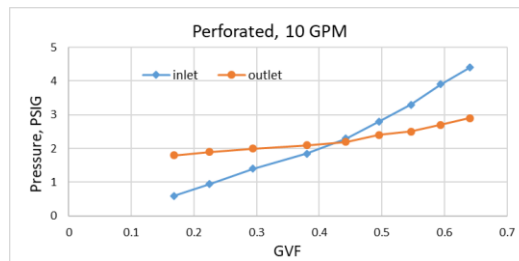
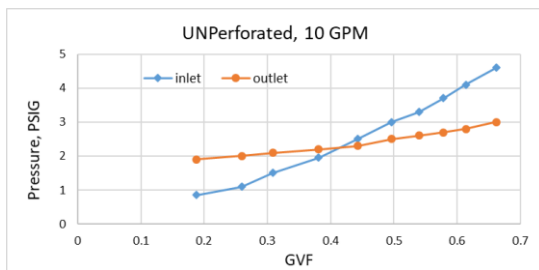
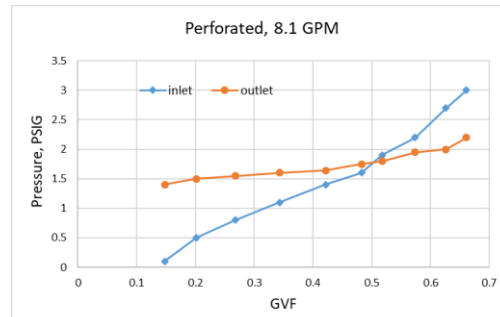
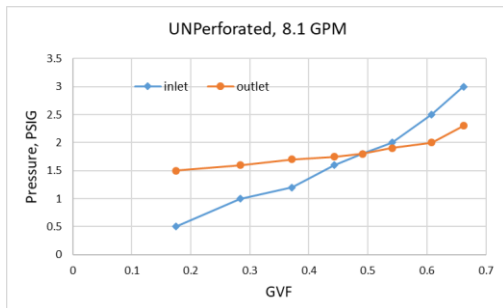
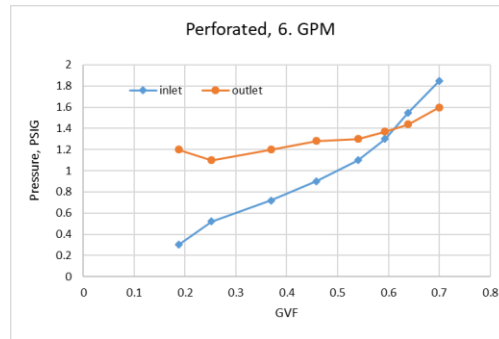
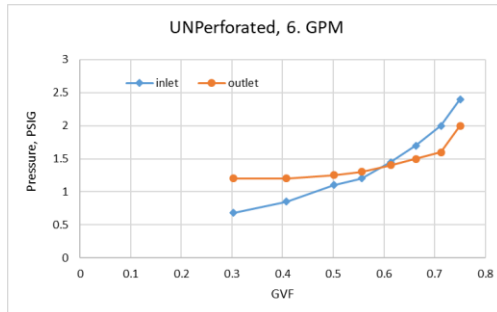
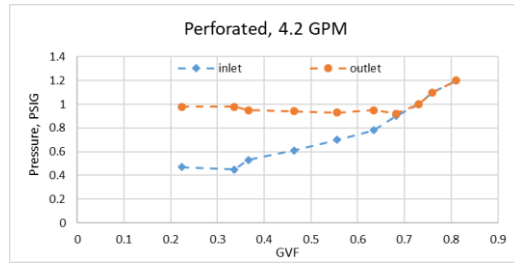
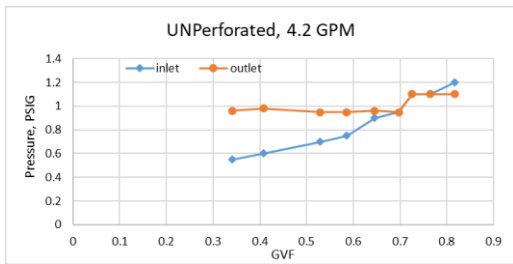
a) Four stage pump with and without the perforated first stage of the impellers at low and high inlet air pressure

(Constant inlet liquid and variable gas flowrates)

The pump was tested under two-phase flow conditions for both cases with no modifications to its first impeller stage and with the first impeller stage being perforated. Two different sets of experiments were conducted, one at low and the other at high inlet air pressure. The air pressure for the low-pressure experiments was varied (depends on the liquid and gas flow rate) from 0.2 to 1 bar with an average value for all runs of 0.44 bar, while the one at high-pressure experiments, the inlet air pressure was from 0.38 to 3 bar with an average value for all runs of 1.377 bar. The system pressure improves the performance of the pump under two-phase flow conditions. Thus, the suction and discharge pressure distribution are presented as well as the increase in the pressure (∇P) across the pump at all GVF. The lowest liquid flow rate was 2 GPM, and the highest liquid flow rate in the case of two-phase flow was 12 GPM.

Figure 4.38 shows the two-phase flow results at low-pressure experiments. It can be observed that perforation of the first stage improves the performance at the lowest liquid flow rate, while small improvements can be seen at a high liquid flow rate





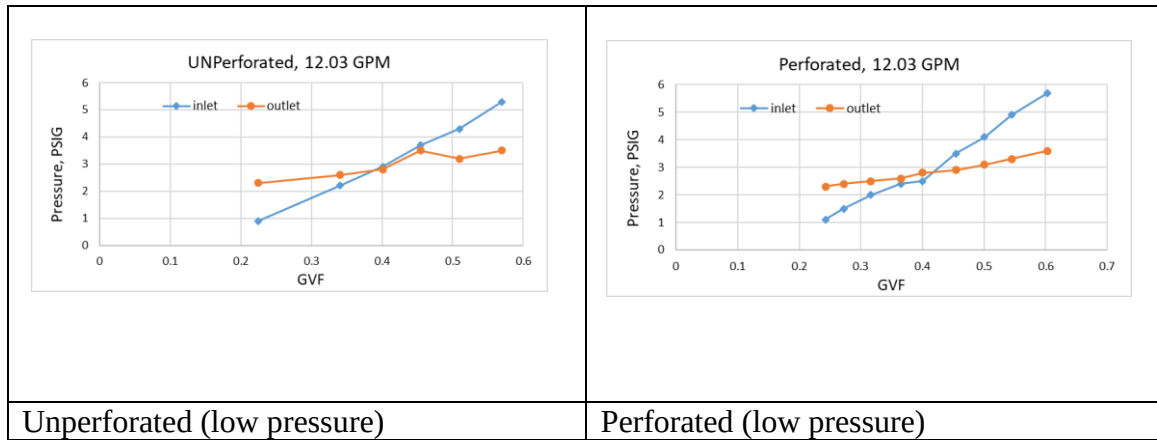
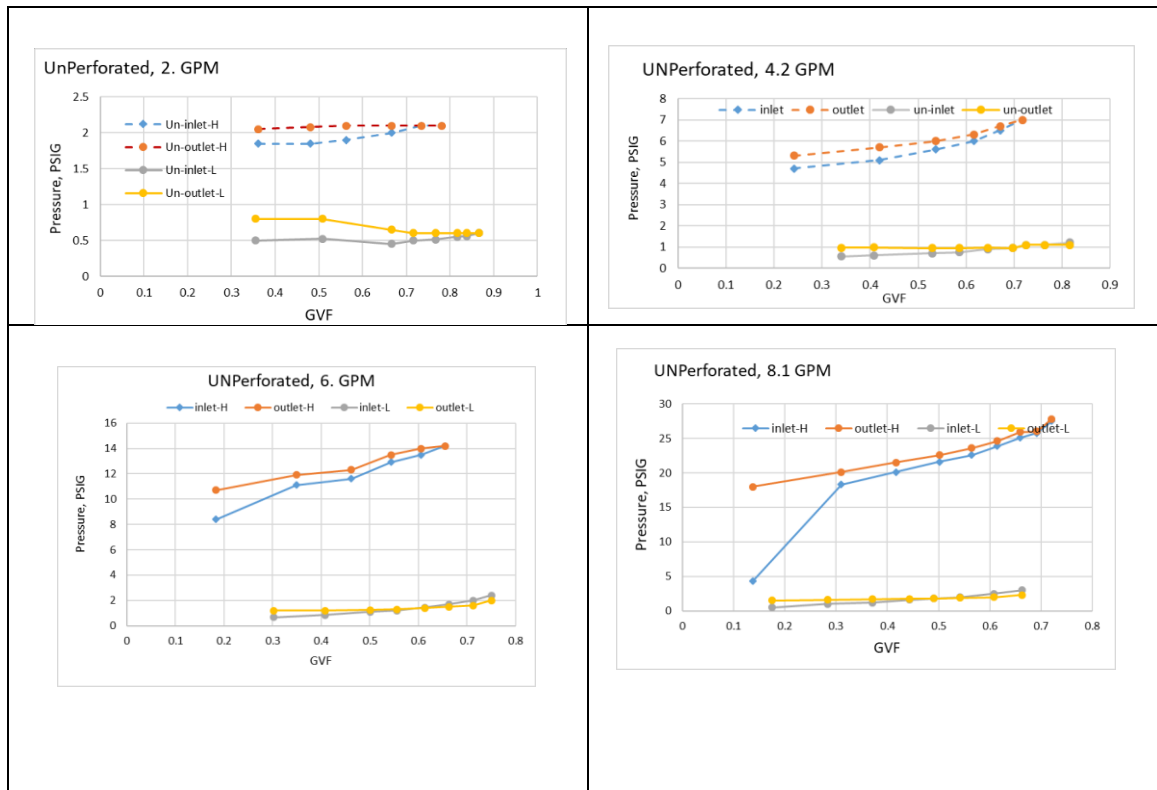


Figure 4.38: Suction and discharge pressure distribution with GVF for unperforated and first stage perforated pump at low system pressure

Figure 4.39 shows the two-phase flow results at high-pressure experiments for the unperforated first stage compared to the low-pressure experiments. It can be noticed that the performance gets worse as the GVF increases especially at high liquid flow rates. Increasing the system pressure improves the pump performance under two-phase flow conditions at all GVF and liquid flow rates.



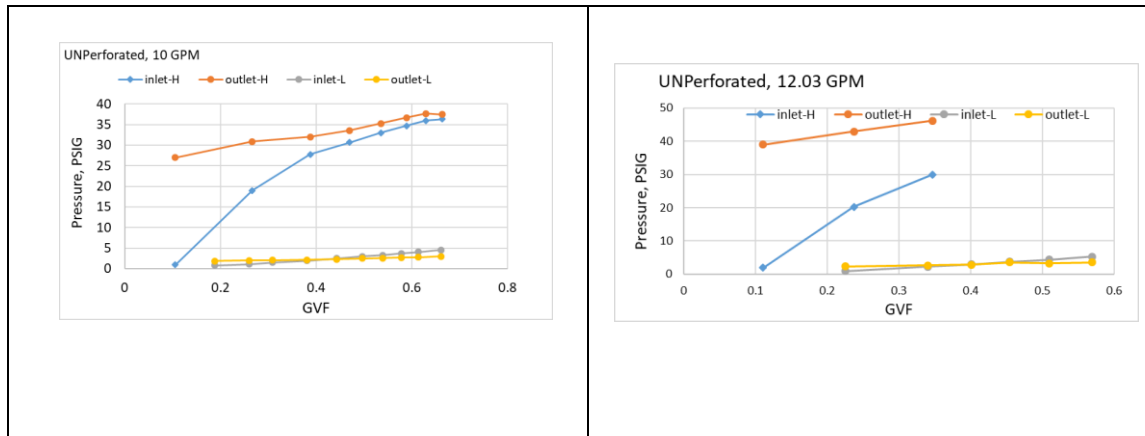
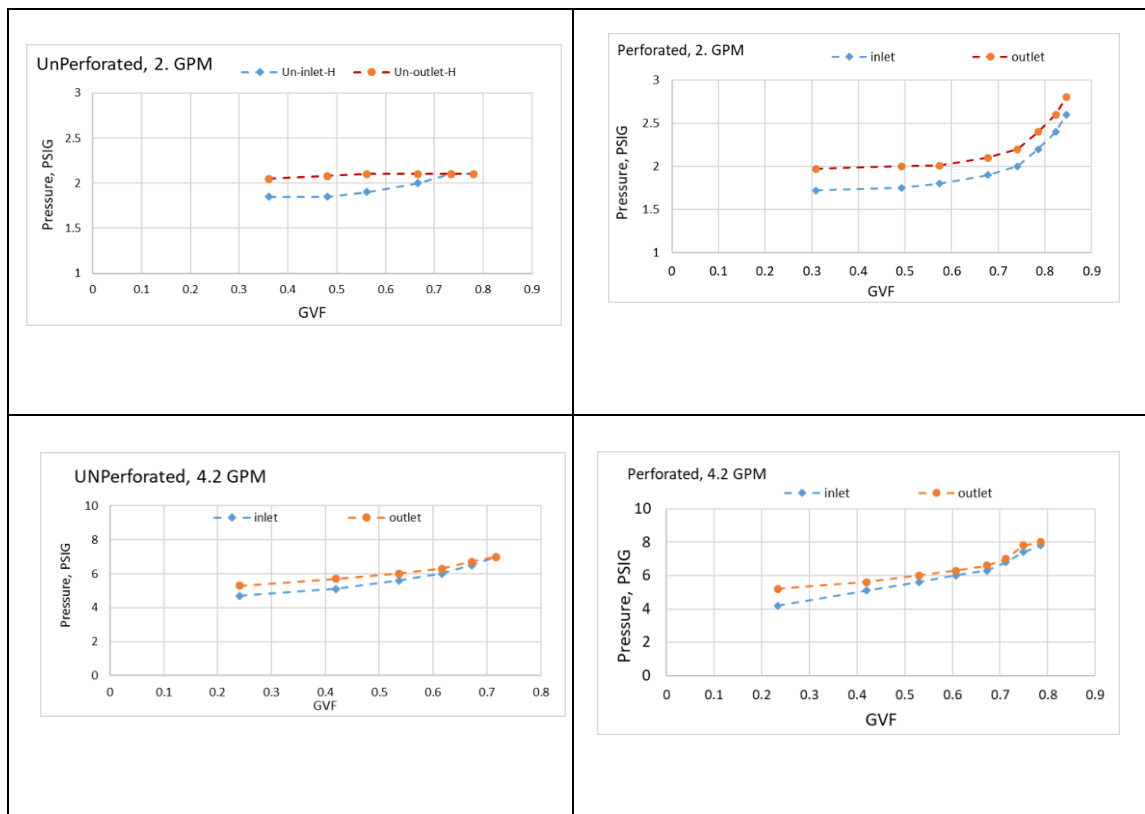


Figure 4.39: Suction and discharge pressure variation with void fraction at low and high system pressure (unperforated impeller)

Figure 4.40 shows a comparison between the perforated first stage and unperforated first stage performance of the 4-stage pump at the high-pressure case. The perforated stage performs better only at the low liquid flow rate only. At a high liquid flow rate, the high-pressure factor dominates the perforation influence, and both perform about the same.



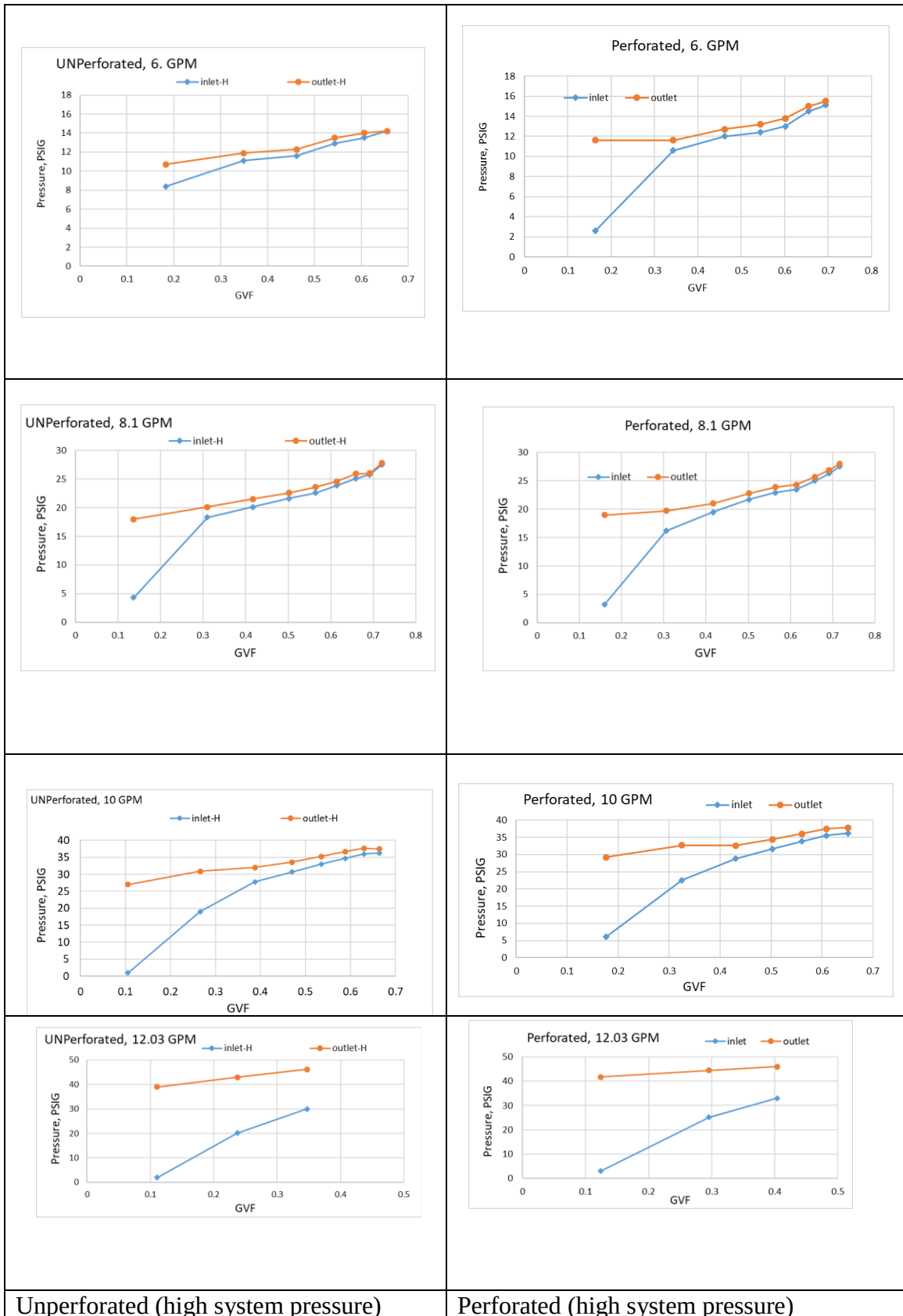
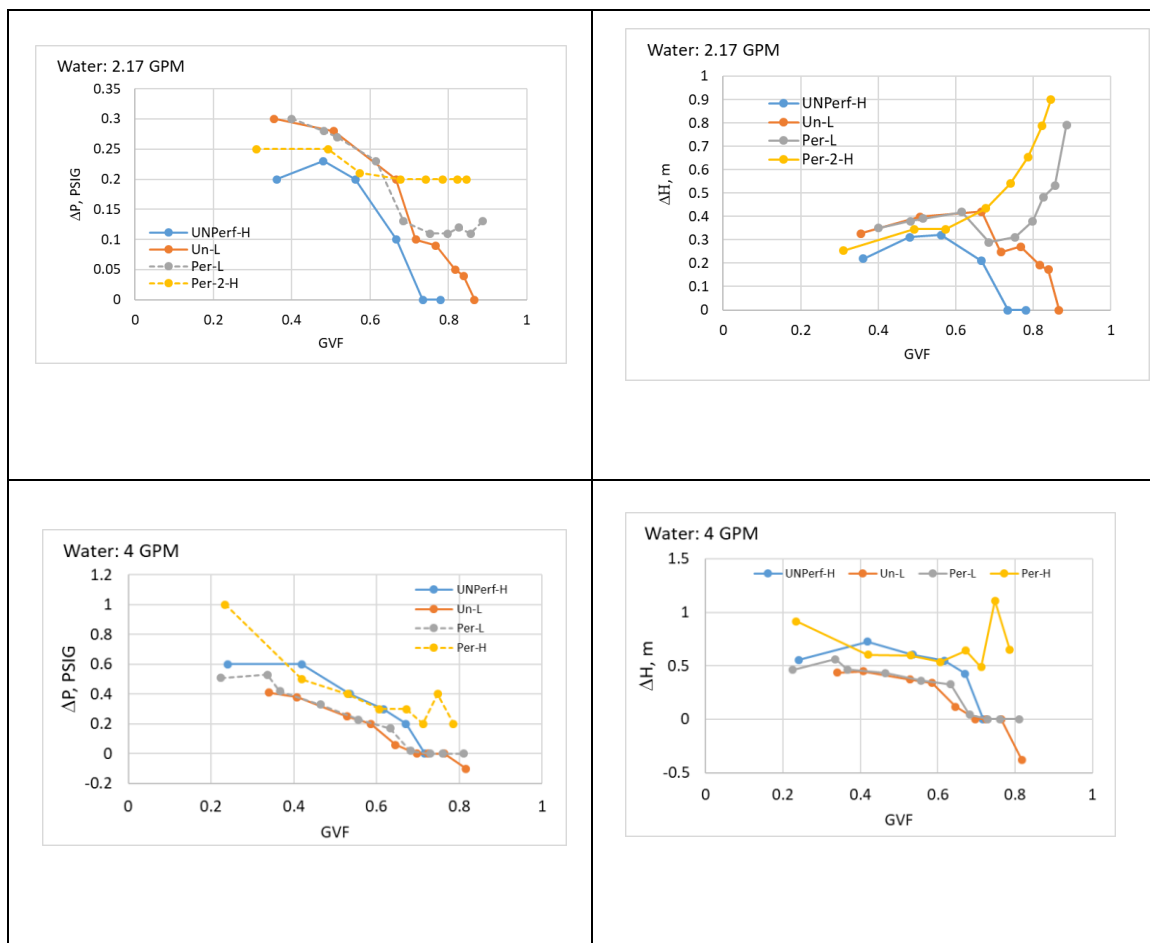


Figure 4.40: Comparison between unperforated pump and pump with perforated first stage at high pressure

Figure 4.41 shows the variation of the pressure increase across the 4-stage pump for low and high-pressure conditions for both cases. The first case; the first stage being perforated, and the second case first stage being unperforated.

It can be concluded that the perforation of the first stage improves the performance only at a low liquid flow rate. The high-pressure case performance is much better than the low-pressure cases. At the high pressure and high liquid flow rate, perforated or unperforated first stage perform almost the same.



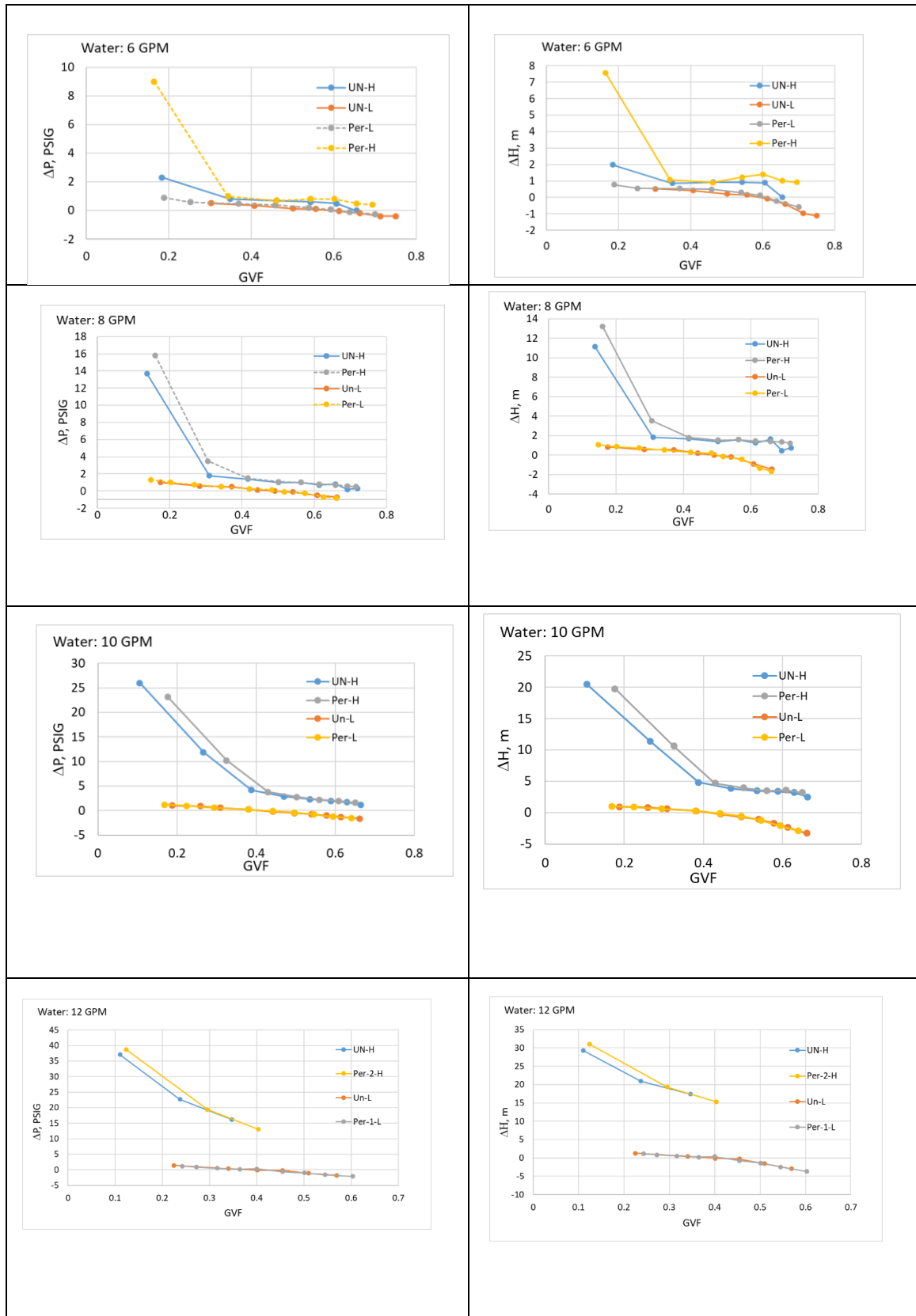


Figure 4.41: Pressure rise and head variations across the pump with void fraction at low and high system pressure

b) Homogenizer followed by 4-stage pump arrangement (constant inlet pressure)

In the following set of experiments, the inlet pressure upstream of the homogenizer was fixed by changing the liquid and gas flow rates. Then the pressure difference across the 4-stage pump was recorded in addition to the gas and liquid flow rate and flow pattern upstream and downstream of the 4-stage pump.

The inlet pressure was fixed at 1, 2, 3, and 4 psig upstream of the homogenizer. The pressure and flow pattern at suction and discharge of the 4-stage pump were recorded. Two different homogenizers were tested. Homogenizer #1 is the previously explained 2-stage perforated vanes of radial type impeller pump (as shown in figure 4.42), and homogenizer #2 is a single-stage open impeller (two vanes) with extensions as shown in figure 4.43



Figure 4.42: Homogenizer#1 details.

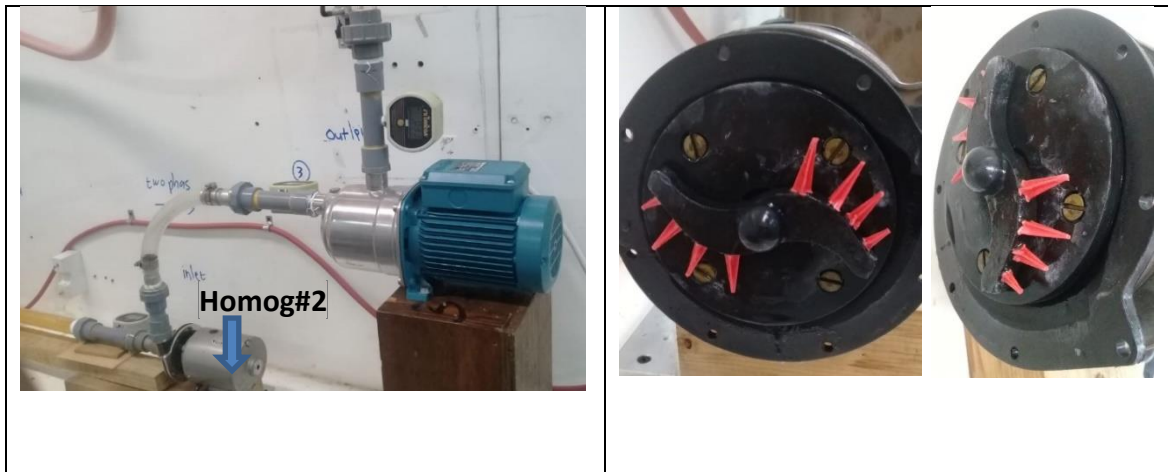


Figure 4.43: Homogenizer#2 details

Figures 4.44 A, B, C and D show the suction and discharge pressure of the 4-stage pump as a function of the GVF at different inlet pressure upstream of Homogenizer #1.

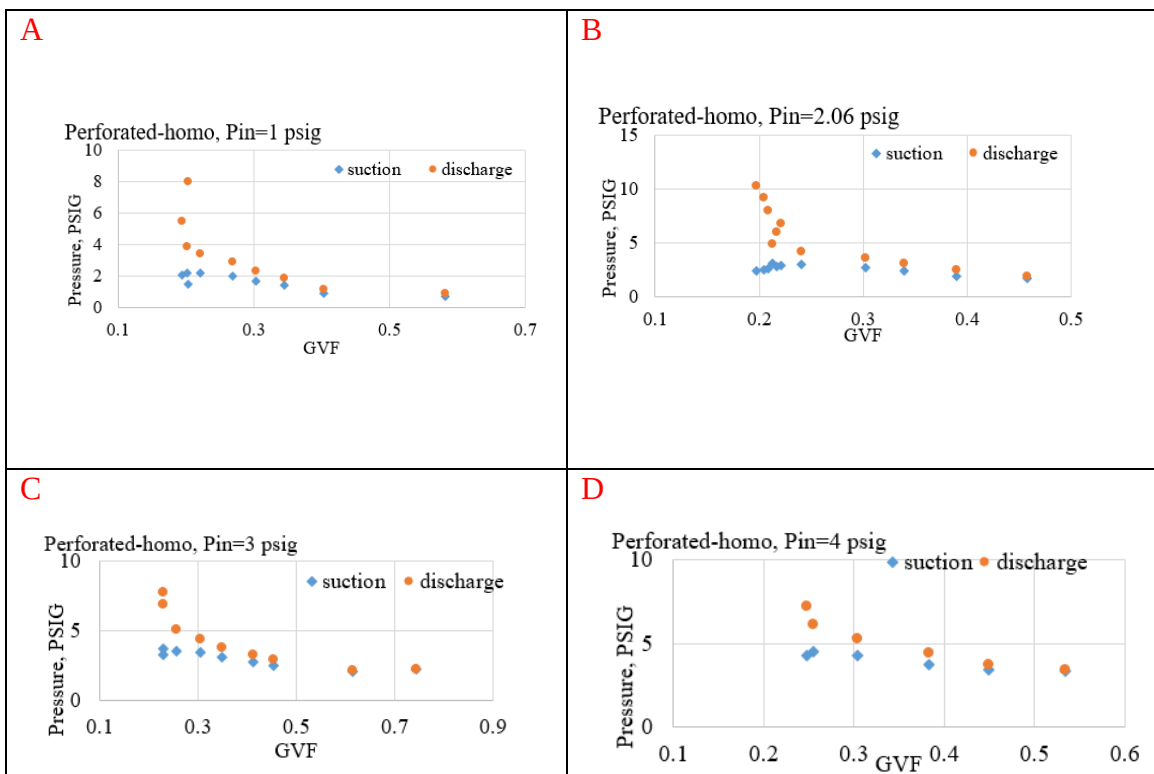


Figure 4.44: Suction and discharge pressure variation with GVF of 4-stage pump downstream of homogenizer#1

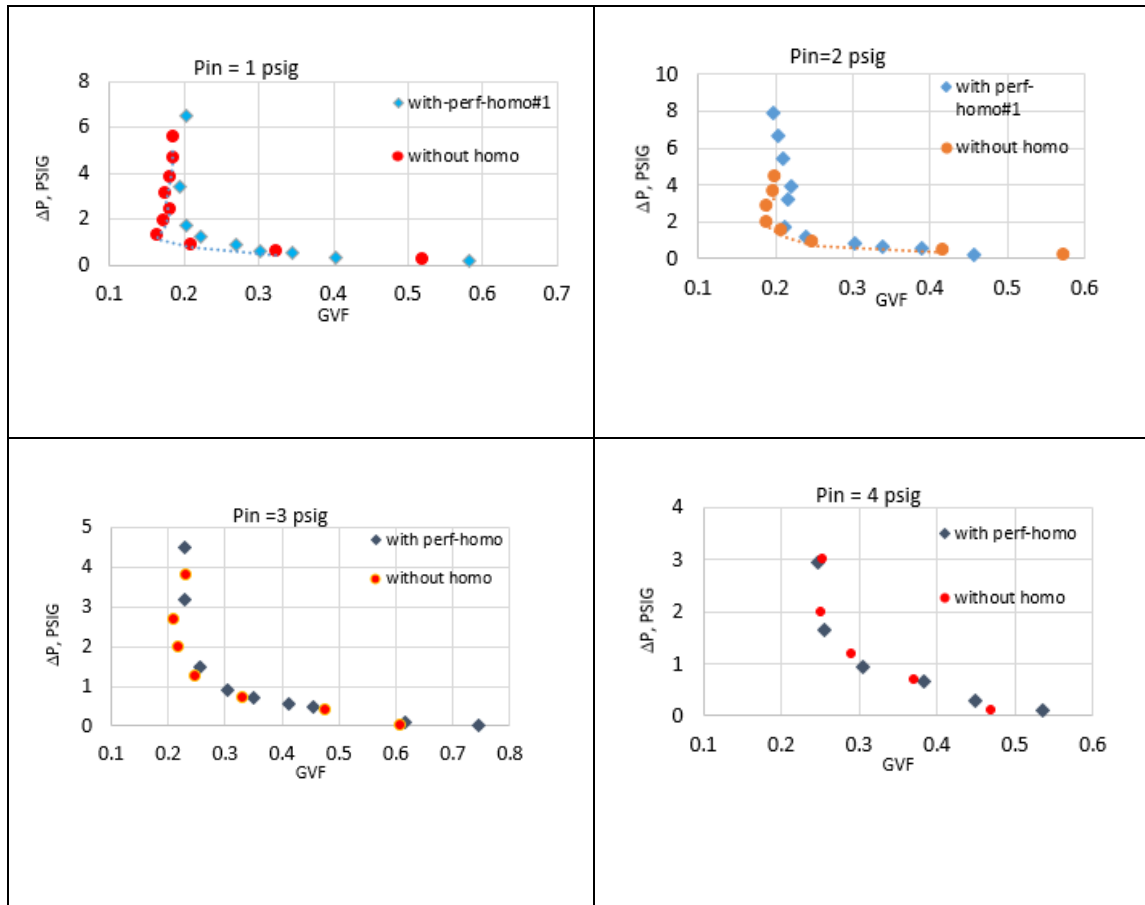


Figure 4.45: Pressure difference across the 4-stage pump downstream of homogenizer#1

At low GVF, the performance of the 4-stage pump when it is installed downstream of the homogenizer 1 is better than when it was alone without any homogenizer at the same inlet pressure.

Figures 4.46 A, B, C and D show the suction and discharge pressure of the 4-stage pump as a function of the GVF at different inlet pressure upstream of Homogenizer #2 (the single-stage open impeller with vanes extensions).

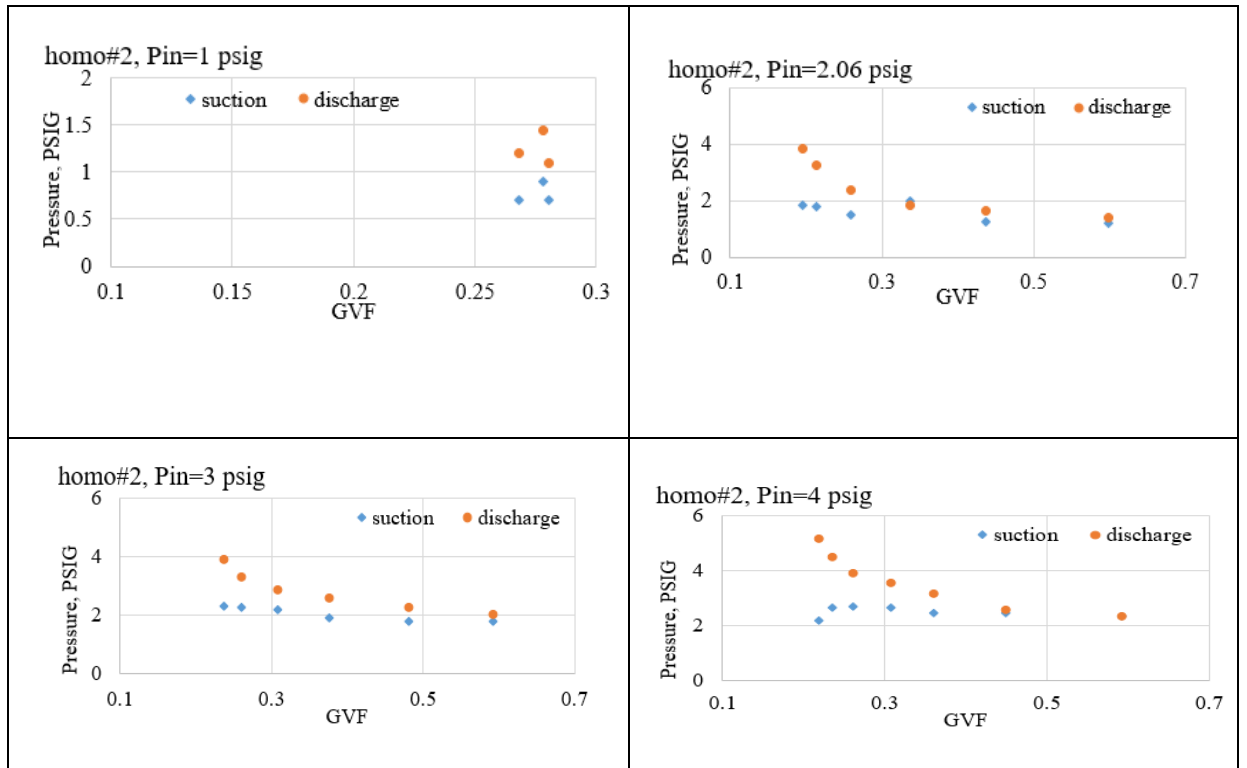


Figure 4.46: Pressure difference across the 4-stage pump downstream of homogenizer#2

In figure 4.46 and at an inlet pressure of 1 psig, the pressure fluctuation level was very high thus; a high GVF data could not be taken.

Figure 4.47 showed a comparison of the pressure difference across the 4-stage pump when it was located downstream of homogenizer 1 (right figure) and homogenizer#2 (left figure). It can be seen that the performance of the 4-stage pump when it sets downstream of the homogenizer is better only at low GVF but as the GVF increases, the type of homogenizer does not matter.

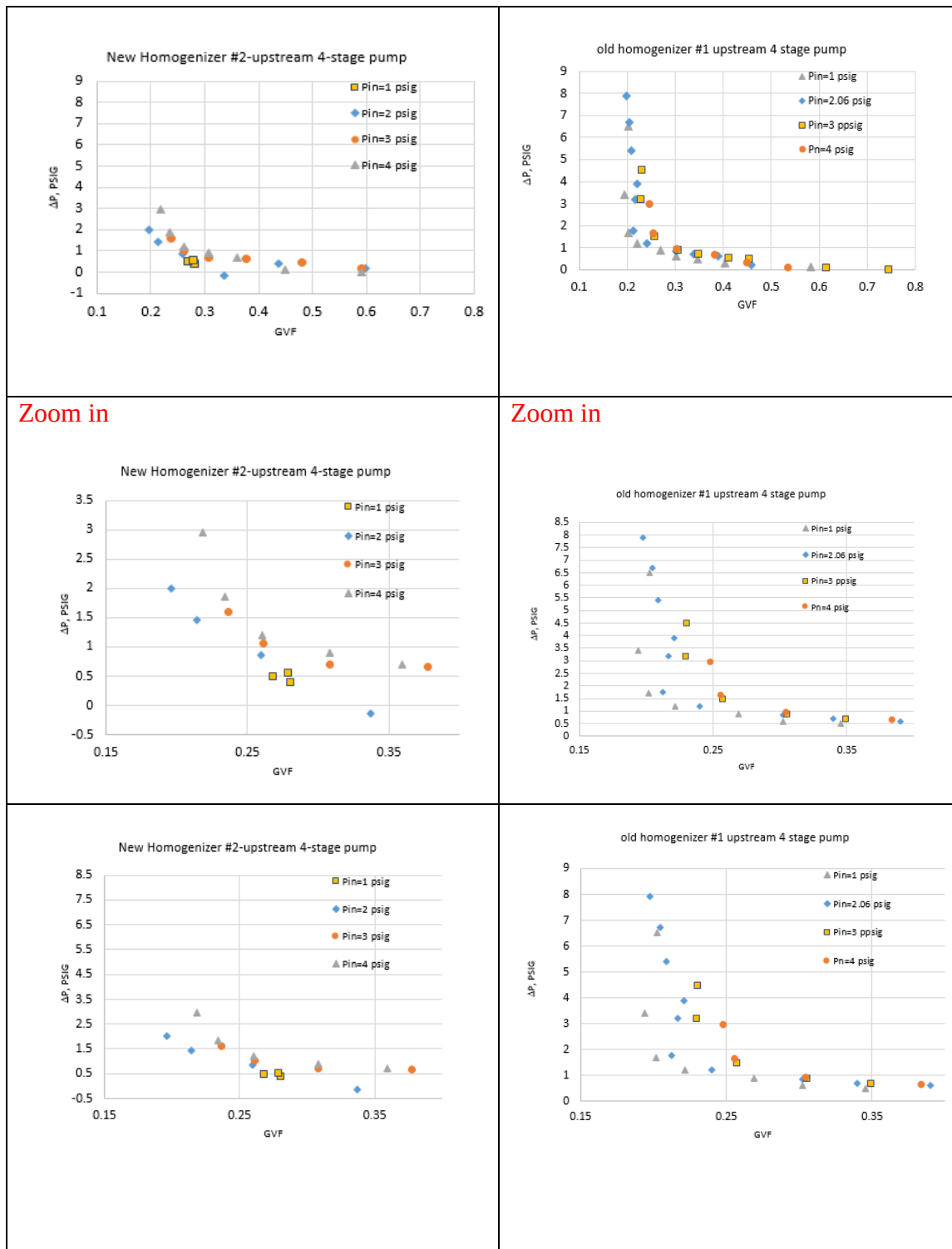


Figure 4.47: Comparison of pressure difference across the 4-stage pump downstream of homogenizer#1 and #2

Figure 4.48 shows the 4-stage pump performance when installed downstream of the old homogenizer (homogenizer#1), and the new homogenizer (homogenizer #2) at constant inlet pressure.

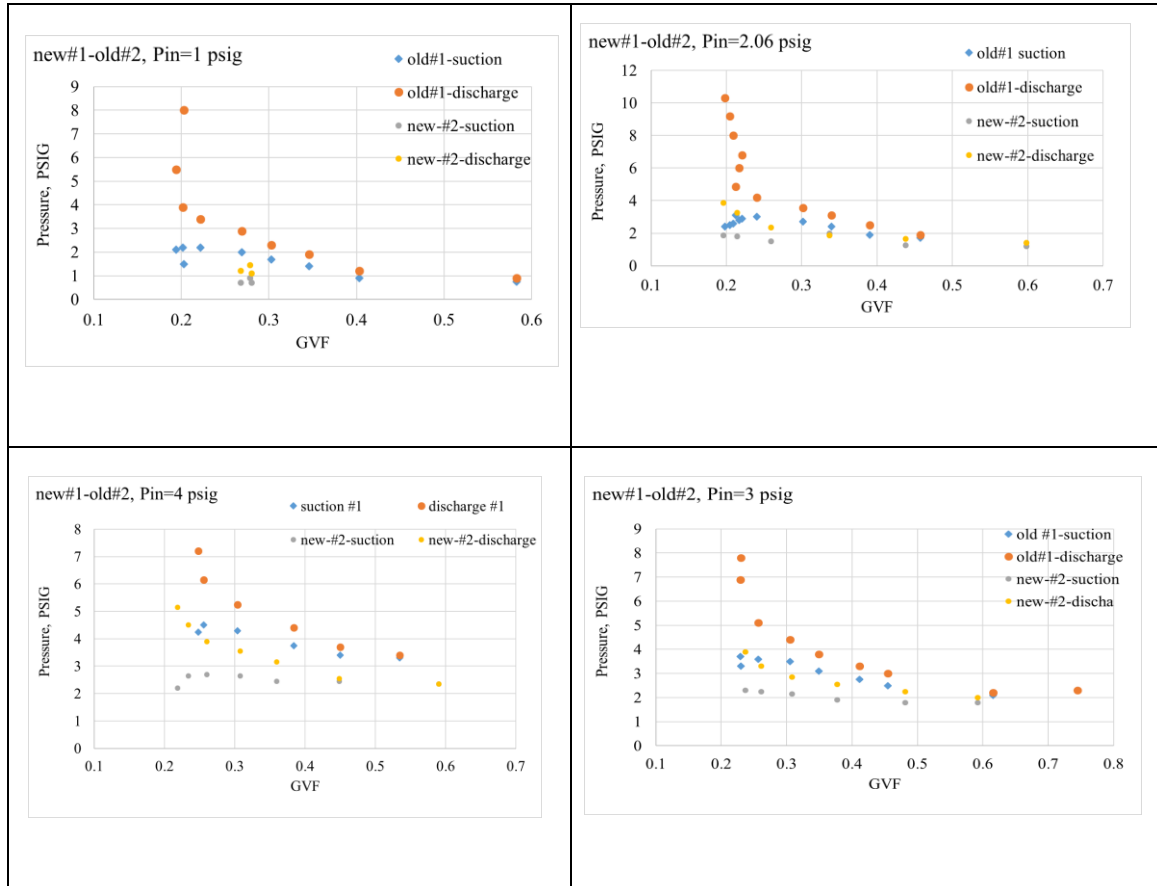


Figure 4.48: Comparison of the suction and discharge pressure of the 4-stage pump downstream of homogenizer#1 and Homogenizer #2

A comparison between the pressure difference across the 4-stage pump installed downstream of the certain homogenizer is shown in figure 4.49. In this figure, the pressure difference of the 4-stage pump when it has no homogenizer is also shown. It can be seen that both homogenizers improve the performance only at low GVF. At high GVF, the performance is about the same.

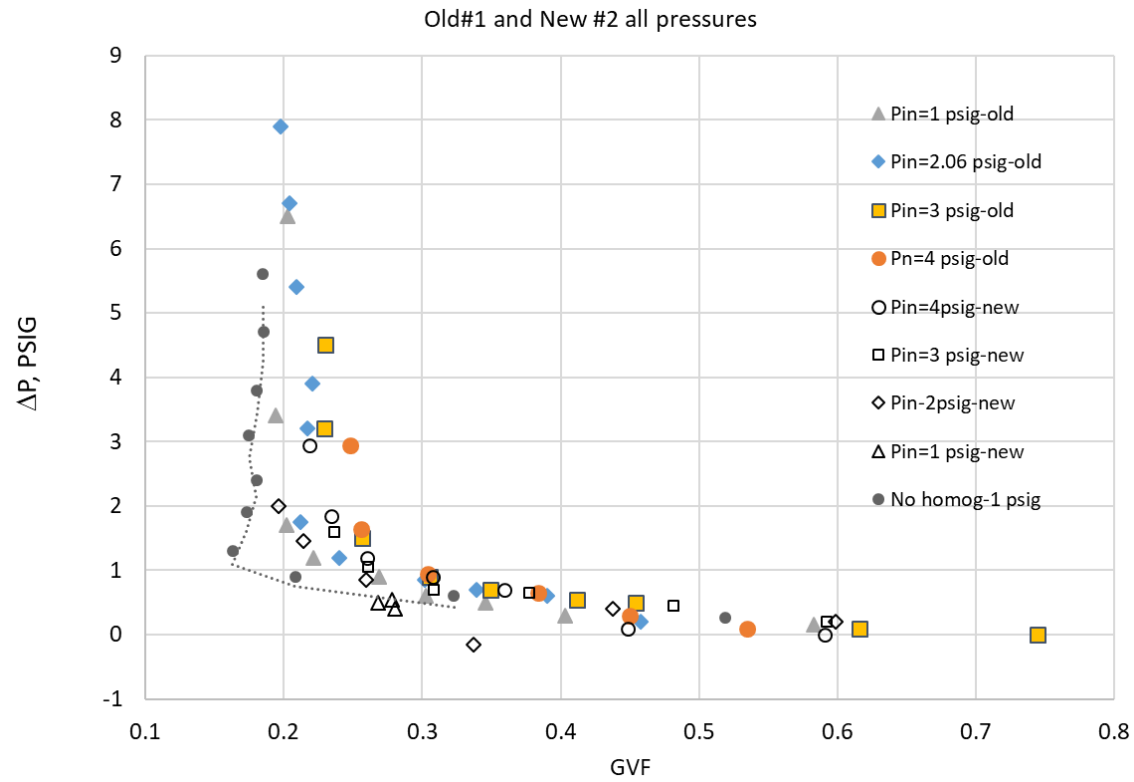


Figure 4.49: Pressure difference across the 4-stage pump with and without homogenizers

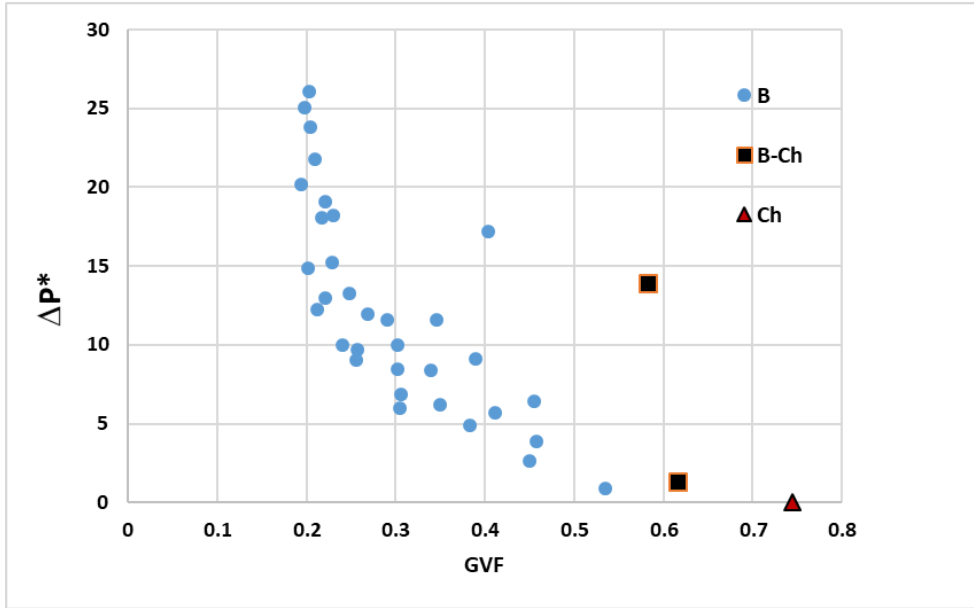


Figure 4.50: Normalized pressure across the 4-stage pump downstream of homogenizer#1 versus GVF. The flow pattern is in the suction of the 4-stage pump and discharge of the homogenizer (all pressure data)

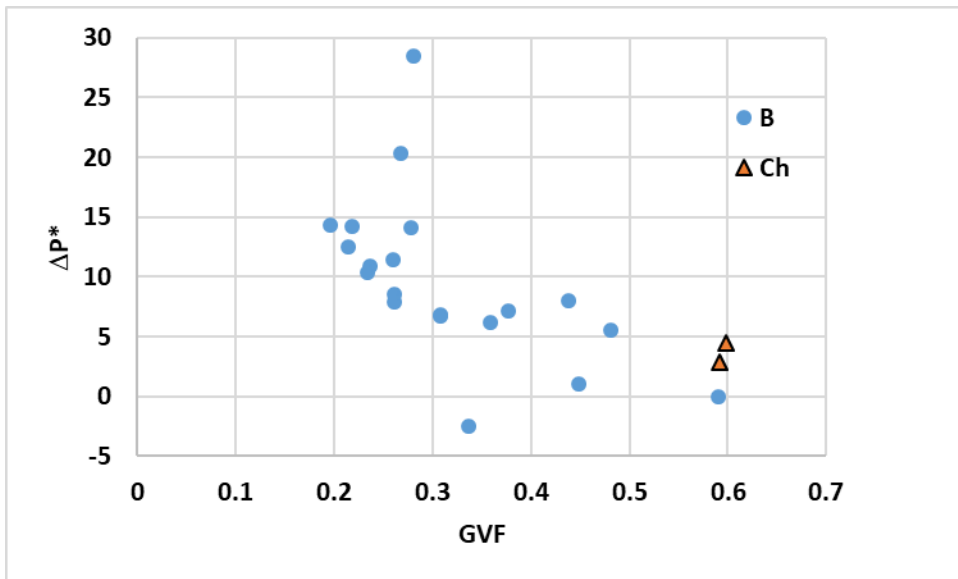


Figure 4.51: Normalized pressure across the 4-stage pump downstream of homogenizer#2 versus GVF. The flow pattern is in the suction of the 4-stage pump and discharge of the homogenizer (all pressure data)

4.9 Evaluation of Homogenizer #1 With Respect To Degree Of Perforation:

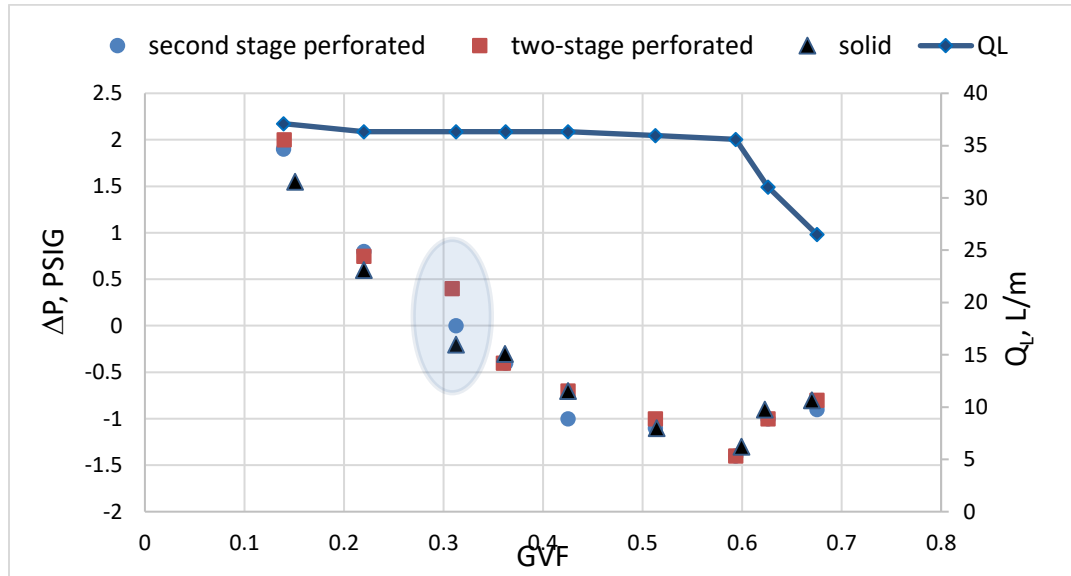


Figure 4.52: Performance comparison at same conditions between perforated second stage and two stages perforated and solid impeller.

Figure 4.52 shows comparison of pressure difference for three combinations of impellers at the same operation conditions for homogenizer #1 (second stage perforated, two stages perforated and two stages solid), it is clear that for GVF less than the critical value 0.35 (the GVF at which the discharge pressure becomes negative) the 2 stages perforated gives better performance. The points shown in oval illustrate how the pressure difference is highest for the case of two stages perforated then the second stage perforated and the least one for the solid or unperforated impeller.

4.10 Observed flow patterns in the experiments

In this part, the flow pattern observed are reported for different cases, each point of the measurements has two photos for the flow pattern before and after the homogenizer, some points are selected as a samples for the flow pattern because each point has two photos for the flow pattern, and hundreds of readings has been done in this work.

At low gas void fraction the pressure difference between the outlet and inlet of the homogenizer is positive means the pump is working fine and adding energy for the fluid so that the pressure after the pump is greater than before, in this case it is observed that the flow pattern after the pump is bubbly flow in most cases. On the other hand when the gas void fraction increases then the outlet pressure decreases and the inlet pressure increases until they are equal at the critical gas void fraction at which the pressure difference is zero. Zero or minus pressure difference occurs at gas void fraction equal or greater than the critical gas void fraction means that the homogenizer fails to deliver the fluid and at this case it is observed that the flow pattern is annular or churn flow.

Some points are presented in this part at the same gas void fraction and three different impeller combinations.

- two impellers are perforated
- one impeller perforated and the other is solid
- two impellers are solid

Table 4.3: Observed flow pattern at inlet and outlet of the homogenizer at GVF=0.2

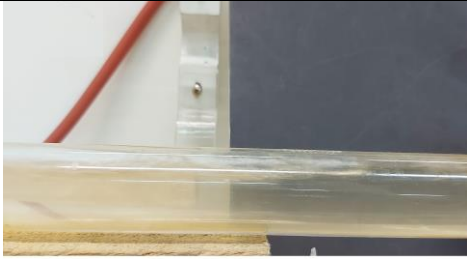
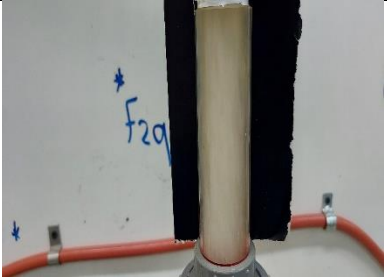
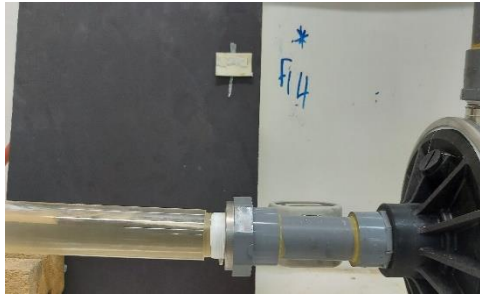
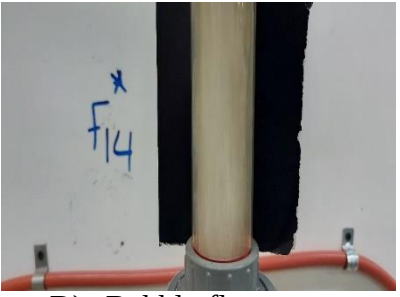
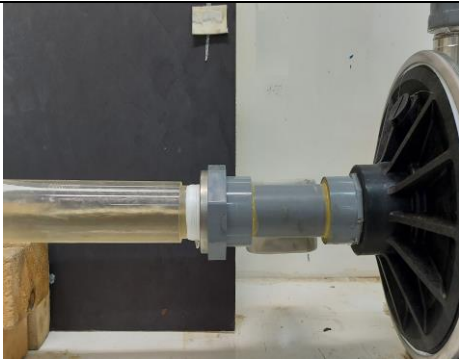
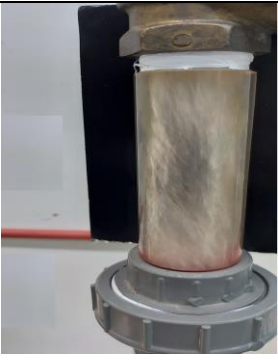
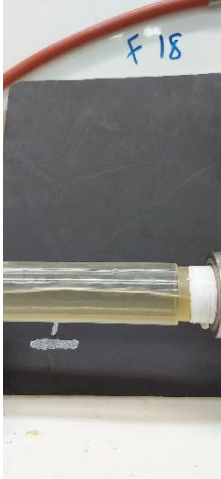
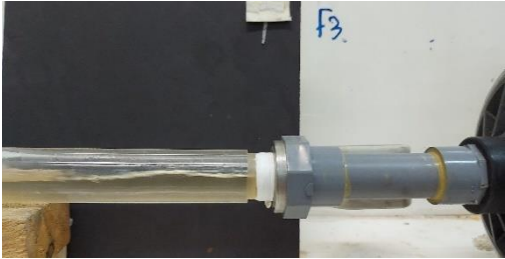
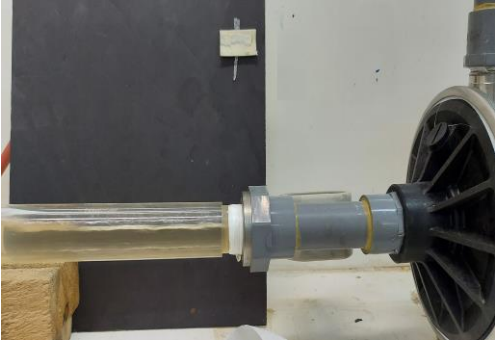
	Inlet flow pattern	Outlet flow pattern
2 stages perforated	 A) Stratified flow	 B) Bubbly flow
One perforated the other is solid	 C) Stratified flow	 D) Bubbly flow
2 stags are solid	 E) Stratified flow	 F) Annular flow

Table 4.4: Observed flow pattern at inlet and outlet of the homogenizer at GVF=0.6

	Inlet flow pattern	Outlet flow pattern
2 stages perforated	 A) Stratified flow	 B) Bubbly-Churn flow
One perforated the other is solid	 C) Stratified flow	 D) Churn flow
2 stags are solid	 E) Stratified flow	 F) Churn flow

4.11 Uncertainty analysis

Experimental uncertainty analysis delivers a technique for predicting the uncertainty of a variable based on its experimental measurement device uncertainties.

Figures 3.6 and 3.7 show the setup used in the experiment, several measurement devices used as it follows:

- Inline digital turbine flow meter to measure water flow rate figure 4.53

Flow ports:	1" NPT or BSPP female inlet and outlet ports
Flow Range:	2 to 35 U.S. GPM / 7.6 to 132.5 LPM
Working pressure:	50 PSI (3.4 Bar)MAX UL
Operating temperature:	-40°F to 176°F (-40°C to 60°C)
Measurement accuracy:	±0.01



Figure 4.53: Digital turbine flowmeter

- Omega digital pressure gauge (DPG3500B-60G) figure 4.54

Pressure range:	0-60 psi
Operating Temperature:	-20 to 85°C (-4 to 185°F)
Measurement accuracy:	±0.01
Material: DPG3500:	ABS/polycarbonate NEMA 4X (IP66) case, rear gasket, polycarbonate label



Figure 4.54: Omega pressure gauge

- Air pressure gauge figure 4.55

Measurement range: 0-6 bar
 Connection: radial - 1/4 "M
 Material: stainless steel case
 Measurement accuracy: ± 0.1 bar



Figure 4.55: Air pressure gauge

- Air flow meter figure 4.56

Measurement range: 0-80 L/m
 Material: Acrylic
 Operating Temperature: +65C
 Measurement accuracy: ± 2.5 L/m



Figure 4.56: Air flowmeter

The uncertainty for the measured variables estimated based on the accuracy of the measuring devices reported by the manufacturer, whereas the uncertainty for the dependent parameter has been calculated. Table 4.5 shows the estimated uncertainties of measured and calculated parameters.

Table 4.5 Estimated uncertainties for measured and calculated values

Parameter	Estimated Uncertainty
Water flowrate	± 0.01 Gl/min
Water inlet and outlet pressure	± 0.01 Psig
Air flowrate	± 2.5 L/min
Air inlet pressure	± 0.1 bar
Pressure difference	± 0.2 bar

CHAPTER 5

CONCLUSION & FUTURE WORK

5.1 Conclusion

5.1.1 Thermo hydraulic modeling concluded remarks

- Different FP map models were used to predict the FP inside the absorber; the flow pattern map used in this study is Wojtan et al. [67] flow pattern map because it was developed for diabatic conditions when there is a heat transfer to the working fluid, which is the case in direct steam generation using solar collectors. The results were compared with the Zhang et al. and Barneia models in table 4.1. In Wojtan et al. flow pattern map, the dry out region appeared because, in this flow pattern map, the effect of heat flux has been considered, which makes this flow pattern map more realistic than the adiabatic flow pattern maps.
- Different HTC correlations were tested and validated for refrigerant R134a and water. The Shah [80] correlation showed a good agreement with the experimental data for water, while the Wojtan–Ursenbacher_Thome model closely agreed with the experimental data for the refrigerant. The error percentage of HTC is fluctuating when the mass quality increases for all models; however, the Wojtan et al. model has the best agreement with the refrigerants experimental data. On the other hand, for water experimental data, the results showed that the Shah [80] correlation agrees more closely with the experimental data than that of Dengeler and Addoms.
- The wall and mixture temperatures are almost constant in this 2-phase region. After that, the quality increases above one, the fluid (steam) is then superheated to a

specific outlet temperature depending on the inlet conditions. In the 2-phase area, the surface temperature depends on the HTC model.

- The void fraction is a function of mass quality, which also increases from zero to one in the 2-phase region; the void fraction is very important in calculating the boiling regime. The slip ratio affects the void fraction for homogenous models. However, the Kroger and Zubair correlation does not give a good prediction for the void fraction at high qualities.
- In this study, numerous models for PD were tested and validated with the experimental data. Lockhart–Martinelli and homogenous models showed a closer agreement with the experimental data than Friedel and Muller, Stienhagen, and Heck. Gronnerud gives a good agreement with the experimental data until a mass quality of 0.5, after which results demonstrate higher disagreement with the experimental data. Different operational conditions were tested with direct application to the DSG process.
- Part of the experimental data used to validate the work on this study for direct steam generation is for the Plataforma Solar de Almeria [58] in which the fluid used is water, pipe diameter=50 mm, L=510 m.
- Thermo-hydraulic model for a direct steam generation was developed using MATLAB interface to calculate all important parameters such as mass quality, void fraction, two-phase pressure drop, two-phase heat transfer coefficient throughout the absorber tube. Moreover, the state of the steam produced is also can be determined by providing some inputs to the developed program, such as the inlet conditions of the water entering and the absorber and the absorber dimensions.

5.1.2 Homogenizer designing, testing and validation concluded remarks

- An intermediate flow homogenizer was designed and examined to improve the performance of pumping a two-phase mixture.
- A flexible experimental setup was designed and implemented in the fluid mechanics' lab in the Mechanical Engineering Department to test and validate the performance of the homogenizer for single and two-phase flow configurations figures 3.6 & 3.7
- Experimental evaluation of the level of mixing and the trade-off between head generation and homogenization investigated in the experimental work as a function of the GVF.
- The experimental facility involves two stages radial impeller centrifugal pump (as a homogenizer) and a 4-stage, radial impeller centrifugal pump (as the main pump).
- Homogenizer #1 (2-stage pump) was tested for different combinations (perforated and unperforated impeller for single and two-phase flow).
 - Two stages unperforated
 - The outer stage perforated with four holes and the inner perforated with seven holes
 - Outer stage perforated (7holes) and the second stage unperforated.
- At the lowest liquid flow rate, the perforated impeller performs better than the unperforated indicated by the higher discharge pressure over the suction pressure up to $GVF = 0.8$. However, when the liquid flow rate increases, the point at which the inlet and outlet pressure are equal moves to a smaller value of GVF.

- For the dimensionless pressure above 100, the outlet flow pattern is bubbly, while above zero and below 100, the flow pattern is Churn or annular.
- Homogenizer #2 (4-stage pump) was tested for different combinations
 - 4-stage pump without homogenizer (single phase)
 - 4-stage pump with the first stage perforated (single phase)
 - 4-stage pump, first stage with and without perforation at low and high inlet air pressure (constant liquid and variable gas flowrate)
 - Homogenizer #1 followed by 4-stage pump (constant inlet liquid pressure)
- At low GVF the performance of the 4-stage pump when it is installed downstream of the homogenizer 1 is better than when it was alone without any homogenizer at the same inlet pressure.
- The performance gets worse as the GVF increases, especially at high liquid flow rates. However, increasing the system pressure improves the pump performance under two-phase flow conditions at all GVF and liquid flow rates.
- Homogenizer improves the performance marginally, not completely, and at low GVF only.
- The pump power must not be rated for liquid then used for gas and liquid. The power needed to pump gas and liquid much larger, and this should be considered carefully (thermodynamics).
- The controlling factors are the inlet pressure, vane design, and speed.

5.2 Future work

- Investigate performance of the two-phase flow with high-speed pump.
- New design of a pump with power rated for gas and liquid.

- Using machine learning and Artificial Neural Network to optimize the power generated using all available correlations and models.

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Publications

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